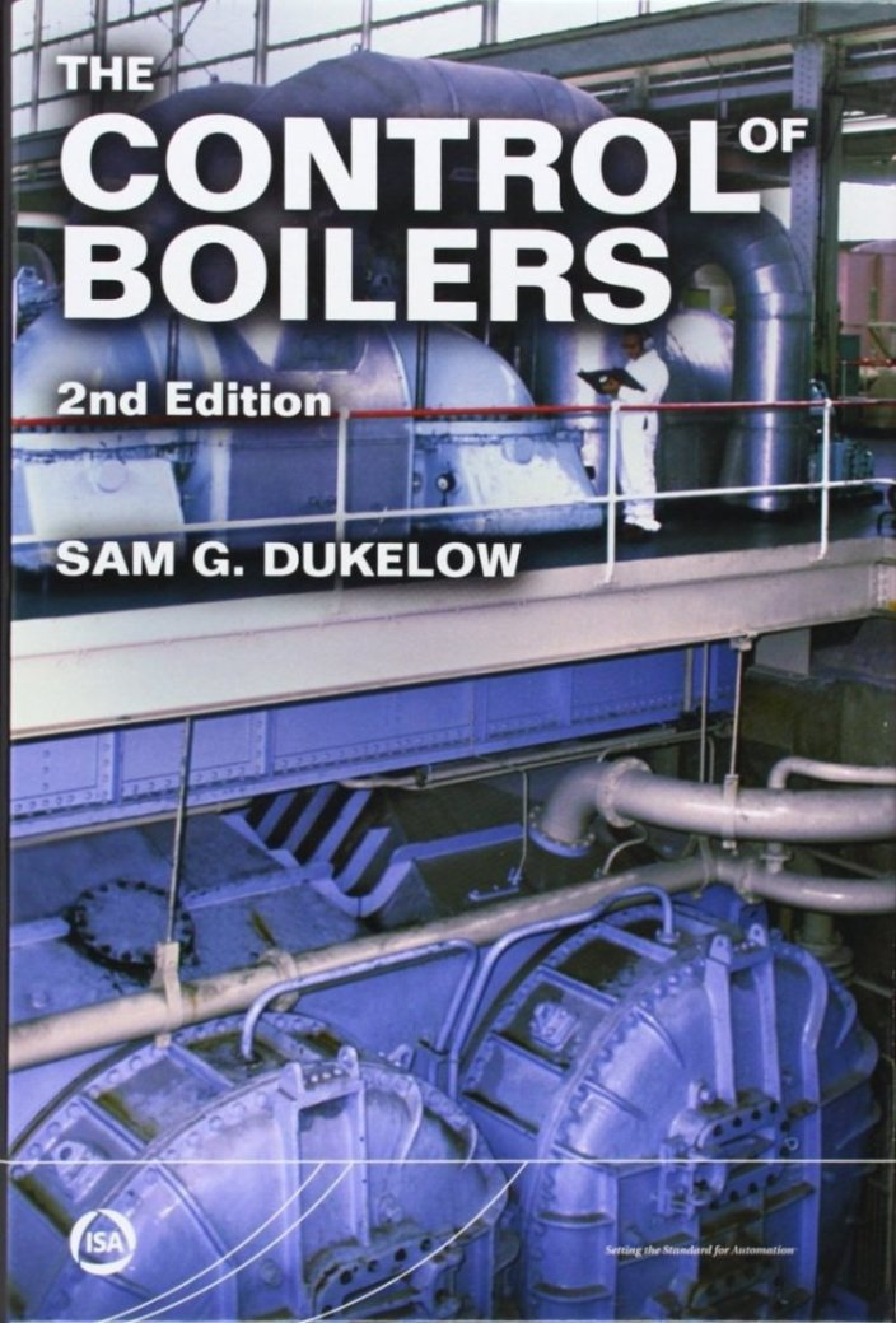
THE CONTROL OF BOILERS

2nd Edition

SAM G. DUKELOW



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THE CONTROL OF BOILERS

2nd Edition

SAM G. DUKELOW

ISA–The Instrumentation, Systems, and Automation Society

THE CONTROL OF BOILERS

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We would like to thank the many suppliers who provided material for this book, and we regret any we may have inadvertently failed to credit for an illustration. On notification we shall insert a correction in any subsequent printings.

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Preface to Second Edition

Five years have passed since the first edition of this book, and I have continued to learn as I have become older and wiser. In the third paragraph of the preface to the first edition, I implied that what I have done in this second edition was impossible. I want to eat those words.

The cartoon by Gus Shaw on the opposite page tells the story. During his work with Bailey Meter Co. (prior to Bailey Controls Co.), Gus made several great cartoons on this theme. A complete study of ''The Control of Boilers'' must include the ''starting up'' phase of the process. One of the purposes of this edition is to include some basic information on that digital phase of the operation in addition to the modulating ''on-line'' operation covered in the original edition.

This results in the sections and subsections covering interlocks, burner start-up and management, and the management of the start-up and operation of pulverizers and other fuel-burning equipment. Along with this is the recognition of applicable safety codes of the National Fire Protection Association (NFPA).

Another purpose of this edition is the extension of the ''on-line'' aspects of boiler control into the arena of the larger-capacity electric utility boilers. The results of this are new sections covering the firing rate demand for utility boilers and steam temperature control. The section covering furnace pressure control has been expanded to include implosion protection. The section covering the control of pulverized coal firing has been expanded to include cyclone furnaces and their control and the compartmented windbox boiler and its control.

And, as I said above, I have become older and wiser in the past five years. My ASME membership card now says 51 years, my ISA membership card says 42 years, and I have had time to rethink some the things I thought I already knew. In this last five years my work has taken me to installations involving chain grate stokers, pulverized coal-fired boilers, gas-fired boilers, the steam power cycle of a nuclear breeder reactor, and process heaters fired with by-product gas. In the past few months, my work with distributed digital system on an electric utility unit demonstrated the differences of working with a digital system as compared to an analog system. Three of the above assignments involved the investigation of furnace explosions that caused major damage. The investigation included considerable dialogue with everybody involved, as I continued to learn.

In addition, the past five years have brought me approximately 25 teaching assignments covering boiler control for various types of boilers. This has involved me with approximately 500 students with various industrial and utility boiler backgrounds. In each of these areas, and in talking to the people involved, I have gained new insights and thus continued my learning.

So now this second edition of *The Control of Boilers*. There is less coverage of the field of ''start-up'' and utility boiler control than these subjects deserve, but I believe that the basics included will help tie the whole subject together. There is still much to be written. It is my understanding that a much more detailed text in the area of burner start-up and management is in process. An expansion in the area of utility boiler control directed at cogeneration, coal gasification combined cycles, low NOx control, flue gas scrubbers, fluidized bed boilers and their control, expert systems, artificial intelligence, and power plant unit performance analysis would be welcome. As indicated in the preface to the first edition of this book, the whole field of energy management for least cost operation of boilers and HVAC should also be given attention.

The boiler control field, and the rest of the I&C field, has had its revolution. It is now a distributed digital ''microprocessor'' world. But as with all control systems, no matter what hardware or software is used, the control application of the job to be done must be the major focus and must be defined. This book is a discussion of that application area. The hardware-software combination of today's world unleashes the control application engineer from the bonds of hardware and hardware installation cost contraints. The engineer can now concentrate on how best to control the boiler and other aspects of the power process.

Again I thank all those who, both knowingly and unknowingly, helped me along the way. I particularly thank Paul Kenny of Forney Engineering Co.; Ollie Durrant, now retired from Babcock & Wilcox Co.; and Russ Beal, now retired from Bailey Controls Co., who furnished me with source documents.

My intent is to emphasize the basic ideas involved in boiler control and thus stimulate the reader to expand his or her knowledge with more detailed study. If this book provides a jump-start to beginners in the field of boiler control application, or adds new insights to those experienced readers, I shall have accomplished my purpose.

S.G. Dukelow
Hutchinson, Kansas

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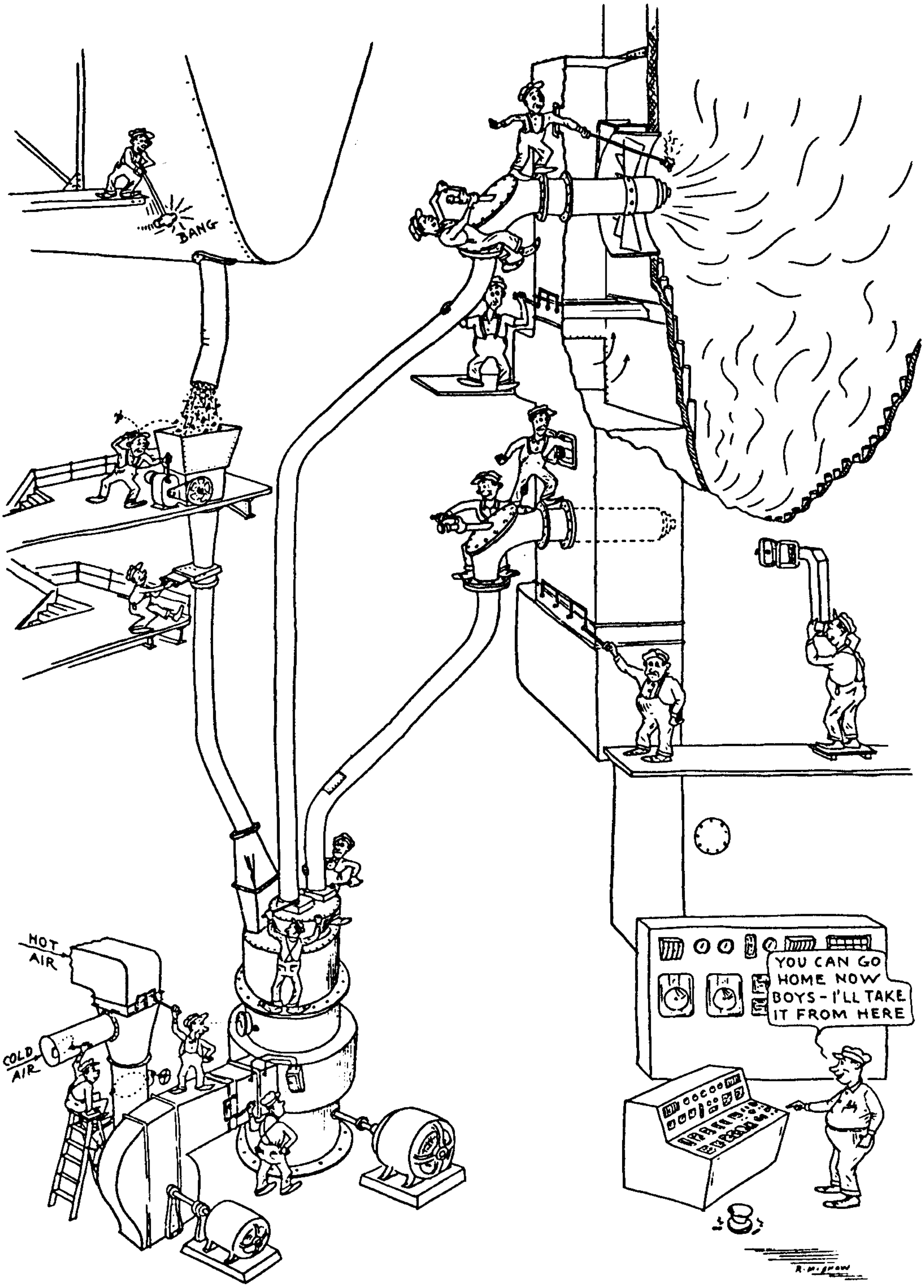

BANG
HOT AIR
COLD AIR
YOU CAN GO HOME NOW BOYS - I'LL TAKE IT FROM HERE

Section 2
Boiler Basics and the Steaming Process

2-1 The Basic Steaming Process

In the conversion of water from its liquid phase to steam (its vapor phase), heat is added to initially increase the water temperature to the boiling point temperature. This heat that raises the temperature of the water is known as sensible heat. The boiling point temperature is 212°F at atmospheric pressure and rises as the pressure in the system is increased. The boiling point temperature is also known as the saturation temperature of the steam that is produced. The relationships between the saturation temperatures and pressures of steam are fixed thermodynamic properties of steam.

As the conversion from the liquid phase (water) to the vapor phase (steam) begins, the temperature no longer changes with the addition of heat. The fluid exists at the saturation temperature-pressure relationship during the entire conversion of the water to steam. The heat that is added in converting from the liquid to the vapor phase at constant temperature is called the latent heat of evaporation. Steam that is fully vaporized but has not been heated to a temperature above the saturation temperature is called dry saturated steam. Steam that is not fully vaporized is called "wet" steam. The percentage by weight of the water droplets in the wet steam is known as the % moisture. The % quality of wet steam is obtained by subtracting the % moisture from 100.

The total amount of heat in a quantity of dry saturated steam includes the amount of sensible heat above 32°F and the latent heat of evaporation. Generally, as the pressure of dry saturated steam increases, the amount of sensible heat increases and the amount of latent heat decreases. The relationships between the various thermodynamic steam properties are shown in Tables 2-1 and 2-2.

By adding additional sensible heat to dry saturated steam, the temperature can be increased above the saturation temperature. Steam that is heated above the saturation temperature is called superheated steam. The effect on the thermodynamic properties by superheating steam is shown in Table 2-3. Note that superheating increases the total heat or enthalpy (h) of the steam. Superheating also causes the steam to expand, increasing its specific volume (ft^3/lb).

2-2 The Basic Boiler

A basic diagram of a boiler is shown in Figure 2-1. This diagram shows that a boiler comprises two separate systems. One system is the steam-water system, which is also called the water side of the boiler. Into this system water is introduced and, upon receiving heat that is transferred through a solid metal barrier, is heated, converted to steam, and leaves the system in the form of steam.

The other system of a boiler is the fuel-air-flue gas system, which is also called the fire side of the boiler. This system provides the heat that is transferred to the water. The inputs to this system are fuel and the necessary air required to burn the fuel.

In this system the fuel and air are thoroughly mixed and ignited in a furnace. The resulting combustion converts the chemical energy of the fuel to thermal or heat energy. The furnace is usually lined with heat transfer surface in the form of water-steam circulating tubes. These tubes receive heat radiating from the flame and transfer it to the water-side system. The gases resulting from the combustion, known as the flue gases, are cooled by the transfer of their heat by what is known as the radiant heat transfer surface. The gases leave the furnace and pass through additional heating surface that is in the form of water-steam circulating tubes. In this area the surfaces cannot "see" the flame, and the heat is transferred by convection. Also

Table 2-1
Saturation: Temperatures

Temp F t	Abs Press. lb / Sq In. p	Specific Volume: Sat. Liquid v_f	Specific Volume: Evap v_{fg}	Specific Volume: Sat. Vapor v_g	Enthalpy: Sat. Liquid h_f	Enthalpy: Evap h_{fg}	Enthalpy: Sat. Vapor h_g	Entropy: Sat. Liquid s_f	Entropy: Evap s_{fg}	Entropy: Sat. Vapor s_g	Temp F t
32	0.08854	0.01602	3306	3306	0.00	1075.8	1075.8	0.0000	2.1877	2.1877	32
35	0.09995	0.01602	2947	2947	3.02	1074.1	1077.1	0.0061	2.1709	2.1770	35
40	0.12170	0.01602	2444	2444	8.05	1071.3	1079.3	0.0162	2.1435	2.1597	40
45	0.14752	0.01602	2036.4	2036.4	13.06	1068.4	1081.5	0.0262	2.1167	2.1429	45
50	0.17811	0.01603	1703.2	1703.2	18.07	1065.6	1083.7	0.0361	2.0903	2.1264	50
60	0.2563	0.01604	1206.6	1206.7	28.06	1059.9	1088.0	0.0555	2.0393	2.0948	60
70	0.3631	0.01606	867.8	867.9	38.04	1054.3	1092.3	0.0745	1.9902	2.0647	70
80	0.5069	0.01608	633.1	633.1	48.02	1048.6	1096.6	0.0932	1.9428	2.0360	80
90	0.6982	0.01610	468.0	468.0	57.99	1042.9	1100.9	0.1115	1.8972	2.0087	90
100	0.9492	0.01613	350.3	350.4	67.97	1037.2	1105.2	0.1295	1.8531	1.9826	100
110	1.2748	0.01617	265.3	265.4	77.94	1031.6	1109.5	0.1471	1.8106	1.9577	110
120	1.6924	0.01620	203.25	203.27	87.92	1025.8	1113.7	0.1645	1.7694	1.9339	120
130	2.2225	0.01625	157.32	157.34	97.90	1020.0	1117.9	0.1816	1.7296	1.9112	130
140	2.8886	0.01629	122.99	123.01	107.89	1014.1	1122.0	0.1984	1.6910	1.8894	140
150	3.718	0.01634	97.06	97.07	117.89	1008.2	1126.1	0.2149	1.6537	1.8685	150
160	4.741	0.01639	77.27	77.29	127.89	1002.3	1130.2	0.2311	1.6174	1.8485	160
170	5.992	0.01645	62.04	62.06	137.90	996.3	1134.2	0.2472	1.5822	1.8293	170
180	7.510	0.01651	50.21	50.23	147.92	990.2	1138.1	0.2630	1.5480	1.8109	180
190	9.339	0.01657	40.94	40.96	157.95	984.1	1142.0	0.2785	1.5147	1.7932	190
200	11.526	0.01663	33.62	33.64	167.99	977.9	1145.9	0.2938	1.4824	1.7762	200
210	14.123	0.01670	27.80	27.82	178.05	971.6	1149.7	0.3090	1.4508	1.7598	210
212	14.696	0.01672	26.78	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	212
220	17.186	0.01677	23.13	23.15	188.13	965.2	1153.4	0.3239	1.4201	1.7440	220
230	20.780	0.01684	19.365	19.382	198.23	958.8	1157.0	0.3387	1.3901	1.7288	230
240	24.969	0.01692	16.306	16.323	208.34	952.2	1160.5	0.3531	1.3609	1.7140	240
250	29.825	0.01700	13.804	13.821	218.48	945.5	1164.0	0.3675	1.3323	1.6998	250
260	35.429	0.01709	11.746	11.763	228.64	938.7	1167.3	0.3817	1.3043	1.6860	260
270	41.858	0.01717	10.044	10.061	238.84	931.8	1170.6	0.3958	1.2769	1.6727	270
280	49.203	0.01726	8.628	8.645	249.06	924.7	1173.8	0.4096	1.2501	1.6597	280
290	57.556	0.01735	7.444	7.461	259.31	917.5	1176.8	0.4234	1.2238	1.6472	290
300	67.013	0.01745	6.449	6.466	269.59	910.1	1179.7	0.4369	1.1980	1.6350	300
320	89.66	0.01765	4.896	4.914	290.28	894.9	1185.2	0.4637	1.1478	1.6115	320
340	118.01	0.01787	3.770	3.788	311.13	879.0	1190.1	0.4900	1.0992	1.5891	340
360	153.04	0.01811	2.939	2.957	332.18	862.2	1194.4	0.5158	1.0519	1.5677	360
380	195.77	0.01836	2.317	2.335	353.45	844.6	1198.1	0.5413	1.0059	1.5471	380
400	247.31	0.01864	1.8447	1.8633	374.97	826.0	1201.0	0.5664	0.9608	1.5272	400
420	308.83	0.01894	1.4811	1.5000	396.77	806.3	1203.1	0.5912	0.9166	1.5078	420
440	381.59	0.01926	1.1979	1.2171	418.90	785.4	1204.3	0.6158	0.8730	1.4887	440
460	466.9	0.0196	0.9748	0.9944	441.4	763.2	1204.6	0.6402	0.8298	1.4700	460
480	566.1	0.0200	0.7972	0.8172	464.4	739.4	1203.7	0.6645	0.7868	1.4513	480
500	680.8	0.0204	0.6545	0.6749	487.8	713.9	1201.7	0.6887	0.7438	1.4325	500
520	812.4	0.0209	0.5385	0.5594	511.9	686.4	1198.2	0.7130	0.7006	1.4136	520
540	962.5	0.0215	0.4434	0.4649	536.6	656.6	1193.2	0.7374	0.6568	1.3942	540
560	1133.1	0.0221	0.3647	0.3868	562.2	624.2	1186.4	0.7621	0.6121	1.3742	560
580	1325.8	0.0228	0.2989	0.3217	588.9	588.4	1177.3	0.7872	0.5659	1.3532	580
600	1542.9	0.0236	0.2432	0.2668	617.0	548.5	1165.5	0.8131	0.5176	1.3307	600
620	1786.6	0.0247	0.1955	0.2201	646.7	503.6	1150.3	0.8398	0.4664	1.3062	620
640	2059.7	0.0260	0.1538	0.1798	678.6	452.0	1130.5	0.8679	0.4110	1.2789	640
660	2365.4	0.0278	0.1165	0.1442	714.2	390.2	1104.4	0.8987	0.3485	1.2472	660
680	2708.1	0.0305	0.0810	0.1115	757.3	309.9	1067.2	0.9351	0.2719	1.2071	680
700	3093.7	0.0369	0.0392	0.0761	823.3	172.1	995.4	0.9905	0.1484	1.1389	700
705.4	3206.2	0.0503	0	0.0503	902.7	0	902.7	1.0580	0	1.0580	705.4

in this area, known as the convection heating surface, additional amounts of heat are transferred to the water side of the boiler. This heat transfer further cools the flue gases, which then leave the boiler.

Since heat transfer depends upon a temperature difference as a "driving force," with the simple boiler described the flue gases can be cooled only to a temperature that is at some level above the temperature of the steam-water system. The temperature of the flue gases determines the amount of heat remaining in these gases, so the heat loss in the boiler flue gases is determined to some extent by the saturation temperature in the steam-water system.

The process of adding heat to convert water to steam has a time constant that depends upon the specific characteristics of the installation. The factors affecting this time constant

Table 2-2
Saturation: Pressures

Abs Press. Lb Sq In. *p*	Temp F *t*	Specific Volume Sat. Liquid v_f	Specific Volume Sat. Vapor v_g	Enthalpy Sat. Liquid h_f	Enthalpy Evap h_{fg}	Enthalpy Sat. Vapor h_g	Entropy Sat. Liquid s_f	Entropy Evap s_{fg}	Entropy Sat. Vapor s_g	Internal Energy Sat. Liquid u_f	Internal Energy Evap u_{fg}	Internal Energy Sat. Vapor u_g	Abs Press. Lb Sq In. *p*
1.0	101.74	0.01614	333.6	69.70	1036.3	1106.0	0.1326	1.8456	1.9782	69.70	974.6	1044.3	**1.0**
2.0	126.08	0.01623	173.73	93.99	1022.2	1116.2	0.1749	1.7451	1.9200	93.98	957.9	1051.9	**2.0**
3.0	141.48	0.01630	118.71	109.37	1013.2	1122.6	0.2008	1.6855	1.8863	109.36	947.3	1056.7	**3.0**
4.0	152.97	0.01636	90.63	120.86	1006.4	1127.3	0.2198	1.6427	1.8625	120.85	939.3	1060.2	**4.0**
5.0	162.24	0.01640	73.52	130.13	1001.0	1131.1	0.2347	1.6094	1.8441	130.12	933.0	1063.1	**5.0**
6.0	170.06	0.01645	61.98	137.96	996.2	1134.2	0.2472	1.5820	1.8292	137.94	927.5	1065.4	**6.0**
7.0	176.85	0.01649	53.64	144.76	992.1	1136.9	0.2581	1.5586	1.8167	144.74	922.7	1067.4	**7.0**
8.0	182.86	0.01653	47.34	150.79	988.5	1139.3	0.2674	1.5383	1.8057	150.77	918.4	1069.2	**8.0**
9.0	188.28	0.01656	42.40	156.22	985.2	1141.4	0.2759	1.5203	1.7962	156.19	914.6	1070.8	**9.0**
10	193.21	0.01659	38.42	161.17	982.1	1143.3	0.2835	1.5041	1.7876	161.14	911.1	1072.2	**10**
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	180.02	897.5	1077.5	**14.696**
15	213.03	0.01672	26.29	181.11	969.7	1150.8	0.3135	1.4415	1.7549	181.06	896.7	1077.8	**15**
20	227.96	0.01683	20.089	196.16	960.1	1156.3	0.3356	1.3962	1.7319	196.10	885.8	1081.9	**20**
30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993	218.73	869.1	1087.8	**30**
40	267.25	0.01715	10.498	236.03	933.7	1169.7	0.3919	1.2844	1.6763	235.90	856.1	1092.0	**40**
50	281.01	0.01727	8.515	250.09	924.0	1174.1	0.4110	1.2474	1.6585	249.93	845.4	1095.3	**50**
60	292.71	0.01738	7.175	262.09	915.5	1177.6	0.4270	1.2168	1.6438	261.90	836.0	1097.9	**60**
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315	272.38	827.8	1100.2	**70**
80	312.03	0.01757	5.472	282.02	901.1	1183.1	0.4531	1.1676	1.6207	281.76	820.3	1102.1	**80**
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1.6112	290.27	813.4	1103.7	**90**
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026	298.08	807.1	1105.2	**100**
120	341.25	0.01789	3.728	312.44	877.9	[illegible]	0.4916	1.0962	1.5878	312.05	795.6	1107.6	**120**
140	353.02	0.01802	3.220	324.82	868.2	1193.0	0.5069	1.0682	1.5751	324.35	785.2	1109.6	**140**
160	[illegible]	0.01815	2.834	335.93	859.2	1195.1	0.5204	1.0436	1.5640	335.39	775.8	1111.2	**160**
180	373.06	0.01827	2.532	346.03	850.8	1196.9	0.5325	1.0217	1.5542	345.42	767.1	1112.5	**180**
200	381.79	0.01839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453	354.68	759.0	1113.7	**200**
250	400.95	0.01865	1.8438	376.00	825.1	1201.1	0.5675	0.9588	1.5263	375.14	740.7	1115.8	**250**
300	417.33	0.01890	1.5433	393.84	809.0	1202.8	0.5879	0.9225	1.5104	392.79	724.3	1117.1	**300**
350	431.72	0.01913	1.3260	409.69	794.2	1203.9	0.6056	0.8910	1.4966	408.45	709.6	1118.0	**350**
400	444.59	0.0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844	422.6	695.9	1118.5	**400**
450	456.28	0.0195	1.0320	437.2	767.4	1204.6	0.6356	0.8378	1.4734	435.5	683.2	1118.7	**450**
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	447.6	[illegible]	1118.6	**500**
550	476.93	0.0199	0.8422	460.8	743.1	1203.9	0.6608	0.7934	1.4542	458.8	[illegible]	1118.2	**550**
600	486.21	0.0201	0.7698	471.6	731.6	1203.2	0.6720	0.7734	1.4454	469.4	[illegible]	1117.7	**600**
700	503.10	0.0205	0.6554	491.5	709.7	1201.2	0.6925	0.7371	1.4296	488.8	627.5	1116.3	**700**
800	518.23	0.0209	0.5687	509.7	688.9	1198.6	0.7108	0.7045	1.4153	506.6	607.8	1114.4	**800**
900	531.98	0.0212	0.5006	526.6	668.8	1195.4	0.7275	0.6744	1.4020	523.1	589.0	1112.1	**900**
1000	544.61	0.0216	0.4456	542.4	649.4	1191.8	0.7430	0.6467	1.3897	538.4	571.0	1109.4	**1000**
1100	556.31	0.0220	0.4001	557.4	630.4	1187.8	0.7575	0.6205	1.3780	552.9	553.5	1106.4	**1100**
1200	567.22	0.0223	0.3619	571.7	611.7	1183.4	0.7711	0.5956	1.3667	566.7	536.3	1103.0	**1200**
1300	577.46	0.0227	0.3293	585.4	593.2	1178.6	0.7840	0.5719	1.3559	580.0	519.4	1099.4	**1300**
1400	587.10	0.0231	0.3012	598.7	574.7	1173.4	0.7963	0.5491	1.3454	592.7	502.7	1095.4	**1400**
1500	596.23	0.0235	0.2765	611.6	556.3	1167.9	0.8082	0.5269	1.3351	605.1	486.1	1091.2	**1500**
2000	635.82	0.0257	0.1878	671.7	463.4	1135.1	0.8619	0.4230	1.2849	662.2	403.4	1065.6	**2000**
2500	668.13	0.0287	0.1307	730.6	360.5	1091.1	0.9126	0.3197	1.2322	717.3	313.3	1030.6	**2500**
3000	695.36	0.0346	0.0858	802.5	217.8	1020.3	0.9731	0.1885	1.1615	783.4	189.3	972.7	**3000**
3206.2	705.40	0.0503	0.0503	902.7	0	902.7	1.0580	0	1.0580	872.9	0	872.9	**3206.2**

include the system heat storage, the heat transfer coefficients in different parts of the system, the masses of metal and refractory and their configuration, and various other factors. For the purpose of control, it is generally enough to understand that the complete time constant is a matter of minutes. Viewing the system as achieving 63 percent of total response in one fifth of the total time constant (a first-order system) is sufficient for most boiler control analysis procedures.

2-3 Heat Recovery from the Flue Gases

If the heat losses in the boiler flue gases are to be reduced, separate heat exchangers must be added to the simple boiler to recover more of the heat and further cool the flue gases. The combustion air preheater is one form of such an added heat exchanger. The application of an air preheater is shown in Figure 2-2. The flue gas leaves the boiler and passes through the

Table 2-3
Superheated Vapor

Abs Press. Lb/Sq In. (Sat. Temp)		Temperature, F 200	300	400	500	600	700	800	900	1000	1200	1400	1600
	v	392.6	452.3	512.0	571.6	631.2	690.8	750.4	809.9	869.5	988.7	1107.8	1227.0
1	h	1150.4	1195.8	1241.7	1288.3	1335.7	1383.8	1432.8	1482.7	1533.5	1637.7	1745.7	1857.5
(101.74)	s	2.0512	2.1153	2.1720	2.2233	2.2702	2.3137	2.3542	2.3923	2.4283	2.4952	2.5566	2.6137
	v	78.16	90.25	102.26	114.22	126.16	138.10	150.03	161.95	173.87	197.71	221.6	245.4
5	h	1148.8	1195.0	1241.2	1288.0	1335.4	1383.6	1432.7	1482.6	1533.4	1637.7	1745.7	1857.4
(162.24)	s	1.8718	1.9370	1.9942	2.0456	2.0927	2.1361	2.1767	2.2148	2.2509	2.3178	2.3792	2.4363
	v	38.85	45.00	51.04	57.05	63.03	69.01	74.98	80.95	86.92	98.84	110.77	122.69
10	h	1146.6	1193.9	1240.6	1287.5	1335.1	1383.4	1432.5	1482.4	1533.2	1637.6	1745.6	1857.3
(193.21)	s	1.7927	1.8595	1.9172	1.9689	2.0160	2.0596	2.1002	2.1383	2.1744	2.2413	2.3028	2.3598
	v		30.53	34.68	38.78	42.86	46.94	51.00	55.07	59.13	67.25	75.37	83.48
14.696	h		1192.8	1239.9	1287.1	1334.8	1383.2	1432.3	1482.3	1533.1	1637.5	1745.5	1857.3
(212.00)	s		1.8160	1.8743	1.9261	1.9734	2.0170	2.0576	2.0958	2.1319	2.1989	2.2603	2.3174
	v		22.36	25.43	28.46	31.47	34.47	37.46	40.45	43.44	49.41	55.37	61.34
20	h		1191.6	1239.2	1286.6	1334.4	1382.9	1432.1	1482.1	1533.0	1637.4	1745.4	1857.2
(227.96)	s		1.7808	1.8396	1.8918	1.9392	1.9829	2.0235	2.0618	2.0978	2.1648	2.2263	2.2834
	v		11.040	12.628	14.168	15.688	17.198	18.702	20.20	21.70	24.69	27.68	30.66
40	h		1186.8	1236.5	1284.8	1333.1	1381.9	1431.3	1481.4	1532.4	1637.0	1745.1	1857.0
(267.25)	s		1.6994	1.7608	1.8140	1.8619	1.9058	1.9467	1.9850	2.0212	2.0883	2.1498	2.2069
	v		7.259	8.357	9.403	10.427	11.441	12.449	13.452	14.454	16.451	18.446	20.44
60	h		1181.6	1233.6	1283.0	1331.8	1380.9	1430.5	1480.8	1531.9	1636.6	1744.8	1856.7
(292.71)	s		1.6492	1.7135	1.7678	1.8162	1.8605	1.9015	1.9400	1.9762	2.0434	2.1049	2.1621
	v			6.220	7.020	7.797	8.562	9.322	10.077	10.830	12.332	13.830	15.325
80	h			1230.7	1281.1	1330.5	1379.9	1429.7	1480.1	1531.3	1636.2	1744.5	1856.5
(312.03)	s			1.6791	1.7346	1.7836	1.8281	1.8694	1.9079	1.9442	2.0115	2.0731	2.1303
	v			4.937	5.589	6.218	6.835	7.446	8.052	8.656	9.860	11.060	12.258
100	h			1227.6	1279.1	1329.1	1378.9	1428.9	1479.5	1530.8	1635.7	1744.2	1856.2
(327.81)	s			1.6518	1.7085	1.7581	1.8029	1.8443	1.8829	1.9193	1.9867	2.0484	2.1056
	v			4.081	4.636	5.165	5.683	6.195	6.702	7.207	8.212	9.214	10.213
120	h			1224.4	1277.2	1327.7	1377.8	1428.1	1478.8	1530.2	1635.3	1743.9	1856.0
(341.25)	s			1.6287	1.6869	1.7370	1.7822	1.8237	1.8625	1.8990	1.9664	2.0281	2.0854
	v			3.468	3.954	4.413	4.861	5.301	5.738	6.172	7.035	7.895	8.752
140	h			1221.1	1275.2	1326.4	1376.8	1427.3	1478.2	1529.7	1634.9	1743.5	1855.7
(353.02)	s			1.6087	1.6683	1.7190	1.7645	1.8063	1.8451	1.8817	1.9493	2.0110	2.0683
	v			3.008	3.443	3.849	4.244	4.631	5.015	5.396	6.152	6.906	7.656
160	h			1217.6	1273.1	1325.0	1375.7	1426.4	1477.5	1529.1	1634.5	1743.2	1855.5
(363.53)	s			1.5908	1.6519	1.7033	1.7491	1.7911	1.8301	1.8667	1.9344	1.9962	2.0535
	v			2.649	3.044	3.411	3.764	4.110	4.452	4.792	5.466	6.136	6.804
180	h			1214.0	1271.0	1323.5	1374.7	1425.6	1476.8	1528.6	1634.1	1742.9	1855.2
(373.06)	s			1.5745	1.6373	1.6894	1.7355	1.7776	1.8167	1.8534	1.9212	1.9831	2.0404
	v			2.361	2.726	3.060	3.380	3.693	4.002	4.309	4.917	5.521	6.123
200	h			1210.3	1268.9	1322.1	1373.6	1424.8	1476.2	1528.0	1633.7	1742.6	1855.0
(381.79)	s			1.5594	1.6240	1.6767	1.7232	1.7655	1.8048	1.8415	1.9094	1.9713	2.0287
	v			2.125	2.465	2.772	3.066	3.352	3.634	3.913	4.467	5.017	5.565
220	h			1206.5	1266.7	1320.7	1372.6	1424.0	1475.5	1527.5	1633.3	1742.3	1854.7
(389.86)	s			1.5453	1.6117	1.6652	1.7120	1.7545	1.7939	1.8308	1.8987	1.9607	2.0181
	v			1.9276	2.247	2.533	2.804	3.068	3.327	3.584	4.093	4.597	5.100
240	h			1202.5	1264.5	1319.2	1371.5	1423.2	1474.8	1526.9	1632.9	1742.0	1854.5
(397.37)	s			1.5319	1.6003	1.6546	1.7017	1.7444	1.7839	1.8209	1.8889	1.9510	2.0084

v—Specific Volume (cu. ft./lb.)
h—Total Heat (BTU/lb.)
s—Entropy

Table 2-3
(continued)

Abs Press. Lb/Sq In. (Sat. Temp)		Temperature, F 500	600	700	800	900	1000	1200	1400	1600
	v	2.063	2.330	2.582	2.827	3.067	3.305	3.776	4.242	4.707
260	h	1262.3	1317.7	1370.4	1422.3	1474.2	1526.3	1632.5	1741.7	1854.2
(404.42)	s	1.5897	1.6447	1.6922	1.7352	1.7748	1.8118	1.8799	1.9420	1.9995
	v	1.9047	2.156	2.392	2.621	2.845	3.066	3.504	3.938	4.370
280	h	1260.0	1316.2	1369.4	1421.5	1473.5	1525.8	1632.1	1741.4	1854.0
(411.05)	s	1.5796	1.6354	1.6834	1.7265	1.7662	1.8033	1.8716	1.9337	1.9912
	v	1.7675	2.005	2.227	2.442	2.652	2.859	3.269	3.674	4.078
300	h	1257.6	1314.7	1368.3	1420.6	1472.8	1525.2	1631.7	1741.0	1853.7
(417.33)	s	1.5701	1.6268	1.6751	1.7184	1.7582	1.7954	1.8638	1.9260	1.9835
	v	1.4923	1.7036	1.8980	2.084	2.266	2.445	2.798	3.147	3.493
350	h	1251.5	1310.9	1365.5	1418.5	1471.1	1523.8	1630.7	1740.3	1853.1
(431.72)	s	1.5481	1.6070	1.6563	1.7002	1.7403	1.7777	1.8463	1.9086	1.9663
	v	1.2851	1.4770	1.6508	1.8161	1.9767	2.134	2.445	2.751	3.055
400	h	1245.1	1306.9	1362.7	1416.4	1469.4	1522.4	1629.6	1739.5	1852.5
(444.59)	s	1.5281	1.5894	1.6398	1.6842	1.7247	1.7623	1.8311	1.8936	1.9513
	v	1.1231	1.3005	1.4584	1.6074	1.7516	1.8928	2.170	2.443	2.714
450	h	1238.4	1302.8	1359.9	1414.3	1467.7	1521.0	1628.6	1738.7	1851.9
(456.28)	s	1.5095	1.5735	1.6250	1.6699	1.7108	1.7486	1.8177	1.8803	1.9381
	v	0.9927	1.1591	1.3044	1.4405	1.5715	1.6996	1.9504	2.197	2.442
500	h	1231.3	1298.6	1357.0	1412.1	1466.0	1519.6	1627.6	1737.9	1851.3
(467.01)	s	1.4919	1.5588	1.6115	1.6571	1.6982	1.7363	1.8056	1.8683	1.9262
	v	0.8852	1.0431	1.1783	1.3038	1.4241	1.5414	1.7706	1.9957	2.219
550	h	1223.7	1294.3	1354.0	1409.9	1464.3	1518.2	1626.6	1737.1	1850.6
(476.94)	s	1.4751	1.5451	1.5991	1.6452	1.6868	1.7250	1.7946	1.8575	1.9155
	v	0.7947	0.9463	1.0732	1.1899	1.3013	1.4096	1.6208	1.8279	2.033
600	h	1215.7	1289.9	1351.1	1407.7	1462.5	1516.7	1625.5	1736.3	1850.0
(486.21)	s	1.4586	1.5323	1.5875	1.6343	1.6762	1.7147	1.7846	1.8476	1.9056
	v		0.7934	0.9077	1.0108	1.1082	1.2024	1.3853	1.5641	1.7405
700	h		1280.6	1345.0	1403.2	1459.0	1513.9	1623.5	1734.8	1848.8
(503.10)	s		1.5084	1.5665	1.6147	1.6573	1.6963	1.7666	1.8299	1.8881
	v		0.6779	0.7833	0.8763	0.9633	1.0470	1.2088	1.3662	1.5214
800	h		1270.7	1338.6	1398.6	1455.4	1511.0	1621.4	1733.2	1847.5
(518.23)	s		1.4863	1.5476	1.5972	1.6407	1.6801	1.7510	1.8146	1.8729
	v		0.5873	0.6863	0.7716	0.8506	0.9262	1.0714	1.2124	1.3509
900	h		1260.1	1332.1	1393.9	1451.8	1508.1	1619.3	1731.6	1846.3
(531.98)	s		1.4653	1.5303	1.5814	1.6257	1.6656	1.7371	1.8009	1.8595
	v		0.5140	0.6084	0.6878	0.7604	0.8294	0.9615	1.0893	1.2146
1000	h		1248.8	1325.3	1389.2	1448.2	1505.1	1617.3	1730.0	1845.0
(544.61)	s		1.4450	1.5141	1.5670	1.6121	1.6525	1.7245	1.7886	1.8474
	v		0.4532	0.5445	0.6191	0.6866	0.7503	0.8716	0.9885	1.1031
1100	h		1236.7	1318.3	1384.3	1444.5	1502.2	1615.2	1728.4	1843.8
(556.31)	s		1.4251	1.4989	1.5535	1.5995	1.6405	1.7130	1.7775	1.8363
	v		0.4016	0.4909	0.5617	0.6250	0.6843	0.7967	0.9046	1.0101
1200	h		1223.5	1311.0	1379.3	1440.7	1499.2	1613.1	1726.9	1842.5
(567.22)	s		1.4052	1.4843	1.5409	1.5879	1.6293	1.7025	1.7672	1.8263
	v		0.3174	0.4062	0.4714	0.5281	0.5805	0.6789	0.7727	0.8640
1400	h		1193.0	1295.5	1369.1	1433.1	1493.2	1608.9	1723.7	1840.0
(587.10)	s		1.3639	1.4567	1.5177	1.5666	1.6093	1.6836	1.7489	1.8083

(Tables 2-1, 2-2, and 2-3 are from *Steam, Its Generation and Use,* © Babcock and Wilcox.)

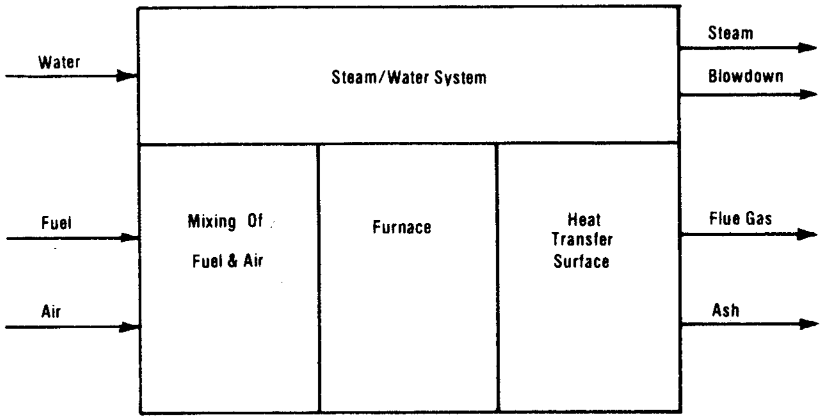
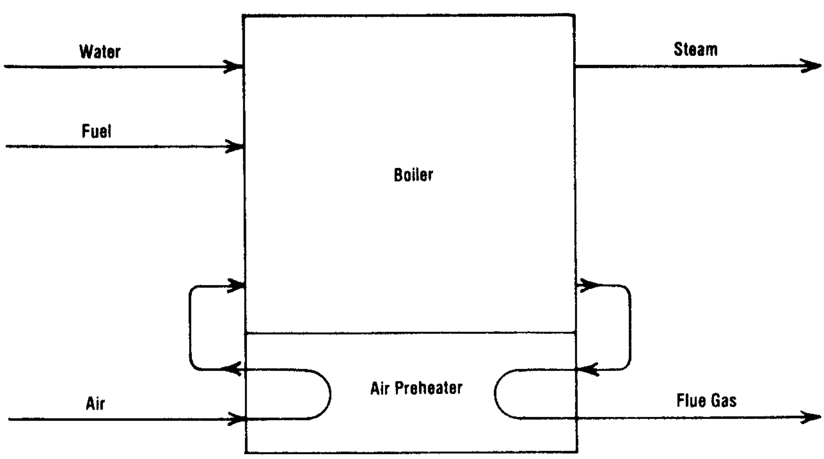


Figure 2-1 Basic Diagram of a Boiler

combustion air preheater. The combustion air also passes through the air preheater before being mixed with the fuel. Since the flue gas temperature is higher than the air temperature, heat is transferred from the flue gas to the combustion air via the convection heat transfer surface of the combustion air preheater.

This transfer of heat cools the flue gas and thus reduces its heat loss. The added heat in the combustion air enters the furnace, enhances the combustion process, and reduces the fuel requirement in an amount equal in heat value to the amount of heat that has been transferred in the combustion air preheater. By the use of an air preheater, approximately 1 percent of fuel is saved for each 40°F rise in the combustion air temperature.

The use of an economizer is another flue gas heat recovery method. The arrangement of this type of additional heat exchanger is shown in Figure 2-3. In the economizer arrangement shown, the flue gas leaves the simple boiler and enters the economizer, where it is in contact with heat transfer surface, in the form of water tubes, through which the boiler feedwater flows. Since the flue gas is at a higher temperature than the water, the flue gas is cooled and the water temperature is increased. Cooling the flue gas reduces its heat loss in an amount equal to the increased heat in the feedwater to the boiler. The increased heat in the feedwater



Air Preheater Purpose—Preheat combustion air and absorb additional heat from flue gases

Figure 2-2 A Simple Boiler plus Combustion Air Preheater

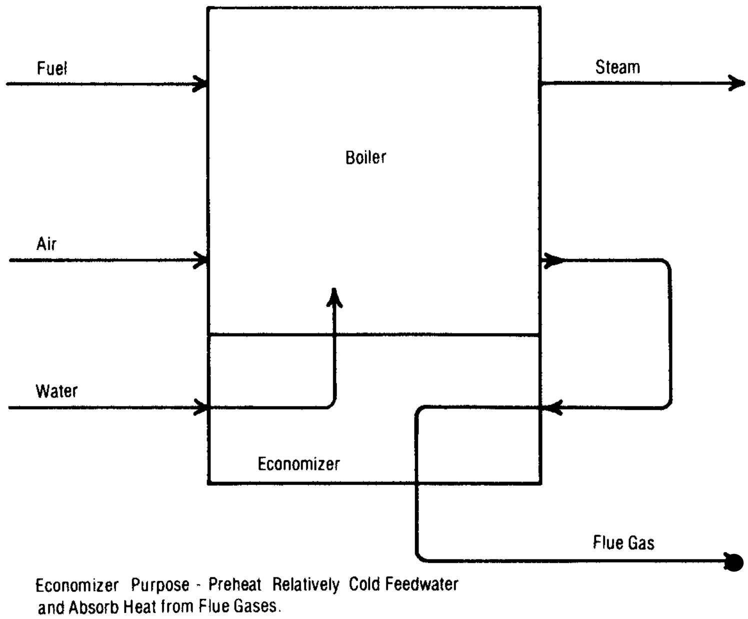


Figure 2-3 A Simple Boiler plus Economizer

reduces the boiler's requirement for fuel and combustion air. Approximately 1 percent of fuel input is saved for each 10°F rise in the feedwater as it passes through the economizer.

Both types of heat exchangers are often used in large boilers. When both an air preheater and an economizer are used, the normal practice consists of passing the flue gases first through the economizer and then through the combustion air preheater.

2-4 Boiler Types and Classifications

There are two general types of boilers: firetube and watertube. In addition, boilers are classified as "high" or "low" pressure and as "steam" boilers or "hot water" boilers.

By definition high pressure boilers are steam boilers that operate at a pressure greater than 15 psig. Because the boiler water temperature rises as the pressure is increased, the flue gas temperature is increased as the pressure increases, increasing the boiler heat losses.

An advantage of using higher pressure is a reduction in physical size of the boiler and steam piping for the same heat-carrying capacity. This is due to the increased density (lower specific volume) of the higher pressure steam. The advantage is particularly important if the boiler is some distance from the heat load. When high pressure boilers are used for space heating, the pressure is usually reduced near the point of steam use.

A particular attribute of high pressure steam is that it contains a significantly greater amount of available energy. Available energy is a term given to the energy that is available to be converted to work in an industrial or electric power generation steam engine or turbine.

A low pressure boiler is one that is operated at a pressure lower than 15 psig. Almost all low pressure boilers are used for space heating. Low pressure boiler systems are simpler since pressure-reducing valves are seldom required and the water chemistry of the boiler is simpler to maintain.

Another boiler classification is the hot water boiler. Strictly speaking, this is not a boiler since the water does not boil. It is essentially a fuel-fired hot water heater in which sensible heat is added to increase the temperature to some level below the boiling point. Because of similarities in many ways to steam boilers, the term "hot water boiler" is generally used to describe this type of unit.

A high temperature hot water (HTHW) boiler furnishes water at a temperature greater than 250°F (121°C) or at a pressure higher than 160 psig. A low temperature hot water boiler furnishes water at a pressure not exceeding 160 psig and at a temperature not exceeding 250°F.

2-5 Firetube Boilers

Firetube boilers constitute the largest share of small- to medium-sized industrial units. In firetube boilers the flue gas products of combustion flow through boiler tubes surrounded by water. Steam is generated by the heat transferred through the walls of the tubes to the surrounding water. The flue gases are cooled as they flow through the tubes, transferring their heat to the water; therefore, the cooler the flue gas, the greater the amount of heat transferred. Cooling of the flue gas is a function of the heat conductivity of the tube and its surfaces, the temperature difference between the flue gases and the water in the boiler, the heat transfer area, the time of contact between the flue gases and the boiler tube surface, and other factors.

Firetube boilers used today evolved from the earliest designs of a spherical or cylindrical pressure vessel mounted over the fire with flame and hot gases around the boiler shell. This obsolete approach has been improved by installing longitudinal tubes in the pressure vessel and passing flue gases through the tubes. This increases the heat transfer area and improves the heat transfer coefficient. The results are the two variations of the horizontal return tubular (HRT) boiler shown in Figures 2-4 and 2-5. A variation of the HRT boiler in Figure 2-4 is the packaged (shop-assembled) firebox boiler shown in Figure 2-6.

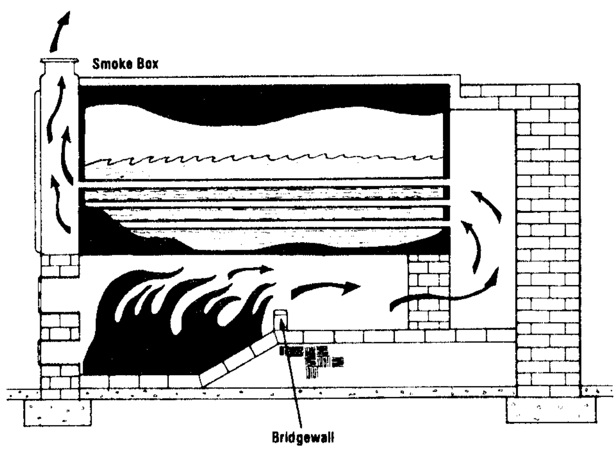


Figure 2-4 Horizontal-Return-Tubular Boiler

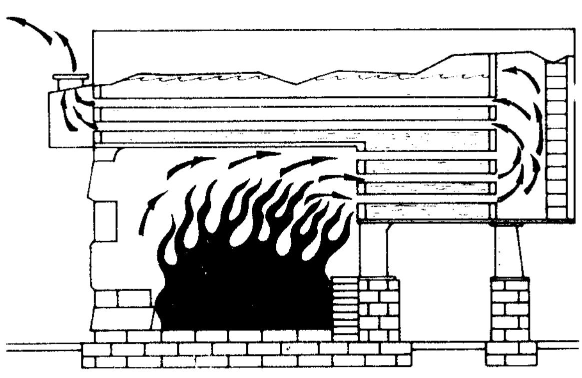


Figure 2-5 Two-Pass Boiler

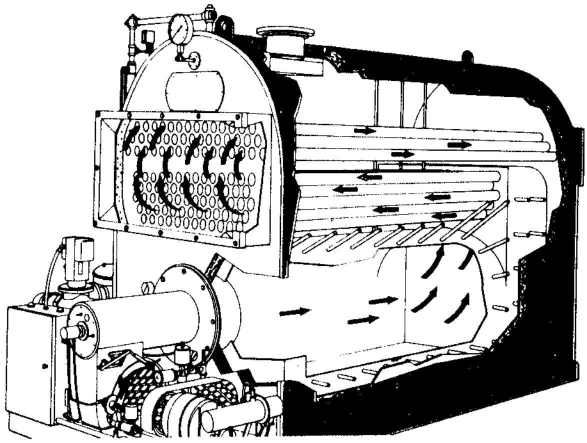


Figure 2-6 Firebox Boiler

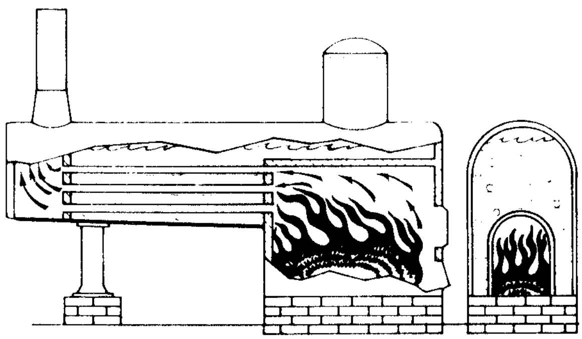


Figure 2-7 Locomotive-type Boiler

A parallel evolution of the firetube boiler was the locomotive boiler designed with the furnace surrounded by a heat transfer area and a heat transfer area added by using horizontal tubes. This type is shown in Figure 2-7.

The Scotch Marine boiler design, as shown in Figure 2-8 with the furnace a large metal tube, combined that feature of the English Cornish boiler of the 1800s and the smaller horizontal tubes of the HRT boiler. This boiler originally was developed to fit the need for compact shipboard boilers. Because the furnace is cooled completely by water, no refractory furnace is required. The radiant heat from the combustion is transferred directly through the metal wall of the furnace chamber to the water. This allows the furnace walls to become a heat

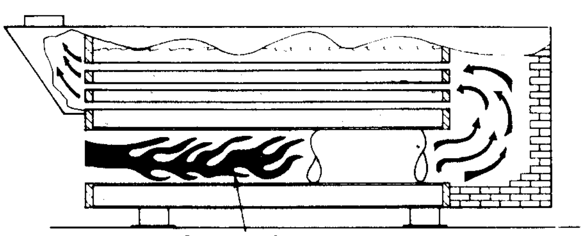


Figure 2-8 Scotch Marine Boiler

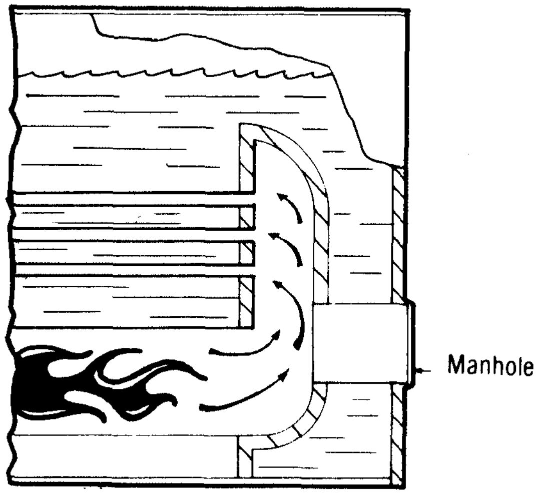


Figure 2-9 Wetback

transfer surface—a surface particularly effective because of the high temperature differential between the flame and the boiler water.

A modified Scotch boiler design, as used in the standard firetube package boiler, is the most common firetube boiler used today. Two variations of the Scotch design, called wetback and dryback, are shown in Figures 2-9 and 2-10. These names refer to the rear of the combustion chamber, which must be either water-jacketed or lined with a high temperature insulating material, such as refractory, to protect it from the heat of combustion.

The wetback boiler gains some additional heating surface; however, it is more difficult to service because access to the back end of the boiler tubes is limited. The only such access normally provided is a 16-inch manhole in the rear water header or through the furnace tube.

The dryback boiler is easy to service because the rear doors may be removed for complete access to the tubes and to the insulating or refractory material. The refractory or insulating lining may deteriorate over a period of time. If this lining is not properly maintained, efficiency may be reduced because the flue gases will bypass heating surface on three- and four-pass designs, the radiation loss through the rear doors will increase, and the metal doors will be damaged.

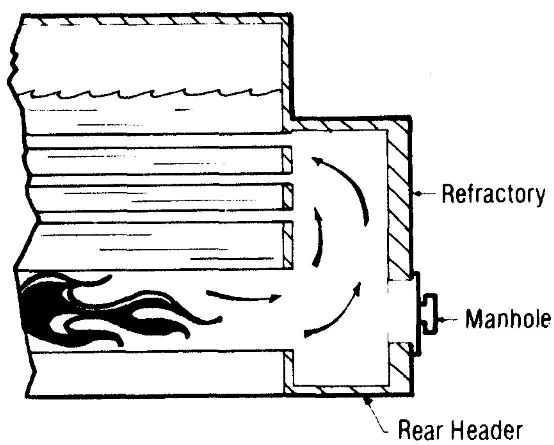


Figure 2-10 Dryback

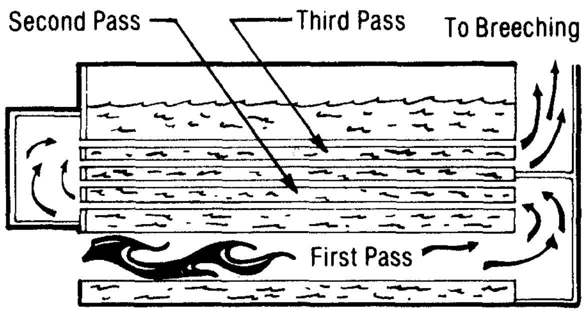


Figure 2-11 Boiler Passes

The number of boiler passes for a firetube boiler refers to the number of horizontal runs the flue gases take between the furnace and the flue gas outlet. The combustion chamber or furnace is considered the first pass; each separate set of firetubes provides additional passes as shown in Figure 2-11.

The number of gas passes in a firetube boiler does not necessarily determine its efficiency characteristic. For the same total number, length, and size of tubes (same tube heating surface), increasing the number of passes increases the length the flue gas must travel because the gases must pass through tubes in series rather than in parallel. This increases the flue gas velocity within the tubes but does little to change the total time for the hot gases to flow from furnace to outlet in contact with the tube heating surface.

The increased gas velocity in some cases may improve heat transfer by increasing the turbulence of the gases as they travel through the tubes. Generally, however, increasing the number of passes and the resultant velocity of the gases increases the resistance to flow and forces the combustion air blower to consume more power.

One additional firetube boiler, generally used only where space is limited and steam requirements are small, is the vertical firetube boiler shown in Figure 2-12. This is a variation

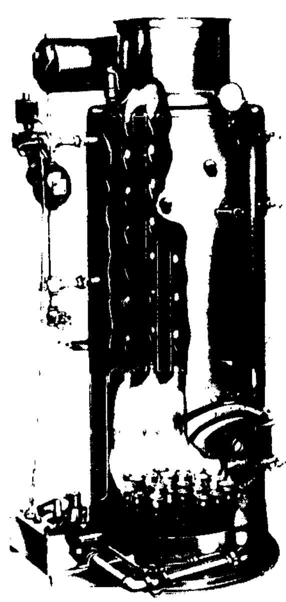


Figure 2-12 Vertical Firetube Boiler

Table 2-4
Firetube Boiler Characteristics
(Approximate)

Boiler Type	Max Pressure	BoHP* Range	Lbs/Hr
HRT	150 psig	30–300	1000–10000
Firebox	200 psig	10–600	350–25000
Pkg. "Scotch"	300 psig	10–1000	350-35000
Vert. Firetube	200 psig	2–300	70–10000

*The term BoHP is discussed in Section 3

of the firebox boiler with the water-jacketed furnace and vertical tubes. Its configuration is similar that of a typical home hot water heater of today.

Characteristics of the various types of firetube boilers relative to operational limitations are approximate in Table 2-4.

2-6 Watertube Drum Boilers

As the name implies, water circulates within the tubes of a watertube boiler. These tubes are often connected between two or more cylindrical drums. In some boilers the lower drum is replaced with a tube header. The higher drum, called the steam drum, is maintained approximately half full of water. The lower drum is filled with water completely and is the low point of the boiler. Sludge that may develop in the boiler gravitates to the low point and can be drawn off the bottom of this lower drum, commonly called the mud drum.

A cross-sectional view of a small field-erected watertube boiler is shown in Figure 2-13. Heating the "riser" tubes with hot flue gas causes the water to circulate and steam to be released in the steam drum. This principle is shown in Figure 2-14. This particular type of boiler has not been built since the 1950s but many are still in service.

Because watertube boilers can be easily designed for greater or lesser furnace volume using the same boiler convection heating surface, watertube boilers are particularly applicable to

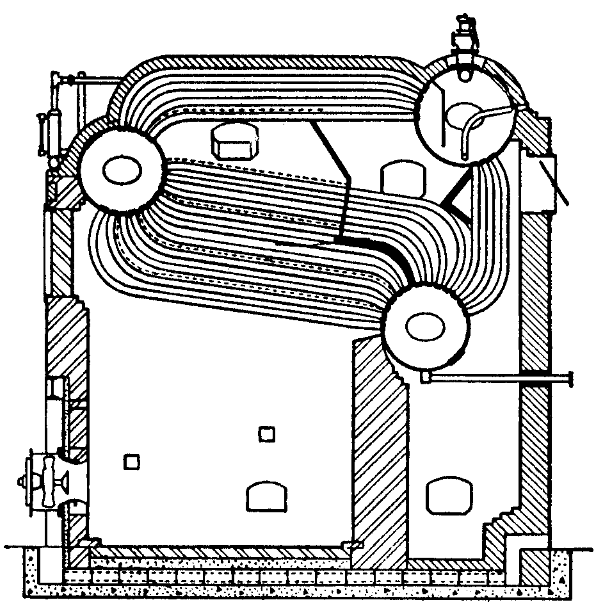


Figure 2-13 Small Field-Erected Watertube Boiler

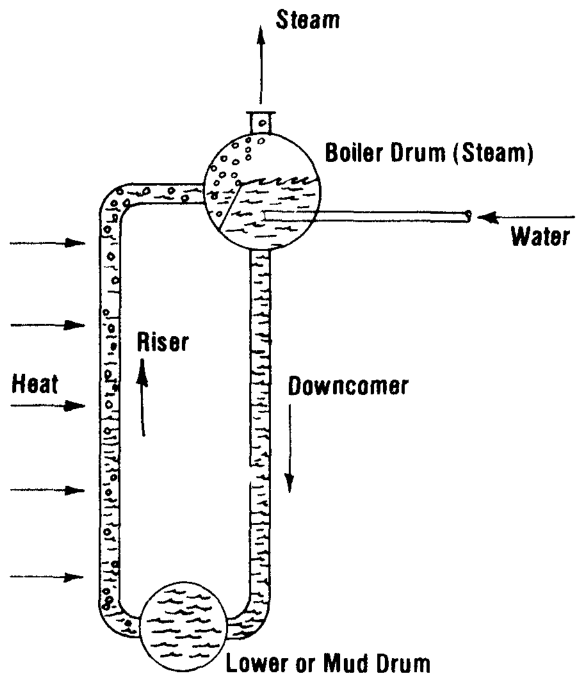


Figure 2-14 Circulation of Watertube Boiler

solid fuel firing. They are also applicable for a full range of sizes and for pressures from 50 psig to 5000 psig. The present readily available minimum size of industrial watertube boilers is approximately 20,000 to 25,000 lbs/hr of steam—equivalent to 600 to 750 BoHP (boiler horsepower). Many watertube boilers operating today are in the 250 to 300 BoHP size range.

A typical industrial watertube boiler for gas and oil firing is the packaged (shop-assembled) boiler shown in Figure 2-15. Such packaged watertube boilers generally have a single burner with up to approximately 125,000 lbs/hr steam flow (approximately 4000 BoHP) but are available in sizes up to approximately 250,000 lbs/hr with more than one burner.

Older designs of watertube boilers, as shown in Figure 2-13, consisted of refractory-lined furnaces with only convection heating surface. Later developments included placing some boiler tubes in the furnace walls where they were exposed to the radiant heat of the flame, as shown in Figure 2-16. This development continued until furnaces became fully water-walled, as in the package boiler shown in Figure 2-15.

Figure 2-16 shows the baffles for directing flue gas. Watertube boilers generally have such gas baffles to assure contact between the hot gases and the maximum amount of the tube heating surface. The baffle design determines the number of gas passes and which tubes act as "risers" and "downcomers," as shown in Figure 2-14. Leakage in the baffles causes hot flue gas to bypass a portion of the heating surface, thereby decreasing the heat being transferred and lowering the boiler efficiency.

This evolution in the design of boilers came about because of the economic benefits resulting from the lower cost of the required radiant heat transfer surface as compared with convection heating surface. The net result was a reduction in the physical size and cost of the boilers and changes in water volume, heat storage, and an improvement in response characteristics. Other effects of the fully water-cooled furnaces were reduced furnace temperatures and the resulting reduction in nitrous oxides (NOx) production. The cooler furnaces also affect the chemistry of the progression of the combustion process from ignition to complete combustion.

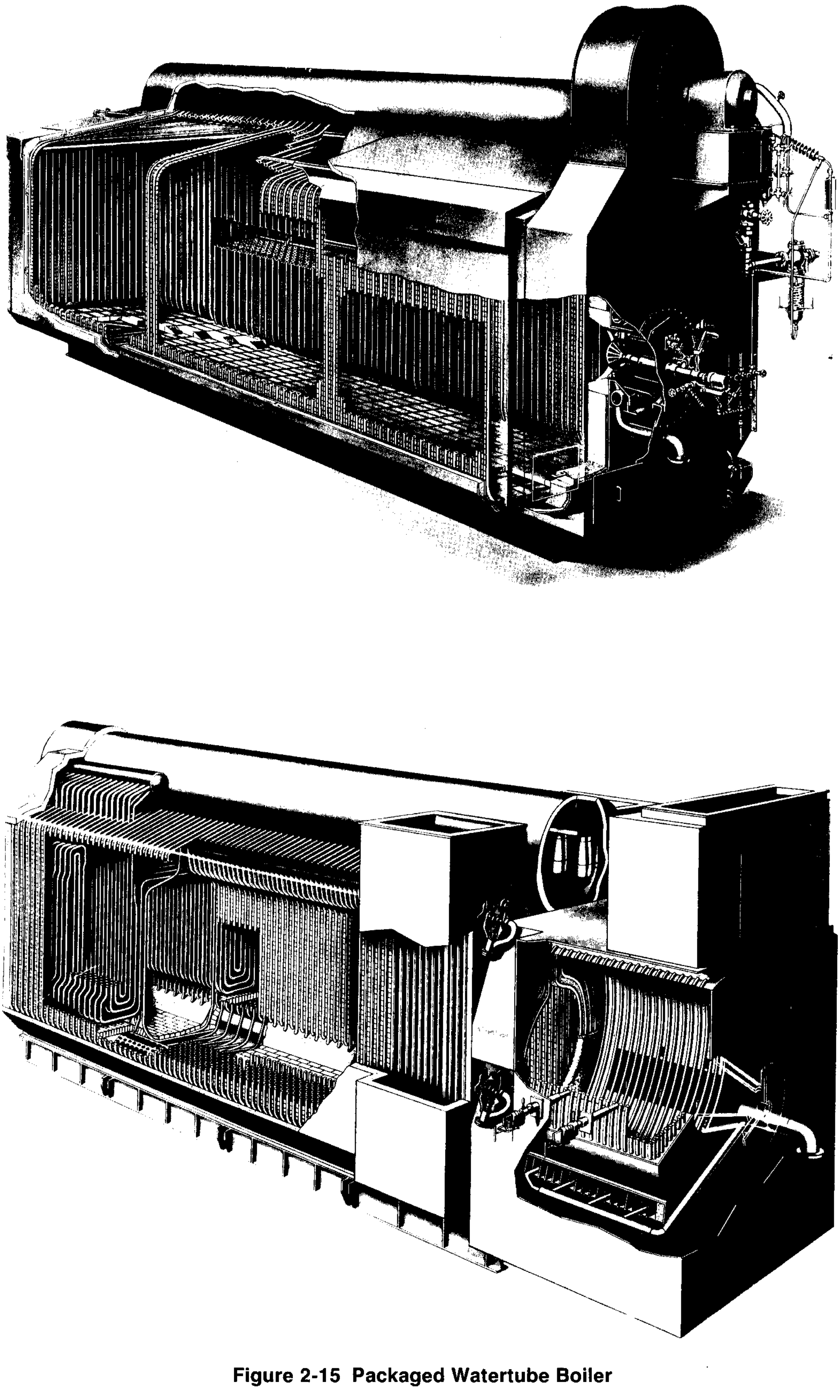


Figure 2-15 Packaged Watertube Boiler

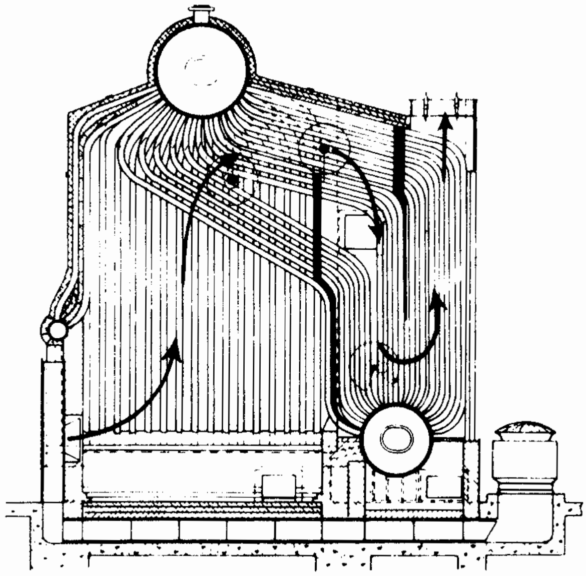


Figure 2-16 Flue Gas Flow and Watertube Boiler

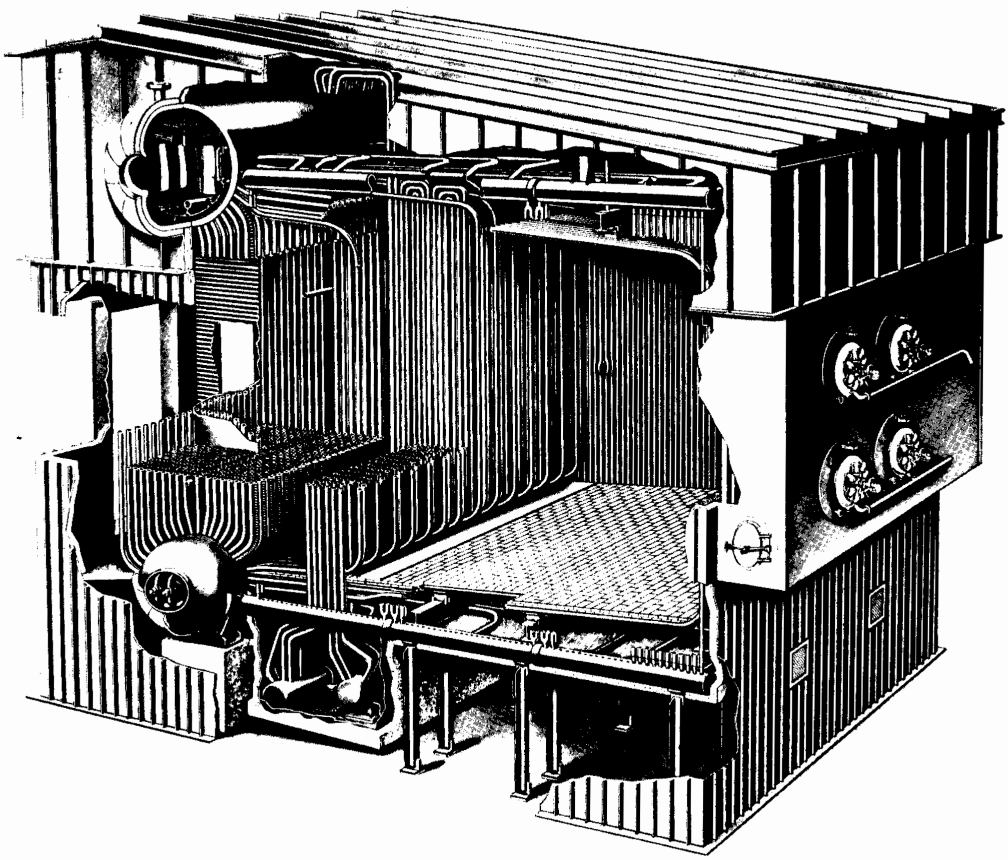


Figure 2-17 Gas- or Oil-Fired Industrial Boiler

(From *Steam, Its Generation and Use* © Babcock and Wilcox)

As industrial boilers are increased in size for liquid and gaseous fuels, the balance between radiant and convection heat transfer surface remains approximately the same as for the package boiler for oil and gas firing shown in Figure 2-15. One such larger gas- or oil-fired industrial boiler is shown in in Figure 2-17.

For solid fuel, however, coal-, wood-, or waste material-fired boilers usually require greater spacing between the boiler tubes. In addition, the furnace volume must be increased. A large industrial boiler for solid fuel is shown in Figure 2-18. These differences make it difficult to convert a gas- or oil-fired boiler to coal and obtain full steam capacity, while a conversion from coal to gas or oil firing can be much more easily accomplished.

As stated earlier, boilers used in electric utility generating plants are generally considerably larger than their industrial counterparts. The steam generating capacity of the largest electric utility boiler is approximately 10 times the capacity of the largest industrial steam boiler. The maximum size of electric utility watertube boilers, at this writing, handles approximately 10,000,000 lbs/hr of steam. In industrial use the largest are in the approximate range of 1,000,000 to 1,500,000 lbs/hr.

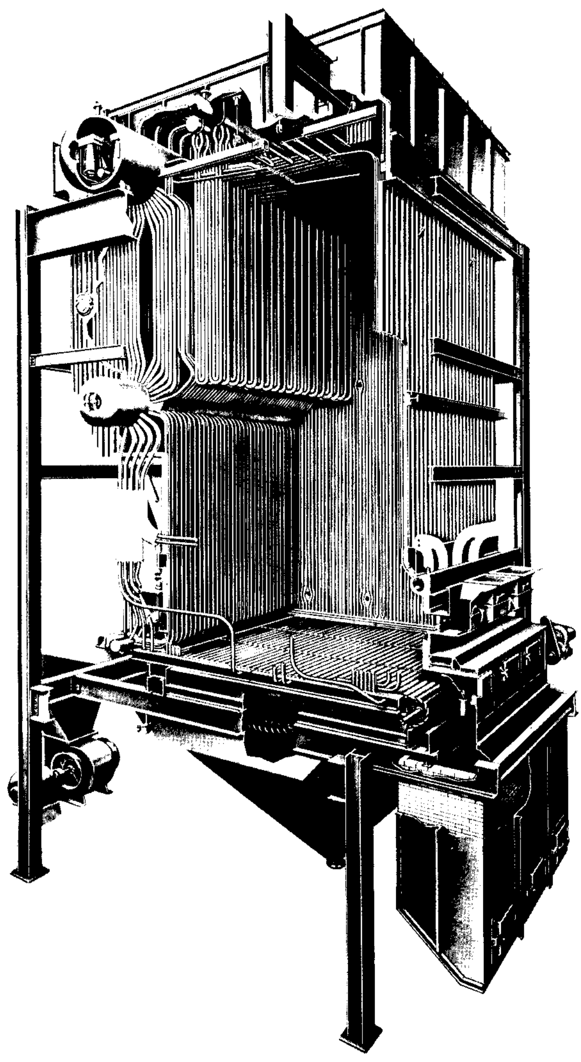


Figure 2-18 Large Industrial Boiler for Solid Fuel

(From Detroit Stoker Company. Used with permission.)

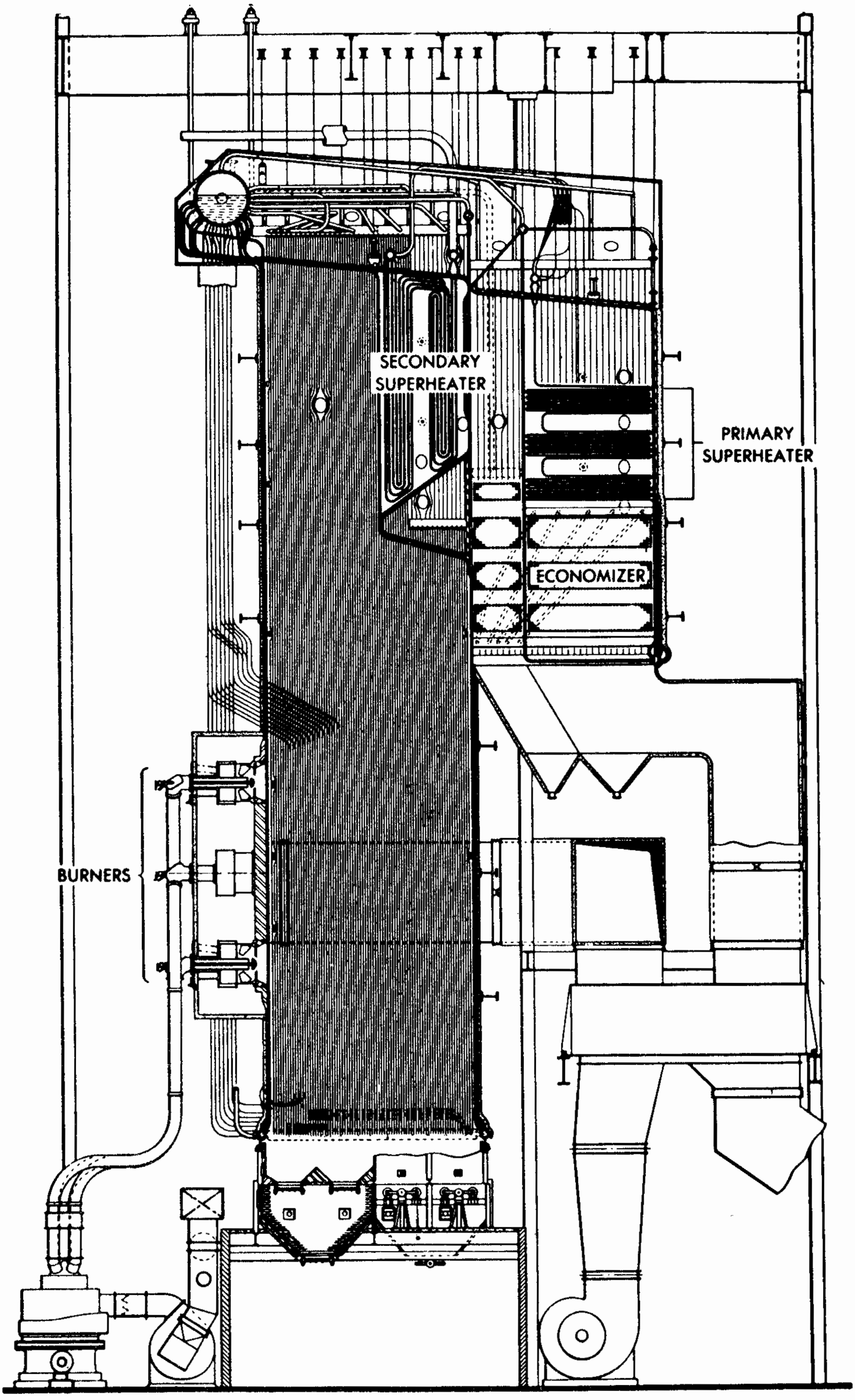


Figure 2-19 Pulverized Coal-Fired Utility Boiler

(From Steam, Its Generation and Use, © Babcock and Wilcox)

Modern utility boilers operate at pressures in the range of 2,000 to 4,000 psig, while their industrial counterparts are generally in the range of 100 to 1000 psig. In generating electric power with a turbogenerator, it is much more efficient to use steam that has been superheated and reheated as is done in the typical electric utility plant. The general practice with industrial boilers is to use saturated steam or only small amounts of superheat unless electric power is being generated in the industrial plant.

A modern electric utility boiler, as shown in Figure 2-19, may have only the steam drum. Because the water used is very pure, chemical sludge does not normally develop and that need for the mud drum is eliminated. The lower end of the circulation loop consists of water headers. Boilers such as shown can be designed to operate up to approximately 2,750 psig. On all drum boilers, the boiler drum acts to decouple the feedwater flow rate from the boiler steam flow rate.

2-7 Watertube Once-Through Boilers

At pressures in the neighborhood of 2750 psig, the circulation driving force of a drum boiler diminishes rapidly because the specific volumes of water and steam are nearly the same. Above these pressures a drumless once-through boiler is used with the feedwater flow past the

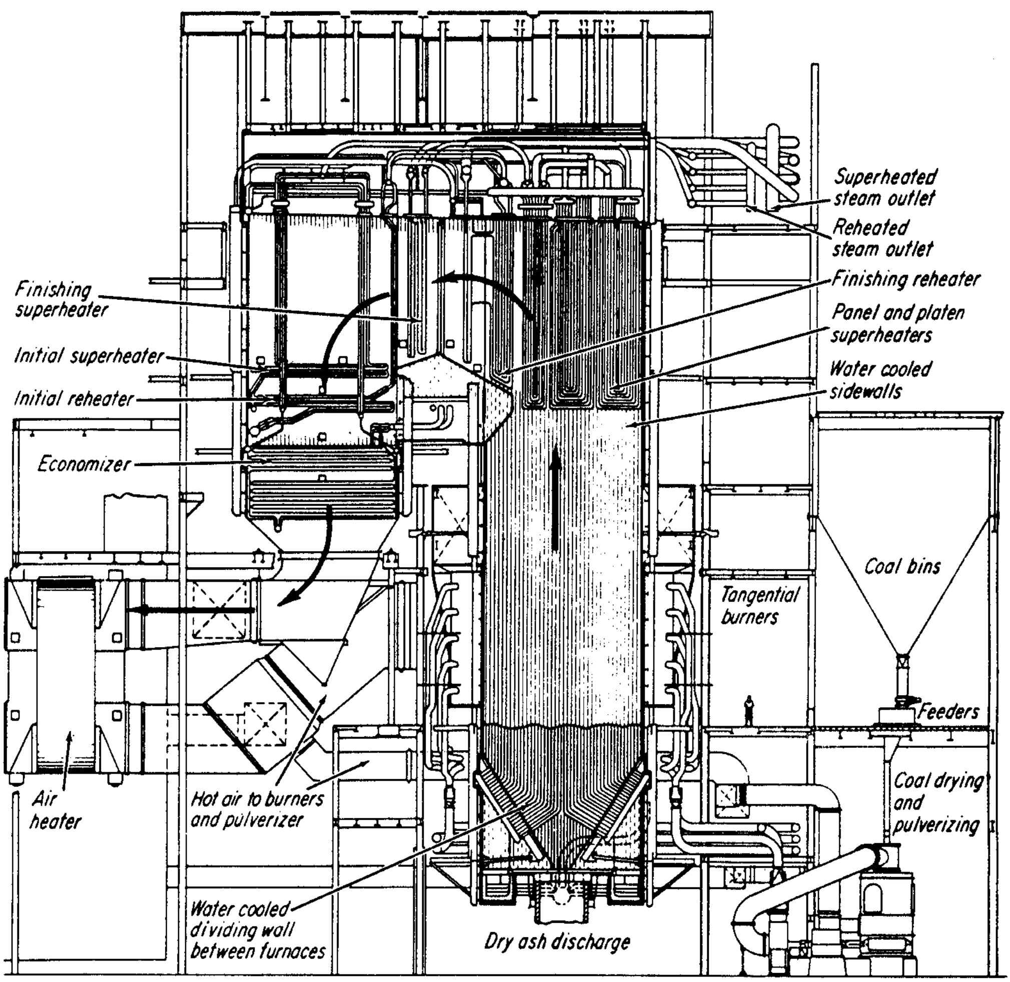


Figure 2-20 Large 3500-psig Combined Circulation Boiler

(Courtesy Combustion Engineering Corporation) heating surfaces provided by the pumping system. In this design, water is pumped into one end of the boiler tubes and superheated steam emerges from the other. Such boilers have been designed to operate up to 5000 psig.

In starting up a cold system, the fluid flow cannot be directed to the turbine until it becomes steam. Consequently, the fluid is initially diverted to a flash tank. As steam starts to develop in the flash tank, it is used to heat the feedwater. The level is maintained in the flash tank by flow of the flash tank bottom liquid to the condenser. At minimum load conditions, the turbine can be started up and operated on flash tank steam flow alone. As pressure is raised above flash tank design pressure, a logic system and series of valves change the steam and water flow circuit arrangements so that the flow of feedwater flows all the way through to the turbine.

The feedwater chemistry is very important, with extremely pure water required, since all feedwater chemicals pass to the turbine. If the water is not pure enough when the unit is started, operating on the flash tank and blowing down the flash tank will gradually increase the water purity.

This type of boiler is used in electric utility installations and would very rarely, if ever, be used for industrial steam. The drumless boiler in Figure 2-20 is typical of such boilers.

Section 3
Performance and Input/Output Relationships

A boiler's performance relates to its ability to transfer heat from the fuel to the water while meeting operating specifications. Boiler performance includes all aspects of the operation. The basic elements are the operating capacity and the boiler efficiency.

Performance specifications include the operating capacity and the factors for adjusting that capacity, steam pressure, boiler water quality, boiler temperatures, boiler pressures, boiler drafts and draft losses, flue gas analysis, fuel analysis, and fuel burned. Additional performance specifications indicate the fan power requirements (boiler flue gas temperatures and draft losses) and the fuel supply assumptions.

The result of a calculation involving the performance specification is a calculated efficiency. Boiler efficiency is presented as a percentage ratio of heat supplied to the boiler and the heat absorbed in the boiler water.

3-1 Capacity and Performance

Packaged firetube boilers generally are described in terms of BoHP (boiler horsepower).* The BoHP rating of a modern firetube boiler is approximately one fifth of the square footage of its heating surface. For example, a boiler of 500 BoHP has approximately 2,500 square feet of heating surface. Although these boilers are described in terms of BoHP, the developed Btu output can be converted easily to lbs/hr of steam. Because the heat content of a pound of steam increases as pressure is increased in firetube boilers, the pounds of steam per BoHP decreases with pressure. Table 3-1 shows this relationship.

Industrial watertube boilers formerly were classified in BoHP by dividing the heating surface by 10. In the past 40 or more years, however, BoHP ratings for new watertube boilers have disappeared, and boiler capacity ratings are specified in terms of pounds of steam per hour with feedwater temperature specified. Existing watertube boilers rated in BoHP can be rated in lbs/hr by using a conversion factor from Table 3-1. Smaller watertube and firetube boilers often are rated in terms of maximum Btu input to the burner with efficiency specified.

3-2 Input Related to Output

Boiler energy inputs generally are thought of as the heat content of the fuel used. The flow of this fuel measured over a period of time multiplied by the heat content of this fuel develops a total Btu input during the time period. Measuring the energy output of a steam boiler involves measuring the steam flow in lbs/hr over a period of time and multiplying by the Btu content of a pound of steam to provide the Btu output. Useful simple relationships of input and output—such as pounds of steam/gallon of fuel oil, pounds of steam/pound of coal, or pounds of steam/standard cubic foot of gas—can be used to track relative efficiency. These relationships, however, are not precise because such factors as fuel Btu content, steam Btu content, feedwater temperature, and blowdown are not considered.

The chief energy loss of most boilers depends on the mass of the flue gases and their temperature as they leave the boiler. To obtain the net energy loss of the flue gas, however, the temperatures of the incoming combustion air and fuel must be considered.

When hydrogen in the fuel burns, it forms water, which leaves the boiler in the form of superheated vapor. The latent heat of this vapor is an energy loss, which is approximately

*The term "boiler horsepower" started because early boilers were used to drive engines with one engine horsepower or one boiler horsepower equivalent to 34.5 pounds of water evaporated from and at 212 degrees. This equals 33,475 Btu, thermal equivalent of one boiler horsepower.

Table 3-1
Conversion Factors—Lbs Steam per BoHP

Boiler pressure, psig	Lbs steam/BoHP	Btu content from 212°F (liquid)
50	33.8	999
75	33.6	1005.2
100	33.4	1009.6
125	33.31	1013
150	33.22	1015.6
175	33.16	1017.6
200	33.1	1019.3
225	33.06	1020.6
250	33.03	1021.7

nine to ten percent for natural gas, five to six percent for fuel oil, and three to four percent for coal. The percentage of hydrogren and moisture in the fuel affects this loss.

Although blowdown is not a useful heat output from the viewpoint of boiler efficiency, it is not considered a loss because the boiler has properly transferred the heat from the fuel to the water.

The useful energy output of boilers is the heat carried by the steam or hot water. In a steam boiler this is usually measured as steam flow at the boiler and adjusted for Btu content by measurements of pressure, temperature, or both. The steam flow can also be obtained by measuring water flow and subtracting the blowdown. For hot water boilers, water flow is measured at the boiler outlet and adjusted for Btu content by measurement of the outlet temperature.

Although these procedures provide information about the useful energy outputs, in themselves they do not determine precisely the contribution of the boiler to this useful energy. To determine the contribution of the boiler, the heat in the incoming feedwater must be subtracted from the heat carried in the boiler output.

3-3 Mass and Energy Balances Involved

The mass balances in a steam boiler are shown in the diagrams of Figures 3-1 through 3-5. For a hot water boiler these diagrams would be slightly different. In Figure 3-1 there is a simple balance on the water side of the boiler between the mass of the feedwater and the mass of the steam plus blowdown.

In this balance, steam is normally 90 to 99 percent of the output. The water and steam plus blowdown is not one to one since the water flow and the steam flow are decoupled by the boiler drum. This decoupling allows a change in boiler water storage at any point in time without a change in steam flow.

Figure 3-2 represents the balance between the mass of combustion air plus fuel and the flue gas and ash output. Ash, of course, would not be present if there were no ash in the fuel. The combustion air is by far the larger input because it may have a mass of more than twelve to eighteen times that of the fuel. There is some amount of decoupling between the fuel-plus-air flow vs. the effluent of total flue product due to the buildup of ash deposits on the boiler heating surfaces. Such buildup must then be periodically removed by soot blowing.

Chemical input and output on the boiler water side also must be considered as one of the

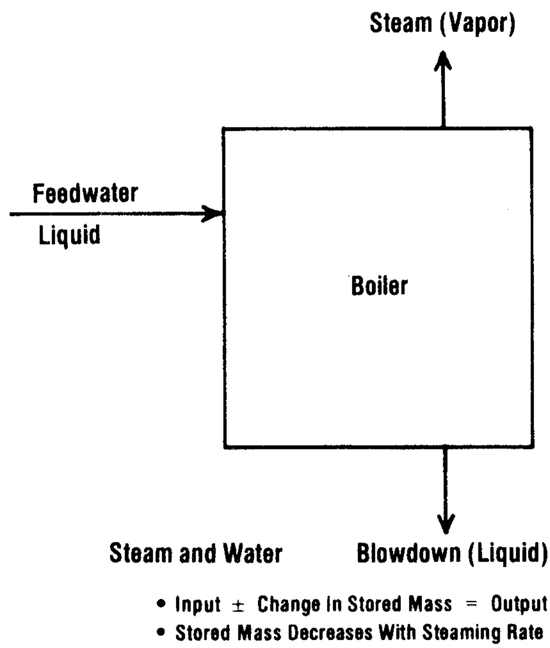


Figure 3-1 Steam-Water Mass Balance

mass balances involved. This is shown in Figure 3-3. In this case there is a mass balance of each individual chemical element present. Steam is expected to be so pure that almost 100 percent of the non-water chemical output is in the boiler blowdown.

The balance of chemical input and output of the combustion process is shown in Figure 3-4. As with the water side chemical balance, this diagram represents a balance of each chemical element although the chemical compounds of the inputs have been changed to different chemical compounds by the combustion process.

The energy balance of the boiler is shown in Figure 3-5. Energy enters and leaves a boiler in a variety of ways. Energy in the steam is the only output considered useful. Fuel energy is by far the major energy input and, unless precise efficiency values are needed, is normally the only energy input considered.

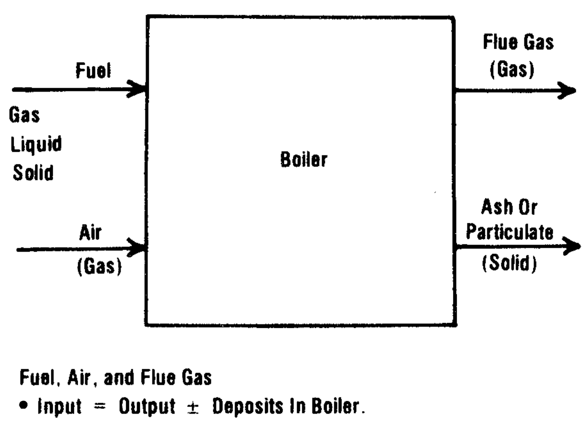


Figure 3-2 Fuel, Air-Flue Gas Mass Balance

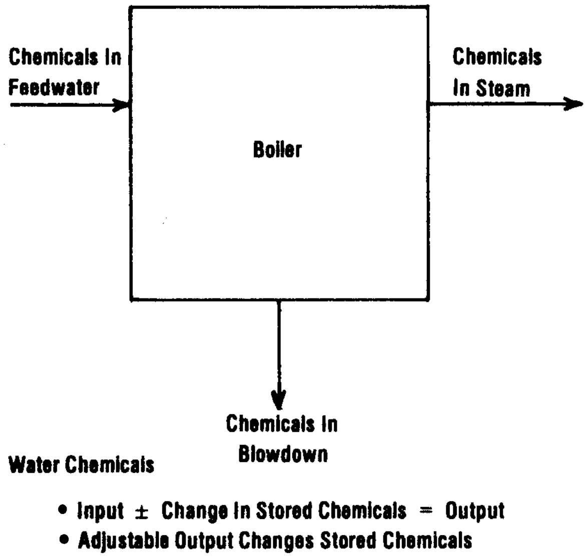


Figure 3-3 Water Side Chemical Mass Balance

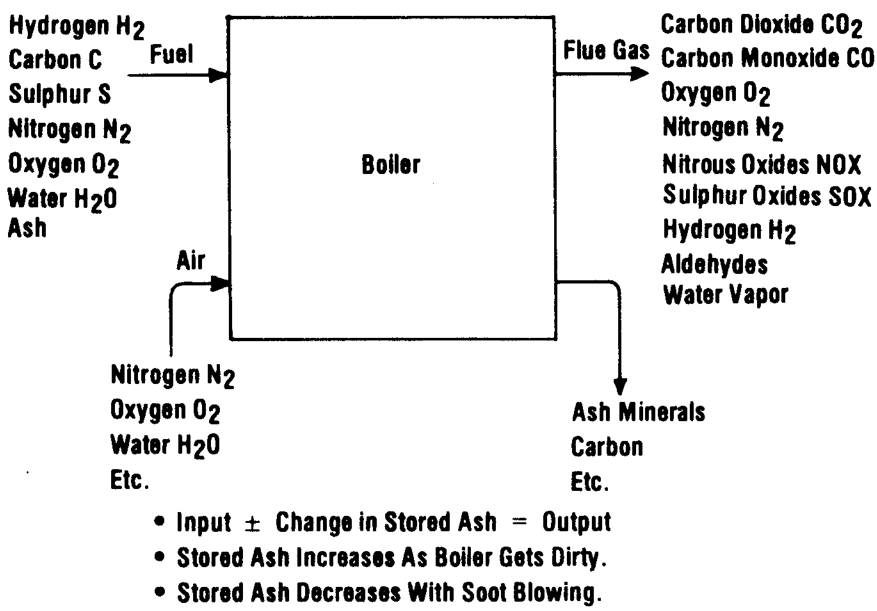


Figure 3-4 Mass Balance—Fuel and Air Chemicals

3-4 Efficiency Calculation Methods

Two methods of calculating the efficiency of a boiler are acceptable. These are generally known as the input/output or direct method and the heat loss or indirect method. Both methods will be covered in more detail in Section 6, Boiler Efficiency Computations.

The input/output method depends on the measurements of fuel, steam, and feedwater flow and the heat content of each.

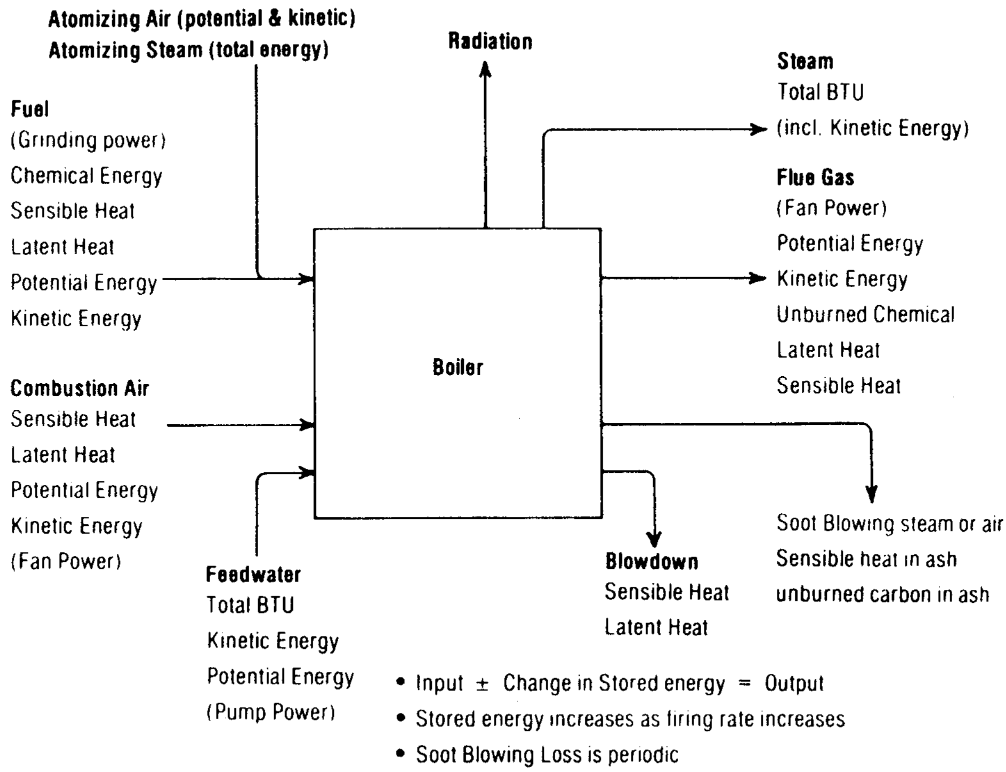


Figure 3-5 Energy Balance—Heat Balance

$$\text{Boiler Efficiency} = \frac{\text{Heat Added to Incoming Feedwater}}{\text{Heat Input (Fuel)} + \text{Heat Input (Combustion Air)}}$$

In this formula, the boiler is credited with the heat added to the blowdown portion of the feedwater. This method yields a decimal number fraction, which is expressed as percent of efficiency.

In the heat loss method, the percentage of each of the major losses is determined. To their total, a small percentage for unaccounted loss is added, and the total obtained is subtracted from 100 percent.

There are eight major losses:

(1) Sensible heat loss in the dry flue gas
(2) Sensible heat loss from water in the combustion air
(3) Sensible heat loss from water in the fuel
(4) Latent heat loss from water in the fuel
(5) Latent heat loss from water formed by hydrogen combustion
(6) Loss from unburned carbon in the refuse
(7) Loss from unburned combustible gas in the flue gas
(8) Heat loss from radiation

3-5 Boiler Control—The Process of Managing the Energy and Mass Balances

The boiler control system is the vehicle through which the boiler energy and mass balances are managed. All the boiler major energy and mass inputs must be regulated in order to achieve

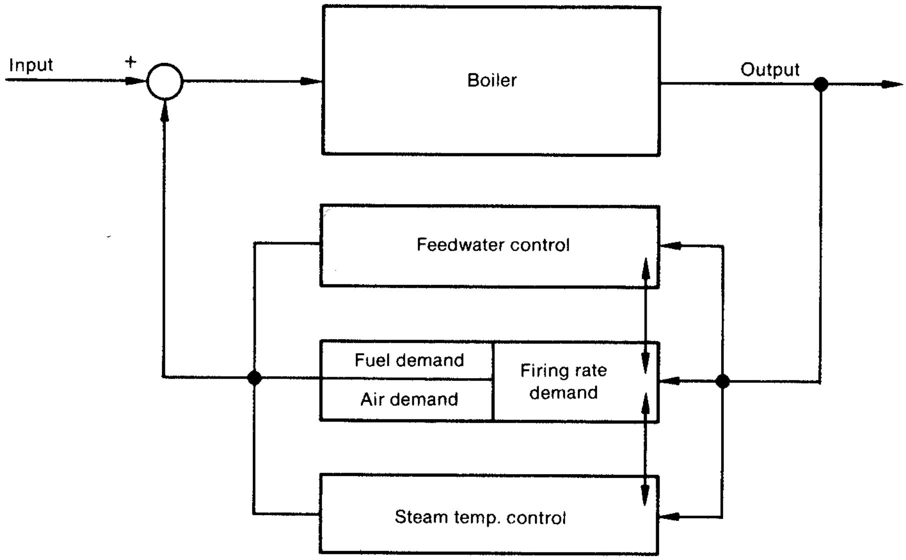


Figure 3-6 Block Diagram of Boiler Control System

the desired output conditions. The measurements of the output process variables furnish the information to the control system intelligence unit. Figure 3-6 is a block diagram showing how the parts of the overall control system are coordinated into the overall boiler control system.

For the energy input requirement, a firing rate demand signal must be developed. This firing rate demand creates the separate demands for the mass of fuel and combustion air. The mass of the water-steam energy carrier must also be regulated, and the feedwater control regulates the mass of water in the boiler. The final steam temperature condition must also be regulated (for boilers generating superheated steam and having such control capability), and this is accomplished by the steam temperature control system. The effects of the input control actions interact, since firing rate also affects steam temperature and feedwater flow affects the steam pressure, which is the final arbiter of firing rate demand. The overall system must therefore be applied and coordinated in a manner to minimize the effect of these interactions. The interactions can be greatly affected by the control system design.

Section 4
Basic Control Loops and Their System Interconnections

Boiler control systems are normally multivariable with the control loops for fuel, combustion air, and feedwater interacting in the overall system. Boiler control systems can be easily understood if one has a good basic knowledge of these loops and their application requirements.

In the descriptions of the loop, the term ''primary variable'' is given to that measurement of a process variable that is to be maintained at a ''set point'' by the control action. The term ''manipulated variable'' is given to that process device that is manipulated in order to achieve the desired condition or set point of the primary variable.

The human body is a perfect example of control loops and interconnected multivariable control systems. The measurements are the sensors: touch, sight, hearing, stability, etc. All of these send signals or measurements via the sensory nerves of the nervous system to the brain or reflex action functions.

The measurements by the body's sensors are processed by the brain (the controller function), which sends motor nerve signals to manipulate the muscles, tendons, and body chemistry functions (the manipulated variables) in order to cause action to take place. All actions of the human body are the result of such manipulative control.

The simple act of standing upright and still results from the body's balance and stability mechanisms sending out control signals. These cause one to shift one's weight on different parts of the feet or from one foot to another in order to remain still and upright — the set points of the control loops.

The two basic types of control, feedforward and feedback, are used as building blocks in forming all types of modulating control action. The body system control function is an example of feedback or ''closed-loop'' control. Anticipatory action before the measured variable changes is called feedforward or ''open-loop.'' In the body system, a brain decision to move the body is feedforward control, which is continuously adjusted to keep the body upright and in stable balance by feedback control.

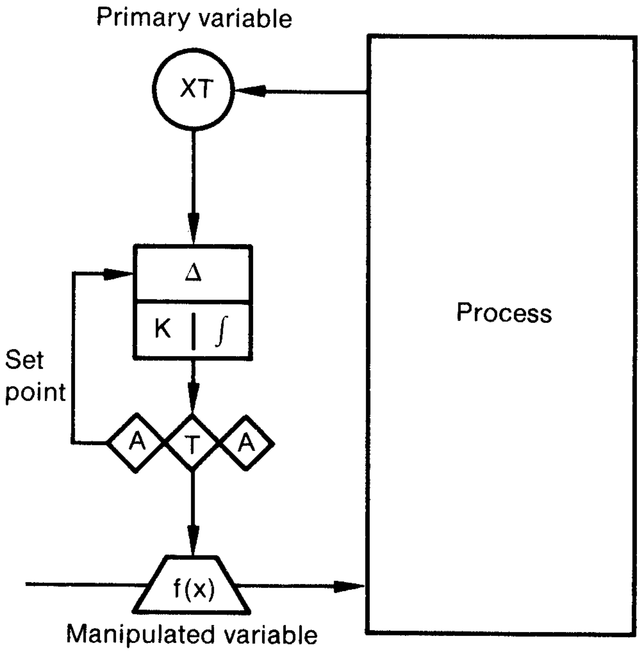
4-1 Simple Feedback Control

Simple feedback control is shown in the control diagram of Figure 4-1A. With this type of loop, changes in the primary variable feed back to a control function, as shown. The control function can be proportional-plus-integral (as shown), proportional-only, proportional-plus-derivative, integral integral-only, or proportional- plus-integral-plus-derivative. In all these cases the controller includes an error detector function, which measures the error between the primary variable and the set point.

The controller output is determined by a combination or summation of the effects of the different control action capabilities that are built into the controller. These are the proportional or gain multiplication of the error magnitude, the difference between the measured amount and the set point, the integral action based on incremental time away from set point multiplied by error magnitude, and the derivative or rate of change of the measured variable.

A change in the controller output changes the manipulated variable, which through action of the process changes the process output selected as the primary variable. This closes the control loop.

A time-based diagram of the control action components for a proportional-plus-integral controller is shown in Figure 4-1B. The first component of the control action is the propor-



(A) Simple Feedback Control

Figure 4-1 Control Functions

tional controller. The error or magnitude of the difference between the process variable and the set point is multiplied by the proportional gain, one of the controller tuning adjustments.

This gain may be expressed directly as a multiplier or in terms such as "proportional band." In the case of proportional band, the value divided into 100 is equal to the gain or error multiplier. As the error increases and decreases, the effect of this control component changes in direct proportion.

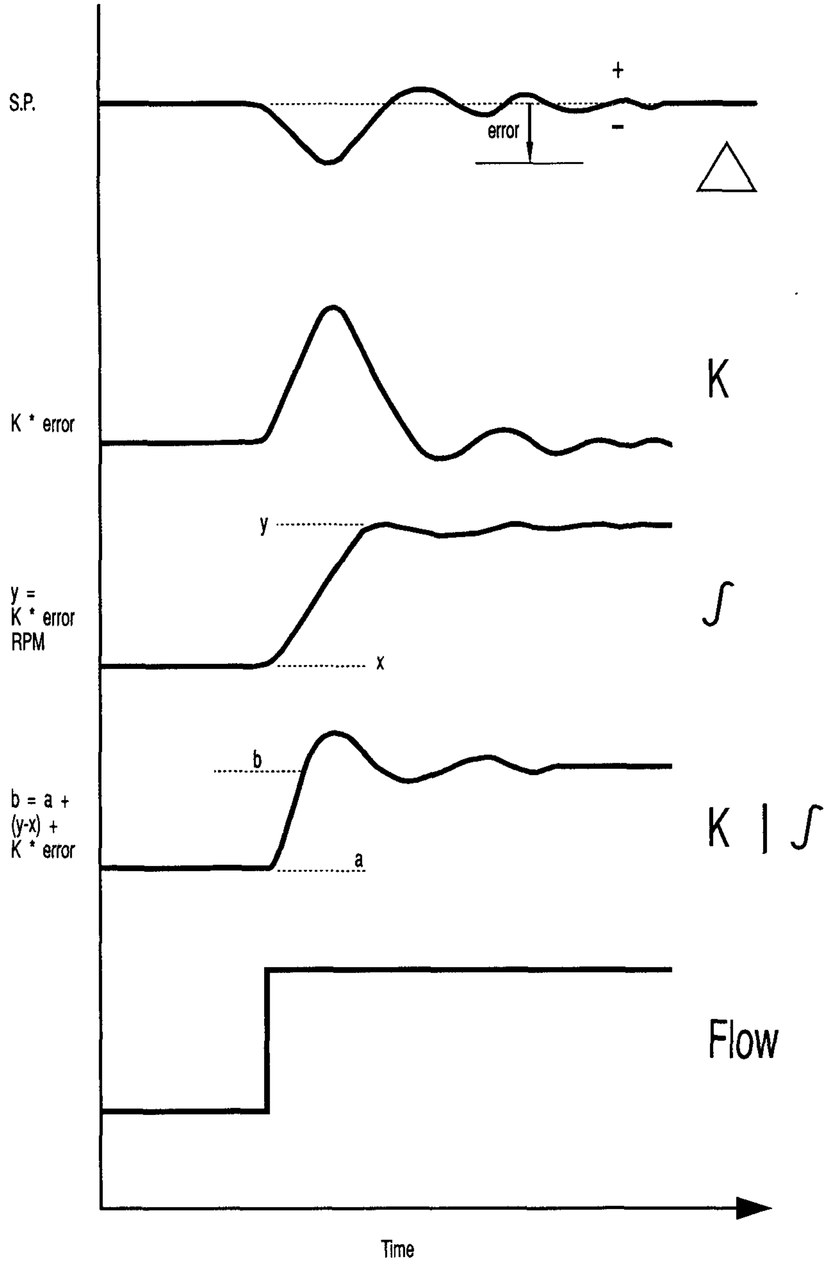
The tuning constants that are set into the controller are affected by the measurement ranges, the dynamics of the control loop, and the capacity of the manipulated variable for effecting change. The measurement range of the measured variable determines the magnitude of the basic error signal with respect to set point. The capacity of the controlled device determines the amount of manipulated variable change for each increment of control signal change.

The proportional gain or multiplier essentially gears the measured and maniulated variables into proper balance so that a measured variable error will produce the desired magnitude of change to the manipulated variable. The desired proportional gain value is the highest value of proportional gain that results in stable conditions of the measured variable, and with change aspects of the manipulated or other variables that also result in most desired conditions.

Nearly perfect steam pressure control of a boiler may be stable but may result in considerable oscillation of the fuel and combustion air, an undesirable operating condition. To reduce these oscillations would require reducing the gain and accepting some deterioration in the steam pressure control.

The second component of the controller output is the integral value. The integral setting value is time-based and closely relates to the process system time constant. Integral values are commonly expressed in repeats per minute, minutes per repeat, or seconds per repeat. There is more than one method of computing (algorithm) for the integral value. The most common is an integration with respect to time of the error multiplied by the gain.

If seconds-per-repeat is used and the controller setting is 10 seconds per repeat, then after



(B) Proportional-plus-Integral Control

Figure 4-1 Continued

10 seconds the controller would have added K (controller gain) times the error from set point to the controller output. Another common algorithm integrates only the error value and does not use the proportional gain.

In the case of Figure 4-1B, assume that, as time starts, the loop is stabilized at set point, the measured variable is pressure, the pressure sensor has a range of 0-400 psig, and the set point has a value of 200 psig. The controller has been tuned, the proportional gain is 2.0, and the integral setting is 0.5 repeat per minute. At the end of (x) time, the system flow capacity changes, the pressure begins to deviate from set point, and a change to the manipulated variable is necessary to return the measured variable to the set point.

This diagram shows that when the controller returns to set point the error is 0; therefore, any effect of the proportional gain is also reduced to zero. The proportional gain effect is, therefore, temporary and has a value only when the process variable deviates from the set point.

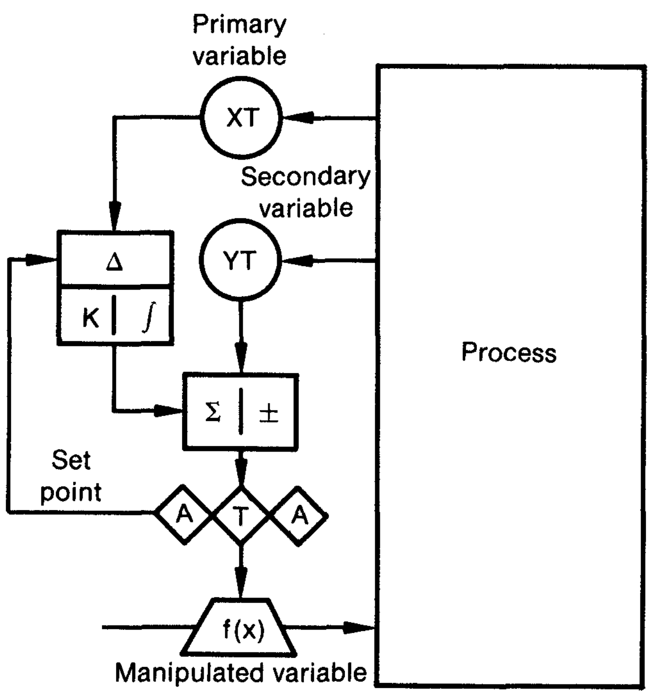
The value of the integral portion of the controller output is, however, permanent and, with the process variable at a steady state and at set point, is equal to the controller output.

If this had been a proportional-only controller, an offset from set point would have to be maintained to satisfy the need for a permanent change in the controller ouput.

4-2 Feedforward-plus-Feedback Control

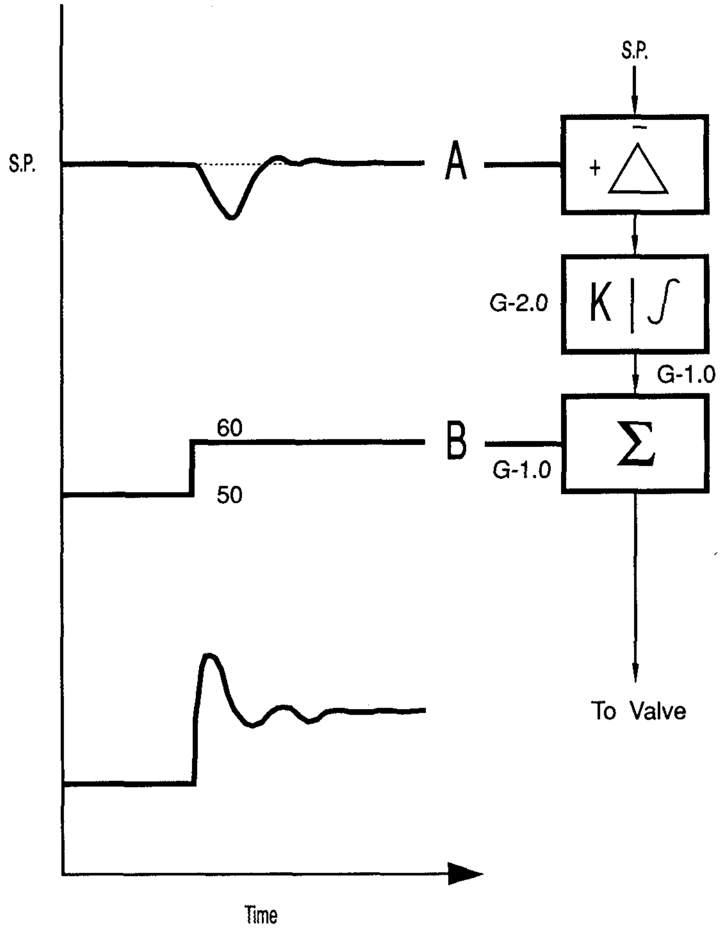
In feedforward-plus-feedback control as shown in Figure 4-2A, a secondary variable that has a predictable relationship with the manipulated variable is connected. In this case a change in the secondary variable causes the manipulated variable to change in anticipation of a change in the primary variable. This reduces the magnitude of the primary variable change due to the more timely control action that originates from the secondary variable. The feedback portion of the loop contains the set point and can contain any of the controller functions of the basic feedback loop. The feedforward gain is adjustable and may be greater than 1.0.

In Figure 4-2B, a time-based diagram based on the use of feedforward-plus feedback control is shown. In this case a change in the signal from the secondary variable acts without waiting for the primary variable to change. The goal is to calibrate the feedforward portion of the control so that the primary variable does not change value. This is done by changing the



(A) Feedforward-plus-Feedback Control

Figure 4-2 Control Functions



Feedforward Control Action

(B) Feedforward-plus-Feedback Control

Figure 4-2 Continued

gain or multiplier of the secondary variable so that its change multiplied by its gain produces a change in the feedforward portion of the controller output that is equal to the permanent steady-state change in the controller output. If this can be accomplished, then there is no permanent effect on the controller output from the feedback portion of the controller.

This net result is very similar to that of proportional-plus-integral control except that the required magnitude of the controller action takes place in less elapsed time. In this case the feedforward action essentially replaces the integral action, so that theoretically (with perfect calibration) the feedback controller needs only proportional action.

In actual practice, it is usually impossible to maintain such a controller in perfect calibration. To overcome that drawback, a small amount of integral action can be used to account for the mismatch between the feedforward effect and the required change in the steady-state output of the controller. In this case the tuning of the integral portion of the controller should not have a relationship to the process time constant but can be arbitrarily set at some very low value such as 0.05 to 0.1 repeat per minute.

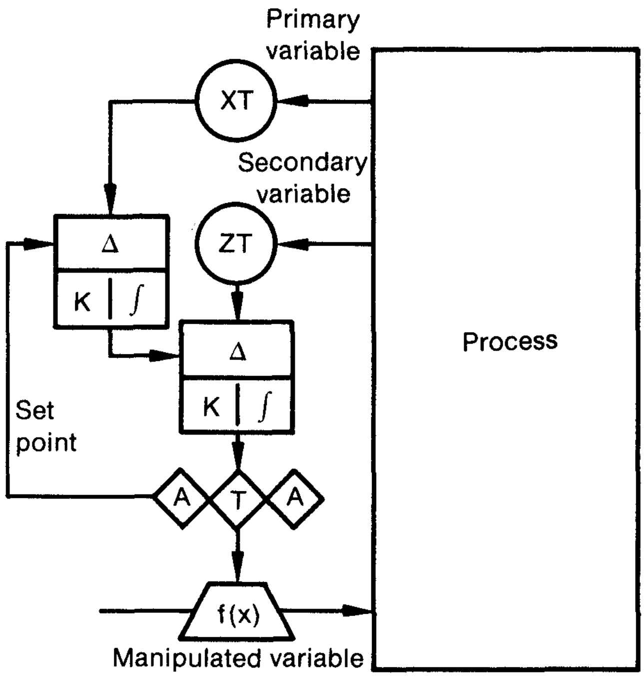


Figure 4-3 Cascade Control

4-3 Cascade Control

Cascade control consists essentially of two feedback control loops connected together with the output of the primary loop acting as set point for the secondary loop. Cascade control (see Figure 4-3) is applied to stabilize the manipulated variable so that a predictable relationship between the manipulated variable and the primary variable can be maintained.

To avoid control instability due to interaction between the two feedback control loops, it is necessary that the response time constants be substantially different. The process response of the secondary control loop should be the faster of the two. A general rule is that the time constant of the primary loop process response should be a minimum of 5 to 10 times that of the secondary loop. The longer time constant of the primary loop indicates a much slower response.

Because of this, a normal application would be temperature control (a normally slow loop) cascading onto flow control (a normally fast loop). Other suitable candidates for cascade control are temperature cascading onto pressure control and level control cascading onto flow control.

4-4 Ratio Control

The fourth of the basic loops involved in boiler control is ratio control, as shown in Figure 4-4. Ratio control consists of a feedback controller whose set point is in direct proportion to an uncontrolled variable. The proportional relationship can be set by the operator of the process, or it can be automatically adjusted by another controller. As shown, the mathematical function is a multiplier. If the ratio is set, the set point of the controlled variable changes in direct proportion to changes in the uncontrolled variable. If the multiplication is changed, the direct proportional relationship or ratio between the controlled and the uncontrolled variables is changed.

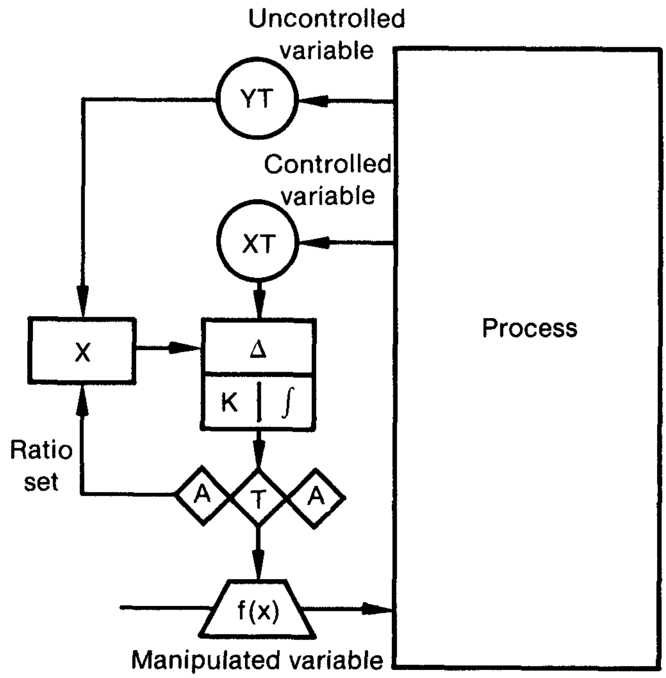


Figure 4-4 Ratio Control

A careful examination of most boiler control applications will show that the overall control system is an interconnected matrix of the four types of control aplication shown in Figures 4-1 through 4-4.

4-5 Some Fundamentals of Control System Application and Design

A first consideration is whether a particular control application merits or requires a multivariable interconnected system. The alternative is the simpler use of an individual independent control loop.

Generally, an independent control loop can be used when the controlled variable does not affect, or is not affected by, measured variables other than the one in the control loop. In boiler control specific examples are fuel supply pressure, pressure-reducing valves, and duct pressure control, assuming constant set points are used for each of these. If the set points are variable, as a function of other variables, then they are not independent loops.

A multivariable input and output system is used when the controlled and measured variables have interdependent effects. In this case the variables affect, or are affected by, more than one measured or manipulated variable.

A system in which control actions are parallel usually is less interacting than one in which the control actions are in series. An example is regulating the fuel and air to a boiler in parallel rather than first regulating the fuel or the air and controlling the other variable in sequential order. In such a case a disturbace in the first variable propagates to the second, causing a disturbance in the second variable.

Another general implementation rule is to develop compensation circuits within the control system so that disturbances in the variables do not enter and disturb the process. An example is control system compensation for the number of coal pulverizers being used so that starting or stopping another pulverizer does not materially change the fuel being fed to the furnace.

4-6 Process Dynamics—Control Response

The reponse characteristics are a combination of resistance, capacitance, and dead time. The major resistance effect is due to the heat transfer coefficients involved. Some of it is caused by the time constants of measurement and control devices and flow inertia and friction. Capacitance effects come from such relationships as mass storage to mass throughput and energy storage to energy throughput. Dead time occurs due to reponse dead time in measurement and control devices and process fluid transport.

While there is some dead time in all control loops, in most cases an assumption of a first-order response is usually adequate for analyzing the performance of boiler control loops. With that assumption, time constants are approximately as follows:

(1) Flow—seconds
(2) Pressure reducing—seconds
(3) Turbine speed—seconds
(4) Level—seconds to minutes
(5) Pressure (heat transfer)—minutes
(6) Temperature—minutes
(7) Coal flow (feeder to furnace)—seconds to minutes
(8) Boiler drafts—seconds
(9) Liquid pressure—milliseconds to seconds

4-7 Process Factors That Affect the Control System or Loop Application

In actual practice, control systems must cope with the mechanical factors of process noise and measurement noise. An example of process noise is the pulsation in furnace pressure. An example of measurement noise is the pressure pulsation of a flow or pressure measurement.

The difficulty in coping with the noise is a function of the magnitude of the noise-to-measurement signal ratio and the noise-to-set point ratio. As these ratios become greater, the controllers must be detuned to reduce noise effects on the controlled variables. Such noise may also dictate a control system redesign to assist in helping to reduce the effects of noise. Other actions that can be used are dampening (inserting an additional time constant), changing the measurement transmitter span, moving the point of measurement, using a nonlinear controller, or using a gap controller.

In the application of level controls to boiler or power plants, a number of other factors may dictate various control strategies. Some of the factors are the linearity of the fluid storage, the throughput/surface area, the control precision needed, the volume change that is acceptable, and the interaction with other control loops.

It can be easily seen that level controls for various fluid-holding equipment (such as the boiler drum, the condensate storage tank, the condenser hotwell, the deaerating heater storage tank, the cooling tower basin, and the stage heater condensate storage) need different control application designs. Some of these have traditionally been controlled with independent control loops. Interaction with other loops and process instability with independent loops make some of the above good candidates for multivariable control systems.

Section 5
Combustion of Fuels, Excess Air, and Products of Combustion

Fuels can generally be classified as gaseous, liquid, or solid. In cases where a solid fuel is finely ground, such as pulverized coal, and can be transported in an air stream, its control characteristics approach those of a gaseous fuel. Liquid fuels, as they are atomized and sprayed into a furnace, also have control characteristics similar to those of a gaseous fuel. The control treatment of a solid fuel that is not finely ground is quite different from that of a gaseous or a liquid fuel.

Whether a fuel is a gas, a liquid, or a solid is determined by the ratio of its two primary chemical ingredients, carbon and hydrogen. Natural gas has an H/C ratio in excess of 0.3. Fuel oil has an H/C ratio above 0.1, and the H/C ratio of coal is usually below 0.07. Since hydrogen is the lightest element and the molecular weight of carbon is approximately six times that of hydrogen, a decrease in the H/C ratio increases the specific gravity and the density of the fuel.

5-1 Gaseous Fuels—Their Handling and Preparation

The most used gaseous fuel is natural gas, but ''waste gas'' or gas produced as a process by-product may allow the replacement of natural gas or other purchased fuels. In the iron and steel industries such gases are coke oven gas and blast furnace gas. In the petroleum refining industry, a mixture of these gases is known as refinery gas. In the petrochemical industry, such gases may be called tail gas or off gas. A characteristic of these gases is a significant difference in heat content and other physical and chemical characteristics as compared to natural gas.

Natural gases vary in their chemical analyses and thus in their heating values. While the average heating value is approximately 1000 Btu per standard cubic foot (scf), Table 5-1 shows that these gases may commonly vary from 950 to over 1100 Btu per scf. Note that in all cases over 80 percent of the gas by volume is methane.

Natural gas is the only major fuel that is delivered by the supplier as it is used. The transfer of this fuel to the user usually occurs at a metering and pressure-reducing station as shown in Figure 5-1. The pressure-reducing valves reduce the pressure from the supplier's pipeline to the pressure required in the user's boiler control system. In addition, the gas is metered for billing purposes, and in many cases a calorimeter is also installed for recording the heating value of the gas that is supplied. No fuel preparation is required before the gas enters the boiler plant except for the reduction in pressure described above.

In the boiler control system the regulation of fuel Btu input can be accomplished with a standard control valve. The design of the valve is based on the capacity requirement of the system, the specific gas properties, and the pressure drop available for control of the gas flow. The supply of the ''waste gas'' streams is usually pressure- and/or flow-controlled based on the ability of the process to produce the particular gases. In almost 100 percent of the cases, some purchased natural gas must be used to meet the total plant demand for steam.

A mixture of propane and air is a fuel alternate for natural gas. It will burn in the same burners and under the same conditions as natural gas. A typical system for mixing the propane and air is shown in Figure 5-2. There are two precautions for properly mixing and burning a propane-air mixture: (1) Such a mixture should have a propane percentage significantly greater than 10.10 percent so that the mixture will not be explosive or flammable; (2), the mixture should act in the burner in much the same manner as the basic natural gas for which the burners

Table 5-1
Selected Samples of Natural Gas from United States Fields

Sample No.		1	2	3	4	5
Source of Gas		Pa.	So. Cal.	Ohio	La.	Okla.
Analyses						
Constituents, % by vol						
H_2	Hydrogen	—	—	1.82	—	—
Ch_4	Methane	83.40	84.00	93.33	90.00	84.10
C_2H_4	Ethylene	—	—	0.25	—	—
C_2H_6	Ethane	15.80	14.80	—	5.00	6.70
CO	Carbon monoxide	—	—	0.45	—	—
CO_2	Carbon dioxide	—	0.70	0.22	—	0.80
N_2	Nitrogen	0.80	0.50	3.40	5.00	8.40
O_2	Oxygen	—	—	0.35	—	—
H_2S	Hydrogen sulfide	—	—	0.18	—	—
Ultimate, % by wt						
S	Sulfur	—	—	0.34	—	—
H_2	Hydrogen	23.53	23.30	23.20	22.68	20.85
C	Carbon	75.25	74.72	69.12	69.26	64.84
N_2	Nitrogen	1.22	0.76	5.76	3.06	12.90
O_2	Oxygen	—	1.22	1.58	—	1.41
Specific gravity (rel to air)		0.636	0.636	0.567	0.600	0.630
Higher heat value						
Btu/cu ft @ 60° F & 30 in Hg		1,129	1,116	964	1,002	974
Btu/lb of fuel		23,170	22,904	22,077	21,824	20,160

(From *Steam, Its Generation and Use,* © Babcock and Wilcox)

are designed. Typically, the gas burner pressure for a given Btu input rate should be as close as possible to the pressure used for the basic natural gas fuel.

The characteristic properties of natural gas, air, and propane are shown in Table 5-2. For a given Btu input rate, the Btu per scf determines the number of cubic feet of gas required, and the specific gravity determines the pressure at the burner orifice.

A mixture of propane and air that will closely approximate the action of 1000 Btu per scf natural gas of 0.6 specific gravity is a mixture of 6 parts propane to 4 parts air. This produces a mixture of approximately 1500 Btu per scf with a specific gravity of 1.34. This fuel-rich mixture is 60 percent propane, well above the minimum flammable limit for a propane-air mixture.

5-2 Liquid Fuels—Their Handling and Preparation

The most common liquid fuel is fuel oil, a product of the oil refining process. While crude oil as produced from the well is sometimes used, the most common fuel oils used for boiler fuel are the lightweight No. 2 fuel oil and the No. 6 grade of heavy residual fuel oil. The normal ranges of analyses of these two fuel oils are shown in Table 5-3.

Other liquid fuels that are used as waste or auxiliary fuels are process by-products such as tar, pitch, or acid sludge, and, in some cases, liquid sewage. In some of these cases the heat content alone may not pay for burning the fuel, and the economics may be based on a comparison with the costs of other methods of waste disposal.

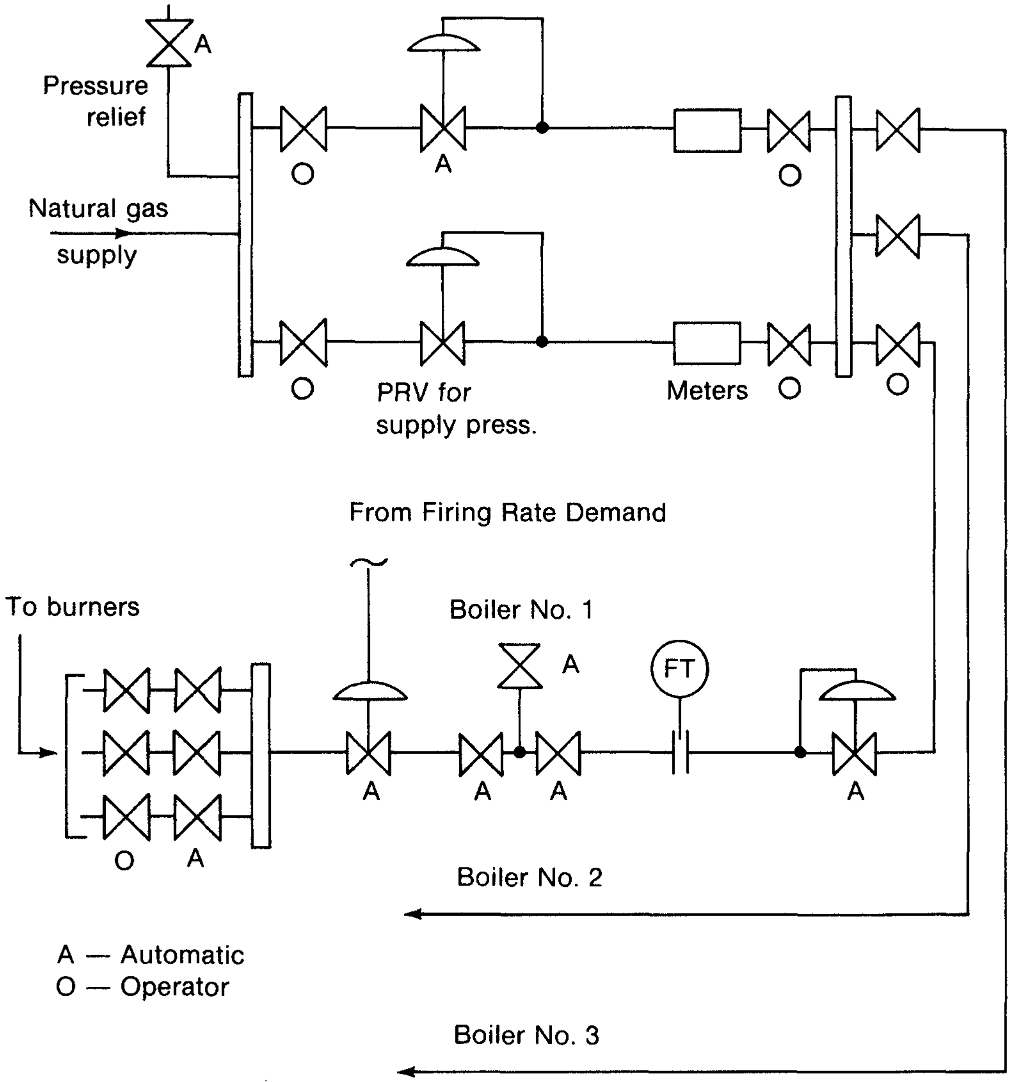


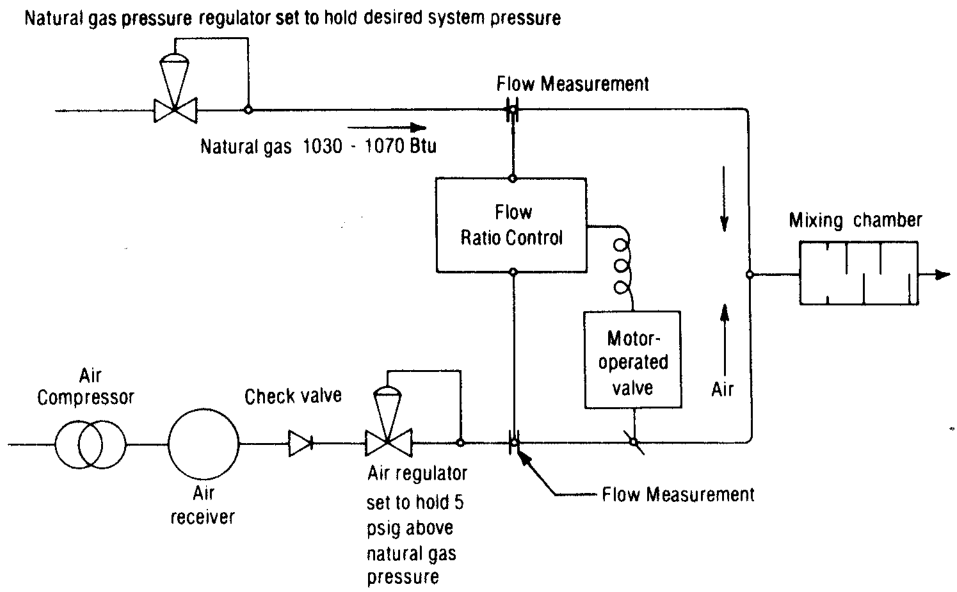
Figure 5-1 Gas Pressure-Reducing and Metering Arrangement

(From Dukelow, *Improving Boiler Efficiency*)

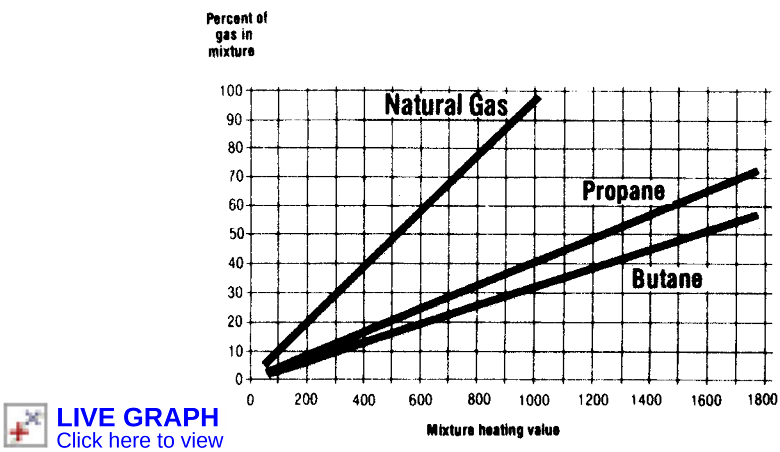
Particularly in pulp and paper mills there is a by-product liquid fuel known as black liquor or red liquor. This liquid is burned in order to recover the chemical content of the "liquor." The heat produced from the combustion of the dissolved "wood" chemicals as the liquor is burned is a bonus. Because the basic process need is chemical recovery, the process is operated to optimize chemical recovery rather than the heat from the combustion of the wood waste content.

If the user is not an oil refinery, the fuel oil is purchased in lots and delivered to the plant by truck, railroad tank car, or oil tanker. The fuel oil is pumped from these delivery vehicles into a user's fuel oil storage tanks and stored there until used. A generic arrangement including the fuel oil preparation is shown in Figure 5-3.

In this arrangement the fuel oil is delivered to the storage tanks. From the storage tanks the fuel may be taken directly to the fuel preparation equipment, or it may be transferred to a smaller tank, sometimes called a day tank. From the day tank, fuel oil pumps provide the pressure necessary for the fuel control and atomizing system.



Simple stabilization control. This type of plant is suitable when the base gas flow is steady or changes very slowly — and when the heating values of constituents change very slowly.



Mixture heating value versus percentage of gas in mixture for various gas-air mixtures.

Figure 5-2 A Typical System for Mixing Propane and Air

(From Dukelow, *Improving Boiler Efficiency*)

Table 5-2
Characteristic Properties

	Btu/scf	Specific Gravity
Natural gas	1000	.6
Propane	2524	1.5617
Air	0	1.0

Flammable limits — propane-to-air 2.10% to 10.10%

Table 5-3
Fuel Oil Analyses

	No. 2	No. 6
Carbon	86.1 to 88.2	86.5 to 90.2
Hydrogen	11.8 to 13.9	9.5 to 12.0
Sulphur	0.05 to 1.0	0.7 to 3.5
Nitrogen	Nil to 0.1	—
Ash	0	0.01 to 0.5
Heating value:		
Btu/lb	19,170 to 19,750	17,410 to 18,990
Water and sediment	0 to 0.01	0.05 to 2.0
Spec. gravity	0.887 to 0.825	1.022 to 0.922
Lb per gal	7.39 to 6.87	8.51 to 7.68

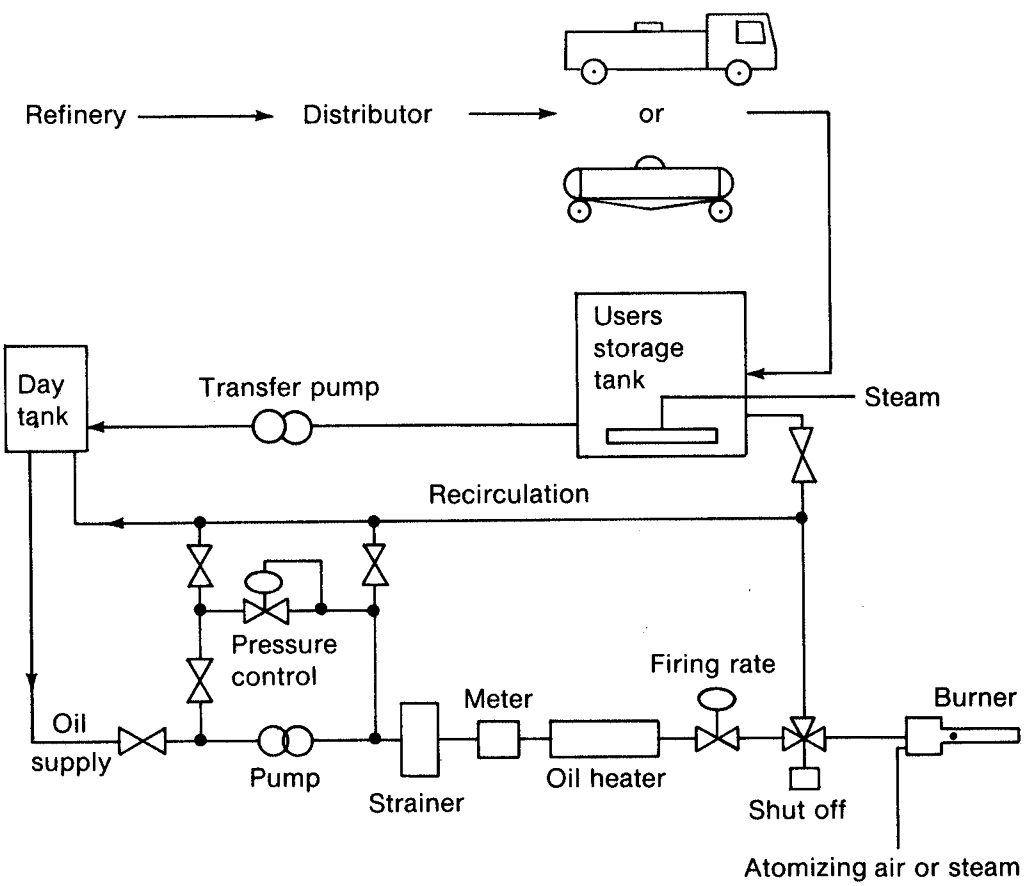


Figure 5-3 Typical Fuel Oil Pumping and Heating Arrangement

If the fuel is No. 2 fuel oil, heating of the fuel is normally unnecessary. If the fuel is a heavy oil such as No. 6, it is usually necessary to heat the oil in the tanks so that it can be easily pumped through the system. If heavy fuel oil in a tank is unused for a period of time, the tank heating may cause the evaporation of some of the lighter constituents, ultimately making the oil too thick to remove from the tank by any normal means.

In some installations water may be present in the oil system. This may be water that has condensed from the atmosphere over a period of time or water originating from cleaning the tanks with water. The mixture of oil and water can be burned with good results if the water is emulsified with the oil before atomizing at the burner. Emulsification forms tiny droplets of the water that are surrounded by a film of oil. As the water droplets enter the furnace, the furnace heat causes the water droplets to suddenly flash to steam - causing fine atomization of the oil film.

As a result of this type of action, water may be intentionally added to oil and emulsified to improve atomization. Due to poor atomization by existing oil burning equipment, this method has been substituted in some installations with improved results.

There is a heat loss penalty for using this method for improving atomization. Heat is lost by vaporization of the water. The heat loss is equal to the latent heat of vaporization (approximately 1040 Btu/lb of water under these conditions) multiplied by the total weight of water used.

Heavy fuel oil must also be heated before burning in order to reduce its viscosity. Figure 5-4 shows the temperature-viscosity relationship of the various grades of fuel oil. Since most

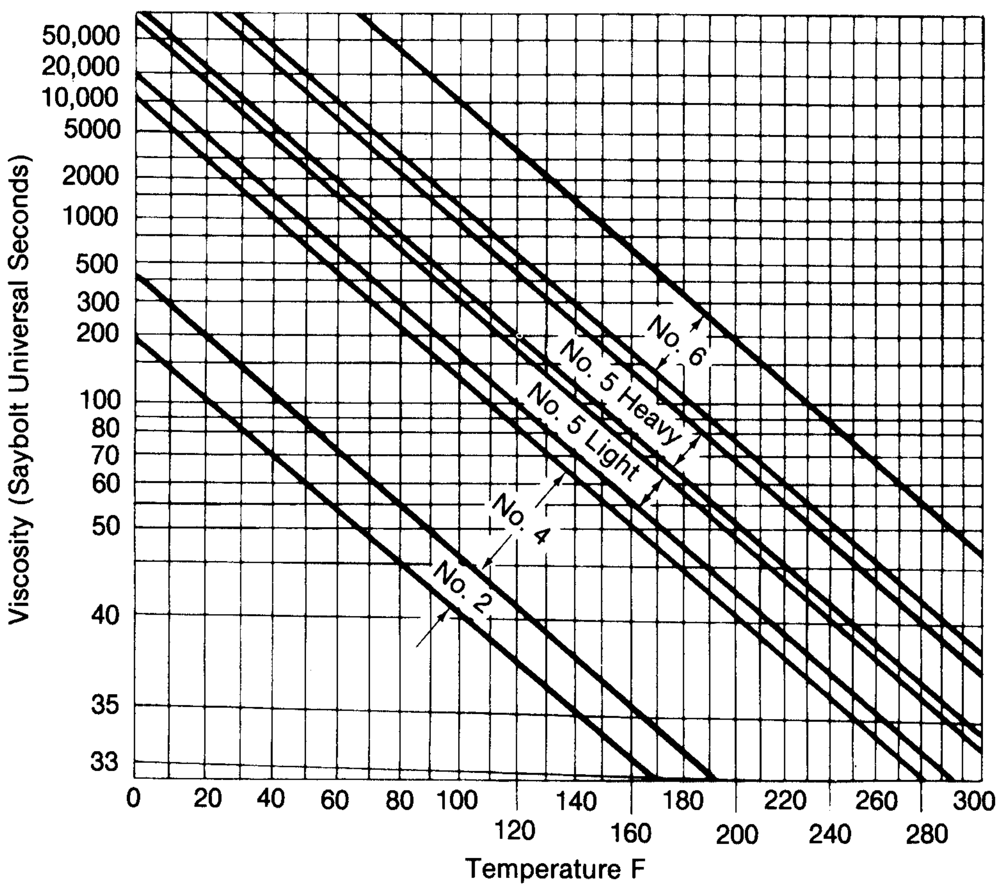


Figure 5-4 Fuel Oil Temperature vs. Viscosity

burners are designed for a viscosity of 135 to 150 Saybolt universal seconds (SSU), the temperature control of the fuel oil is set to produce the desired viscosity for whatever fuel is being used. Note that specifications for No. 6 oil cover a band of viscosities. Because of this, the correct temperature set point necessary to produce the desired viscosity may vary depending upon the specific fuel characteristics.

In the boiler control system the regulation of fuel Btu input can be accomplished by a standard control valve. The design of the valve is based on the capacity requirement of the system, the specific fuel oil properties, and the pressure drop available for control of the fuel oil flow. As with the gas-fired systems, the use of the waste or auxiliary liquid fuels is usually dependent upon the ability of the process to produce such by-product fuels. Also as in the gas-fired system, the use of such fuels may be based on disposal needs rather than the heat of combustion available.

5-3 Solid Fuels—Their Handling and Preparation for Firing

The most commonly purchased solid fuel is coal. Coal is available in a number of grades and classifications that vary from bituminous coal with almost 15,000 Btu per lb to a low-grade coal called lignite of less than 7,000 Btu per lb. There are two different analyses of coal.

Table 5-4
Coal Analyses on As-Received Basis (From *Steam, Its Generation and Use*, © Babcock and Wilcox)

Proximate analysis		Ultimate analysis	
Component	Weight, %	Component	Weight, %
Moisture	2.5	Moisture	2.5
Volatile matter	37.6	Carbon	75.0
Fixed carbon	52.9	Hydrogen	5.0
Ash	7.0	Sulfur	2.3
Total	100.0	Nitrogen	1.5
		Oxygen	6.7
Heating value		Ash	7.0
Btu/lb	13,000	Total	100.0

(From *Steam, Its Generation and Use*, © Babcock and Wilcox)

Table 5-5
Coal Analysis from Different Localities

		% by weight						
	Btu lb	Ash	S	H_2	C	H_2O	N_2	O_2
Alabama	13,350	8.0	1.06	4.87	74.8	3.5	1.46	6.32
Illinois	11,200	9.0	3.79	4.26	61.78	12.0	1.21	7.96
Indiana	11,300	10.7	4.2	4.35	61.52	10.8	1.25	7.18
Kansas	10,950	12.0	4.1	4.09	60.65	12.0	1.16	6.0
Kentucky	12,100	9.0	2.92	4.5	67.24	7.5	1.32	7.52
Ohio	12,550	9.0	3.18	4.74	69.41	5.0	1.36	7.31
Pennsylvania	13,850	8.0	1.96	4.63	77.87	3.0	1.57	2.97
Colorado	9,670	5.4	0.6	3.2	57.6	20.8	1.2	11.2
North Dakota	6,940	5.9	0.6	2.7	41.1	37.3	0.4	12.0

One of these, called proximate analysis, is used primarily for ranking coal. The other, called ultimate analysis, is the analysis of the chemical constituents by weight percentage. The ultimate analysis is the one used in combustion calculations. Table 5-4 demonstrates the difference between the two analysis methods. Table 5-5 shows a comparison of typical coals produced by mines in different localities.

Auxiliary or waste solid fuel is usually a process by-product that is burned either for its heating value or for disposal purposes. The most frequently used solid auxiliary fuel is some form of wood waste from wood product manufacturing processes such as lumber saw mills and pulp and paper manufacture. Typical of these are the bark from the pulpwood that is made into paper pulp and sawdust from a sawmill. The residue of sugar cane (called bagasse) is used in the sugar industry, and coffee grounds are used in plants that manufacture instant coffee. Other solid auxiliary fuel is solid waste such as municipal garbage or other refuse.

5-4 Handling and Delivery of Solid Fuels

The various steps in the coal-handling process are shown in Figure 5-5. Many ''mine mouth'' power plants for electric power generation burn the coal essentially as it is mined, along with all the dirt and rock that come along with the coal. Coal used in other electric power generation boilers and in industrial boilers generally follows the steps shown in Figure 5-5. The coal is cleaned and sized as in the process shown. Since such coal must be shipped to the point of use, this cleaning reduces freight charges. For proper burning of coal with a stoker, the size of the coal lumps is important. The coal bunkers admit coal by gravity directly to the stoker hoppers.

One coal preparation method crushes the coal and mixes it with water so that it can be transported in a pipeline as a slurry. The slurry is then dewatered with a centrifuge at the point of use before being pulverized. Such slurry pipelines over 200 miles in length are in service in the Arizona-Nevada area. Much longer ones are planned.

The largest volume of non-coal solid fuel is wood or wood waste and bark from the paper or lumber industry. The steps in its preparation and feed to the furnace are shown in Figure 5-6. The ''hogger'' device chops the wood waste to a somewhat uniform size so that it can be easily handled and burned by a stoker. Other solid waste fuel is bagasse, the fibrous residue from sugar cane, coffee grounds (which remain after the process of making instant coffee), and other solid refuse. The handling of these fuels is similar to that of wood waste. A characteristic of industrial solid waste is its high moisture content, at times in the 50 to 60% range.

Coal is delivered to the user in lump or chunk form by means of trucks or rail cars. The user must have sufficient space and mechanical equipment for storing and handling the coal prior to use. The fuel is often furnished on a sized basis after preliminary preparation near the mine. The sized coal is transferred from coal storage to coal ''bunkers.'' These bunkers, or short-term storage bins, are located at an elevation that allows the coal to feed to the boiler system by gravity.

There are three basic coal-burning methods. Of these, ''on a grate'' and ''in suspension'' have been used for many years. The third is ''fluidized bed,'' a relatively new coal-burning development that is emerging as a future major burning method. For grate-type burning, coal needs no further preparation and flows by gravity to a stoker hopper. In many cases a weighing system is installed to weigh the amount of coal that is admitted to the stoker hopper.

A fluidized bed system uses an adjustable rate feeder to admit the coal to the fluidized bed. This feeder can be either a volumetric (by volume) or a gravimetric (by weight) type of device.

If coal is to be burned in suspension, this implies that it is to be pulverized or very finely ground before being admitted to the furnace. Figure 5-7 demonstrates this method of fuel preparation. There are different types of pulverizers, but the basic principle is the same. An adjustable coal feeder delivers the coal to a motor-driven pulverizer (also called a mill). A primary air fan connected to the pulverizer forces or induces air flow through the pulverizer.

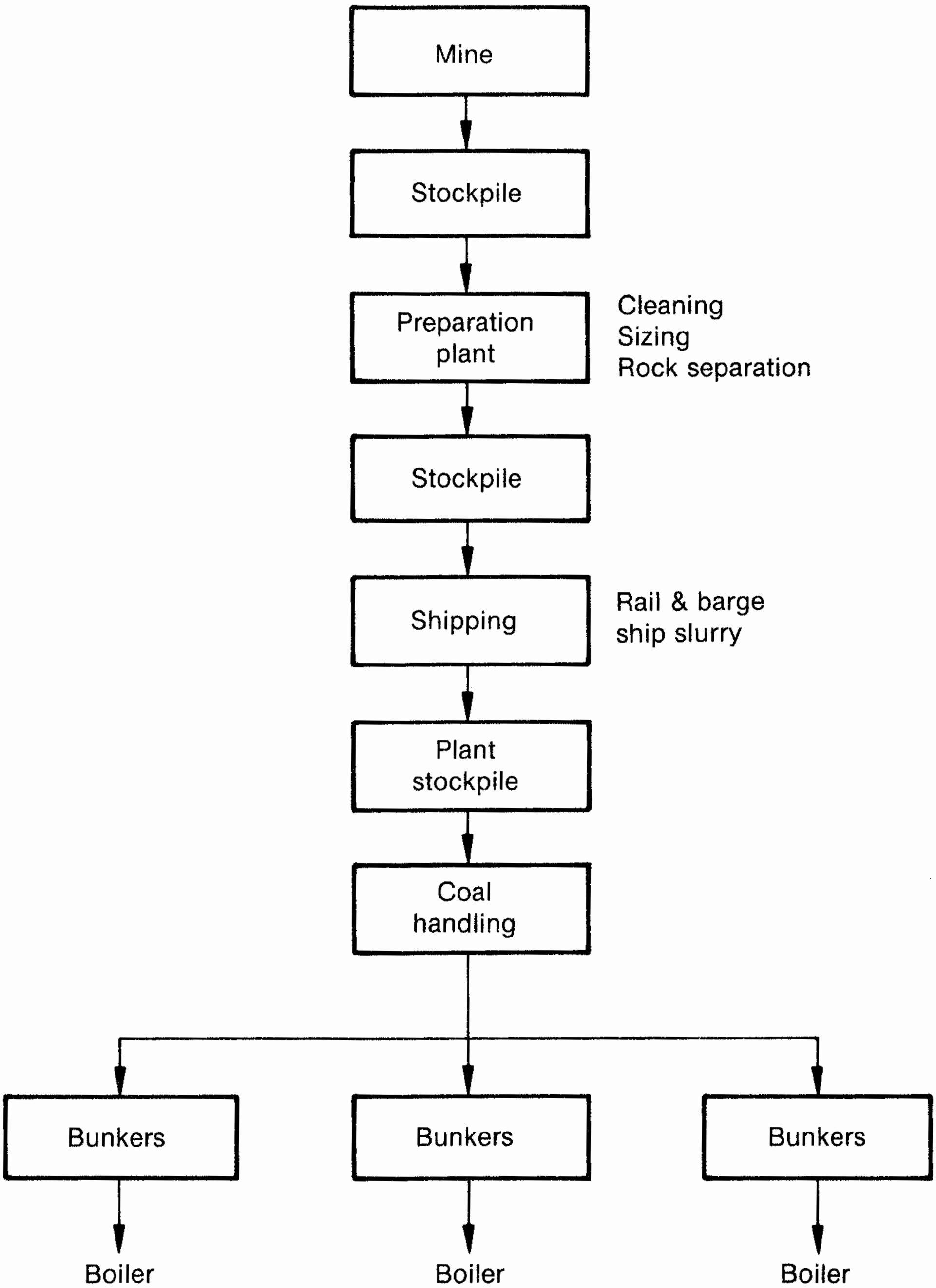


Figure 5-5 Fuel Preparation and Handling—Coal

As the coal is ground to a fine powder, the primary air stream transports the coal to the burner. To dry the moisture from the coal, the temperature of the coal-air mixture is controlled by mixing hot air and ambient air at the inlet to the pulverizer.

Most auxiliary or waste solid fuels are burned on a grate. The design of the grate-burning system is based on the particular fuel to be burned. Some such fuels may be prepared briquettes that can be burned very similarly to coal on a coal stoker. For other fuels, depending on the

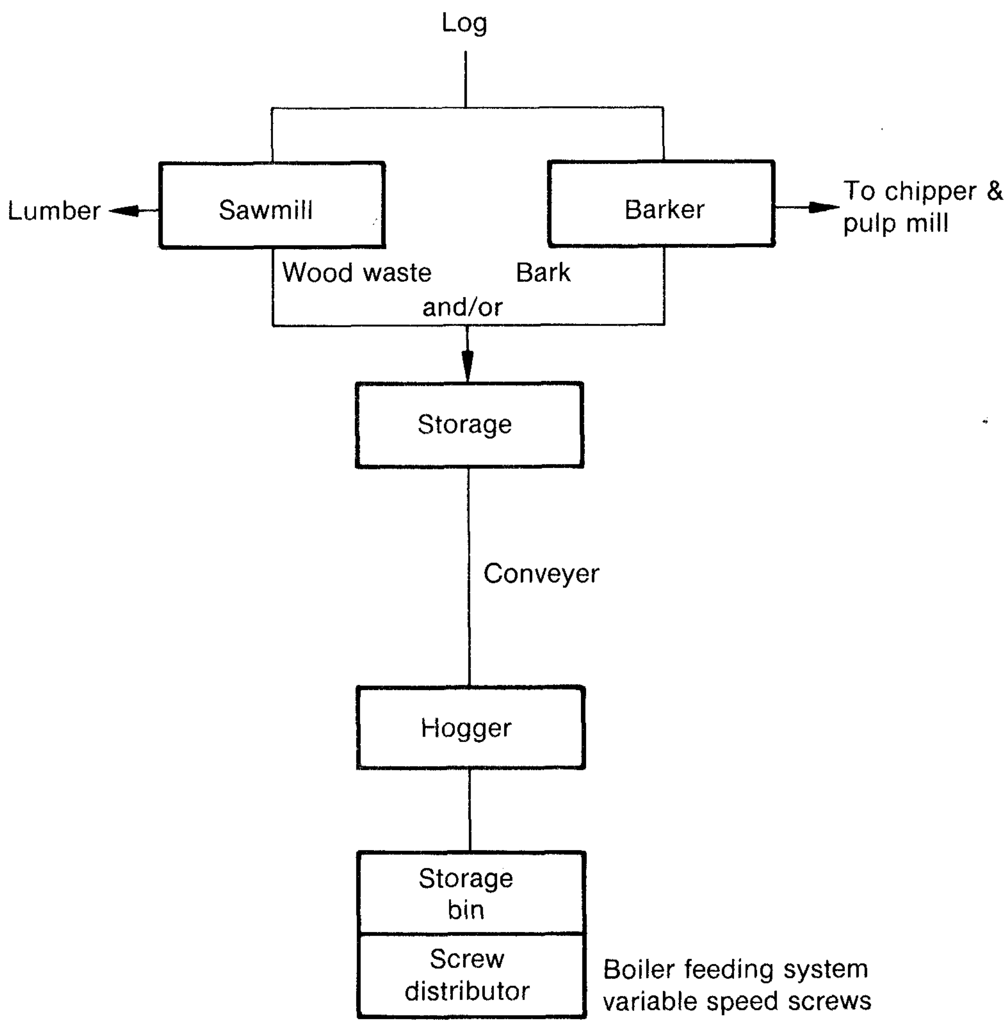


Figure 5-6 Fuel Preparation—Waste Wood or Bark

condition of the fuel or the type of grate-burning system used, the fuel may or may not need further preparation before being admitted to the furnace.

A common preparation for wood waste is called "hogging." A hogger shreds the wood waste to a somewhat uniform size to improve the performance of the fuel burning system. After hogging, the fuel is conveyed to a bin from which variable-speed screw feeders or other adjustable devices deliver the fuel to the furnace.

5-5 Fuel Mixtures—Coal-Oil, Coal-Water

To reduce fuel costs or when a preferred fuel is in short supply, fuels can be burned in combination. Burning of gaseous, liquid, or solid fuels in separate systems but in combination burning requires the same fuel preparation as if the fuel were burned alone. For such burning, boiler designs must be the same as if the fuels were burned alone.

The existence of thousands of gas- and/or oil-designed installations has spurred an effort to find ways of burning coal in boilers designed for oil firing. A mixture of finely ground coal in a colloidal mixture with oil produces a fuel that can be burned in an oil furnace in a manner very similar to that of No. 6 fuel oil. This mixture contains up to approximately 50 percent

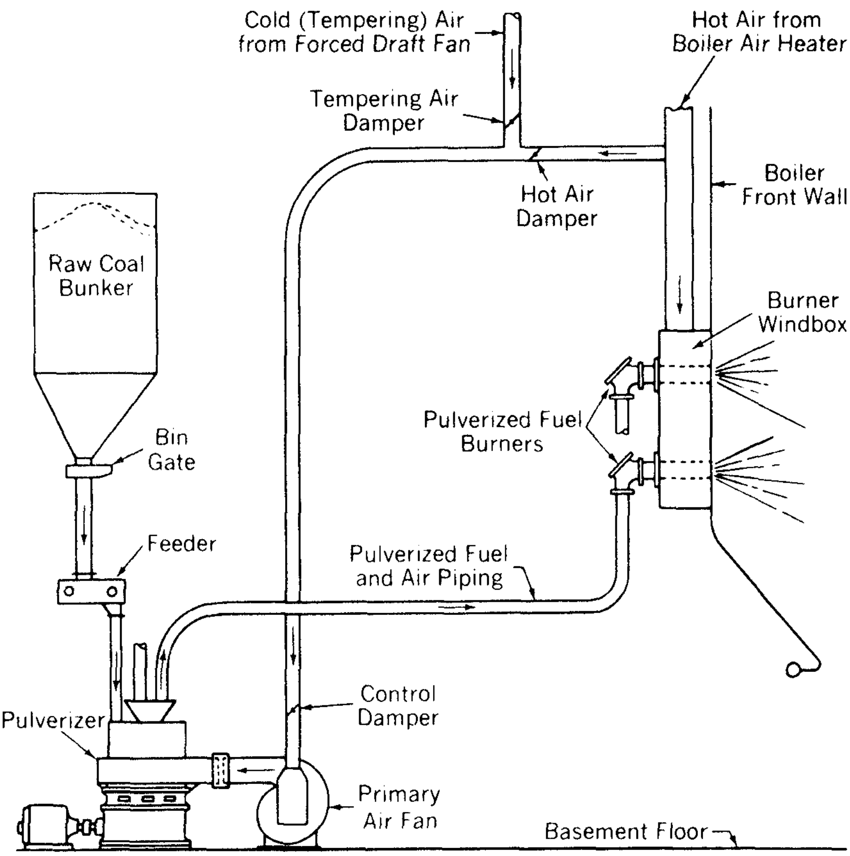


Figure 5-7 Direct Firing System for Pulverized Coal

(From *Steam, Its Generation and Use*, © Babcock and Wilcox)

coal. Additives in the mixture help to keep the coal in suspension so that it will not settle out in the storage tanks. The advantage of this process is the reduction in the cost of fuel, since the price of coal for a given heat content is much less than that of fuel oil.

A similar developing fuel that substitutes for fuel oil is a mixture of water and pulverized coal (approximately 70 percent coal) in which the coal is kept in suspension with chemical additives. The advantage of this fuel is the complete elimination of the higher priced fuel oil. A disadvantage is the resulting lower boiler efficiency due to the increased latent heat loss of the water vapor in the flue gas. Both the coal-oil mixture and the coal-water mixture require a pulverizing system, a mixing system, and a storage and handling system similar to that for heavy fuel oil.

5-6 Physical Combustion Requirements

Combustion is the rapid oxidation of fuel in a mixture of fuel and air with heat produced and carried by the mass of flue gas generated. Combustion takes place, however, only under the conditions shown in Figure 5-8.

Time, temperature, and turbulence are known as the three "T"s of combustion. A short period of time, high temperature, and very turbulent flame indicates rapid combustion. Turbulence is a key because the fuel and air must be mixed thoroughly if the fuel is to be completely burned. When fuel and air are well mixed and all the fuel is burned, the flame temperature will be high and the combustion time will be shorter. When the fuel and air are not

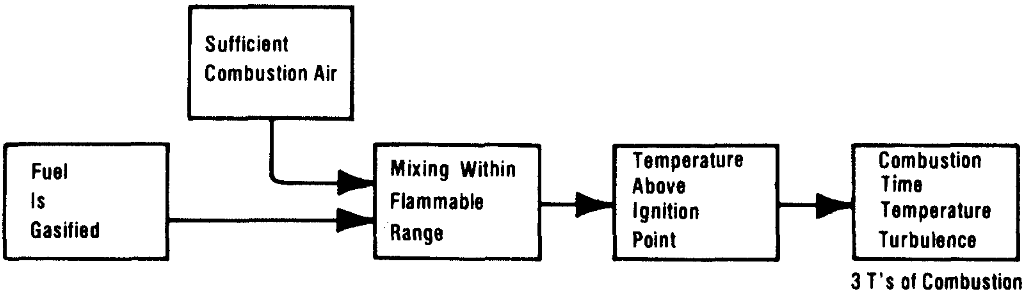


Figure 5-8 Combustion Requirements

(From Dukelow, *Improving Boiler Efficiency*)

well mixed, complete combustion may not occur, the flame temperature will be lower, and the fuel will take longer to burn.

Less turbulence and longer burning has, however, been found to produce fewer nitrous oxides (NOx). In some cases, combustion is delayed or staged intentionally to obtain fewer nitrous oxides or to obtain desired flame characteristics.

The fuel must be gasified. In the case of natural gas, this is automatically true. For oil, the fuel must be atomized so that the temperature present can turn it into a gas. When coal is burned, the coal must be pulverized so that it can be gasified by the furnace temperature or distilled in suspension or on the grate by the furnace temperature if a stoker is used.

The ignition temperature and also the flame temperature are different for different fuels if all other conditions are the same. Table 5-6 lists the flammable limits of the "standard" fuels

Table 5-6
Flammable Limits and Ignition Temperatures (From *Flame Safeguard Controls* © Honeywell, Inc.)

Percentage of Stoichiometric Air

	Minimum	Maximum
Natural gas	64	173
Oil	30	247
Coal (pulverized)	8	425

Ignition Temperatures of Fuels

Fuel	Degrees F
Kerosene	500
Light fuel oil	600
Gasoline	735
N-Butane	760
Heavy fuel oil	765
Coal	850
Propane	875
Natural gas	1000
Hydrogen	1095
Carbon monoxide	1170
Natural gas	1200

(From *Flame Safeguard Controls*, © Honeywell, Inc. Used with permission.)

and the ignition temperatures of a number of fuels. Of these, when properly mixed with air, the gases have the highest temperature required for ignition. Various liquid fuels if properly atomized and mixed with air have the lowest ignition temperatures.

The table of flammable mixture limits identifies that coal may continue burning or be ignited with as little as 8 percent of the theoretical air required for combustion. Natural gas, on the other hand, becomes "fuel-rich" and cannot be ignited or burned if less than 64 percent of the complete combustion theoretical air is mixed with the fuel. This is consistent with the fact that a coal fire can be "banked" with a very small amount of combustion air present, and that combustion will continue at a very low rate until a larger amount of combustion air is admitted.

Combustion may take place when the physical requirements are within the limits but may not proceed to completion due to insufficient combustion air and/or insuffiient turbulence for complete mixing of the fuel and air. This can also occur if the gases are chilled by heat being withdrawn before the combustion proceeds to completion. The chilling may occur if the flame impinges on relatively cold boiler heat transfer surfaces in the furnace. It can also occur if the furnace volume is too low and allows too little time for complete combustion to take place before the gases are chilled by the convection heating surfaces of the boiler. Figure 5-9 demonstrates some of these effects as they relate to the products of combustion.

5-7 Combustion Chemistry and Products of Combustion

For all fuels, the actual chemical process is the oxidation of the hydrogen and the oxygen in the fuel by combining them with oxygen from the air. The nitrogen from the air and any other non-combustibles in the fuel pass through the process with essentially no chemical change. A minimal amount of nitrogen in the air combines with oxygen to form nitrous oxides (NOx), which pollute the air. Some fuels contain a small percentage of sulphur, which—when burned—results in sulphur oxides that pollute the air. These may also corrode the boiler if the flue gas containing them is allowed to cool below the dew point.

Figure 5-10 demonstrates the basic chemical process and the chemical elements and compounds involved in complete and incomplete combustion. For any fuel, a precise amount of

Insufficient Air

$2C + O_2$	$2CO$ + heat energy
$4H_2 + O_2$	$2H_2O + 2H_2$ + heat energy

Insufficient Time, Temperature, Turbulence

$4CH_4 + 8O_2$	$2CO + 2CO_2 + 6H_2O + 2H_2 + 2O_2$ + heat energy
$8CH_4 + 16O_2$	$4CO_2 + 12H_2O + 2\,CH_3CHO^* + 5O_2$ + heat energy

NOTE:	$2CO + 2H_2 + 2O_2$	$2CO_2 + 2H_2O$ + heat energy
	$2CH_3CHO + 5O_2$	$4CO_2 + 4H_2$ + heat energy

$^*CH_3CHO$ — Acetaldehyde

Figure 5-9 Incomplete Combustion Examples

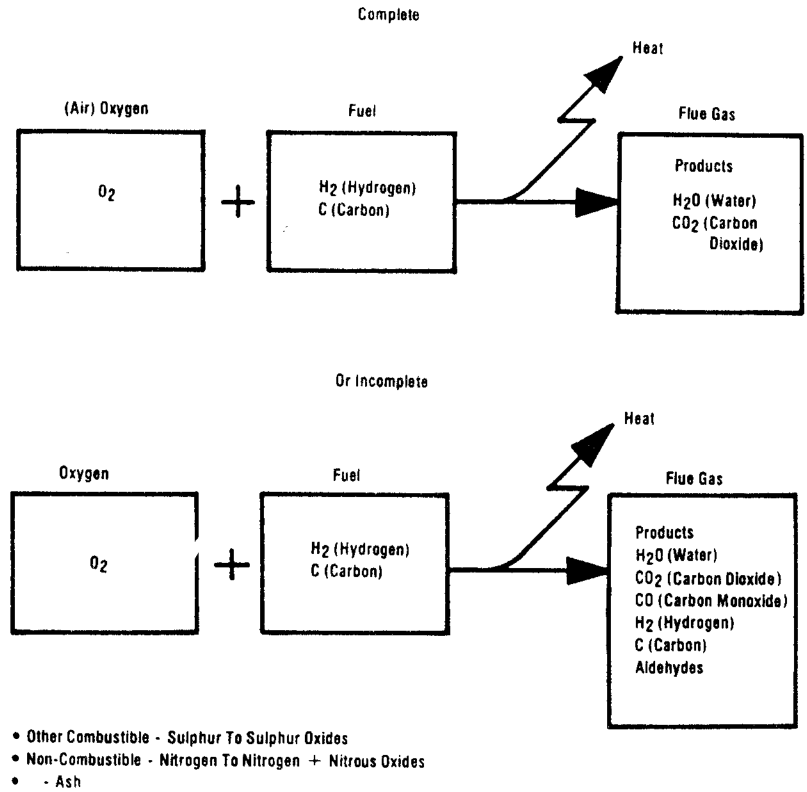


Figure 5-10 Basic Combustion Chemistry and Products of Combustion

combustion air is needed to furnish the oxygen for complete combustion of that fuel's carbon and hydrogen.

The precise amount of combustion air is called the theoretical air for that particular fuel. If the fuel analysis is known, the theoretical air requirements can be calculated easily.

The amounts of carbon and oxygen for complete combustion of carbon are represented by the formula:

$$\text{(carbon)} + \text{(oxygen)} = \text{(carbon dioxide)}$$
$$C + O_2 = CO_2 + 14{,}100 \text{ Btu/lb C}$$
$$12 \text{ lbs} + 32 \text{ lbs} = 44 \text{ lbs}$$

Weights equivalent to the molecular weight in pounds combine. One molecule of carbon containing one atom of carbon combines with one molecule of oxygen containing two atoms of oxygen to form one molecule of carbon dioxide containing one atom of carbon and two atoms of oxygen.

The formula for the combustion of hydrogen is represented:

$$\text{(hydrogen)} + \text{(oxygen)} = \text{(water)}$$
$$2H_2 + O_2 = 2H_2O + 61{,}100 \text{ Btu/lb } H_2$$
$$4 \text{ lbs} + 32 \text{ lbs} = 36 \text{ lbs}$$

As with the carbon combustion, weights equivalent to the molecular weights in pounds combine. Two molecules of hydrogen, each containing two atoms of hydrogen, and one molecule of two atoms of oxygen make two molecules of water, with a total of four atoms of hydrogen and two atoms of oxygen.

A simple example of the many incomplete combustion reactions resulting in intermediate hydrocarbon compounds is the partial combustion of carbon, resulting in carbon monoxide rather than carbon dioxide. In this case some of the potential heat energy from the carbon remains in the carbon monoxide.

$$\begin{aligned} \text{(carbon)} + \text{(oxygen)} &= \text{(carbon monoxide)} \\ 2C + O_2 &= 2CO + 4{,}000 \text{ Btu/lb C} \\ 24 \text{ lbs} + 32 \text{ lbs} &= 56 \text{ lbs} \end{aligned}$$

With the right conditions of time, temperature, and turbulence, and by adding more oxygen to the carbon monoxide, the carbon monoxide will further oxidize to carbon dioxide, releasing the second part of the heat energy from the original carbon.

$$\begin{aligned} \text{(carbon monoxide)} + \text{(oxygen)} &= \text{(carbon dioxide)} \\ 2CO + O_2 &= 2CO + 4{,}345 \text{ Btu/lb CO} \\ 56 \text{ lbs} + 32 \text{ lbs} &= 88 \text{ lbs} \end{aligned}$$

The common chemical reactions in combustion are shown in Table 5-7, with the heat energy resulting from the combustion reaction.

Figures 5-9 and 5-10 and Table 5-7 identify those products of combustion that are produced by the oxidation of the hydrogen, carbon, or sulfur present in the fuel. As indicated, the combustion process produces heat, but a low percentage of the heat produced is not useful in transferring heat to the boiler water. As hydrogen combines with oxygen during the combus-

Table 5-7
Common Chemical Reactions in Combustion (From *Steam, Its Generation and Use* © Babcock and Wilcox)

Combustible	Reaction	Moles	Pounds	Heat of combustion (high) Btu/lb of fuel
Carbon (to CO)	$2C + O_2 = 2CO$	2 + 1 = 2	24 + 32 = 56	4000
Carbon (to CO_2)	$C + O_2 = CO_2$	1 + 1 = 1	12 + 32 = 44	14,100
Carbon monoxide	$2CO + O_2 = 2CO_2$	2 + 1 = 2	56 + 32 = 88	4,345
Hydrogen	$2H_2 + O_2 = 2H_2O$	2 + 1 = 2	4 + 32 = 36	61,100
Sulfur (to SO_2)	$S + O_2 = SO_2$	1 + 1 = 1	32 + 32 = 64	3,980
Methane	$CH_4 + 2O_2 = CO_2 = 2H_2O$	1 + 2 = 1 + 2	16 + 64 = 80	23,875
Acetylene	$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$	2 + 5 = 4 + 2	52 + 160 = 212	21,500
Ethylene	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	1 + 3 = 2 + 2	28 + 96 = 124	21,635
Ethane	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	2 + 7 = 4 + 6	60 + 224 = 284	22,325
Hydrogen sulfide	$2H_2S + 3O_2 = 2SO_2 + 2H_2O$	2 + 3 = 2 + 2	68 + 96 = 164	7,100

(From *Steam, Its Generation and Use,* © Babcock and Wilcox)

Coal:	5 to 7%; higher efficiency than gas 1 to 3%; higher efficiency than oil
Oil:	3 to 5%; higher efficiency than gas 1 to 3%; lower efficiency than coal
Gas:	5 to 7%; lower efficiency than coal 3 to 5%; lower efficiency than oil

Figure 5-11 Products of Combustion and Effect on Boiler Efficiency

tion process to form water, the combustion temperature vaporizes the water into superheated steam. This vaporization absorbs the latent heat for producing the steam from the hot combustion gases. As the gases pass through the boiler and exit from the system, the gases retain the vaporized water in the form of superheated steam, and the latent heat and any remaining sensible heat are lost from the process.

The amount of latent heat loss is determined by the hydrogen content of the fuel. If the fuel is natural gas and thus is higher in % hydrogen, the latent heat loss is greater than if the fuel were coal, which is lower in % hydrogen. The effect on boiler efficiency for different fuels is shown in Figure 5-11.

- Gross Btu/lb as measured by calorimeter, 14,100 Btu/lb

Ultimate analysis;

C	—	80.31%
H_2	—	4.47%
S	—	2.85%
N_2	—	1.38%
H_2O	—	2.9%
Ash	—	6.55%

- BUT 0.0447 lb H_2 forms water vapor and 0.029 lb water is vaporized. All water vapor discharged in flue gas.

- Water vapor formed from H_2: $0.0447 \times \frac{18.02}{2.016} = 0.3996$ lb

H_2O in fuel .... = 0.029 lb

Total H_2O/lb. in fuel .... = .4286 lb.

- Heat loss due to vaporizing H_2O: $0.4286 \times 1040^* = 445.7$ Btu

Net Btu/lb = Gross Btu/lb - Loss vaporizing H_2O or
14,100 - 445.7 = 13.654.3 Btu/lb

- Loss vaporizing H_2O = 445.7/14,100 = 3.16%

*Standard ASTM procedure uses 1040. Actually, value changes with fuel analysis due to change in partial pressure of H_2O in the flue gases.

Figure 5-12 Gross and Net Btu Content (higher and lower heating value)

Since this latent heat is not useful to a combustion process, the fuel is said to have a "gross" and a "net" heating value or a higher (HHV) or a lower (LHV) heating value. It is important to keep in mind that combustion air must be furnished for the total combustion or on the basis of the HHV, while only the LHV has any effect on the heat transfer of the system. Figure 5-12 demonstrates with a coal analysis how the difference between these two heating values can be calculated.

5-8 Theoretical Air Requirements and Relationship to Heat of Combustion

Using the combustion chemistry formulas, if the fuel analysis is known, the theoretical amount of oxygen can be calculated. The amount of oxygen can easily be converted to a quantity of combustion air due to the known content of oxygen in air. An example of this calculation using a formula developed from the combustion equations and the known content of oxygen in air is given in Figure 5-13. In this example the amount of air theoretically required to produce 10,000 Btu is also shown. Table 5-8 is a tabulation of combustion constants

(A) Theoretical Requirements for Combustion Air

- Lbs air/lb fuel $= 11.53\,C + 34.34\left(H_2 - \dfrac{O_2}{8}\right) + 4.29\,S$

for 10,000 Btu

$$\text{Lb air/10,000 Btu} = \frac{10{,}000\left[11.53C + 34.34\left(H_2 - \dfrac{O_2}{8}\right) + 4.29\,S\right]}{\text{Btu/lb}}$$

C, H_2, O_2, S are decimal equivalent to ultimate analysis % by weight.

(B) Theoretical Air Example

- Assume coal; Ultimate Analysis:

C	—	80.31%
H_2	—	4.47%
S	—	1.54%
N_2	—	1.38%
H_2O	—	2.9%
Ash	—	6.55%
O_2	—	2.85%

- Lbs air/lb fuel =

$$(11.53 \times 0.8031) + 34.34\left(0.0447 - \frac{0.0285}{8}\right) + (4.29 \times 0.0154)$$

$$9.259 + 1.413 + 0.0661 = 10.7381$$

- Lbs air/10,000 Btu $= \dfrac{10{,}000 \times 10.7381}{14{,}100} = 7.6156$

Figure 5-13 Combustion Equations and Theoretical Air Examples

Table 5-8
Combustion Constants (From *Steam, Its Generation and Use* © Babcock & Wilcox)

No.	Substance	Formula	Molecular Weight	Lb per cu ft	Cu ft per lb	Sp Gr Air=1.0000	Heat of combustion: Btu per cu ft, Gross (High)	Btu per cu ft, Net (Low)	Btu per lb, Gross (High)	Btu per lb, Net (Low)	For 100% total air, moles per mole of combustible or cu ft per cu ft of combustible: Required for combustion, O_2	Required, N_2	Required, Air	Flue products, CO_2	Flue products, H_2O	Flue products, N_2	For 100% total air, lb per lb of combustible: Required for combustion, O_2	Required, N_2	Required, Air	Flue products, CO_2	Flue products, H_2O	Flue products, N_2
1	Carbon*	C	12.01	—	—	—	—	—	14,093	14,093	1.0	3.76	4.76	1.0	—	3.76	2.66	8.86	11.53	3.66	—	8.86
2	Hydrogen	H_2	2.016	0.0053	187.723	0.0696	325	275	61,095	51,623	0.5	1.88	2.38	—	1.0	1.88	7.94	26.41	34.34	—	8.94	26.41
3	Oxygen	O_2	32.00	0.0846	11.819	1.1053	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
4	Nitrogen (atm)	N_2	28.01	0.0744	13.443	0.9718	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
5	Carbon monoxide	CO	28.01	0.0740	13.506	0.9672	321	321	4,347	4,347	0.5	1.88	2.38	1.0	—	1.88	0.57	1.90	2.47	1.57	—	1.90
6	Carbon dioxide	CO_2	44.01	0.1170	8.548	1.5282	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
	Paraffin series																					
7	Methane	CH_4	16.04	0.0425	23.552	0.5543	1012	911	23,875	21,495	2.0	7.53	9.53	1.0	2.0	7.53	3.99	13.28	17.27	2.74	2.25	13.28
8	Ethane	C_2H_6	30.07	0.0803	12.455	1.0488	1773	1622	22,323	20,418	3.5	13.18	16.68	2.0	3.0	13.18	3.73	12.39	16.12	2.93	1.80	12.39
9	Propane	C_3H_8	44.09	0.1196	8.365	1.5617	2524	2342	21,669	19,937	5.0	18.82	23.82	3.0	4.0	18.82	3.63	12.07	15.70	2.99	1.63	12.07
10	n-Butane	C_4H_{10}	58.12	0.1582	6.321	2.0665	3271	3018	21,321	19,678	6.5	24.47	30.97	4.0	5.0	24.47	3.58	11.91	15.49	3.03	1.55	11.91
11	Isobutane	C_4H_{10}	58.12	0.1582	6.321	2.0665	3261	3009	21,271	19,628	6.5	24.47	30.97	4.0	5.0	24.47	3.58	11.91	15.49	3.03	1.55	11.91
12	n-Pentane	C_5H_{12}	72.15	0.1904	5.252	2.4872	4020	3717	21,095	19,507	8.0	30.11	38.11	5.0	6.0	30.11	3.55	11.81	15.35	3.05	1.50	11.81
13	Isopentane	C_5H_{12}	72.15	0.1904	5.252	2.4872	4011	3708	21,047	19,459	8.0	30.11	38.11	5.0	6.0	30.11	3.55	11.81	15.35	3.05	1.50	11.81
14	Neopentane	C_5H_{12}	72.15	0.1904	5.252	2.4872	3994	3692	20,978	19,390	8.0	30.11	38.11	5.0	6.0	30.11	3.55	11.81	15.35	3.05	1.50	11.81
15	n-Hexane	C_6H_{14}	86.17	0.2274	4.398	2.9704	4768	4415	20,966	19,415	9.5	35.76	45.26	6.0	7.0	35.76	3.53	11.74	15.27	3.06	1.46	11.74
	Olefin series																					
16	Ethylene	C_2H_4	28.05	0.0742	13.475	0.9740	1604	1503	21,636	20.275	3.0	11.29	14.29	2.0	2.0	11.29	3.42	11.39	14.81	3.14	1.29	11.39
17	Propylene	C_3H_6	42.08	0.1110	9.007	1.4504	2340	2188	21,048	19,687	4.5	16.94	21.44	3.0	3.0	16.94	3.42	11.39	14.81	3.14	1.29	11.39
18	N-Butene	C_4H_8	56.10	0.1480	6.756	1.9336	3084	2885	20,854	19,493	6.0	22.59	28.59	4.0	4.0	22.59	3.42	11.39	14.81	3.14	1.29	11.39
19	Isobutene	C_4H_8	56.10	0.1480	6.756	1.9336	3069	2868	20,737	19,376	6.0	22.59	28.59	4.0	4.0	22.59	3.42	11.39	14.81	3.14	1.29	11.39
20	n-Pentene	C_5H_{10}	70.13	0.1852	5.400	2.4190	3837	3585	20,720	19,359	7.5	28.23	35.73	5.0	5.0	28.23	3.42	11.39	14.81	3.14	1.29	11.39
	Aromatic series																					
21	Benzene	C_6H_6	78.11	0.2060	4.852	2.6920	3752	3601	18,184	17,451	7.5	28.23	35.73	6.0	3.0	28.23	3.07	10.22	13.30	3.38	0.69	10.22
22	Toluene	C_7H_8	92.13	0.2431	4.113	3.1760	4486	4285	18,501	17,672	9.0	33.88	42.88	7.0	4.0	33.88	3.13	10.40	13.53	3.34	0.78	10.40
23	Xylene	C_8H_{10}	106.16	0.2803	3.567	3.6618	5230	4980	18,650	17,760	10.5	39.52	50.02	8.0	5.0	39.52	3.17	10.53	13.70	3.32	0.85	10.53
	Miscellaneous gases																					
24	Acetylene	C_2H_2	26.04	0.0697	14.344	0.9107	1477	1426	21,502	20,769	2.5	9.41	11.91	2.0	1.0	9.41	3.07	10.22	13.30	3.38	0.69	10.22
25	Napthalene	$C_{10}H_8$	128.16	0.3384	2.955	4.4208	5854	5654	17,303	16,708	12.0	45.17	57.17	10.0	4.0	45.17	3.00	9.97	12.96	3.43	0.56	9.97
26	Methyl alcohol	CH_3OH	32.04	0.0846	11.820	1.1052	868	767	10,258	9,066	1.5	5.65	7.15	1.0	2.0	5.65	1.50	4.98	6.48	1.37	1.13	4.98
27	Ethyl alcohol	C_2H_5OH	46.07	0.1216	8.221	1.5890	1600	1449	13,161	11,917	3.0	11.29	14.29	2.0	3.0	11.29	2.08	6.93	9.02	1.92	1.17	6.93
28	Ammonia	NH_3	17.03	0.0456	21.914	0.5961	441	364	9,667	7,985	0.75	2.82	3.57	—	1.5	3.32	1.41	4.69	6.10	—	1.59	5.51
29	Sulfur*	S	32.06	—	—	—	—	—	3,980	3,980	1.0	3.76	4.76	SO_2 1.0	—	3.76	1.00	3.29	4.29	SO_2 2.00		3.29
30	Hydrogen sulfide	H_2S	34.08	0.0911	10.979	1.1898	646	595	7,097	6,537	1.5	5.65	7.15	1.0	1.0	5.65	1.41	4.69	6.10	1.88	0.53	4.69
31	Sulfur dioxide	SO_2	64.06	0.1733	5.770	2.2640	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
32	Water vapor	H_2O	18.02	0.0476	21.017	0.6215	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
33	Air	—	—	0.0766	13.063	1.0000	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

*Carbon and sulfur are considered as gases for molal calculations only.

All gas volumes corrected to 60° F and 30 in. Hg dry.

(From *Steam, Its Generation and Use*, © Babcock and Wilcox)

Theeoretical Air Example

Fuel – natural gas: 85% methane, 15% ethane

Btu/scf = $(0.85 \times 1012) + (0.15 \times 1773) = 1126.15$

Cu ft air required/cu ft fuel = $(0.85 \times 9.53) + (0.15 \times 16.68) = 10.607$

Cu ft air required/10000 Btu = $10.607 \times (10000/1126.15) = 94.277$

Lbs air required/10000 Btu = $94.277/13.063 = 7.217$

Btu/lb. air = $10000/7.217 = 1386$ Btu

Figure 5-14 Theoretical Air Calculations by Using Table of Combustion Constants

that is useful in simplifying such calculations. Figure 5-14 demonstrates using the table of combustion constants for a gaseous fuel of 85 percent methane and 15 percent ethane.

Note that the amount of combustion air required to produce 10,000 Btu is nearly the same for coal and natural gas. If the reciprocals are taken, the result is Btu/lb of air. Table 5-9 shows that for coal, oil, or gas the Btu/lb of air is approximately the same even though the Btu/lb of the fuels is radically different. The difference between the Btu/lb of air on a "net" basis for these fuels is smaller than that shown in the table. The fact that combustion air requirements can be closely approximated, based on the heat requirement, is an important concept used in the application of combustion control logic.

5-9 The Requirement of Excess Combustion Air

In actual practice gas-, oil-, coal-burning, and other systems do not do a perfect job of mixing the fuel and air even under the best achievable conditions of turbulence. Additionally, complete mixing may take too much time—so that the gases pass to a lower temperature area not hot enough to complete the combustion—before the process is completed.

If only the amount of theoretical air were furnished, some fuel would not burn, the combustion would be incomplete, and the heat in the unburned fuel would be lost. To assure complete combustion, additional combustion air is furnished so that every molecule of the fuel can easily find the proper number of oxygen molecules to complete the combustion.

Table 5-9
Fuel and Air Heating Value Comparison

Fuel	Btu/lb Fuel	Btu/lb Air
Bituminous Coal	12,975	1,332
Subbituminous Coal	9,901	1,323
#6 Oil	18,560	1,351
#2 Oil	19,410	1,376
Natural Gas	23,170	1,393

This additional amount of combustion air that is furnished to complete the combustion process is called excess air. Excess air plus theoretical air is called total air. Having this necessary excess air means that some of the oxygen will not be used and will leave the boiler in the flue gases, as shown in Figure 3-4, which describes the fuel and air chemicals' mass balance. The oxygen portion of the flue gas can be used to determine the percentage of excess air.

If the percentage of excess air is increased, flame temperature is reduced and the boiler heat transfer rate is reduced. The usual effect of this change is the increase in the flue gas temperature, as shown in Figure 5-15.

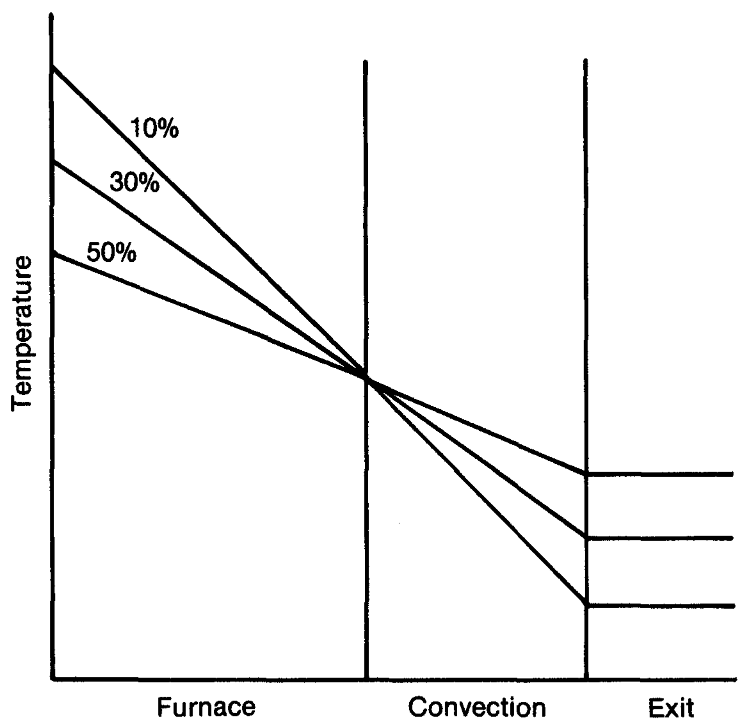
If oxygen in the flue gas is known or can be measured and no table or curve is available, the following empirical formula provides a close approximation of the percentage of excess air. This formula is based on "dry basis" percentage oxygen. The excess air calculation based on "wet basis" percentage oxygen is a complex formula based on the "wet basis" percentage oxygen and the complete fuel analysis by weight basis percentages of its constituents.

$$\text{Excess air (\%)} = K\left(\frac{21}{21 - \%\ \text{oxygen}} - 1\right) \times 100$$

where K = 0.9 for gas

0.94 for oil

0.97 for coal



- Reduced excess air improves heat transfer.
- Reduced excess air reduces mass of flue gases.

Figure 5-15 Effect of Excess Air on Temperatures

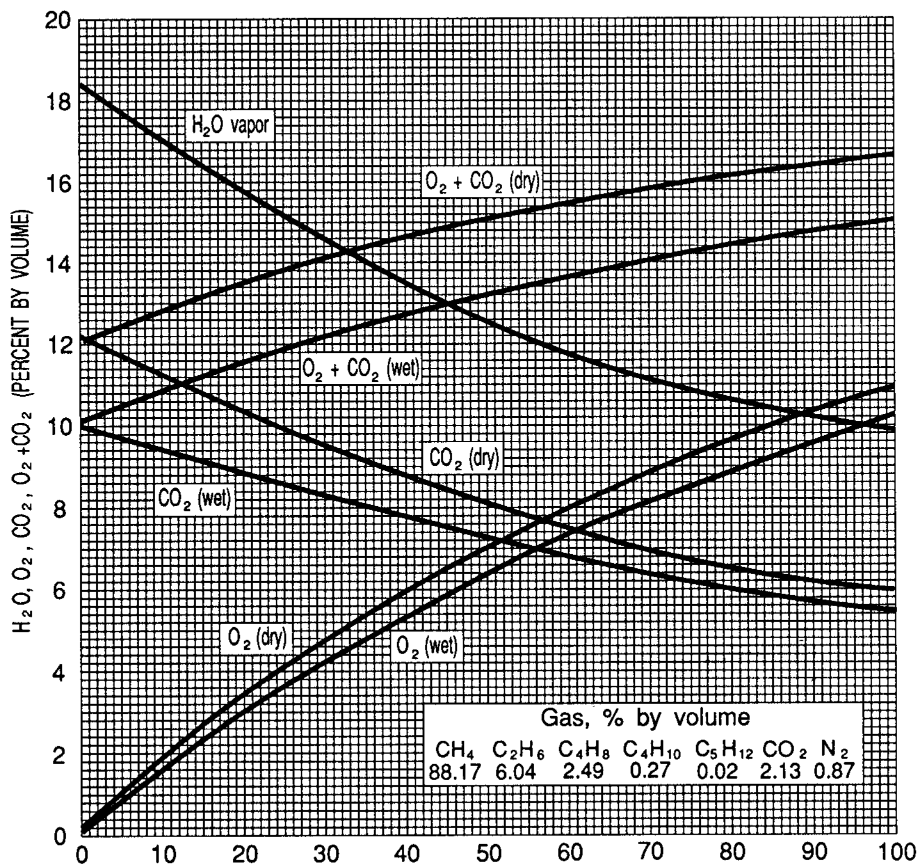
Measurements of either percentage of carbon dioxide or the percentage of oxygen in the flue gas or both are used to determine percentage of excess air, but the percentage of oxygen is preferred for the following reasons:

(1) Oxygen is part of the air—if oxygen is zero, then excess air is zero. The presence of oxygen always indicates that some percentage of excess air is present.

(2) The percent of carbon dioxide rises to a maximum at minimum excess air and then decreases as air is further reduced. It is thus possible, with the same percentage of carbon dioxide, to have two different percentages of total combustion air. For this reason, the per-

Table 5-10
Excess Air Required at Full Capacity

Fuel	% Oxygen in flue gas	% Excess air, minimum
Natural gas	1.5 to 3	7-15
Fuel oil	0.6 to 3	3-15
Coal	4.0 to 6.5	25-40



LIVE GRAPH
Click here to view

Figure 5-16 Excess Air as a Function of Flue Gas Composition for Natural Gas Fuel (Heating value is approximately 22,810 Btu/lb)

centage of carbon dioxide cannot be used alone as a flue gas analysis input to a combustion control system.

(3) To determine excess air with the same precision, greater precision of measurement is required for the percentage of carbon dioxide method than for the percentage of oxygen method.

(4) The relationship between the percentage of oxygen and the percentage of excess air changes little as fuel analysis or type of fuel changes, while the percentage of carbon dioxide-to-excess air relationship varies considerably as the percentage of carbon-to-hydrogen ratio of the fuel changes.

The heat loss in the flue gases essentially depends upon the difference between the temperature of the flue gases and that of the incoming combustion air, the amount of excess air, and the fuel analysis. There is an optimum amount of excess air because less air will mean unburned fuel from incomplete combustion, and more air will mean complete combustion but more heat loss in the flue gas due to the greater mass of the flue gases.

The amount of excess air required depends upon the type of fuel, burner design, fuel characteristics and preparation, furnace design, capacity as a percent of maximum, and other factors. The amount for any installation should be determined by testing that particular unit. An approximate amount of excess air required for full capacity is shown in Table 5-10.

The charts in Figures 5-16, 5-17, and 5-18 show the relationship between the flue gas analysis by volume and the percentage of excess air for natural gas, fuel oil, and coal. While

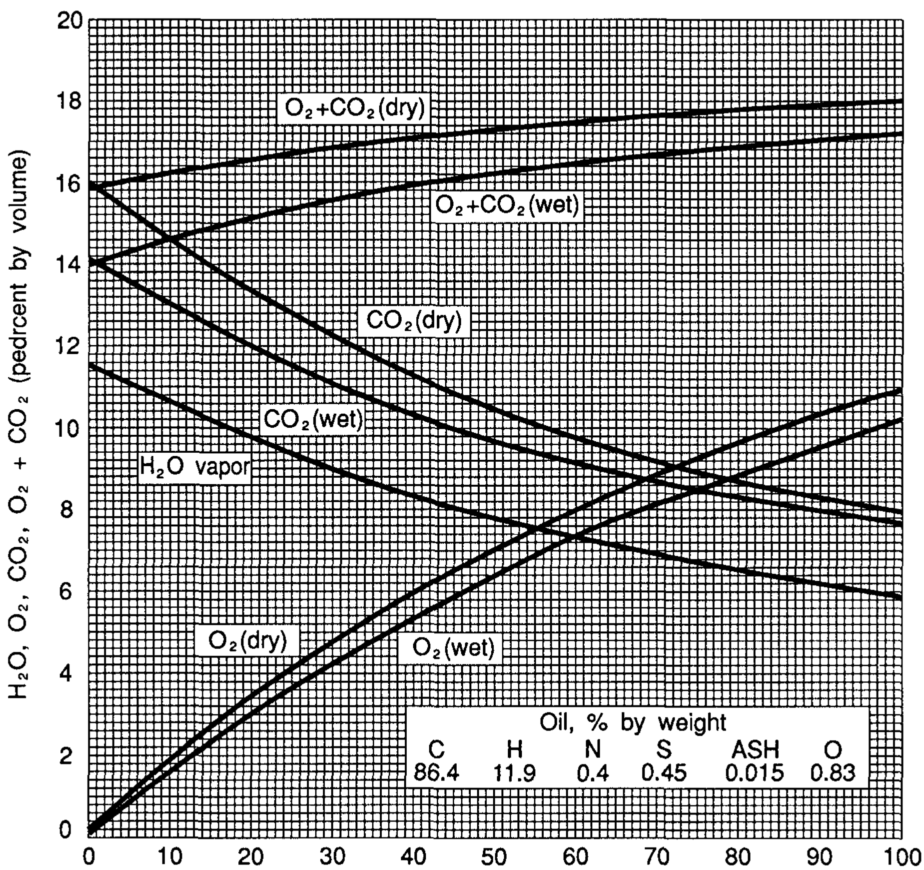


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Figure 5-17 Excess Air as a Function of Flue Gas Composition for Oil Fuels (Heating value is approximately 18,700 Btu/lb)

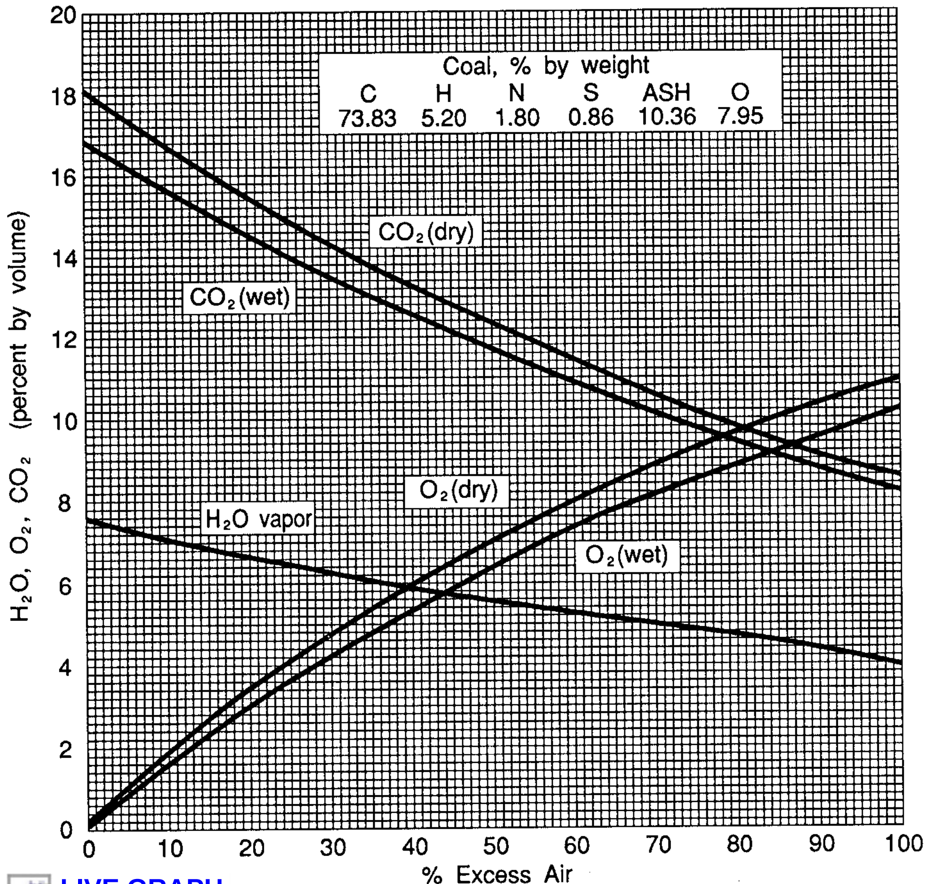

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Figure 5-18 Excess Air as a Function of Flue Gas Composition for Coal Fuels (Heating value is approximately 13,320 Btu/lb)

these curves are for fuels with specific fuel analyses, the curves for % oxygen vs. excess air are quite similar, while the % carbon dioxide vs. excess air curves are quite different for the different fuels.

These curves also show the difference between the flue gas analysis depending upon the presence or removal of the water vapor that is formed by the combustion process. This difference is important to note for two reasons. Since approximately 1970, flue gas analyzers for % oxygen using the zirconium oxide fuel cell principle have been marketed. This type of % oxygen analyzer, which analyzes the flue gas on the ''wet'' basis, is now the standard method for permanently installed flue gas analysis equipment. On the other hand, % oxygen vs. excess air formulas, including the one in the text above, are based on the ''dry'' basis, which was universally used until the early 1970s.

In addition, these newer zirconium oxide analyzers normally measure the % oxygen on a ''net'' basis. If combustible gases such as CO are present, the high temperature and catalytic action of the measuring cell complete the combustion by subtracting a portion or all of the oxygen passing the analysis cell. It is thus not necessary to subtract CO_2 from the % oxygen before it is used in the older formulas. Since these formulas are on the dry basis, however, it is necessary to convert the wet basis analyzer readings to dry basis before using them in the older equations for combustion calculations.

In analyzing flue gases to determine % excess air, it is useful to have a check on the

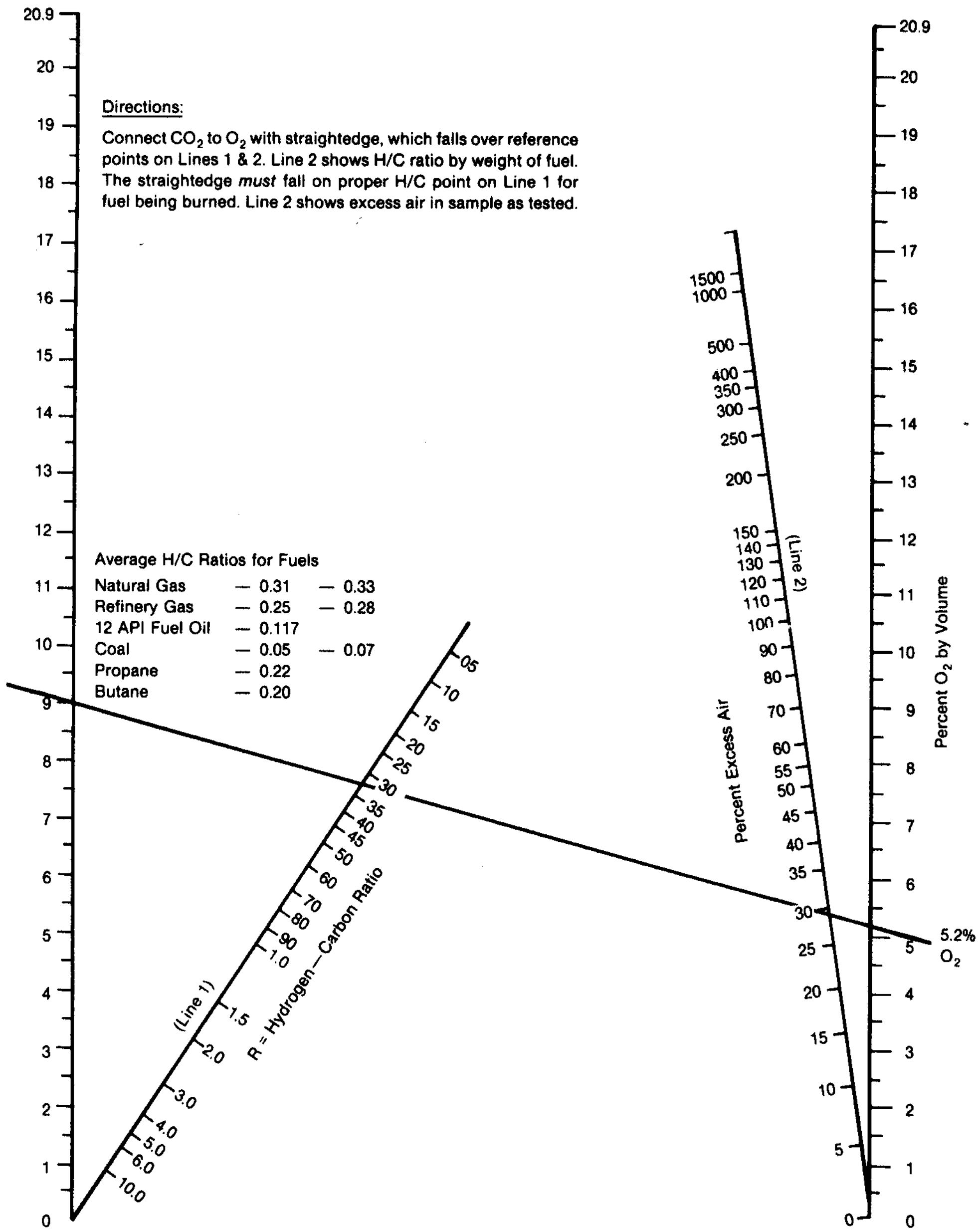


Figure 5-19 Properties of the Products of Combustion

accuracy of the analysis. Figure 5-19, which is based on dry basis analysis, can be used for this purpose. By drawing a straight line, as shown, between the % oxygen and the % carbon dioxide values, there is an intersection with the hydrogen-carbon ratio line. When using a particular fuel of a certain H/C ratio, the intersection should always be at the same point on the line. If it is not, the fuel has changed or the results of the analysis are incorrect. If the fuel analysis is known, the measurement of % oxygen can be used to determine the correct % carbon dioxide, or the % carbon dioxide can be used to determine the correct % oxygen. If boiler tests are being made and the fuel analysis is constant, any data (within limits of measurement accuracy) that doesn't measure up to this kind of examination should be thrown out.

Section 6
Efficiency Calculations

Two different methods are used for calculating the efficiency of boilers. Both of these methods are useful, but it is generally agreed that the heat loss method provides more accurate results. The input/output or direct method is more easily understood and for that reason is often preferred.

6-1 Input/Output or Direct Method

The basic calculation for the input/output method is shown in Figure 6-1. This formula is based on a knowledge of the flows of steam from the boiler, feedwater to the boiler, blowdown from the boiler, and fuel to the boiler. In addition, the unit heating value of each of these flows must be known.

All the flows above can be determined from flowmeters. If steam flow alone or feedwater flow alone is known, blowdown flow can often be determined from the chemical relationships between feedwater entering the boiler and the water in the boiler.

If the pressure, temperature, or % moisture of the steam is known, the heat content of the steam, feedwater, or blowdown can be determined from the ''Thermodynamic Properties of Steam and Water'' tables or a suitable computer program. The heat content of the fuel must be determined by the use of a fuel calorimeter. For gaseous fuels the calorimeter may be either a laboratory device or an on-line recording and/or indicating instrument. For liquid or solid fuels such as oil or coal, a laboratory test is necessary.

In the formula shown in Figure 6-1, the output of the boiler is credited with all the heat that is added to the incoming feedwater. This heat added includes the non-useful heat added to the blowdown flow in increasing its temperature from feedwater temperature to the saturation temperature of the boiler water.

At least three flows are used in this calculation method. The best normal operation flowmeter accuracy that can be expected for an individual meter is plus or minus 1-2%. Users often estimate that actual accuracy is in the range of 3-5%.

The potential for erroneous results because of flowmeter inaccuracy is the primary weakness of this calculation method. In addition to flowmeter inaccuracy, more inaccuracies can be introduced in the determination of the heat content of the fuel.

It is also normal that a steam, water, or fuel flowmeter may not be operating under its design condition of pressure, temperature, specific gravity, etc. Unless particular care is taken in correcting the flowmeter results to design conditions, further inaccuracies can easily be introduced. With gas flowmeters, the basic design conditions of the meter may not match the conditions of the calorimeter. Particular care must be taken to see that the heating value and fuel flow are on the same basis.

Figure 6-2 is a worksheet for determination of boiler efficiency by the input/output method. Note that completion of this worksheet forces correction to design conditions for the flowmeters and calorimetric determination. The three formulas shown are based on different combinations of flowmeter results and the particular flowmeters that are available.

The usefulness of this calculation method is derived from its ease of understanding and its ease of calculation. Though precise results may not be attainable, inaccuracies will probably repeat closely from day to day. This method is therefore useful in tracking daily performance.

- EFFICIENCY = $\dfrac{\text{Heat Added to Incoming Feedwater}}{\text{Heat Input (Fuel)}}$

  Heat Added to Steam-Water circuit:

  (Steam Flow) (Steam enthalpy–Feedwater enthalpy) +
  (Blowdown Flow) (Blowdown enthalpy–Feedwater enthalpy)

  Heat Input:

  (Fuel Flow) (Fuel Higher Heating Value)

- LIMITATIONS

  Accuracy of flow meters

  Accuracy of fuel weight (solid fuel)

  Heat content of fuel (measurement accuracy)

Figure 6-1 Input-Output

A.	Steam Flow, lbs	______	Design ______
B.	Pressure, psig	______	
C.	Temperature, °F	______	
D.	Corrected Steam Flow, lbs	______	
E.	Enthalpy of Steam, Btu/lb	______	
F.	Feedwater Flow, lbs	______	Design ______
G.	Feedwater Temperature, °F	______	
H.	Corrected Feedwater Flow, lbs	______	
I.	Enthalpy of Feedwater, Btu/lb	______	
J.	Blowdown Flow, lbs*	______	Design ______
K.	Blowdown Temperature, °F	______	
L.	Corrected Blowdown Flow, lbs	______	
M.	Enthalpy of Blowdown, Btu/lb	______	
N.	Gas Flow, scf at std.	______	Design ______
O.	Gas Pressure, psig	______	
P.	Gas Temperature, °F	______	
Q.	Corrected Gas Flow, scf at std.	______	
R.	Heat Content of Gas at std.	______	

$$\%\ \text{Efficiency} = \frac{(D)(E) + (L)(M) - (H)(I)}{(Q)(R)} \times 100 \text{ or}$$

$$\%\ \text{Efficiency} = \frac{(D)(E-I) + (L)(M-I)}{(Q)(R)} \times 100 \text{ or}$$

$$\%\ \text{Efficiency} = \frac{(H-L)(E-I) + (L)(M-I)}{(Q)(R)}$$

* Blowdown is measured, (H)–(D), or computed from chemical analysis.

Figure 6-2 Boiler Efficiency, Input-Output

6-2 Heat Loss or Indirect Method

The heat loss method is used to calculate individual losses, totalize them, and determine boiler efficiency by subtracting the total losses from 100%. The individual losses are identified in Figure 6-3.

Figure 6-3 also identifies the method of determining the losses. In this method, a particular fuel unit, 1 mole or 100 moles, 1 lb or 100 lbs of fuel, is used. For the fuel unit the mass of the flue gases is determined. The sensible heat losses are determined by multiplying the flue gas mass flows by the mean specific heat (M_{cp}) between the flue gas temperature and the combustion air inlet temperature. The result is then multiplied by the difference in temperature between the flue gases and the combustion air.

Latent heat losses are determined by multiplying the mass flow of the water that is vaporized (fuel moisture plus moisture from the combustion of hydrogen) by an average figure of 1040 Btu/lb. Each loss is calculated in terms of heat units (Btu, for example), the heat units totalized, and the percentage of the heating value of the total fuel unit determined.

For calculation of the heat losses involved, the two accepted methods are the "mole" method, and the "weight" method. Because of the greater involvement of the combustion chemistry in the "mole" method, many engineers who are not trained in chemistry prefer the "weight" method. Almost identical results can be obtained from either method if the calculations are carefully made.

The "weight" method is used in the "Indirect Method" of the ASME Power Test Code, which is generally used for contractual purposes. Its use does not require the knowledge of combustion chemistry and can be accomplished with a "cookbook" approach using the stan-

Calculate Various Losses and Subtract from 100%

- Sensible heat loss in dry flue gas:
  Mass (dry gas) $\times \Delta t \times M_{cp}$*
- Sensible heat loss from H_2O in combustion air:
  Mass (air moisture) $\times M_{cp} \times \Delta t$
- Sensible heat loss from H_2O in fuel:
  Mass (H_2O in fuel) $\times M_{cp} \times \Delta t$
- Latent heat loss from H_2O in the fuel:
  Mass (H_2O in fuel) $\times$ 1040**
- Latent heat loss from H_2O formed by combustion of H_2:
  Mass (water vapor) $\times$ 1040
- Heat loss from unburned carbon in the refuse:
  Mass (per fuel unit) $\times$ 14,100 Btu/lb
- Heat loss from unburned combustible gas in flue gases:
  Mass (per fuel unit) $\times$ Btu content specific gases

Plus

- Radiation and unaccounted for losses

* M_{cp} = *mean specific heat*
** Standard value of Btu/lb

Figure 6-3 Heat Loss Method

SUMMARY SHEET

ASME TEST FORM
FOR ABBREVIATED EFFICIENCY TEST

PTC 4.1-a (1964)

	TEST NO.	BOILER NO.	DATE
OWNER OF PLANT	LOCATION		
TEST CONDUCTED BY	OBJECTIVE OF TEST		DURATION
BOILER, MAKE & TYPE		RATED CAPACITY	
STOKER, TYPE & SIZE			
PULVERIZER, TYPE & SIZE		BURNER, TYPE & SIZE	
FUEL USED MINE	COUNTY	STATE	SIZE AS FIRED

PRESSURES & TEMPERATURES

1	STEAM PRESSURE IN BOILER DRUM	psia	
2	STEAM PRESSURE AT S. H. OUTLET	psia	
3	STEAM PRESSURE AT R. H. INLET	psia	
4	STEAM PRESSURE AT R. H. OUTLET	psia	
5	STEAM TEMPERATURE AT S. H. OUTLET	F	
6	STEAM TEMPERATURE AT R.H. INLET	F	
7	STEAM TEMPERATURE AT R.H. OUTLET	F	
8	WATER TEMP. ENTERING (ECON.) (BOILER)	F	
9	STEAM QUALITY % MOISTURE OR P. P. M.		
10	AIR TEMP. AROUND BOILER (AMBIENT)	F	
11	TEMP. AIR FOR COMBUSTION (This is Reference Temperature) †	F	
12	TEMPERATURE OF FUEL	F	
13	GAS TEMP. LEAVING (Boiler) (Econ.) (Air Htr.)	F	
14	GAS TEMP. ENTERING AH (If conditions to be corrected to guarantee)	F	

UNIT QUANTITIES

15	ENTHALPY OF SAT. LIQUID (TOTAL HEAT)	Btu/lb	
16	ENTHALPY OF (SATURATED) (SUPERHEATED) STM.	Btu/lb	
17	ENTHALPY OF SAT. FEED TO (BOILER) (ECON.)	Btu/lb	
18	ENTHALPY OF REHEATED STEAM R.H. INLET	Btu/lb	
19	ENTHALPY OF REHEATED STEAM R. H. OUTLET	Btu/lb	
20	HEAT ABS/LB OF STEAM (ITEM 16 – ITEM 17)	Btu/lb	
21	HEAT ABS/LB R.H. STEAM (ITEM 19 – ITEM 18)	Btu/lb	
22	DRY REFUSE (ASH PIT + FLY ASH) PER LB AS FIRED FUEL	lb/lb	
23	Btu PER LB IN REFUSE (WEIGHTED AVERAGE)	Btu/lb	
24	CARBON BURNED PER LB AS FIRED FUEL	lb/lb	
25	DRY GAS PER LB AS FIRED FUEL BURNED	lb/lb	

HOURLY QUANTITIES

26	ACTUAL WATER EVAPORATED	lb/hr	
27	REHEAT STEAM FLOW	lb/hr	
28	RATE OF FUEL FIRING (AS FIRED wt)	lb/hr	
29	TOTAL HEAT INPUT $\frac{\text{(Item 28 X Item 41)}}{1000}$	kB/hr	
30	HEAT OUTPUT IN BLOW-DOWN WATER	kB/hr	
31	TOTAL HEAT OUTPUT $\frac{\text{(Item 26}\times\text{Item 20)+(Item 27}\times\text{Item 21)+Item 30}}{1000}$	kB/hr	

FLUE GAS ANAL. (BOILER) (ECON) (AIR HTR) OUTLET

32	CO_2	% VOL	
33	O_2	% VOL	
34	CO	% VOL	
35	N_2 (BY DIFFERENCE)	% VOL	
36	EXCESS AIR	%	

FUEL DATA

	COAL AS FIRED PROX. ANALYSIS	% wt
37	MOISTURE	
38	VOL MATTER	
39	FIXED CARBON	
40	ASH	
	TOTAL	
41	Btu per lb AS FIRED	
42	ASH SOFT TEMP.* ASTM METHOD	

	COAL OR OIL AS FIRED ULTIMATE ANALYSIS	
43	CARBON	
44	HYDROGEN	
45	OXYGEN	
46	NITROGEN	
47	SULPHUR	
40	ASH	
37	MOISTURE	
	TOTAL	

	COAL PULVERIZATION	
48	GRINDABILITY INDEX*	
49	FINENESS % THRU 50 M*	
50	FINENESS % THRU 200 M*	
64	INPUT-OUTPUT EFFICIENCY OF UNIT %	$\frac{\text{ITEM 31} \times 100}{\text{ITEM 29}}$

	OIL	
51	FLASH POINT F*	
52	Sp. Gravity Deg. API*	
53	VISCOSITY AT SSU* BURNER SSF	
44	TOTAL HYDROGEN % wt	
41	Btu per lb	

	GAS	% VOL
54	CO	
55	CH_4 METHANE	
56	C_2H_2 ACETYLENE	
57	C_2H_4 ETHYLENE	
58	C_2H_6 ETHANE	
59	H_2S	
60	CO_2	
61	H_2 HYDROGEN	
	TOTAL	
	TOTAL HYDROGEN % wt	
62	DENSITY 68 F ATM. PRESS.	
63	Btu PER CU FT	
41	Btu PER LB	

	HEAT LOSS EFFICIENCY	Btu/lb A. F. FUEL	% of A. F. FUEL
65	HEAT LOSS DUE TO DRY GAS		
66	HEAT LOSS DUE TO MOISTURE IN FUEL		
67	HEAT LOSS DUE TO H_2O FROM COMB. OF H_2		
68	HEAT LOSS DUE TO COMBUST. IN REFUSE		
69	HEAT LOSS DUE TO RADIATION		
70	UNMEASURED LOSSES		
71	TOTAL		
72	EFFICIENCY = (100 – Item 71)		

Figure 6-4 Summary Sheet, ASME Test Form for Abbreviated Efficiency Test

ASME TEST FORM

CALCULATION SHEET — **FOR ABBREVIATED EFFICIENCY TEST** — PTC 4.1-b (1964)

	OWNER OF PLANT — TEST NO. — BOILER NO. — DATE			
30	HEAT OUTPUT IN BOILER BLOW-DOWN WATER = LB OF WATER BLOW-DOWN PER HR × $\left[\dfrac{\text{ITEM 15} - \text{ITEM 17}}{1000}\right]$ = ... kB/hr			
24	*If impractical to weigh refuse, this item can be estimated as follows* DRY REFUSE PER LB OF AS FIRED FUEL = $\dfrac{\text{\% ASH IN AS FIRED COAL}}{100 - \text{\% COMB. IN REFUSE SAMPLE}}$ CARBON BURNED PER LB AS FIRED FUEL = $\dfrac{\text{ITEM 43}}{100} - \left[\dfrac{\text{ITEM 22} \times \text{ITEM 23}}{14{,}500}\right]$ = ... NOTE: IF FLUE DUST & ASH PIT REFUSE DIFFER MATERIALLY IN COMBUSTIBLE CONTENT, THEY SHOULD BE ESTIMATED SEPARATELY. SEE SECTION 7, COMPUTATIONS.			
25	DRY GAS PER LB AS FIRED FUEL BURNED = $\dfrac{11CO_2 + 8O_2 + 7(N_2 + CO)}{3(CO_2 + CO)}$ × (LB CARBON BURNED PER LB AS FIRED FUEL + $\frac{3}{8}$ S) = $\dfrac{11 \times \text{ITEM 32} + 8 \times \text{ITEM 33} + 7(\text{ITEM 35} + \text{ITEM 34})}{3 \times (\text{ITEM 32} + \text{ITEM 34})} \times \left[\text{ITEM 24} + \dfrac{\text{ITEM 47}}{2.67}\right]$ = ...			
36	EXCESS AIR† = $100 \times \dfrac{O_2 - \frac{CO}{2}}{.2682N_2 - (O_2 - \frac{CO}{2})} = 100 \times \dfrac{\text{ITEM 33} - \frac{\text{ITEM 34}}{2}}{.2682(\text{ITEM 35}) - (\text{ITEM 33} - \frac{\text{ITEM 34}}{2})}$ = ...			
	HEAT LOSS EFFICIENCY	**Btu/lb AS FIRED FUEL**	**$\frac{\text{LOSS}}{\text{HHV}}$ × 100 =**	**LOSS %**
65	HEAT LOSS DUE TO DRY GAS = LB DRY GAS PER LB AS FIRED FUEL × C_p × $(t_{lvg} - t_{air})$ = ITEM 25 × 0.24 × (ITEM 13) − (ITEM 11) = ... Unit	......	$\frac{65}{41}$ × 100 =	......
66	HEAT LOSS DUE TO MOISTURE IN FUEL = LB H_2O PER LB AS FIRED FUEL × [(ENTHALPY OF VAPOR AT 1 PSIA & T GAS LVG) − (ENTHALPY OF LIQUID AT T AIR)] = $\frac{\text{ITEM 37}}{100}$ × [(ENTHALPY OF VAPOR AT 1 PSIA & T ITEM 13) − (ENTHALPY OF LIQUID AT T ITEM 11)] = ...	......	$\frac{66}{41}$ × 100 =	......
67	HEAT LOSS DUE TO H_2O FROM COMB. OF H_2 = $9H_2$ × [(ENTHALPY OF VAPOR AT 1 PSIA & T GAS LVG) − (ENTHALPY OF LIQUID AT T AIR)] = 9 × $\frac{\text{ITEM 44}}{100}$ × [(ENTHALPY OF VAPOR AT 1 PSIA & T ITEM 13) − (ENTHALPY OF LIQUID AT T ITEM 11)] = ...	......	$\frac{67}{41}$ × 100 =	......
68	HEAT LOSS DUE TO COMBUSTIBLE IN REFUSE = ITEM 22 × ITEM 23 = ...	......	$\frac{68}{41}$ × 100 =	......
69	HEAT LOSS DUE TO RADIATION* = $\dfrac{\text{TOTAL BTU RADIATION LOSS PER HR}}{\text{LB AS FIRED FUEL — ITEM 28}}$	......	$\frac{69}{41}$ × 100 =	......
70	UNMEASURED LOSSES **	......	$\frac{70}{41}$ × 100 =	......
71	TOTAL	......	......	......
72	EFFICIENCY = (100 − ITEM 71)	......	......	......

Figure 6-5 Calculation Sheet, ASME Test Form for Abbreviated Efficiency Test

Start/Oil Firing

Component	Lbs/ Fuel Unit	Mol Wt. Div.	Moles Fuel Unit	O_2 Mult.	O_2 Moles	CO_2 SO_2	O_2	N_2	H_2O
C-CO_2	87.9*	12	7.32	1.0	7.32	7.32			
H_2	10.3*	2	5.15	0.5	2.58				5.15
S	1.2*	32	0.04	1.0	0.04	0.04			
O_2(Ded)	0.5*	32	0.02	1.0	(.02)				
N_2	0.1*	28	0.00					0.00	
CO_2									
H_2O		18							
	100.0 lbs		12.53		9.92				
Theor. O_2	- 9.92								
O_2 Excess	- 1.19 (12% of 9.92)						1.19		
Total O_2	- 11.11								
N_2 (3.76 × O_2)	- 41.77							41.77	
Dry Air	- 52.88								
H_2O in Air	- 0.536 (0.01015 Moles/Mole)								0.536
Air incl. H_2O	- 53.416					7.36	1.19	41.77	5.686

Dry Flue Gas Heat Loss –

	Mcp		Moles		(T-t)		
CO_2 –	*10.0	×	(7.32)	×	420	=	30,744
SO_2 –	*10.4	×	(0.04)	×	420	=	175
O_2 –	* 7.25	×	(1.9)	×	420	=	3,624
N_2 –	* 7.05	×	(41.77)	×	420	=	123,681
							158,224 Btu

Heat Loss from H_2O in Comb. Air –

* 8.2 × (0.536) × 420 = 1,846 Btu

Heat Loss from H_2O in Fuel

Moles

* 8.2 × (5.15) × 420 = 17,737 Btu

Latent Heat Loss – H_2O in Fuel + H_2O formed from H_2

Moles

1040 × 18 × 5.15 = 96,408 Btu

Total Losses	=	274,215 Btu
* Total Input 100 × 18,500 Btu/lb	=	1,850,000 Btu
% Heat Loss	–	14.82%
% Comb Eff. 100 – 14.82	–	85.18%

Note—Boiler Eff = Comb Eff. – Radiation

Figure 6-6 Worksheet for "Mole" Method of Boiler Efficiency Calculation

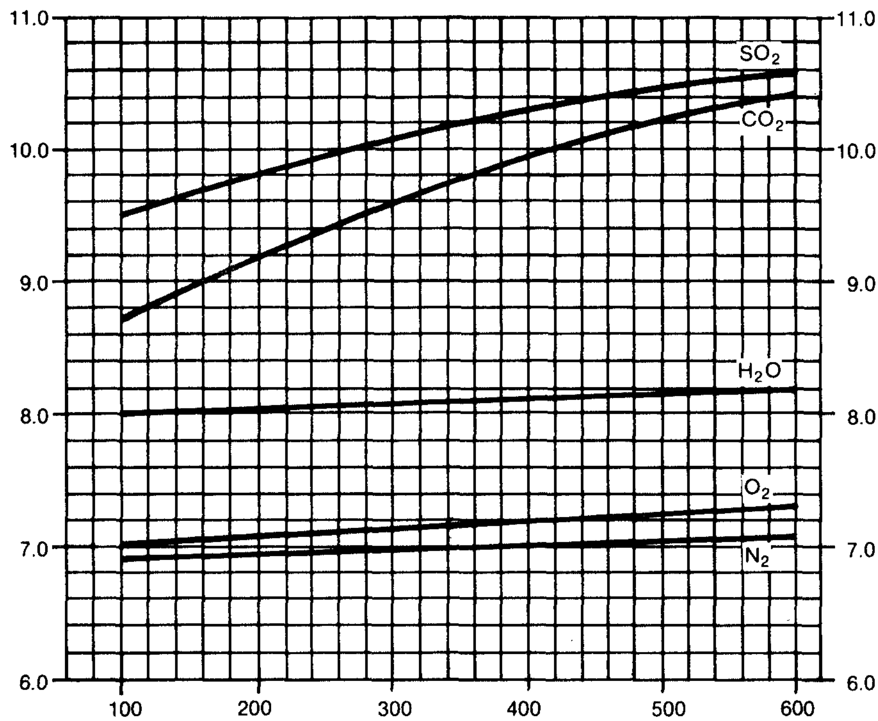
dardized forms that are shown in Figures 6-4 and 6-5. Note that in the formula in line no. 65 a standard mean specific heat of 0.24 for the dry flue gas is used.

With the "mole" method, the combustion chemistry formulas are used to determine the number of moles of each flue gas constituent, and the individual specific heat of that constituent for the particular flue gas and combustion air temperature difference is also used. For this reason, the "mole" method is slightly more precise than the standard ASME method shown. If a precisely calculated specific heat value is used instead of the standard 0.24 value, then the overall result of the computed losses is of similar precision. For precise results both methods require knowledge of the fuel analysis and the flue gas analysis.

A worksheet for using the "mole" method with a fuel unit of 100 lbs of fuel is shown in Figure 6-6. This worksheet has been completed for a specific analysis by weight of a particular fuel oil but can be used for any fuel for which the analysis by weight is known.

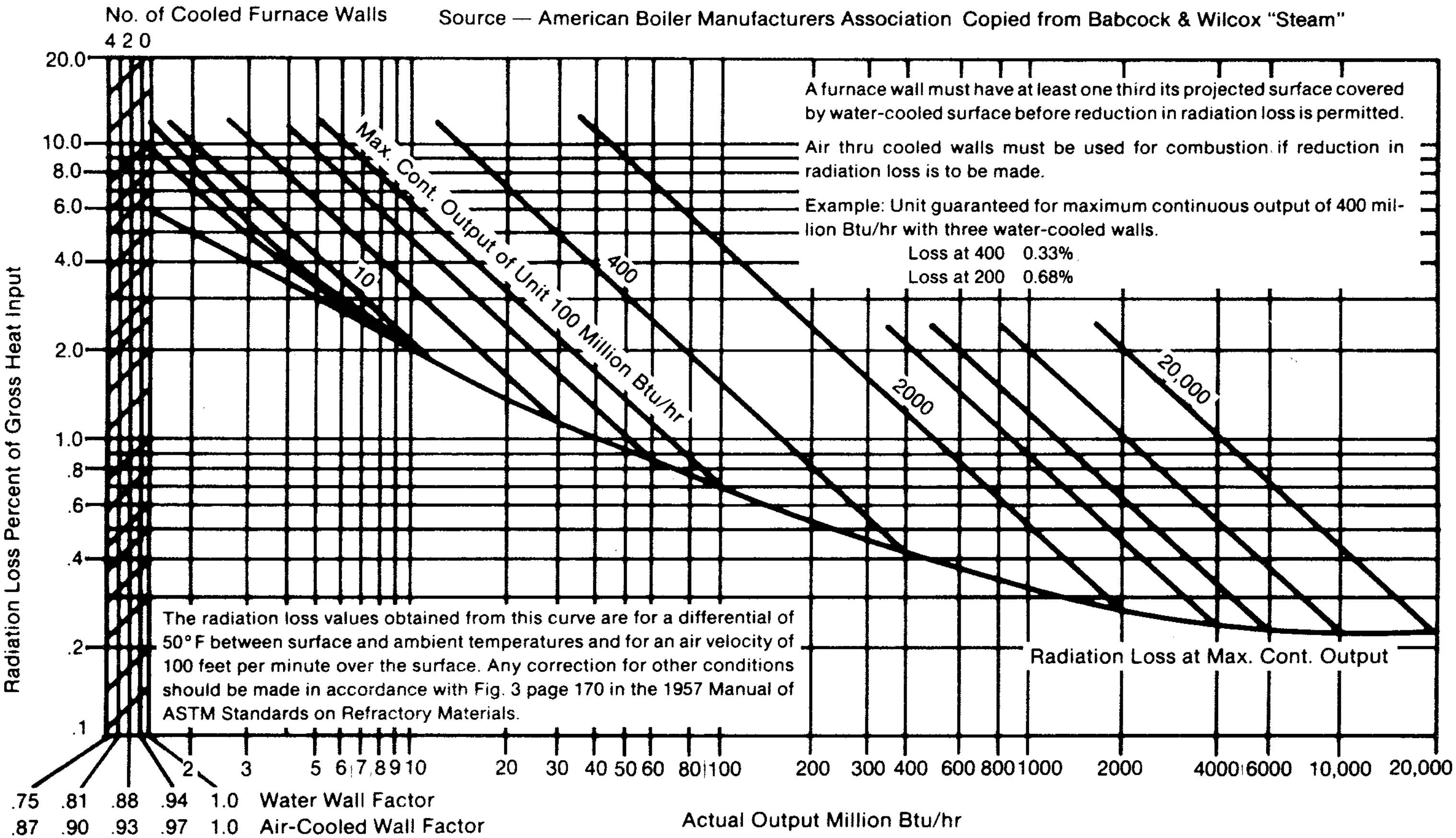
In column 1 the percentage by weight for the fuel components is entered. Column 2 shows the molecular weights of the components. Column 3 is column 1 divided by column 2. Column 4 is the relationship between the moles of oxygen and the moles of the constituent in the particular chemical formula involved. Column 5 is column 3 multiplied by column 4. Column 5 divided by column 4 yields the values in the four right-hand columns.

The excess air (12 percent in this example) is determined from flue gas analysis. The moles per mole of water vapor in the combustion air are determined from the relative humidity and temperature of the combustion air. The mean specific heat (M_{cp}) can be determined for each



T = Flue Gas Temp.
Chart Shows Mean Molal Specific Heat between
Temperature T and Air Temperature, t = 60°

Figure 6-7 Mean Molal Specific Heat (M_{cp})



- Radiation Essentially Constant Quantity for All Boiler Ratings
- Radiation % Doubles @ 50% Rating;
  5 Times @ 20%;
  10 Times @ 10%

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Figure 6-8 Radiation Loss

flue gas constituent from the curves in Figure 6-7. With the tabulation shown, the first 5 of the 8 losses listed in Figure 6-3 are determined.

Heat loss from carbon in the refuse is calculated from the weight of ash per fuel unit and the percent carbon in the refuse. This determines the weight of carbon per fuel unit, which is then multiplied by 14,100, a commonly used value for the heating value of carbon.

For determining the loss for unburned combustible gas per fuel unit, the particular unburned gas and its percentage by volume in the flue gases must be known. If it is CO, the percent by volume as analyzed in the flue gas multiplied by the total number of moles in column 3 provides the number of moles of CO per fuel unit. This value multiplied by 28 (molecular weight of CO) provides the weight (lbs of CO). This weight multiplied by 4347 (the gross and also net heating value of CO) gives the unburned gas loss.

For a boiler fired with solid fuel, an unaccounted for loss of 1.5 percent is commonly used, but it can be any number agreed on by the contracting parties. For a gaseous or liquid fuel boiler, the commonly used value is 1 percent. The radiation loss is taken from the standard ABMA (American Boiler Manufacturers Association) curve shown in the upper part of Figure 6-8. Note that the percent radiation loss increases as the boiler is operated at reduced load. An analysis of these percentage values demonstrates that the heat loss from radiation for a particular boiler is essentially constant regardless of the boiler rating at the time.

The heat loss method involves a number of measurements and a much more involved calculation than the input/output method. Because of the manner in which measurements are used in the calculations, measurement errors do not cascade as in the input/output calculation

Fuel		Bituminous Coal % by Wt.	No. 6 Fuel Oil % by Wt.	Natural Gas % by Vol.	Commercial Propane % by Vol.
Carbon	C	73.6	86.0	—	—
Hydrogen	H_2	5.3	11.0	—	—
Carbon Monoxide	CO	—	—	—	—
Methane	CH_4	—	—	82.9	—
Ethane	C_2H_6	—	—	14.9	2.2
Propane	C_3H_8	—	—	—	97.3
Pentane	C_4H_{10}	—	—	—	0.5
Illuminants	C_2H_4	—	—	—	—
Oxygen	O_2	10.0	1.0	—	—
Nitrogen	N_2	1.7	0.2	2.2	—
Carbon Dioxide	CO_2	—	—	—	—
Sulphur	S	0.8	0.8	—	—
Moisture	H_2O	0.6	1.0	—	—
Ash	—	8.0	—	—	—
		100.0	100.0	100.0	100.0
Dry Products of Perfect Combustion — cft of CO_2 and N_2		125.8	168.5	9.4	21.7
Air Requirement for Perfect Combustion — cft of Air		129.4	178.8	10.4	23.7
Ultimate CO_2 % by Volume		18.4	16.1	12.1	13.7
Heat Content Btu		13,640 per lb	18,873 per lb	1,107 per cft	2,576 per cft

Reference Temp. — 60°F, dry air basis.

Figure 6-9 Fuel Analysis for Combustion Efficiency Charts

CO_2		12.1	11.5	11.0	10.4	9.8	9.2	8.7	8.1	7.5	6.9	6.4	5.8
Excess Air		0	4.5	9.5	15.1	21.3	28.3	36.2	45.0	55.6	67.8	82.2	99.3
Oxygen		0	1	2	3	4	5	6	7	8	9	10	11
	°F												
	300	85.6	85.4	85.2	85.0	84.7	84.5	84.2	83.9	83.5	83.0	82.4	81.7
	350	84.6	84.3	84.1	83.8	83.5	83.2	82.8	82.4	81.9	81.3	80.6	79.8
	400	83.5	83.2	82.9	82.6	82.2	81.8	81.4	80.9	80.3	79.6	78.8	77.8
	450	82.5	82.1	81.8	81.4	81.0	80.5	80.0	79.4	78.7	78.9	77.0	75.9
	500	81.4	81.0	80.6	80.2	79.7	79.1	78.6	77.9	77.1	76.2	75.2	73.9
	550	80.3	79.9	79.4	79.0	78.4	77.8	77.2	76.4	75.5	74.5	73.4	71.9
	600	79.2	78.7	78.2	77.7	77.1	76.4	75.7	74.9	73.9	72.8	71.5	69.9
	650	78.1	77.6	77.1	76.5	75.8	75.1	74.3	73.4	72.3	71.1	69.7	67.9
	700	77.0	76.5	75.9	75.3	74.5	73.7	72.9	71.9	70.7	69.4	67.8	65.9
	750	75.9	75.4	74.7	74.1	73.2	72.4	71.5	70.4	69.1	67.7	66.0	63.9
	800	74.8	74.2	73.5	72.8	71.9	71.0	70.0	68.8	67.5	65.9	64.1	61.9
	850	73.7	73.1	72.3	71.6	70.6	69.7	68.6	67.3	65.9	64.2	62.3	59.9
	Loss per Percent Combustibles												
		2.8	3.0	3.2	3.4	3.7	4.0	4.3	4.6	5.0	5.5	6.1	6.8

Figure 6-10 Combustion Efficiency Chart—Gas

CO_2		16.1	15.3	14.5	13.8	13.0	12.2	11.5	10.7	9.9	9.2	8.4	7.7
Excess Air		0	4.7	9.9	15.7	22.2	29.5	37.7	47.1	58.0	70.7	85.7	103
Oxygen		0	1	2	3	4	5	6	7	8	9	10	11
	°F												
	300	89.7	89.5	89.2	89.0	88.8	88.5	88.1	87.8	87.3	86.7	86.1	85.4
	350	88.7	88.4	88.1	87.9	87.3	87.2	86.8	86.4	85.8	85.1	84.4	83.5
	400	87.6	87.3	87.0	86.7	86.3	85.9	85.4	84.9	84.2	83.4	82.6	81.5
	450	86.6	86.2	85.9	85.6	85.1	84.7	84.1	83.5	82.7	81.8	80.8	79.6
	500	85.5	85.2	84.8	84.4	83.8	83.3	82.7	82.0	81.1	80.1	79.0	77.6
	550	84.5	84.1	83.7	83.2	82.6	82.0	81.3	80.6	79.6	78.4	77.2	75.6
	600	83.4	83.0	82.5	82.0	81.3	80.7	79.9	79.1	78.0	76.7	75.4	73.6
	650	82.4	81.9	81.4	80.8	80.1	79.4	78.5	77.7	76.5	75.1	73.6	71.7
	700	81.3	80.8	80.2	79.6	78.8	78.1	77.1	76.2	74.9	73.4	71.8	69.4
	750	80.3	79.7	78.1	78.4	77.6	76.8	75.7	74.7	73.3	71.6	70.0	67.7
	800	79.2	78.6	77.9	77.2	76.3	75.4	74.3	73.2	71.7	70.0	68.1	65.7
	850	78.1	77.5	76.7	76.0	75.0	74.1	72.9	71.7	70.1	68.3	66.2	63.7
	Loss per Percent Combustibles												
		2.9	3.0	3.2	3.4	3.6	3.8	4.1	4.4	4.7	5.0	5.4	6.0

Figure 6-11 Combustion Efficiency Chart—No. 6 Oil

CO_2		18.4	17.5	16.6	15.7	14.9	14.0	13.1	12.2	11.4	10.5	9.6	8.7
Excess Air		0	4.9	10.2	16.2	22.9	30.4	38.8	48.5	59.5	72.9	88.4	107
Oxygen		0	1	2	3	4	5	6	7	8	9	10	11
	°F												
	300	91.5	91.3	91.1	90.8	90.5	90.2	89.9	89.5	89.0	88.5	87.8	87.1
	325	91.0	90.8	90.6	90.3	89.9	89.6	89.3	88.8	88.3	87.7	86.9	86.1
	350	90.5	90.3	90.0	89.7	89.3	89.0	88.6	88.1	87.5	86.8	86.0	85.1
	375	90.0	89.8	89.5	89.1	88.7	88.4	87.9	87.4	86.7	86.0	85.1	84.1
	400	89.5	89.2	88.9	88.5	88.1	87.7	87.2	86.6	85.9	85.1	84.2	83.1
	425	89.0	88.7	88.4	88.0	87.5	87.0	86.5	85.9	85.2	84.3	83.3	82.1
	450	88.5	88.2	87.8	87.4	86.9	86.4	85.9	85.2	84.4	83.4	82.4	81.1
	475	88.0	87.7	87.3	86.8	86.4	85.7	85.2	84.5	83.7	82.6	81.5	80.1
	500	87.5	87.1	86.7	86.2	85.7	85.1	84.5	83.7	82.9	81.8	80.6	79.1
	525	87.0	86.6	86.2	85.6	85.0	84.4	83.8	83.0	82.1	80.9	79.7	78.1
	550	86.5	86.1	85.6	85.0	84.4	83.8	83.1	82.2	81.3	80.0	78.7	77.1
	575	86.0	85.5	85.0	84.4	83.9	83.1	82.4	81.5	80.5	79.2	77.8	76.1
	600	85.4	85.0	84.4	83.8	83.2	82.5	81.7	80.7	79.7	78.3	76.8	75.1
	625	84.9	84.5	83.9	83.3	82.6	81.8	81.0	80.0	78.9	77.5	75.9	74.1
	650	84.4	83.9	83.3	82.7	82.0	81.1	80.3	79.3	78.1	76.6	75.0	73.1
	675	83.9	83.4	82.8	82.1	81.4	80.5	79.6	78.6	77.3	75.8	74.1	72.1
	700	83.3	82.8	82.2	81.5	80.7	79.8	78.9	77.8	76.5	74.9	73.2	71.1
	725	82.8	82.3	81.7	80.9	80.1	79.2	78.2	77.1	75.7	74.1	72.3	70.1
	750	82.3	81.7	81.1	80.3	79.5	78.5	77.5	76.3	74.9	73.2	71.3	69.1
	775	81.8	81.2	80.5	79.7	78.9	77.9	76.8	75.6	74.1	72.4	70.4	68.1
	800	81.2	80.6	79.9	79.1	78.2	77.2	76.1	74.8	73.3	71.5	69.4	67.0
	825	80.7	80.1	79.4	78.5	77.6	76.6	75.4	74.1	72.5	70.7	68.5	66.0
	850	80.1	79.5	78.8	77.9	77.0	75.9	74.7	73.3	71.7	69.8	67.6	65.0
	875	79.6	79.0	78.2	77.3	76.4	75.3	74.0	72.6	70.9	69.0	66.7	64.0
	Loss per Percent Combustibles												
		3.0	3.1	3.3	3.5	3.7	3.9	4.2	4.5	4.8	5.2	5.7	6.3

Figure 6-12 Combustion Efficiency Chart—Coal

(As used in Figures 6-10, 6-11, and 6-12, combustion efficiency is boiler efficiency before radiation loss and unaccounted for loss are deduced.)

method. Since the measurements used affect only the losses, a comparatively low percentage of the total heat input, measurement errors have a much smaller effect on the resulting boiler efficiency.

From a practical basis, a standard fuel analysis can usually be used without affecting the boiler efficiency more than approximately 1 percent. A practical method for quick use in the field is a simplified heat loss method that is based on curve fits of precalculated efficiency tables for a standard fuel. The fuel analysis and the precalculated tables are given in Figures 6-9, 10, 11, and 12. This shortcut method is shown in Figure 6-13. If the fuel analysis is unknown, the result of this shortcut method may be as precise as the detailed calculation using a standard fuel analysis. The simplified method requires a minimum of information. The essentials are % oxygen, flue gas and combustion air temperatures, type of fuel, boiler size, and percent rating.

The curve in Figure 6-14 demonstrates the large influence of the radiation loss as boiler size and percent boiler rating are reduced. As shown on this example for a boiler of approximately 20,000 lbs/hr, an efficiency loss of 10 or more percentage points can occur if the boiler is operated over a 10 to 1 turndown range. The radiation loss is built into the boiler design, and the user cannot operate the boiler in any manner to reduce this loss.

% Sensible Heat Loss (all flue gas)

$\Delta t \times [0.023 + 0.00011\ (\%\ O_2 + 1)^2]$ *

% Latent Heat Loss

Natural gas, 9%

No. 2 oil, 5.5%

No. 6 oil, 5.0%

Coal, $[2.7 + 12.8\ (M^2 + M)]\%$

M equals fuel moisture % as a decimal number

% Radiation Loss	20,000 lb/hr blr	100,000 lb/hr blr
Full load	1.0%	.4%
50% load	2.0%	.8%
25% load	4.0%	1.6%

% Boiler Efficiency

100 − % Sensible Heat Loss −
% Latent Heat Loss −
% Radiation Loss

* Δt, Flue Gas Temperature − Combustion Air Temperature

Figure 6-13 Heat Loss Method Shortcut

(S.G.Dukelow Formula—completely empirical method)

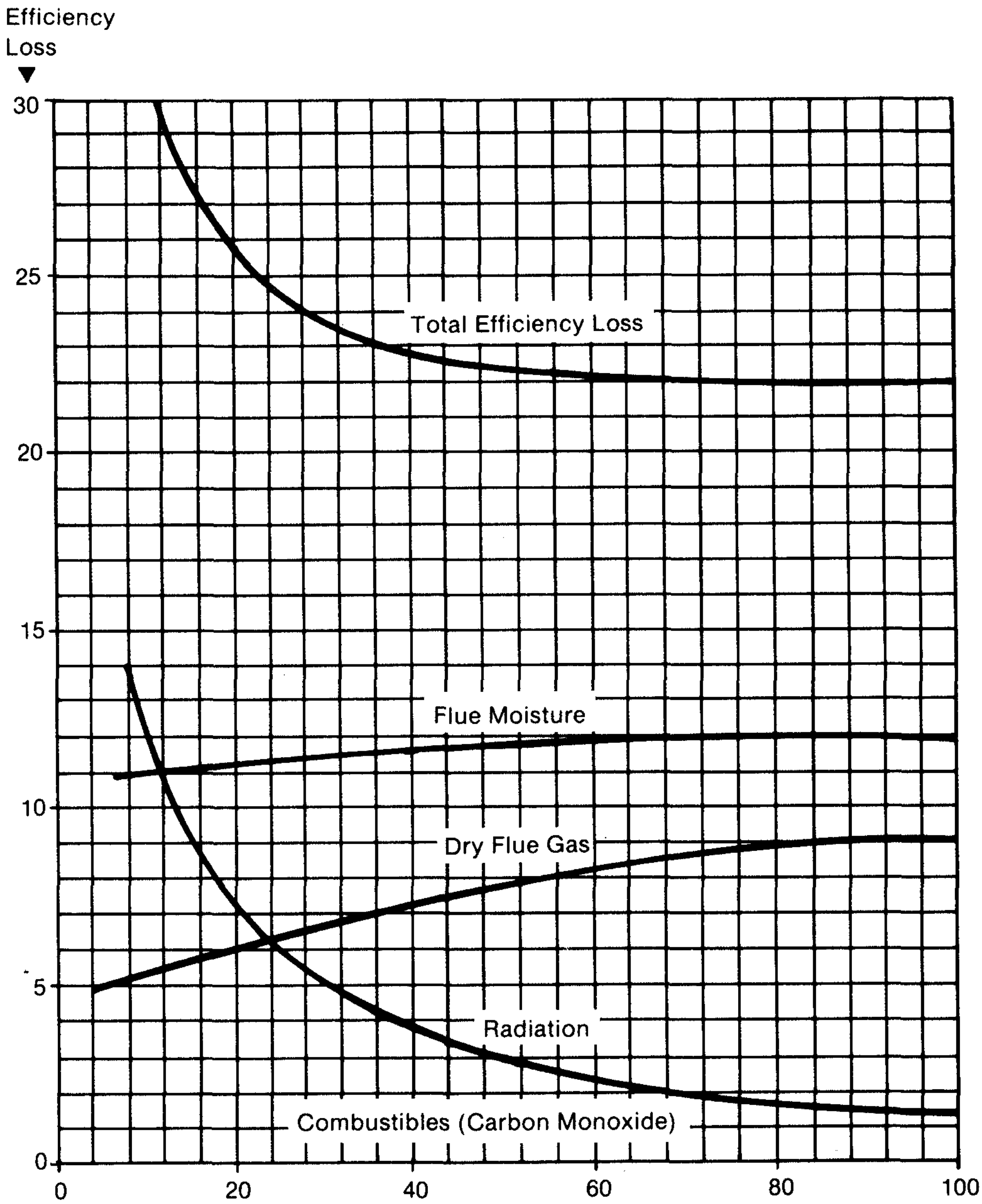

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Figure 6-14 Effect of Boiler Capacity on Various Losses

Section 7
The Steam Supply System

7-1 Saturated Steam Moisture Elimination

In Section 2 the water circulation of watertube boilers was explained. The steam that is collected in the steam drum contains some droplets of boiler water that must be removed before the steam is delivered from the drum. Figure 7-1 demonstrates in a general way how the water and steam are separated.

As the steam-water mixture rises to the drum, internal baffles separate this mixture from the water that goes on to recirculate through the downcomer tubes. The steam is liberated from the mixture and collects in the upper part of the drum. Before passing out of the drum, the steam passes through mechanical separation devices. These ''scrubbers'' or ''separators,'' which may be of many different designs, return the moisture droplets to the water in the drum and allow dry steam to pass out of the drum.

The most sophisticated of these separators uses a centrifugal action that whirls the mixture with the steam emitted from the center and the heavier water from the outside. When very dry steam is required, many of these devices will be found inside the drum along with simpler scrubber equipment. While it is not evident from the outside, the inside of the drum is often filled with such devices. A typical arrangement is shown in Figure 7-2.

After passing through the scrubbing devices for eliminating moisture droplets, the steam leaves the drum. If the boiler has no superheater, the steam flows to a distribution header. All boilers feeding such a header must have a non-return valve at the boiler outlet. If the boiler has an integral superheater, the steam will pass through the superheater before it leaves the boiler.

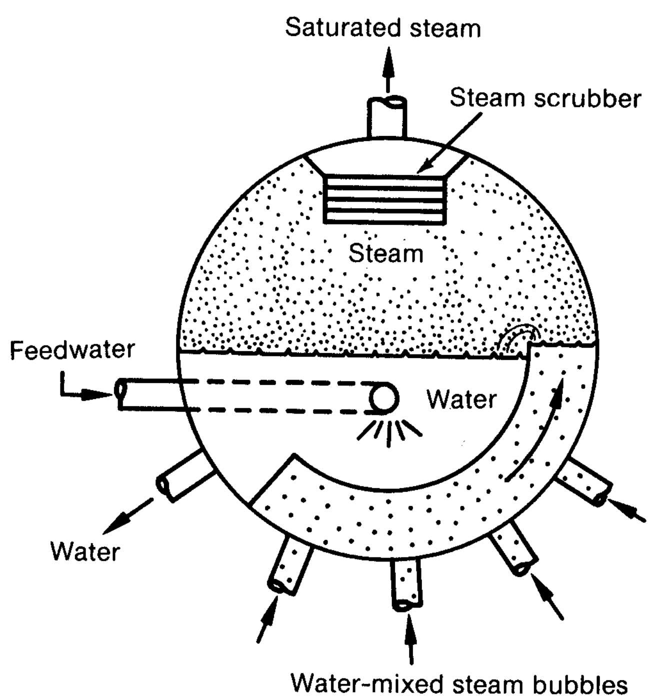


Figure 7-1 Boiler Steam Drum

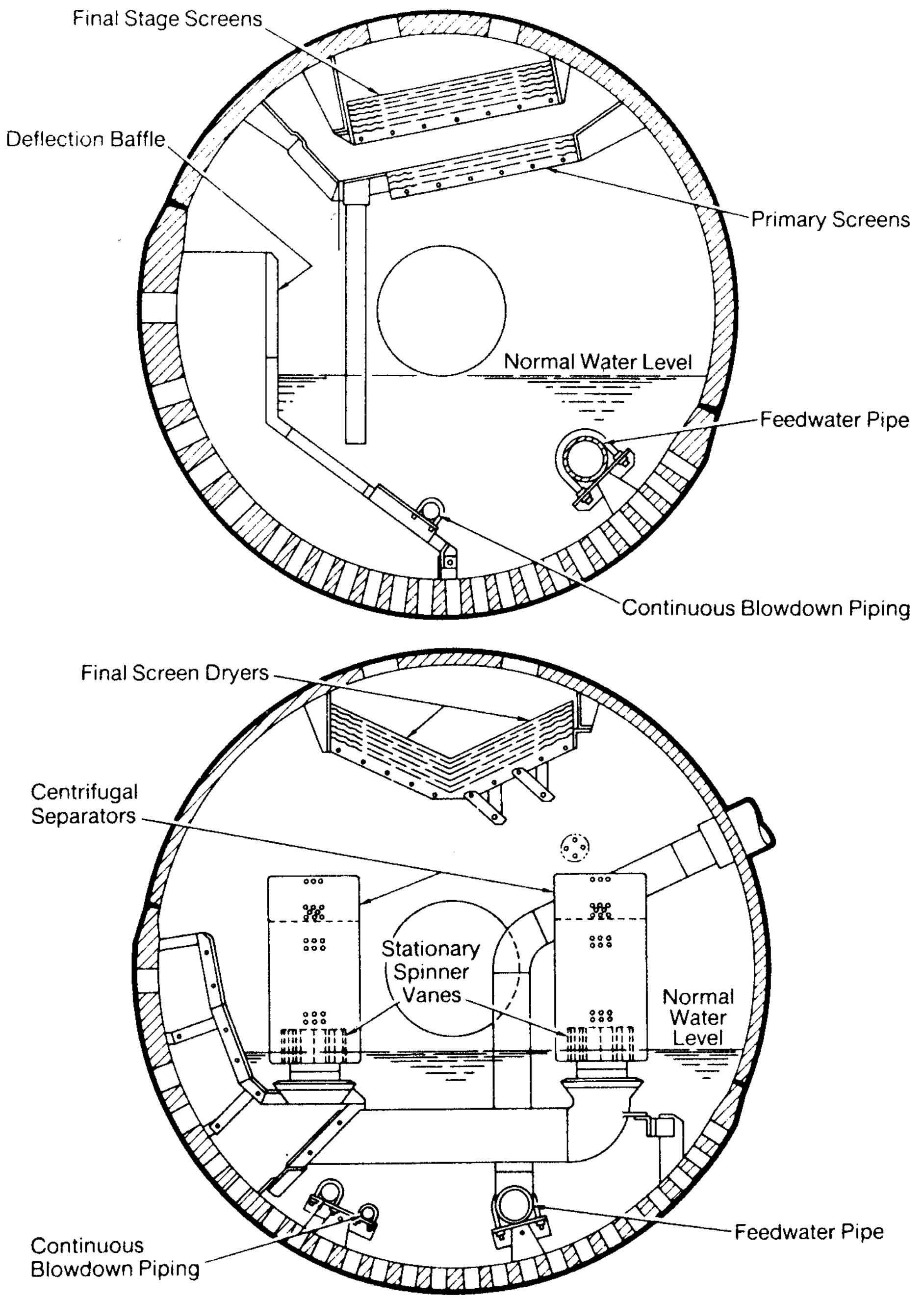


Figure 7-2 Steam Drum Internals

(From *Fossil Power Systems*, © Combustion Engineering Co., Inc., 1981)

Note that Figure 7-2 shows longitudinal piping for feedwater and continuous blowdown. The feedwater and continuous blowdown piping usually enter the boiler drum at one end. Perforations in this piping distributes these flows longitudinally.

7-2 Steam Supply Systems

A generic steam supply system is shown in the diagram in Figure 7-3. This diagram shows two saturated steam boilers connected to a steam header. The normal measurement points of

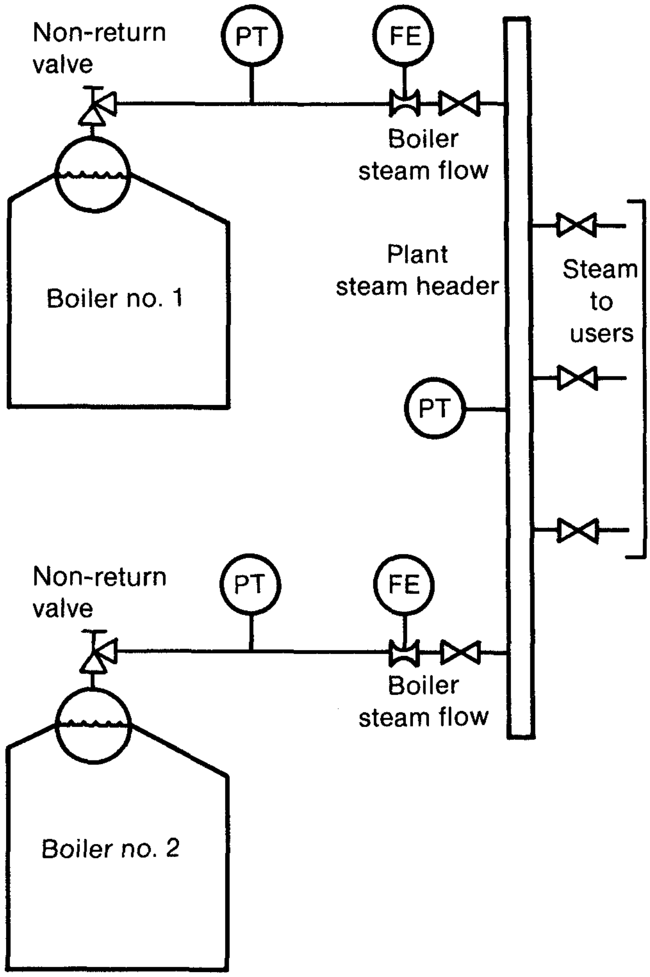


Figure 7-3 Steam Header for Multiple Boilers

a flow nozzle or orifice plate for measuring steam flow and a connection for steam pressure measurement are shown. If these were superheated steam boilers, a steam temperature measurement would also be shown. At the outlet of each boiler is a non-return valve that prevents steam from the header from entering the boiler. The steam header is the collecting point for the steam from more than one boiler and the location from which the steam is distributed to the steam users. While two boilers are shown here, there could be multiple boilers operating at the same pressure and feeding steam to the same header.

The pressure in the steam header is also normally measured. On the outlet side of the steam header, the steam may travel through many steam lines in its transit to the users of the steam. The basic need of the steam user is for heat energy. Steam is a convenient carrier of that energy. When the header, the boiler steam leads, and all the steam lines are pressured to the normal operation pressure, a certain quantity of steam and, therefore, heat energy is stored in that system.

7-3 Heat Energy and Water Storage

Figure 7-4 is a diagram somewhat typical of the heat energy storage in the boiler. Heat energy is stored in the water, steam, metal, refractory material, and insulation material of the boiler. When the boiler is at normal pressure and corresponding temperature but with no steam

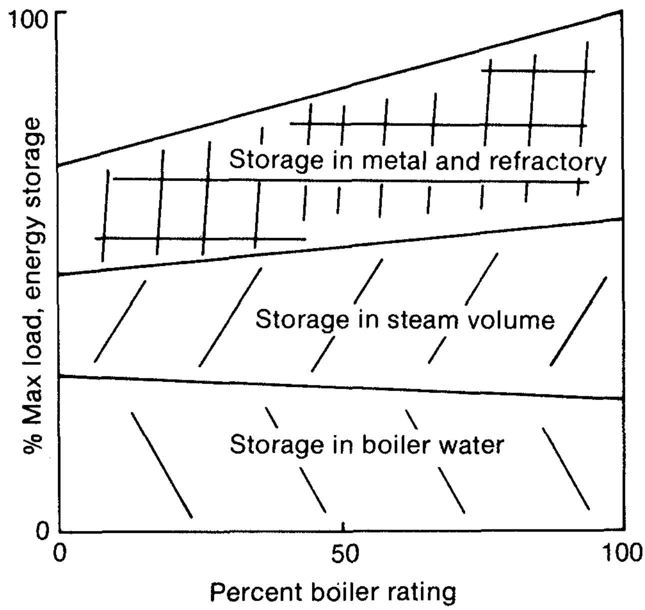


Figure 7-4 Characteristics of Boiler Energy Storage

output, the diagram shows that a large percentage of the heat energy storage at maximum boiler load already exists.

The boiler steam space is filled with steam, which represents heat energy storage in the steam. In this case there are no steam bubbles underneath the surface of the boiler water since no new steam is being produced. Heat energy is also stored in the boiler water. Since there are no steam bubbles, the entire water space below the normal level in the boiler drum is water at saturation pressure and temperature. At this time the boiler is also losing heat by radiation.

As additional fuel and air are burned, the boiler steam output may be increased to 100 percent or more of its rated value. Note that as firing rate increases, the heat energy storage in metal, refractory, etc., increases due to the higher operating temperature of these parts. The steam space of the boiler is increased due to the steam bubbles below the steam drum water surface, and this increases the energy storage in the boiler steam. The steam bubbles subtract from the water volume, and this subtracts from the boiler water energy storage as boiler steam flow is increased. Another aspect of the reduction in boiler water volume is that the mass of water in the boiler is less at higher boiler steam flow rates. This is an important factor in the proper control of boiler feedwater.

The energy storage in the boiler acts the same as a flywheel, which is also an energy storage device. Table 7-1 can be used for demonstration.

Table 7-1
Energy Storage Relationships

Steam at 200 psig	1198.3 Btu/lb
Water at 200 psig sat.	355.5 Btu/lb
Latent heat of evap.	842.8 Btu/lb
Water at 180 psig sat.	346.2 Btu/lb
Heat released by dropping pressure (200 to 180 psig)	9.3 Btu/lb

As pressure is reduced, the water cannot exist at 355.5 Btu/lb. It immediately releases 9.3 Btu/lb, thus cooling the water to the saturation temperature for 180 psig. The heat released causes an immediate generation of steam that carries a corresponding amount of heat. In this way heat is released from storage as the pressure drops and must be replaced in storage as the pressure increases.

Assume that the boiler contains 50,000 lbs of water and that the pressure drop above occurs in 10 seconds. The Btu in the water that becomes steam is 9.3 * 50,000 or 450,000 Btu. This is the Btu value of approximately 450 lbs of steam. Since this change occurs in ten seconds, the hourly rate of steam flow change is 450 * 3600/10 or 162,000 lbs per hour. On an increase in capacity, this increase, which is due to the withdrawal of energy from storage, temporarily helps to maintain pressure while the boiler control system is acting to increase fuel and air.

There are slight delays in the actions of the control equipment. If the boiler steaming rate were 160,000 lbs per hour, then the steaming rate demand could be doubled in 10 seconds with no increase in fuel and combustion air. The drop in steam pressure would be less than 20 psi since the steam volume and boiler parts would also give up energy.

When increasing or decreasing load, the amount of energy in storage must change. Over-firing and under-firing on a load change restores borrowed energy and readjusts energy storage to the proper level for the new load.

If a boiler is not equipped with a superheater, the steam output is saturated steam. The superheater is a secondary heat exchanger that accepts saturated steam from the boiler and, by extracting heat from the flue gases, is able to add more sensible heat to the steam. A superheater alters the control characteristics of a boiler operating at a given pressure. By changing the specific volume of the output steam, the steam storage in lbs in a particular steam system is less while the energy storage per lb increases. The change in the effect on steam pressure is related to the change in specific volume.

Section 8
Firing Rate Demand for Industrial Boilers

8-1 Relationships

The demand or a change in demand on the boiler system is generated by the steam users' requirements for energy flow. As they open valves to get more of the energy locked into the steam energy carrier, the pressure drops in the total storage system, triggering the release of some of the heat energy from storage.

The magnitude of the pressure drop depends on the relationship between the boiler water volume, total volume of the steam system, the magnitude of the steam demand, and the magnitude of the change in steam demand. If the water volume is high, energy from the water is released to slow down the change in steam pressure. If the system volume is relatively low, the steam pressure change will be relatively high and vice versa.

The steam header pressure is the energy balance point between the energy demands of the steam users and the supply of fuel and air to the boilers to replenish the energy to the header system. At a constant steam flow or energy requirement, a constant pressure in the steam header indicates that energy supply and demand are in balance. While the actual requirements are for energy, control systems work on the physical properties of pressure and temperature. A 1:1 relationship between steam flow and energy flow is normally assumed. This is not true if the pressure and temperature change significantly. With this caveat, the balance is represented by the statements that follow.

(1) Steam demand = steam flow plus or minus (K) * (pressure error). K is a function of the system volume and steam specific volume related to the demand flow rate.

(2) Supply side = fuel, air, and water energy to the boiler plus the change in energy storage.

(3) Demand side = steam to users.

(4) Balance point = steam header pressure.

(5) Pressure at set point—demand equals supply when energy storage is constant.

(6) Pressure increasing—supply exceeds demand (may equal demand if energy storage is decreasing)

(7) Pressure decreasing—demand exceeds supply (may equal supply if energy storage is increasing)

8-2 Linking the Steam Pressure Change to Changes in Firing Rate

The combustion, feedwater control, and steam temperature control systems determine how a boiler actually operates and whether it achieves its efficiency potential. The controls should be designed to regulate the fuel, air, and water to a boiler and maintain a desired steam pressure or hot water temperature while simultaneously optimizing the boiler efficiency.

During either normal or abnormal operation, the greater the sophistication of the controls, the greater the efficiency potential of the total boiler system. A control system can usually be upgraded in its functions by adding additional components or software. Improving a control system is usually a cost-effective way to improve the operating efficiency of any boiler.

Generally the boiler controls can be classified in two main groups: on/off and modulating. On/off controls are subdivided into basic on/off (full on and off) and high/low/off, which has a high and low fire "on" condition plus the "off" condition. Modulating controls are subdivided into two basic classes: positioning and metering.

The simplest, most basic, and least costly control and the one used to control firing rate on only the smaller firetube and watertube boilers is on/off. The control is initiated by a steam pressure or hot water temperature switch. As the pressure or temperature drops to the switch setting, the gas valve is opened (or the fuel pump started) along with the combustion air fan motor. The fire is ignited usually with a continuous pilot flame. The fuel and air continue operating at full firing rate capacity, and the pressure or temperature rises until the switch contact is opened.

Although such a system may maintain steam pressure or hot water temperature within acceptable limits, combustion is not controlled because combustion efficiency (while firing) is a result of mechanical burner adjustment. When the burner is on, the excess air is subject to the following variations in the fuel supply.

- Pressure and temperature of the fuel
- Btu content of the fuel (hydrogen/carbon ratio)
- Fuel specific gravity
- Fuel viscosity
- Mechanical adjustment tolerances

The excess air is also subject to variations in the combustion air supplied.

- Air temperature and relative humidity
- Air supply pressure
- Barometric pressure
- Mechanical adjustment tolerances

In addition, each time the burner is off, cold air passes through the boiler carrying heat up the stack unless the flue damper is closed. Using this system, the "on" fire is at full firing rate and the flue gas temperature is at maximum. Figure 8-1 represents how the on/off system works.

The other on/off system is the high/low/off control in which the burner system has two firing rates called "high fire" and "low fire." If the Btu requirements are between those of high fire and low fire, the burner will stay on all the time, cycling between high and low. Unless the load is below low fire input, this eliminates the "off" heat losses caused by cold air through the boiler. Such a system has three steam pressure or hot water temperature settings:

(1) Stop fire or off
(2) Start boiler and go to low fire or stop high fire and go to low fire
(3) Start high fire

This system will hold the steam pressure or the hot water temperature within closer tolerances of the desired steam pressure or hot water temperature. It will have a lower weighted average flue gas temperature than a straight on/off system but a higher weighted average flue gas temperature than a fully modulating control.

The system can be tuned to burn the fuel efficiently when the burner is "on" at any time. It will get out of tune when any of the fuel and air conditions change from those present when the burner was set. Compromising some of the mechanical adjustments may be necessary in trying to optimize combustion at both the "low fire" and "high fire" settings of the high/low/off system.

Using on/off control to add water to a boiler—based on the water level in the boiler—intermittently cools and heats the boiler water, causing increased on/off or high/low cycling action of the firing rate control.

The action of the high/low/off control under the same load conditions as Figure 8-1 is shown in Figure 8-2.

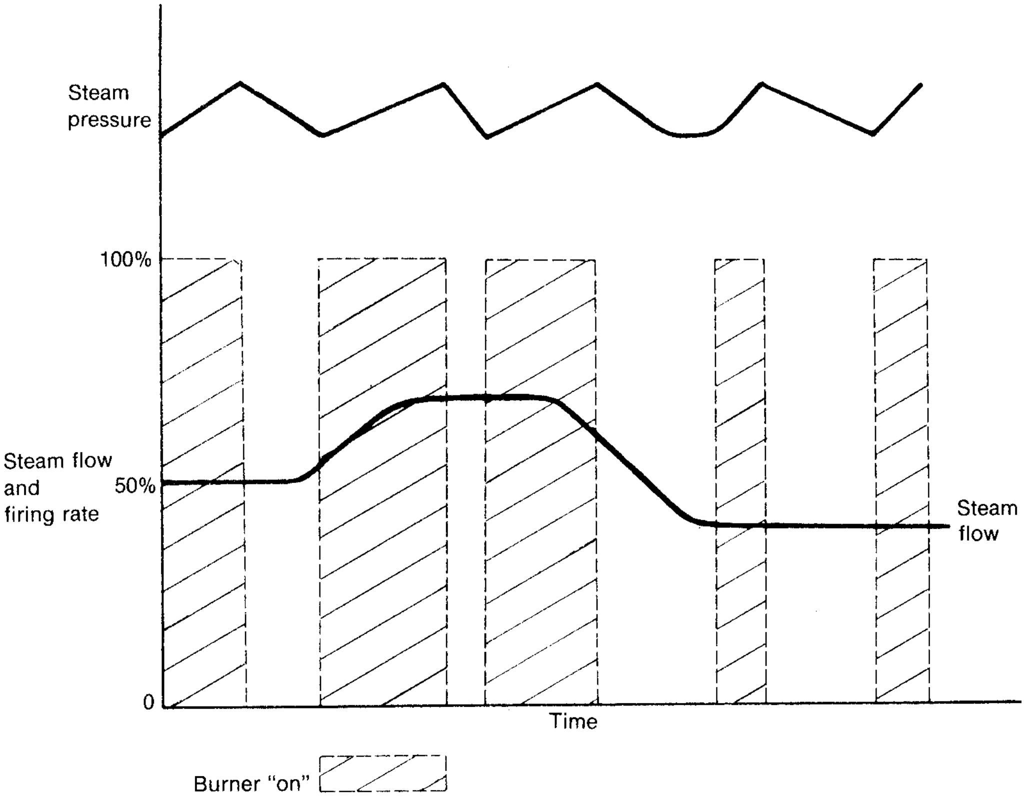


Figure 8-1 On/Off Control Action

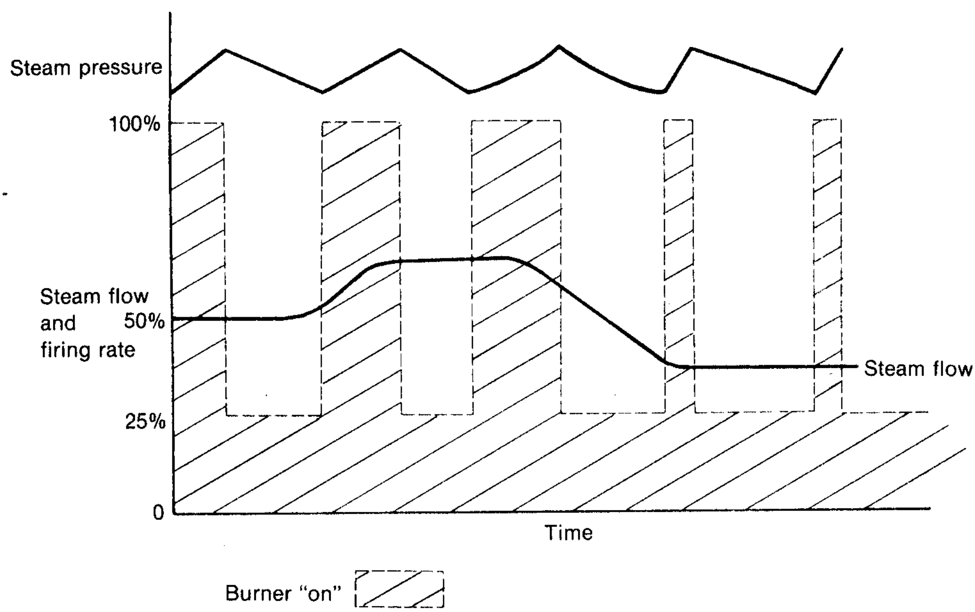


Figure 8-2 High/Low/Off Control Action

Modulating control is a basic improvement in controlling combustion and feedwater. A continuous control signal is generated by a controller connected to the steam or hot water piping system. Reductions in steam pressure or hot water temperature increase the output signal, which calls for a proportionate increase in firing rate.

Modulating control is an improvement because the fuel and air Btu input requirements and the Btu of the steam or hot water output of the boiler are continuously matched. The action of such a control system under the same load conditions of Figure 8-1 and Figure 8-2 is shown in Figure 8-3.

Because matching the input and output Btu requirements is improved, the steam pressure or hot water temperature is maintained within closer tolerances than is possible with the previously discussed control systems. The weighted average flue gas temperature is lower, so boiler efficiency is greater. Table 8-1 compares the efficiency of boilers with the different systems while operating under the indicated load conditions. The influences of changes in the condition of fuel and air have been eliminated by assuming a 10 percent excess air and a constant flue gas temperature for each of the loads or firing rates that would occur.

Although each boiler will have its own characteristics of excess air as opposed to flue gas temperature, the table is typical for a gas-fired boiler with the efficiency calculations based on 450 to 600 degrees flue gas temperature over the 25 percent to 100 percent load range. The table represents control systems operating ideally with no variation in excess air. Excess air should be higher at loads less than 50 percent, and efficiencies would be lower than those shown for the high/low/off and modulating systems.

At 25 percent load, high/low/off and modulating have the same efficiency. Twenty-five percent would be considered low fire, which would be "on" full time, or the same continuous firing rate for both systems. At 100 percent, all systems would be "on" full time at 100 percent firing rate. Therefore, results are the same for all systems. In the middle range, efficiency clearly improves as control sophistication is increased. The benefit of modulating control is clearly established in Table 8-1. The type of modulating control and how it is implemented in developing the "firing rate demand" signal is next examined.

8-3 Steam Pressure or Steam Flow Feedback Control

Assuming that the equipment for generating a firing rate demand signal is of the modulating type, several different methods and considerations are involved. For the simpler systems, a

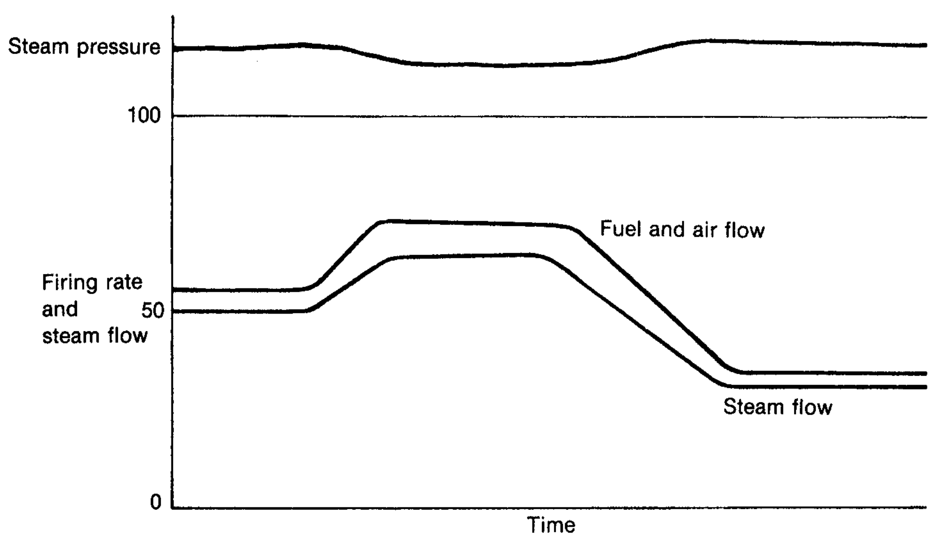


Figure 8-3 Modulating Control Action

Table 8-1
Control System Performance Comparison

Type of Control	Efficiency at % load 25%	50%	75%	100%
On/off	70.28	74.28	75.61	76.28
On/off with flue damper	73.28	75.28	75.95	76.28
Hi/low/off	76.88	76.48	76.35	76.28
Modulating	76.88	77.68	77.15	76.28

simple proportional or proportional-plus-integral feedback controller may be used. Figure 8-4 demonstrates this method for regulating the firing rate using steam pressure. In some installations a constant steam flow may be required for one or more boilers in combination, while other boilers connected to the same header are used for controlling steam pressure.

It is also possible to arrange the system as shown in Figure 8-5 so that the control for a particular boiler can be switched between steam pressure and steam flow control. The switching procedure would require the boiler operator to switch the control to manual, adjust the set point to the desired value of the variable being switched to, operate the transfer switch, and then transfer the control back to automatic operation. In either steam pressure or steam flow, a change in these variables is equated to a system demand for energy.

Figure 8-6 is a diagram of a change in steam flow rate, firing rate, and steam pressure with respect to time. This diagram is useful in analyzing the tuning requirements of the loop. Suppose, as shown, the steam pressure transmitter has a range of 0 to 300 psig for a 0 to 100 percent output signal and that at a constant steam flow rate the firing rate is 90 percent of its range when steam flow is 100 percent of its range. If 10 psig is the maximum desired pressure deviation for a 10 percent change in steam flow rate, such a deviation will produce a (10/300) 0.033 or 3.3 percent change in the signal from the steam pressure transmitter.

This signal must be amplified to produce a change in the firing rate. Under steady-state conditions the firing rate change for the 10 percent steam flow rate change is approximately 9 percent of its range.

As the load is increased, even a step change as shown here, the steam pressure will change

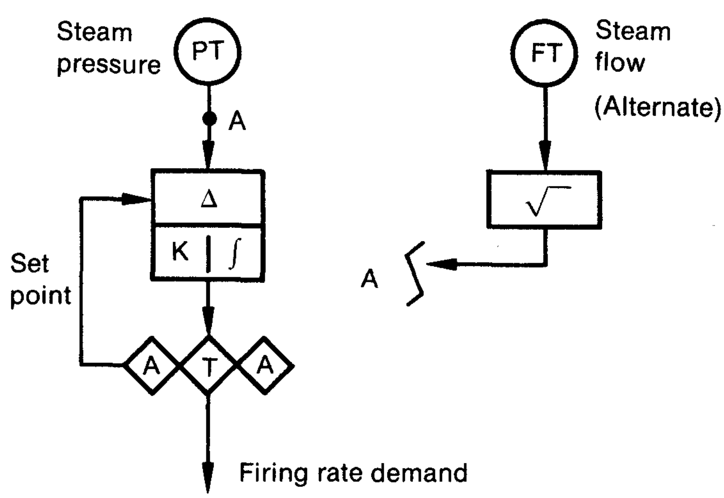


Figure 8-4 Steam Pressure or Steam Flow Feedback Control

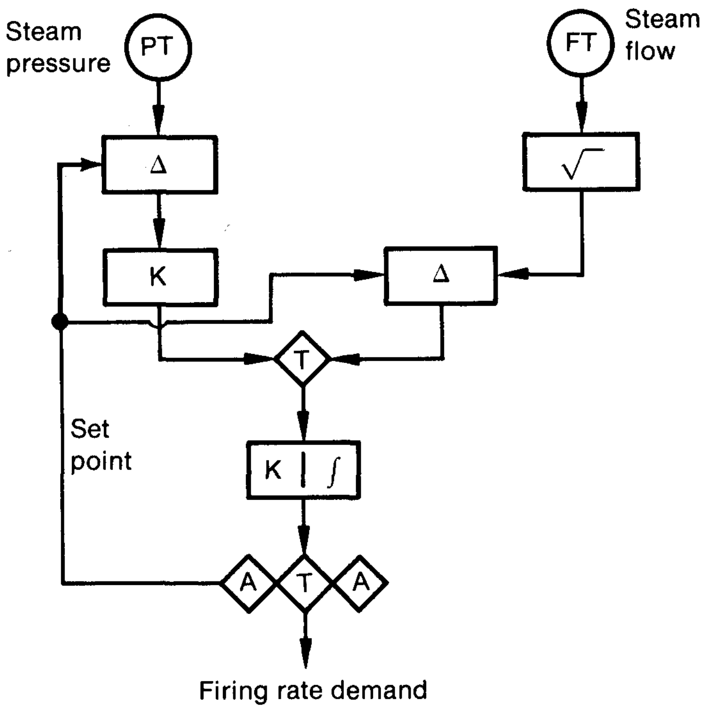


Figure 8-5 Steam Pressure or Steam Flow Feedback Control

at a slower rate. This is the result of the initial load change being partly satisfied from energy storage. The initial firing rate change is thus a slightly delayed and reduced effect, since energy cannot be withdrawn from storage without a drop in pressure. In addition, the process takes time to convert the fuel energy and transfer it to the water in the boiler. Until the rate of withdrawal from storage balances the effect of increased firing rate, the steam pressure will continue to fall. When the pressure stabilizes, the net result is that energy storage has been reduced.

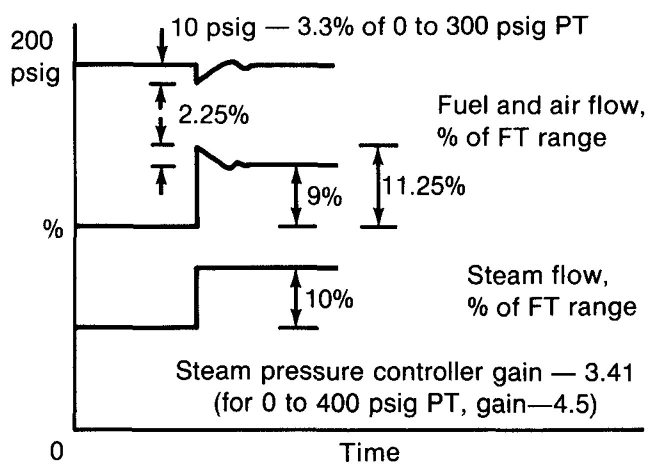


Figure 8-6 Changes in Steam Flow Rate, Firing Rate, and Steam Pressure with Respect to Time

This withdrawal from storage must be paid back by temporary overfiring. A load increase also requires overfiring to add the required additional energy storage that will allow the pressure to return to its set point. The magnitude of the desired temporary overfiring is usually a minimum of approximately 25 percent of the steady-state firing rate change, or 2.25 percent in this case. The 9 percent steady-state firing rate increase is thus increased to a needed 11.25 percent of maximum. Since the multiplier is (11.25/3.3) 3.41, a gain of 3.41 would be applied to the steam pressure controller. Note that if the steam pressure transmitter had a maximum range of 0 to 400 psig, the deviation would have been 0.025 or 2.5 percent and the multiplier would have been (11.25/2.5) 4.5. Upon a reduction in steam flow, underfiring would have been necessary to adjust the system energy storage.

For tuning such a steam pressure controller, a typical controller gain of 4.0 is a reasonable starting point with an integral setting of 0.25 repeats per minute. The optimum gain will be determined by the ratio of boiler capacity to boiler-header energy capacity and to scaling factors of the transmitters used.

The optimum integral setting will be approximately equal to the time constant (one fifth of total time) of the particular steam generation process. In this case, since the time constant is several minutes, the integral value in repeats per minute will be less than 1.0.

The optimum controller tuning may not be that which will produce the optimum steam pressure pattern. In many cases it is possible to obtain improved steam pressure control at the expense of boiler operating efficiency. Increasing the controller gain may produce oscillations of the fuel flow and the air flow that may improve steam pressure control while decreasing efficiency due to the oscillations. The degree to which this may occur is different for different installations. The resolution of this question must be based on the judgment of the process-experienced individual who is responsible for the controller tuning.

If steam flow is the controlled variable, the gain of the controller can be greater than that of a typical flow control loop because the integral time is much longer. In this case a good starting point is 1.125 (11.25/10) for the gain setting. Since the integral value is related to the process time constant, it will approximate the value of the pressure control integral value.

The output of the controller is called the firing rate demand or the energy input demand for the boiler. Development of a proper firing rate demand signal is of primary importance since all control of fuel and air is directed from this master signal. The controller is therefore called the master controller. Because of the importance of this controller, a number of more sophisticated configurations are available. The use of these more complex arrangements results in greater precision of the firing rate demand (input energy demand) control signal and improvement in the steam pressure control.

8-4 Feedforward-plus-Feedback — Steam Flow plus Steam Pressure

A feedforward-plus-feedback arrangement is often used. One of the two most frequently used variations is shown in Figure 8-7. In this arrangement the steam flow (a) is the feedforward demand. The proportional multiplier function (b) is adjusted at the input of summer (c) so that a change in steam flow will produce the correct steady-state change in firing rate demand. The steam pressure controller (d) provides the correct adjustment of the firing rate demand for the necessary overfiring or underfiring to adjust energy storage.

With any such feedforward system, the fuel flow signal change is directly linked to the steam flow change. This results in immediate and faster action on the fuel flow change since the full fuel change occurs before appreciable change in the steam pressure. This results in less energy withdrawal from storage. Since energy withdrawal from storage is directly related to drop in steam pressure, less withdrawal means that there has been a smaller steam pressure deviation from set point.

Using the previous example, the gain of the proportional multiplier (b) would be adjusted to a minimum (0.09/0.1) 0.9. The gain of the pressure controller would be (2.25/3.3)0.68. In

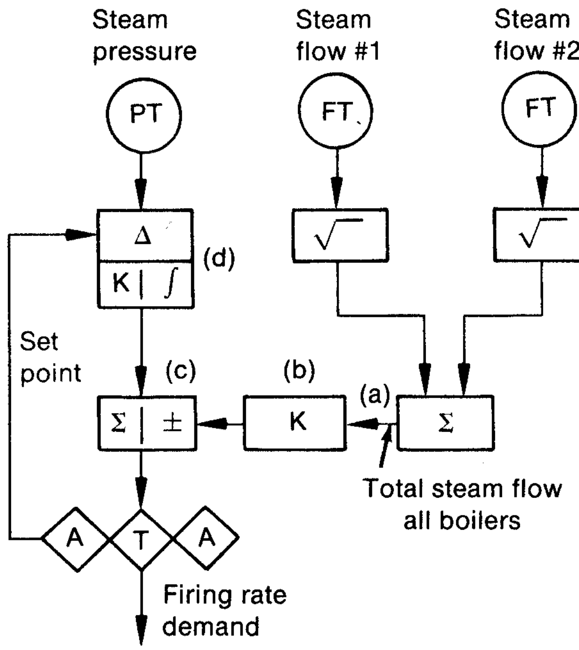


Figure 8-7 Feedforward-plus-Feedback Master Control

some installations, better performance may be obtained with a higher steam pressure controller gain to provide an increase in the over- and underfiring rates.

In this type of system, most of the control action that would be supplied by the integral in a simple feedback control loop is supplied by the feedforward signal. Since the feedback portion should essentially produce no effect under steady-state conditions, it is necessary that the steam pressure controller be essentially proportional in nature. Because of the dynamics involved, the calculated settings given for this type of system may not be optimum.

In order that the steam pressure will eventually return to set point when the steam flow versus firing rate relationship is imperfect, a small amount of integral is needed. This should be an amount less than that indicated by the process time constant to avoid developing an unwanted integral signal during the steam pressure deviation. An integral setting of 0.05 to 0.1 repeats per minute is suggested.

Note that the steam flow signal is shown as the total steam flow for all boilers. Whenever more than one boiler feeds steam to a steam header, arranging one individual boiler feedforward from the steam flow of that boiler alone would result in unstable control due to positive feedback. If all boiler steam flow signals are compared in a high select function, an individual high select output can be used as the feedforward signal for control of all boilers.

In the arrangement above, a change in the steam flow feedforward signal provides the necessary magnitude of the steady-state change in firing rate demand. An alternate feedforward application shown in Figure 8-8 exchanges the functions of the steam flow and steam pressure signals.

In this case the derivative input from steam flow into summer (c) is adjusted to provide the temporary overfiring or underfiring, with the steam pressure controller (d) adjusted to provide the necessary changes in the steady-state firing rate demand. The calibration of summer (c) does not include a bias, since under steady-state operation the output of summer (c) should equal the input from controller (d) with the steam flow derivative signal returning to zero.

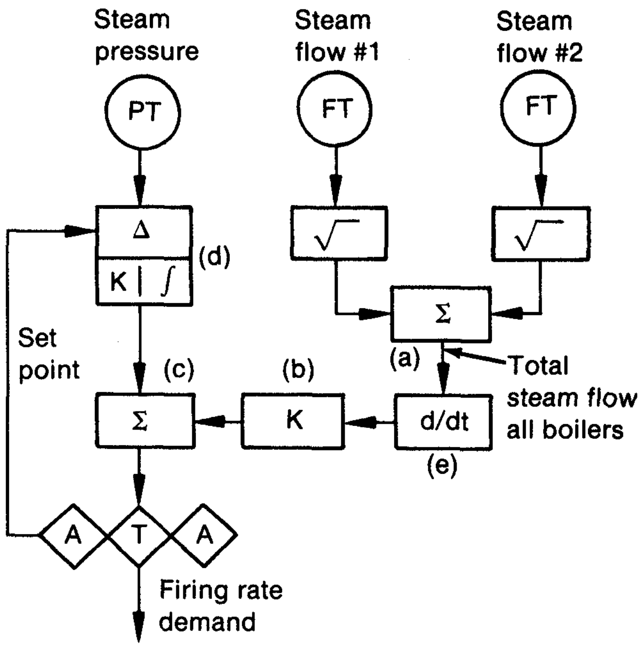


Figure 8-8 Alternate Feedforward-plus-Feedback Master Control

Using the previous example, the steam pressure controller would have a gain of (9/3.3)2.73. If the integral of the simple feedback controller were 0.25 repeat per minute, the integral would be 0.25 also for this arrangement. The derivative from steam flow would be adjusted so that the 2.25 percent magnitude under- or overfiring, adjusted by proportional (b), and the necessary time duration of that overfiring or underfiring, adjusted by derivative (e), would be correct.

While the selection between the two alternates is user choice, a high pressure drop between the boiler and the steam header would indicate that the alternate in Figure 8-8 would probably be a better choice. Under this condition the boiler pressure changes significantly with load even though the steam header may be controlled at a constant pressure. This is often the case with electric utility boilers.

In the normal industrial installation, the change in steam header pressure is almost entirely due to the change in steam flow rate on the user demand side of the header. If the pressure drop between boiler and steam header is high, then a change in steam flow rate may cause a larger change in steam header pressure due to changes in the supply side pressure drop. These two pressure changes mean different things to the system. A pressure change on the demand side means that firing rate should be changed because the user wants more steam. A change in pressure drop on the supply side may be an indication of a needed change in firing rate to change the stored energy that is represented by boiler pressure.

A third arrangement of a feedforward system decouples the steam pressure and the steam flow actions in the firing rate demand control. This arrangement is shown in Figure 8-9.

In this arrangement the deviation between steam pressure and its set point (steam pressure error) multiplies the feedforward signal. Thus, on a major demand for a change in firing rate, the initial effect comes from the feedforward signal without the delay in steam pressure change. As the steam pressure changes to create the steam pressure error, the feedforward magnitude is reduced and replaced by the steam pressure controller output. The steam pressure controller is tuned in a manner similar to that when only feedback control is used with respect to the

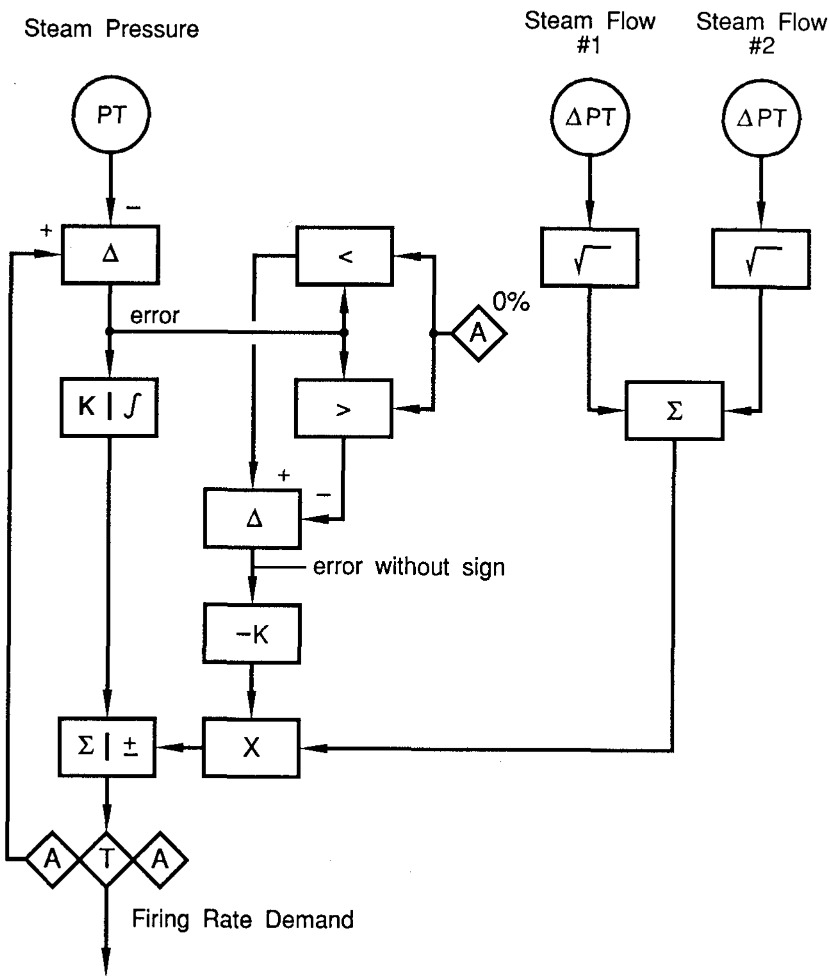


Figure 8-9 Decoupling the Feedforward and Feedback Portions of the Firing Rate Demand Master Control

steam pressure effect. On small firing rate demand requirements, the action is essentially feedforward. The advantage of this is that an extra dimension in the tuning is added, and the tuning constants are not as fixed in their dynamic relationship to each other.

The arrangement shown in Figure 8-10 adds compensation for variation in the heat content of the fuel. Changes in the steam flow signal act to change the steady-state firing rate demand. Overfiring or underfiring is added as required by the controller (d). The total proportional-plus-integral controller is comprised of the three logic functions marked (d). If fuel heating value is constant and at the design value, the analysis of this system is the same as that shown in Figure 8-7.

If the heating value of the fuel were to change under steady-state steam flow, the steam pressure would start to change due to a mismatch of heat energy input and output. The integral function (d) would begin to generate a change in its output due to the steam pressure error from set point multiplied by the gain of the steam pressure controller. The output of the integral function (d) would then change the multiplication in the multiplier (f) to adjust for the change in the fuel heating value.

When the multiplication is correct so that the fuel heat energy input is in proper balance with the steam heat energy output, the steam pressure returns to set point, the steam pressure

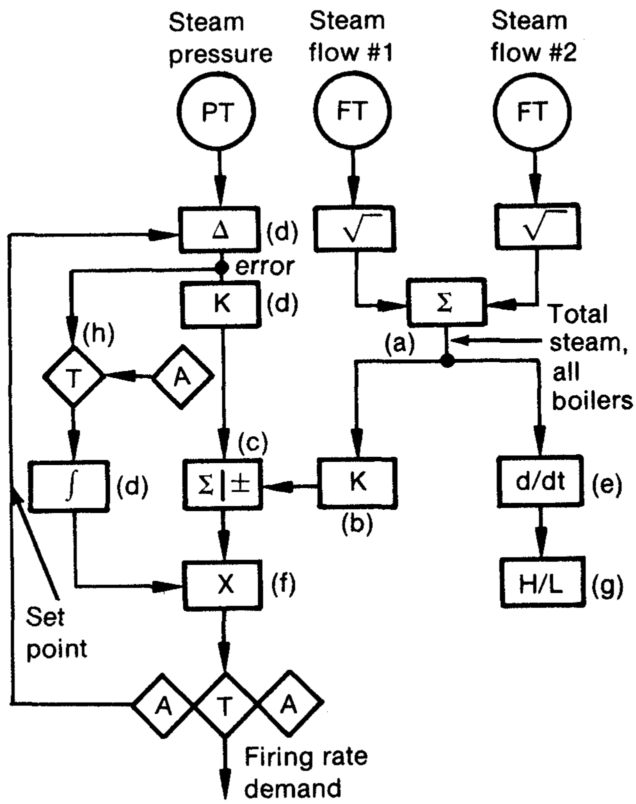


Figure 8-10 Feedforward-plus-Feedback Master Control (with automatic adaptive feed-forward gain)

error becomes 0, and the action of the integral function (d) ceases to change the multiplication value.

In this application the integral tuning should be very slow, with its output changing only on a sustained pressure deviation. The steam pressure deviations from normal steam flow changes should have no effect on the output of this integral function. The interlock functions (e), (g), and (h) are included to block integral action when the steam flow rate is changing. In effect, the boiler is being used as a calorimeter to automatically keep the system in correct calibration. This is sometimes referred to as use of a calibrating integral, a form of adaptive control.

8-5 Load Sharing of Multiple Boilers

The firing rate demand generated by the master controller raises and lowers the firing rate of all boilers connected to a steam header. The next control problem encountered is the proper sharing of the load between boilers. The simplest and most often used method is to leave the boiler load allocation to the judgment of the operator.

In practice a 0 to 100 percent firing rate demand signal is sent to a boiler master manual-automatic station. Incorporated into this station is a bias function. The bias function enables the boiler operator to add to or subtract from the master firing rate demand signal and thus alter this signal before it is transmitted to the individual boiler controls. Figure 8-11 shows the control functions that are involved for two boilers. In a similar manner this can be extended to three or more boilers.

The operator should be instructed to maintain a 0 percent bias on one of the boiler master stations with input and output of equal value. On other stations the operator enters a desired bias signal in order that the output boiler firing rate will deviate from the master firing rate demand by the amount of the plus or minus bias signal.

8-6 Automatic Compensation for the Number and Size of Boilers Participating

In the arrangement of Figure 8-11, an assumption is made in tuning the master steam pressure controller that the number and size of the boilers will always be the same. In many cases as the total load increases, additional boilers of indeterminate size may be put in operation. As the number and/or average size of the boilers connected change, the effect is to change the gain of the system. In this circumstance the optimum tuning of the master controller changes due to the change in total fuel flow capability. By modifying the load allocation functions as shown in Figure 8-12, automatic compensation for the number and average size of the boilers is obtained.

In the first edition of this book and as shown here, item (c) uses proportional-plus-integral control logic. This procedure, when used with analog control and a high gain and integral controller, performs very well.

For the control logic shown in Figure 8-12, assume that three boilers of the same capacity are feeding the steam header and that total steam flow and firing rate demand are at 60 percent. The feedback to the controller (c) from the summer (b) must also be at 60 percent, and the output of the controller (c) would also be 60 percent boiler firing rate. Under this condition, three 60 percent signals are inputs to the summer (b). Since the boilers are equal in size, a gain of 0.33 on each summer input will maintain the system in balance.

If one boiler is shut down with no change in total steam flow, one of the 60 percent inputs is removed from the summer (b). This immediately drops the summer output to 40 percent, causing an immediate increase in controller (c) output to 90 percent. Two 90 percent inputs to summer (b) with 0.33 input gain produces a summer output of 60 percent. The control loop is back in balance with a 60 percent input and 90 percent output from controller (c). Since there are now two boilers instead of three, each 1 percent of span change in master firing rate

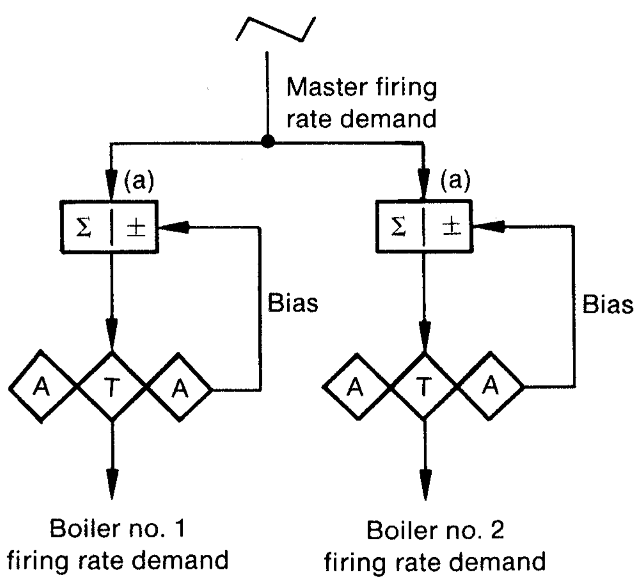


Figure 8-11 Manual Boiler Load Allocation

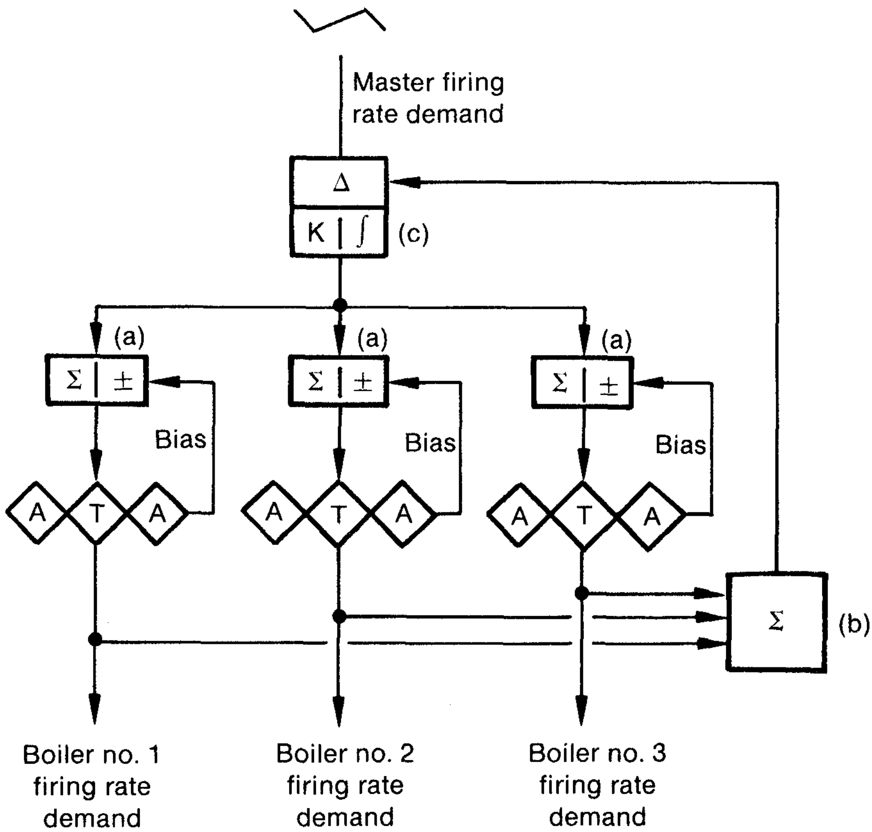


Figure 8-12 Automatic Compensation for Number and Size of Boilers Participating

demand will result in 1.5 percent change in boiler firing rate demand. The arrangement produces correct results whether boilers are added or shut down.

Should one or more boilers be placed on manual control, its firing rate is constant with its input to summer (b) at a fixed value. Any change in master firing rate demand must then be balanced by an output change of summer (b) that does not include the input from the boiler on manual control. The relationship between the input and output changes of the controller (c) will then be the same as if the manually controlled boiler had been removed from service.

Should the boilers be of different sizes, gain values on the inputs of summer (b) are adjusted in accordance with boiler size. If three boilers are used and one is twice the capacity of the other two, then the summer gain used with the larger boiler would be 0.5, and the other two summer inputs would have gains of 0.25. Under the 60 percent load example, removing one of the smaller boilers from service would cause an immediate firing rate change on the two remaining boilers to 80 percent of full load. Whether the boilers are of the same or equal size, any adjustment of load allocation bias by the operator will cause an immediate readjustment to the firing rate of other operating boilers.

The combination of gain and integral should be 50 or more to avoid delay between an input and output change of item (c) that would cause the control performance to deteriorate. This technique applied to pneumatic and electronic analog control systems, and using the gain and integral values above, has produced excellent results.

Information feedback from actual installations indicates that such results may not always be obtainable when using this technique with digital distributed control systems. Due to all of

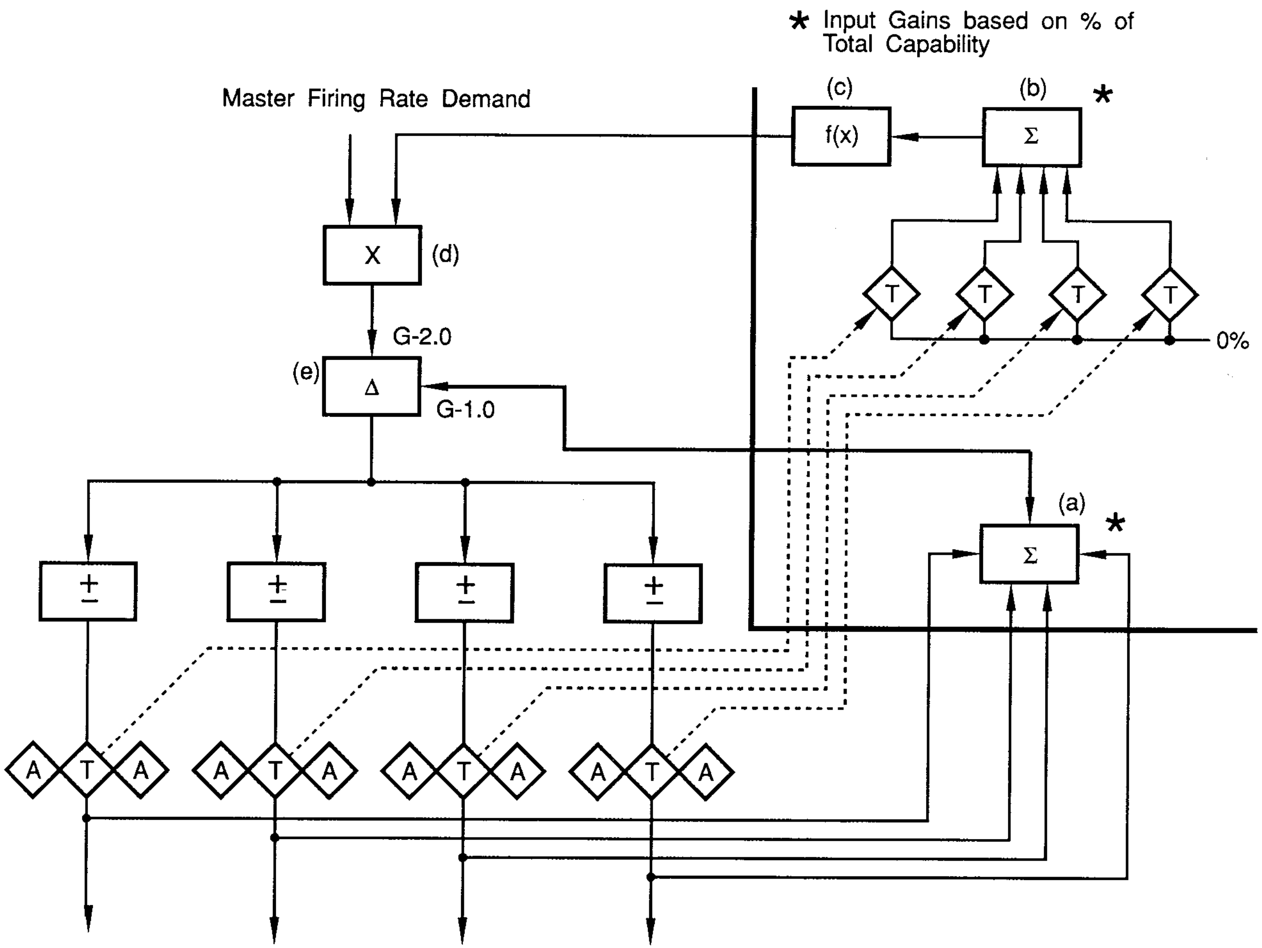


Figure 8-13 Automatic Compensation for Number and Size of Participating Boilers

the system timing and data transfer actions, the loop stability requirements for the gain and integral rate of item (c) may slow the action of item (c) to the detriment of overall control performance. This arrangement can be seen to be the secondary controller of a cascade system. For stability, such a controller must be an order of magnitude faster than the primary controller. The arrangement shown in Figure 8-13 is suggested as an alternate for distributed digital systems if difficulty is encountered with the arrangement in Figure 8-12.

The control function is, however, necessary. If a distributed digital system is being used, many of the PID algorithmns include an adaptive port so that the gain and integral values can be changed automatically. If such a controller is not available with the particular system being used, the technigue for adaptive controller tuning in Figure 8-13 can also be used to automatically compensate for the numbers and capacities of connected sources. In such an adaptive control arrangement, the controller (c) of the previous application is eliminated. Its function is provided by automatically adjusting the gain with the variable multiplication (d).

The individual boiler firing rate demand signals, and whether these signals are on automatic or manual, feed into a an off-line logic block. The individal demand signals feed into summer (a), which has individual input gain adjustment. With four boilers of equal capacity, the individual gain values would be 0.25 and would total 1.0. If the boilers were of different capacities, the input gain values would be different but still total 1.0. The total from the summer (a) thus represents 0-100 percent of total capability.

The output of multiplier (d) with a gain of 2.0 into the delta (e) represents double the 0-100 percent capability and the output of delta (e) represents 0 to 100 percent of capability. If a control station is on manual, signalled through the dotted line connections, and the output adjusted or the bias value of any individual unit is adjusted, the weighted value of the change in relation to the total is adjusted to obtain the correct value for the automatically controlled outputs. Any manual control participation change into summer (a) immediately adjusts the automatically controlled outputs in order that the process see no change. Since such an arrangement does not affect the crispness of control performance, it may be desirable to furnish an adjustable rate of change to the multiplication and subtraction factors.

These same functions can also used for applications that involve multiple fuels, multiple fans, multiple pumps, and multiple coal pulverizers. Where these applications are covered later in this book, the arrangement in Figure 8-12 with the controller (c) will be used, since that arrangement may be more universal to the needs of other applications for this function.

8-7 Preallocation of Boiler Load Based on Test Results

In some installations it may be desirable to remove the load allocation decision from the operator's function. The simplest method of automatically allocating load between the boilers is shown in Figure 8-14. In this control arrangement a function generator has been added for each boiler. The output signal of each function generator is some function of the input signal. Note that this application and the next one to be described may also be implemented with the distributed control-oriented, adaptive gain logic previously described.

In practice, the boilers would be tested, and the best overall load allocation would be determined. The individual boiler function generators would then be calibrated based on the desired function curves that were determined. The arrangement shown also has the capability to automatically compensate for the number of boilers in service. Should a boiler be removed from or added to service or placed in the manual control mode, the loading allocation of the boilers in operation should be approximately correct relative to the others in operation at the time.

8-8 Energy Management by Boiler Load Allocation on a Least Cost Basis

A current state-of-the-art technique for energy use efficiency is to allocate boiler loads on an incremental cost basis. Such control is a part of the control system that is generally called

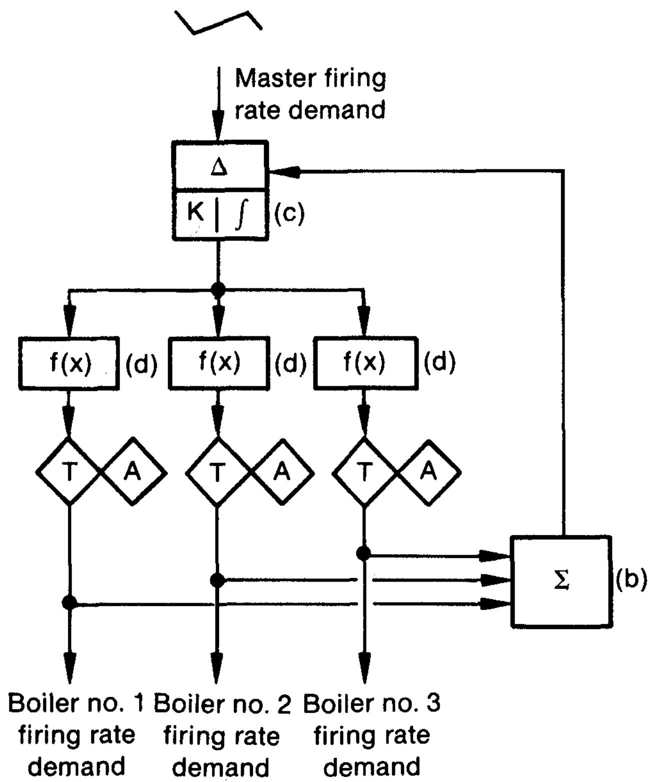


Figure 8-14 Precalibrated Boiler Load Allocation

"energy management" control. The system computations are implemented with a digital computer or in digital calculations as part of a digital distributed control system.

The computing device sends boiler load allocation signals to modify the boiler firing rate demand signals. Figure 8-15 demonstrates the control logic of using the load allocation signals from a computer. The computer develops a bias signal to raise or lower the basic demand for boiler firing rate. A summer function (a) is used to combine the signals. The summer outputs are indicative of boiler load allocation on a least-cost or other "as-desired" basis. All summer input gain adjustments are 1.0.

The computations to develop the least-cost boiler load allocation signal include all applicable factors of the cost of boiler operation. These include the unit fuel cost, any special operation cost factors for a particular boiler, plus the incremental cost of input fuel energy relative to steam energy output. The boiler with the highest efficiency may not be the proper boiler for adding the next increment of total boiler load. The correct result is to load the boilers at equal incremental rates. The incremental heat rate is determined by the derivative of the input-output energy curve.

Figure 8-16 demonstrates the logic of this approach. Assume that there are two boilers and that test data produces curves of efficiency vs. boiler load as shown. Boiler 1 shows an efficiency that is at all loads higher than that of boiler 2. From this it appears that it would always be more economical to load boiler 1 to 100 percent before loading boiler 2.

The input (fuel) vs. output (steam) curve in Figure 8-16 shows clearly the error of this approach. For least-cost it is always necessary to obtain the next increment at lowest cost. Assume the minimum load on both boilers is 20 percent. In this case this would indicate loading boiler 1 from 20 percent to 55 percent, while the load of boiler 2 remains at 20 percent. If additional steam is required, the load of boiler 2 should then be increased up to 100 percent

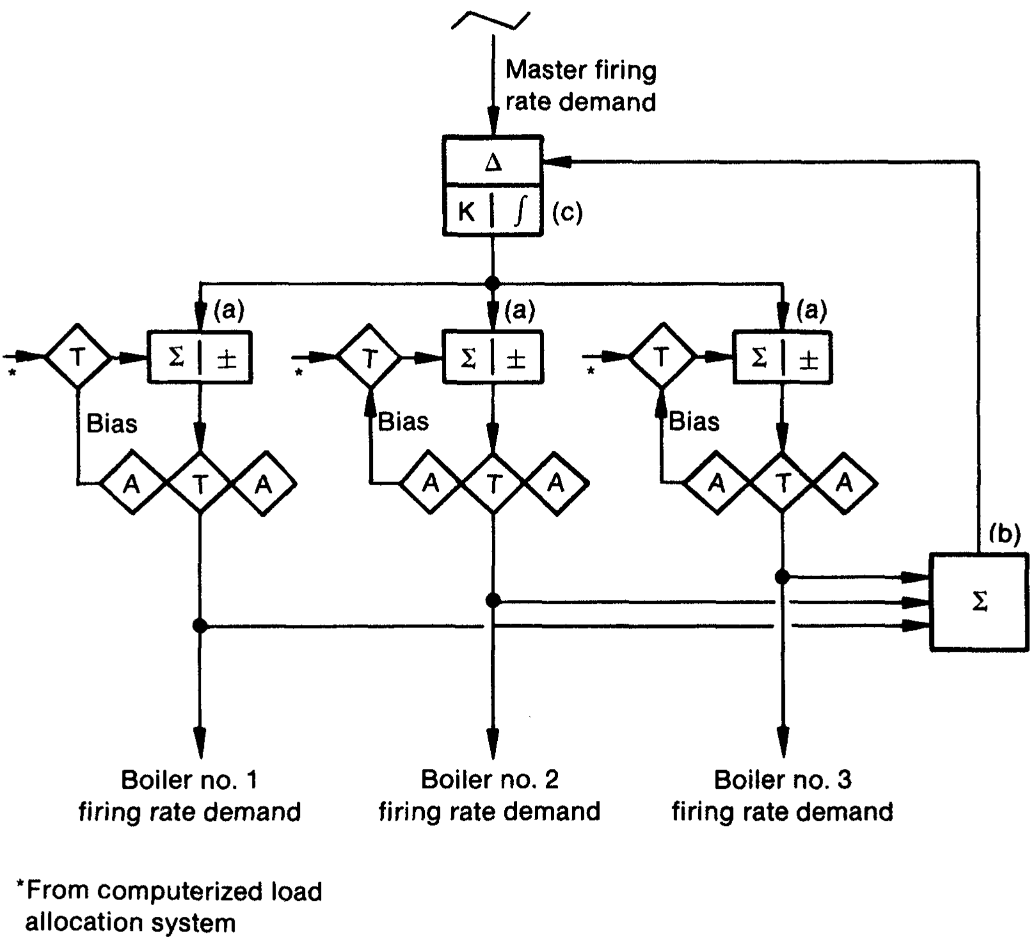


Figure 8-15 Boiler Load Allocation on a Least-Cost Basis

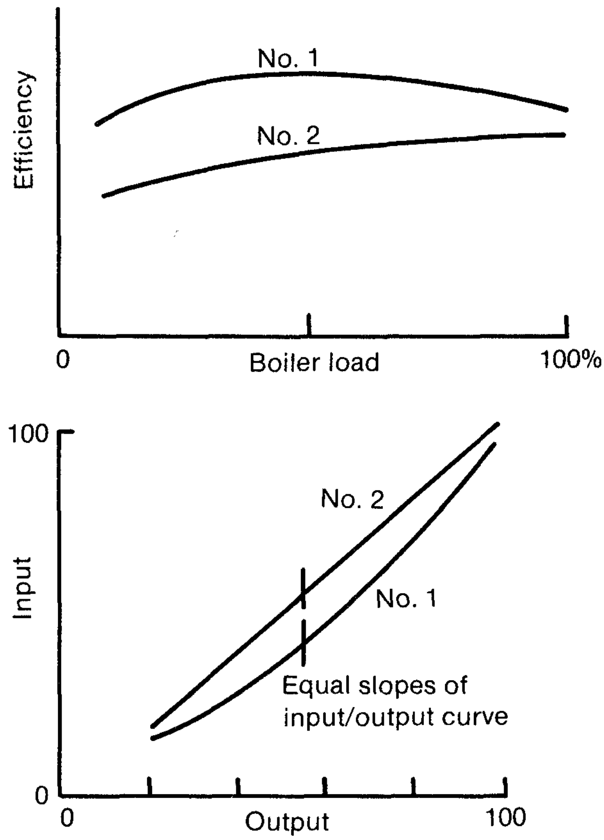
before any further changes are made to the load of boiler 1. Additional steam from this point is then obtained by loading boiler 1 from 55 percent to 100 percent.

If there were a greater number of boilers, loading them all at equal incremental cost rates would always result in the lowest additional increment of cost. Energy quantity is not the whole story. Different fuels may be used with the different boilers with different cost factors. Special operating cost factors may also bias the cost results between boilers. In general, for identical boilers that burn the same fuel, there is usually not enough potential gain to make the described technique of economic benefit.

Real-time calculations of incremental fuel usage cannot be made. The boiler operates at only one load at a time, so the entire load-fuel curve at any one time is unknown. The general practice is to develop the shape of the curve and apply the single load value to bias the curve up or down. Over a period of time as the boiler operates at various loads, the curve shape can be gradually updated.

8-9 Energy Management Involving Cogeneration Networks

Another aspect of energy management is economic loading of industrial turbines or turbogenerators. The design of these turbines and their arrangement in the overall heat cycle presents another opportunity for overall control simplicity and overall total energy management.



Correct least-cost loading:

Hold Boiler No. 2 at 20%; load No. 1 from 20% to 55%.
Hold Boiler No. 1 at 55%; load No. 2 from 20% to 100%.
Load Boiler No. 1 from 55% to 100%.

Figure 8-16 Least-Cost Boiler Loading

Several industrial turbine arrangements are shown in Figure 8-17. Assume a boiler operating at 600 psig. This steam has the potential for its energy to be used for heating or to be converted to work in a steam engine or steam turbine. Of this energy, only a certain portion called the "available energy" can be converted to work. This leaves the steam with the largest portion of its energy intact after it has produced work by flowing through the turbine.

If this steam is taken out of the turbine before the turbine exhaust, it is called "extracted" or "bleed" steam. If the steam is extracted at a fixed pressure, the turbine is a controlled extraction turbine. Other types are backpressure turbines and condensing turbines. An extraction turbine acts as two separate turbines in series with a backpressure control regulating the steam flow to either another controlled extraction in that or another turbine, a backpressure turbine, or a condensing turbine.

The heat balance diagram of an industrial power system often shows steam headers of more than one pressure value and a network of turbines of different types, plus bypassing and other pressure-reducing valves. This is commonly called "cogeneration," a concept that dates back at least to the early part of this century. Some of the largest cogeneration power plants operating today were built in the 1920s. They involve cooperation between a large refinery and an electric utility company.

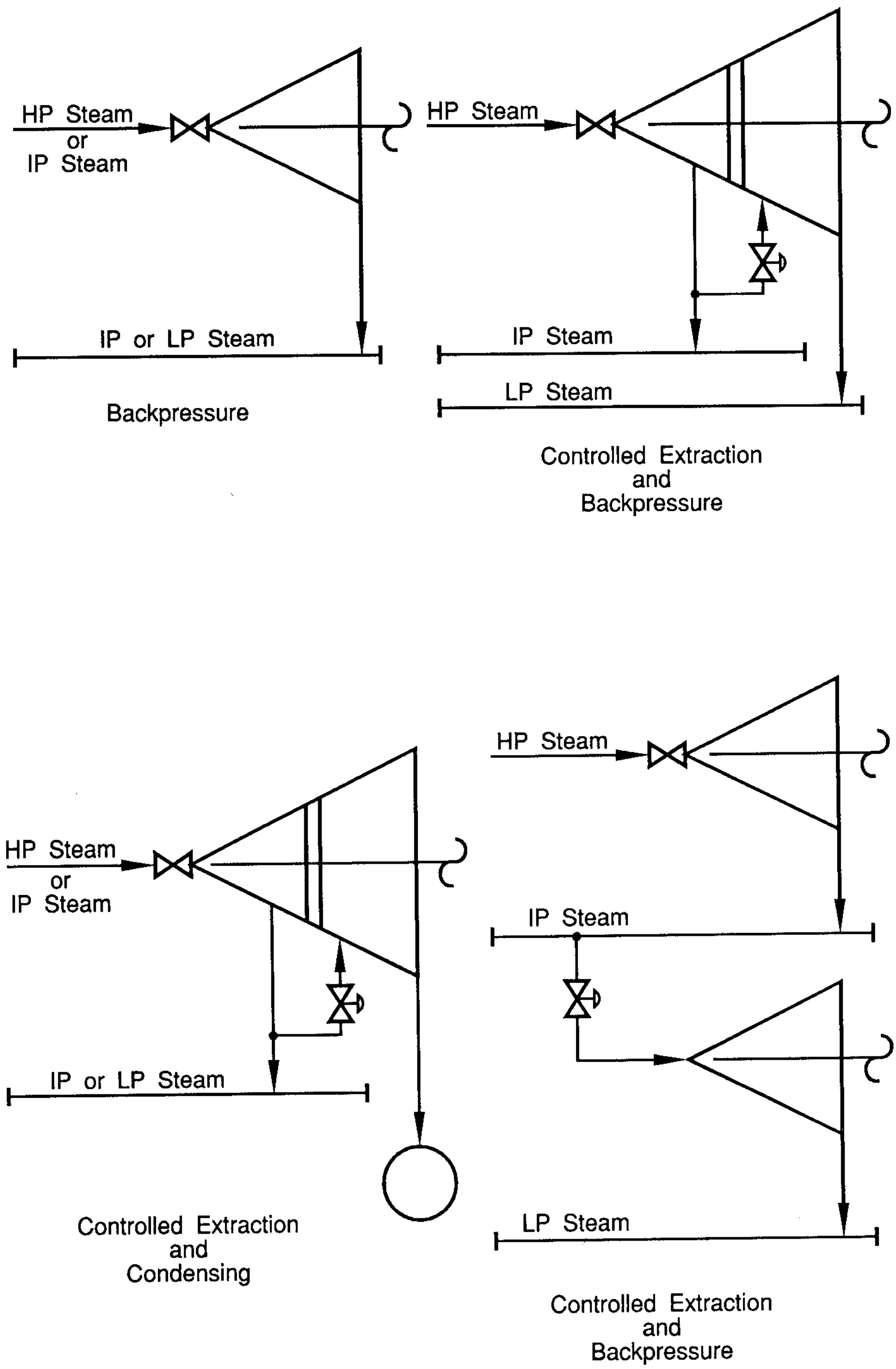


Figure 8-17 Arrangements of Controlled Extraction, Condensing, and Backpressure Turbines

Fifty years ago before the existence of the national power grid, a very large percentage of small users generated their own power with steam engines or turbines. Exhaust steam or "bleed" steam (extraction) was used for space heating, refrigeration, and process heat. The percentage of the total energy needs produced by cogeneration fifty years ago was much larger than it is today. Industry has thus tended to regress in total energy efficiency in spite of the great strides in efficiency improvement of boilers, turbines, and basic "first law" heat cycles.

In the case of steam loads at more than one pressure plus a power generation load, the overall system must be balanced and all pressures maintained at stable values. A relatively very simple example of such a boiler-turbine heat balance network is shown in Figure 8-18. Such a network can be controlled with a single master pressure controller connected to the

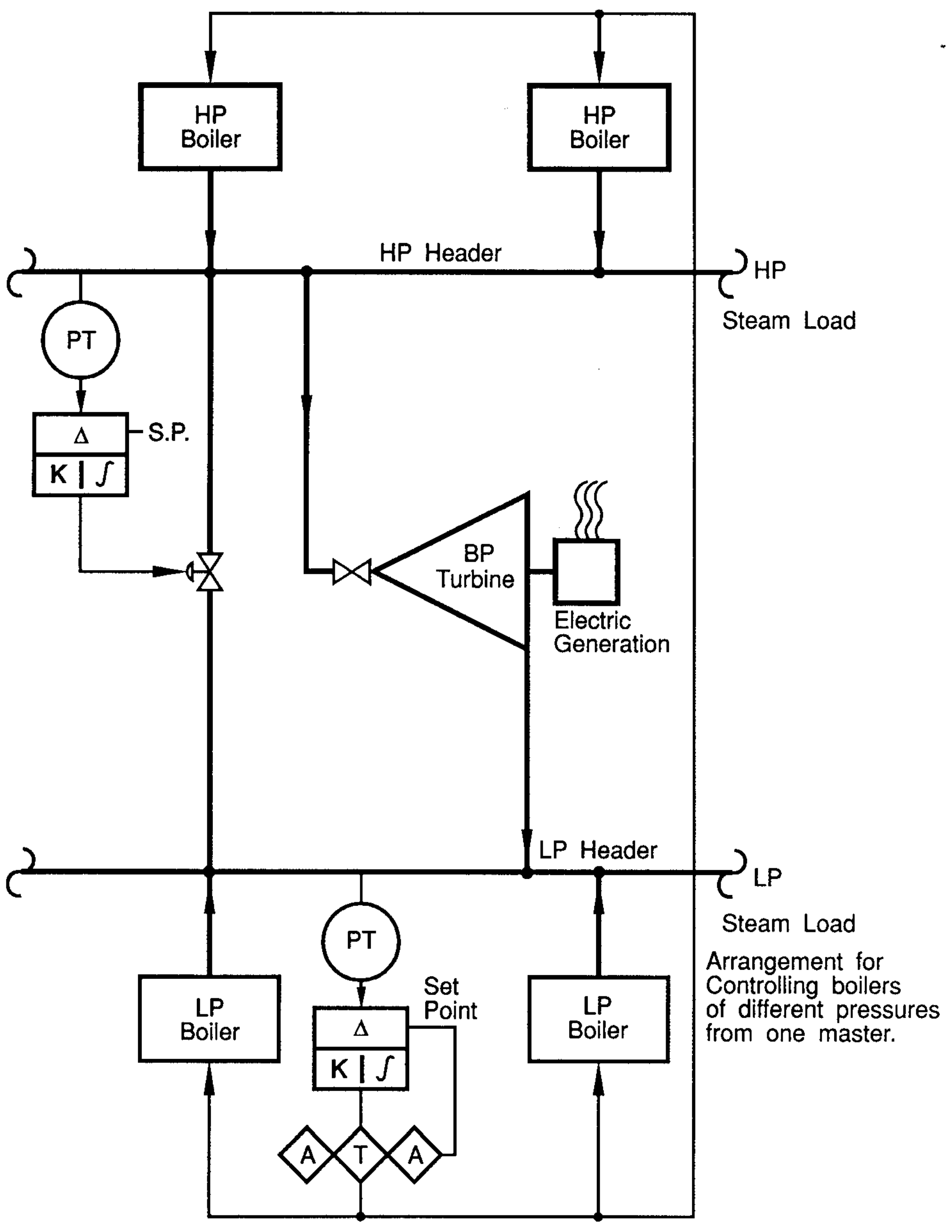


Figure 8-18 Simple Industrial Power Cogeneration Network

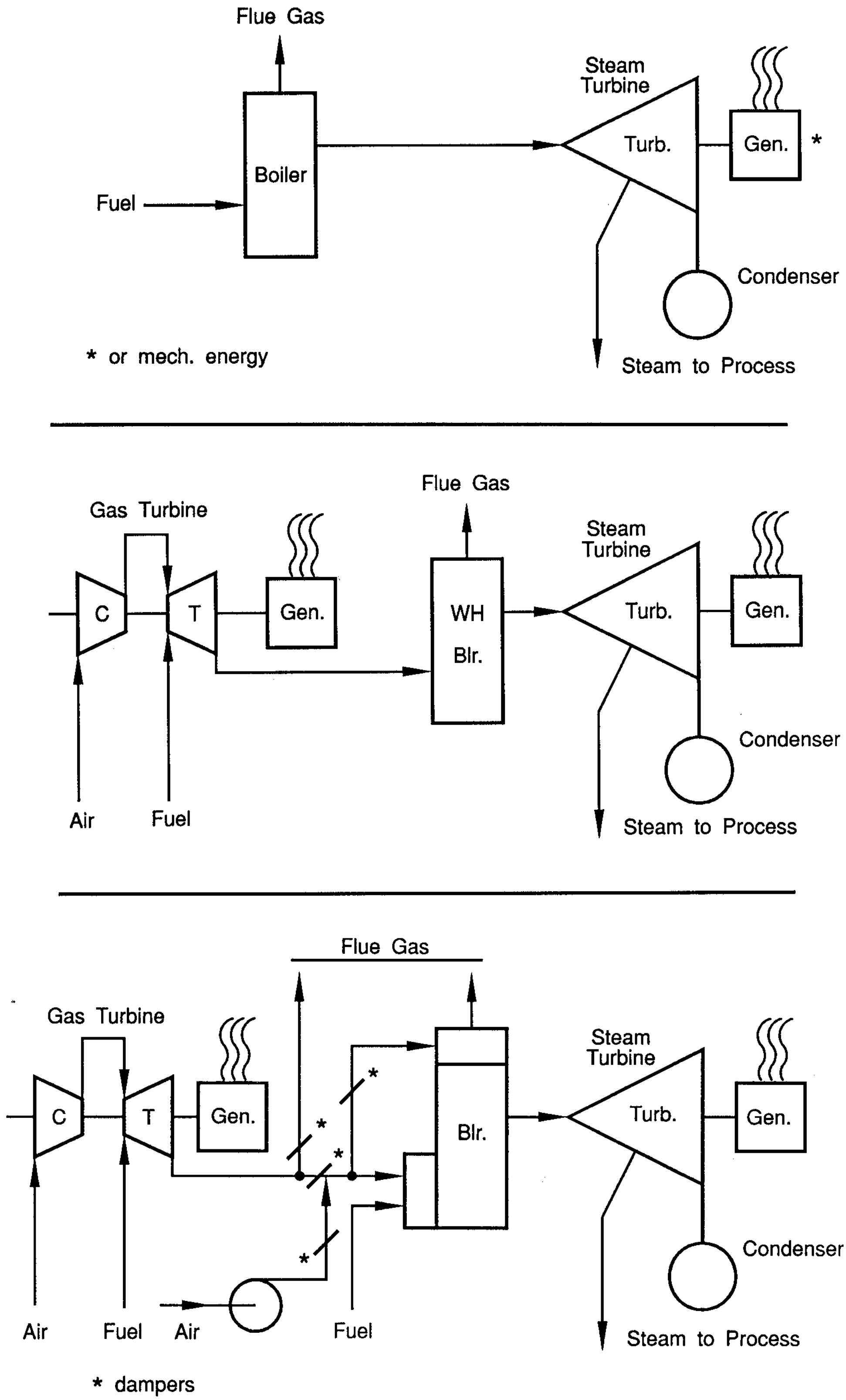


Figure 8-19 Typical Cogeneration Arrangements

lower pressure header even though the boilers may be operating at different pressures. In such a case, the steam loads on the two headers can be completely independent of each other while receiving their firing rate demand from a controller on only one header.

An electrical generation load change can be satisfied by the backpressure turbine. A steam load on the lower pressure header can be satisfied by either the backpressure turbine, the backpressure control valve, or the firing of the low pressure boilers.

The pressure on the higher pressure boilers is maintained at a set point by the backpressure controller. The pressure on the lower pressure boilers is controlled by the master controller connected to their header. A change in steam load on the higher pressure header causes a pressure change on this header and a change in the steam transfer to the lower pressure steam header. The result is a change in steam pressure on the lower pressure header and appropriate action to change the firing rate on all boilers.

More complex networks may include more pressure levels, more fired boilers, waste heat boilers, controlled extraction and condensing turbines, and gas turbines. Figure 8-19 shows several arrangements whereby process heat and electric power generation are combined.

A large, complex cogeneration network offers very significant opportunities for fuel savings through economic loading of the various elements. In most cases these systems have some sort of electric connection or tie line with an electric utility system. Short-term demand in excess of the contract may make the utility power very expensive. With this and other cost factors relating to "make" versus "buy," the "in-house" electric generation can be controlled on a least-cost basis. The analysis usually requires some form of digital processor.

Once that decision on electric generation has been made, the next decision is the allocation of the electric generation between units. This involves computations that include the incremental cost of producing the power on each power generation unit. The result is used for assigning the steam load to each turbogenerator on a least-cost basis. The next decision involves the allocation of the process heat load to the higher or lower pressure boilers, followed by the decision on how the boilers on each header are to be loaded.

These calculations can usually be made as a part of a distributed microprocessor-based control system or, alternately by a computer that acts in a supervisory fashion. In either case, the control level of the specific energy conversion device can be biased or multiplied upward or downward. The interface between the computer and the basic control system can be as shown in Figure 8-15.

Section 9
Firing Rate Demand for Utility Boilers

The control systems structure for electric utility plant individual units can be represented by Figure 9-1. This hierarchial diagram of the combined operator and automation function shows how all the various control actions fit together in the overall scheme.

The unit management takes instruction from the load requirement much as the operating officers of a company take instruction from the board of directors and owners. Unit management then gives instruction down through the line organization. Such instruction is the control action. Simultaneously, data, capability, and status are transmitted upward in order that the instructions do not exceed the capability of the overall system to perform. In this section, the unit management, boiler management, and turbine management functions are developed.

Industrial boilers are almost always connected to a steam header with the various heat (steam carrier) loads connected to the header. The header acts to multiplex the various steam loads to the steam boilers.

Unlike industrial boilers, almost all electric utility boilers that have been installed in the last 50 years are unit systems with a single boiler connected to a single turbogenerator. Utility plants usually have more than one such unit. The turbogenerators in a plant are connected electrically to a common bus, with this bus as the load distributing point for the plant. The desired electrical load on the plant and on the individual units is usually determined at a remote power-dispatching center.

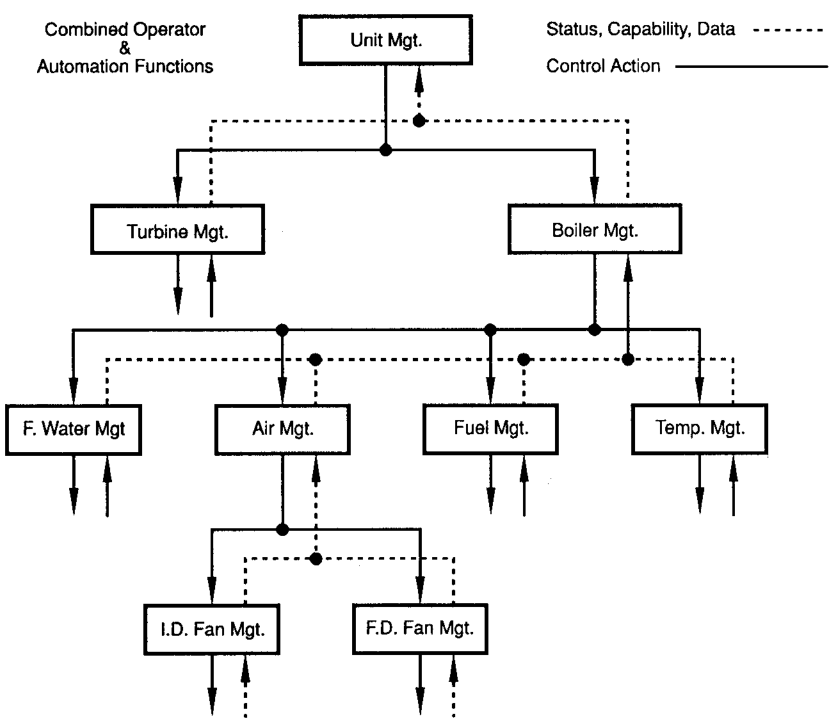


Figure 9-1 Hierarchical Arrangement of Plant Control

9-1 Matching Firing Rate Demand to Electrical Load (Boiler-Turbine Coordination)

The energy balance for an individual boiler-turbine unit must always be satisfied. This balance decrees that energy input from the fuel and air is always equal to the energy output plus or minus the change in energy storage. The energy output consists of (1) energy in the boiler flue gases; (2) of a small, relatively constant percentage of heat losses in the turbine and generator; (3) the electrical production from the generator; and (4) the heat loss to the condenser cooling water. The heat losses of the turbogenerator are fallouts that cannot be controlled on any dynamic real-time basis. A general arrangement of the turbine, condenser, and an extraction feedwater heater is shown in Figure 9-2.

The total energy balance of a unit boiler-turbine system must be kept stable by the boiler control system. A version of such a heat balance is shown in Figure 9-3. The system is so interractive that changing any value on this diagram will result in every other value changing as the operating parameters are again balanced. A diagram of a reheat turbine flow path is shown in Figure 9-4.

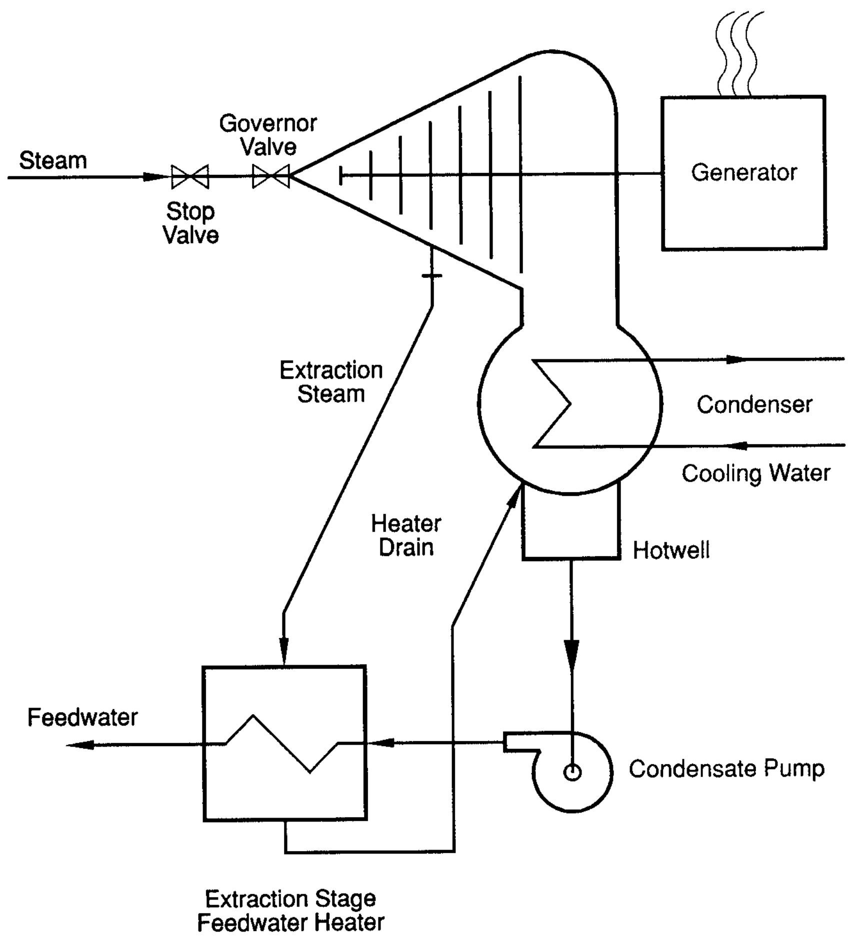


Figure 9-2 Typical Arrangement of a Condensing Turbine with Uncontrolled Extraction

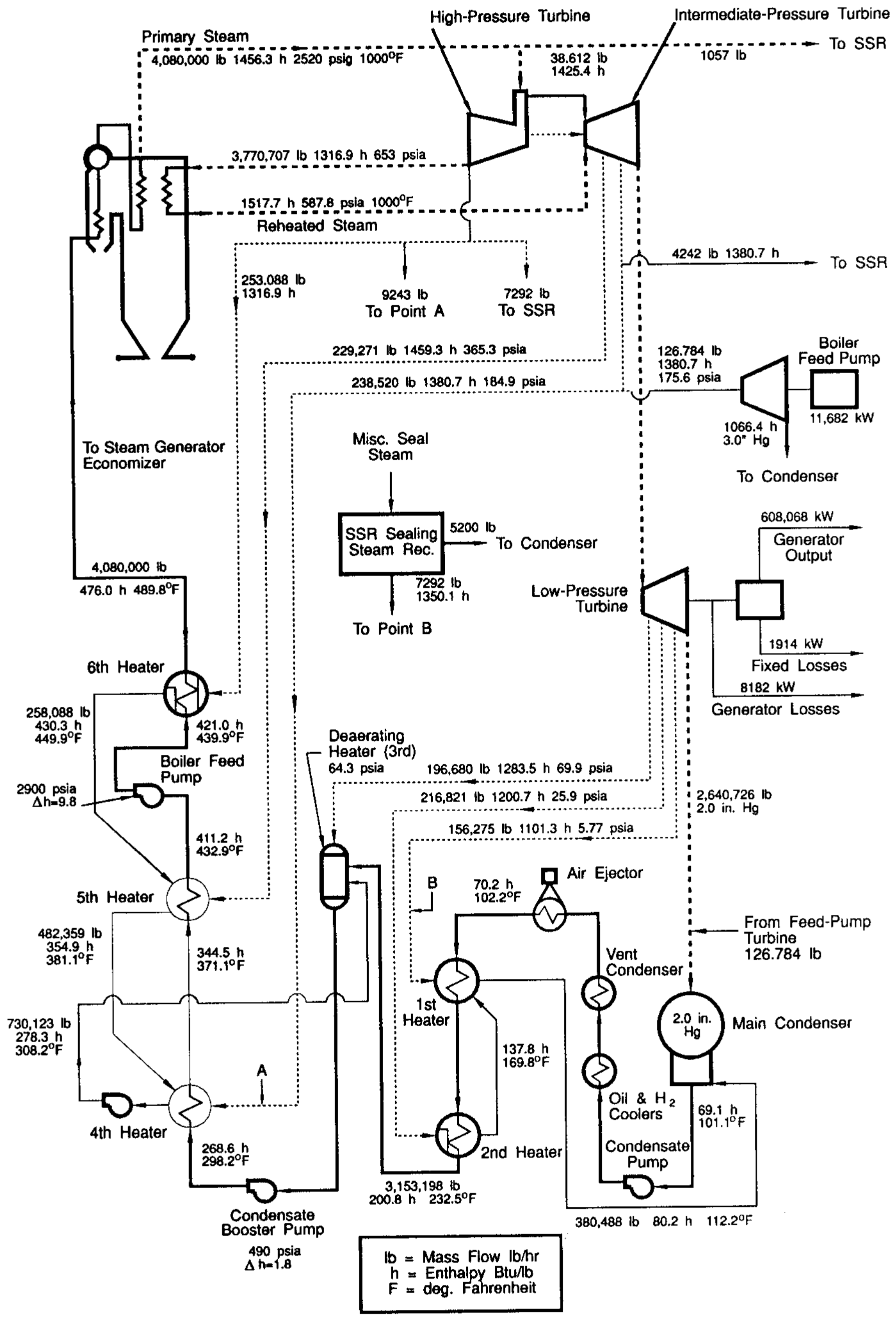


Figure 9-3 Boiler-Turbine Heat Balance for Electric Utility Unit

(From *Fossil Power Systems*, © Combustion Engineering, Inc.)

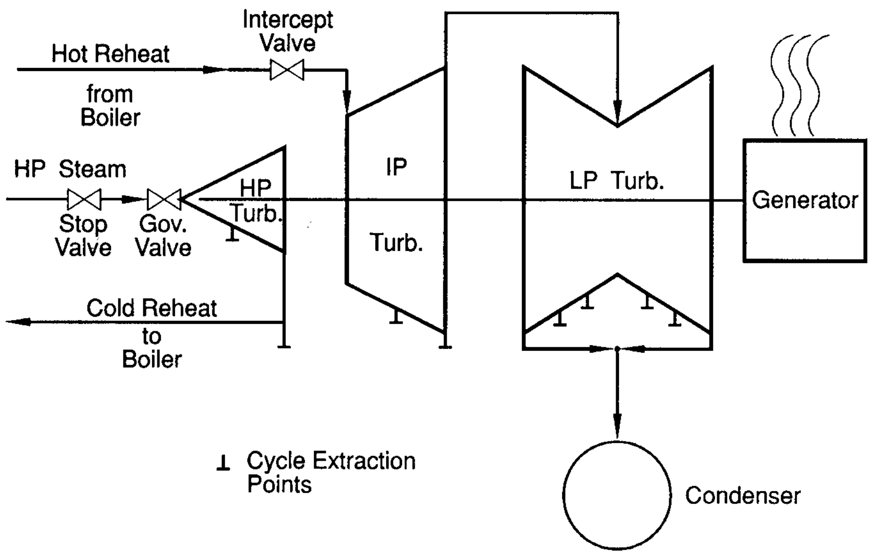


Figure 9-4 Typical Arrangement of Reheat Turbine

9-2 Boiler Load Measurement

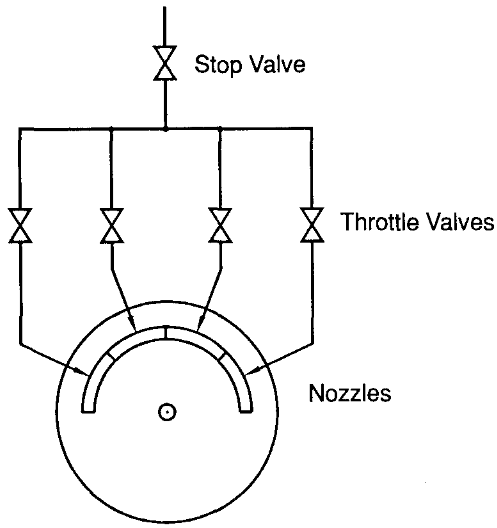
Though not precisely correct, the steam flow from an industrial boiler is assumed to be the heat load on that boiler. To be more precise, this value would require adjustments based on feedwater temperature, blowdown flow, rate of change of pressure, and radiation effects. Unlike an industrial boiler, it is normally not practical on a large utility boiler to measure the steam flow with the conventional combination of orifice or flow nozzle primary element and differential pressure-measuring secondary element. The capacities, pipe sizes, pipe wall thickness, and pressures preclude this in modern day units.

The general arrangement of the throttle valves on a electric utility-type (condensing) turbine is shown in Figure 9-5. In order to minimize the thermodynamic loss due to "throttling," the valves are normally opened sequentially and are individually designed for partial load capacity. They feed individual steam nozzles, which are physically arranged in an arc. At the lowest load the steam is directed at one point on the radius of the turbine wheel; at full load, the steam is directed at multiple points. A common number of these valves and nozzles is four. Until all the valves are open or partly opened, the admission of steam is called "partial arc." With all valves opened is it called "full arc."

Because of the multiple valve arrangement, the nozzle pressure cannot be used for steam flow measurement. The alternative steam flow measurement is the use of turbine stage pressures to measure steam flow.

Figure 9-6 demonstrates that, as the steam flows from the turbine throttle and through the turbine to the condenser, it is analogous to flowing through a series of critical velocity nozzles. Flow through a critical velocity nozzle varies directly with the absolute upstream pressure. By using a curve supplied by the turbine manufacturer, steam flow to a turbogenerator can thus be determined by measuring the first-stage shell pressure. The relationship between energy flow to the turbine and first stage shell pressure is nearly a straight line, (it deviates from a straight line by approximately 5% at the midpoint). Similarly, the flow through any stage of the turbine can be determined by measuring the pressure ahead of that stage.

The specific relationship varies with the pressure-temperature (specific volume and energy content) relationship. The pressure measurement is thus related directly to energy flow. If this energy flow measurement is used to control energy (i.e., fuel and air), a direct measurement



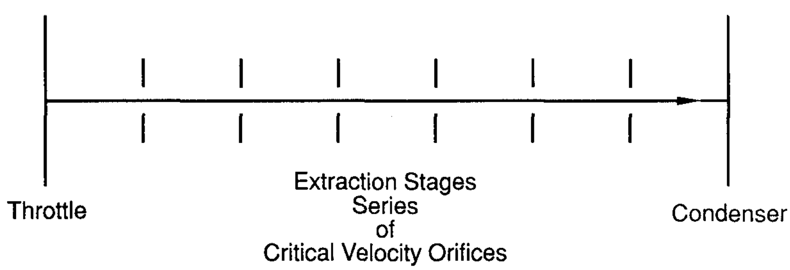
Full Arc - All throttle valves

Partial Arc - At least one throttle valve closed

Throttle valves may be programmed to operate in any desired sequence or in unison

Figure 9-5 Turbine Governor Valve Operation

Turbine Analogy



Stage Pressure - Essentially straight line relationship to flow following that stage

Pressure Ratios - Essentially constant over load range

- May be exception for controlled extraction turbine if ΔP goes subcritical

Critical pressure - flow proportional to upstream pressure

Subcritical pressure - flow proportional to $\sqrt{\ }$ of ΔP

Figure 9-6 Diagram of a Simple System Simulating the Flow Characteristics of a Multistage Turbine

(From *Steam Turbines and Their Cycles*, © J. Kenneth Salisbury) of pressure should be used. If, however, the measurement is used in a mass flow control loop (i.e., feedwater flow), it should be corrected for variations due to deviation from design temperature.

The firing rate demand signal is the primary control system signal that attempts to match the heat input of the fuel and combustion air to the useful output (electrical load on the turbogenerator). A mismatch results in a higher or lower than desired boiler steam pressure or steam temperature. If an initial control signal calculation nearly matches input to outout, any remaining errors from the desired values of pressure and temperature are used as inputs to intregral control action. Such control trimming biases or multiplies the basic firing rate demand (fuel and air demand) control signal and returns the fuel and air to the desired values.

It is necessary that these variables be closely controlled. A variation in steam pressure causes the turbogenerator to manufacture more or less electrical load than desired. Since the electricity cannot be stored, any amount of undesirable generation interracts with other electrical generating units, causing a change in their generation.

A variation in the steam temperature may cause unequal expansion and contraction in the turbine parts. Rapid and excessive changes in temperature can result in damage to the turbine. Steam temperatures that are significantly higher than design can shorten the life of the turbine metal parts. Such temperature variations also cause a change in the unit electrical generation.

Various factors may cause the relationship between energy input and electricity generated to change. The load on the unit is one factor. Figure 9-7 shows the shape of a design turbine heat rate (efficiency) curve related to load. Deviations to this curve result from off-design operating factors such as condenser backpressure, extraction loads, turbine engine efficiencies, and inlet pressures and temperatures.

A requirement is that the electrical load of any user must be immediately satisfied. As the user's load changes, more or less electrical load on the generator causes the turbogenerator to change speed. The turbine governor immediately changes the turbine steam control valve position to provide more or less steam to the turbine. This causes the boiler steam output pressure to change. Since the turbine speed is changed marginally, the frequency of the electrical generation system is also changed.

Demand for load changes are met by slightly shifting the speed setting of the proportional control turbine governor in order to obtain the desired incremental changes to the governor valve positions. In some cases, coordinated boiler-turbine control systems move the valves directly. Except for minor load changes resulting from small frequency changes, load changes

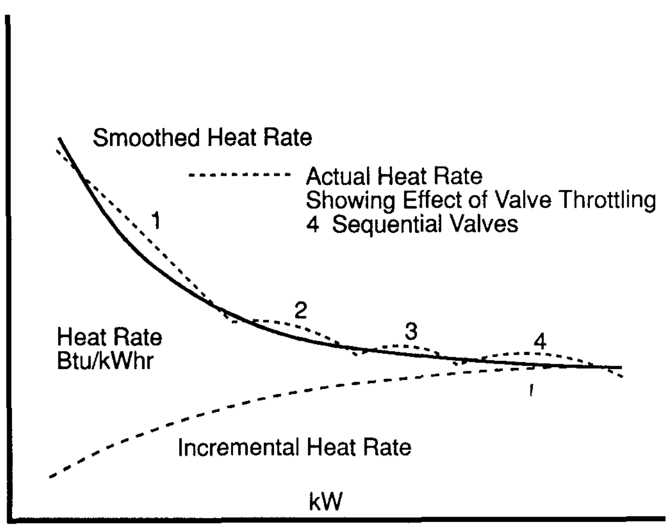


Figure 9-7 Turbine Heat Rate Curves

are initiated in two ways: the load is controlled by automatic signals from the remote dispatch center or manually from a plant control center. The rate of change of load and the magnitude of the total change are controlled.

The development of a firing rate demand control signal is an attempt to match firing rate to the energy demands of such changes. There are three basic methods for the development of the firing rate demand signal. One of these is called "boiler following." This method initiates the boiler firing rate demand based on the steam pressure at the turbine throttle. The boiler control system is thus decoupled from the control of the turbogenerator. Other methods are called turbine following, coordinated control (integrated master and direct energy balance—DEB, are particular manufacturer names). The coordinated control subsystem couples the boiler and turbine control systems in order to better coordinate the overall control action.

These methods use additional information such as turbine valve position, energy flow (first-stage pressure), frequency, and the dispatch center's requirement for electrical energy. In modern digital systems all of these subsystems may be software-installed in the same physical system. In this case they, or various versions that leave out certain inputs, may be programmed to be selectable by the control room operator.

9-3 Unit Load Demand Development

"Unit load demand" is a name often given to the control subsystem that develops the megawatt demand of the total system. The megawatt demand is an input to the turbine demand and the boiler demand. The unit load demand signal usually incorporates a maximum limit on rate of load change, load limits, and runback inputs. Runbacks are necessary to prevent demanding more of the boiler than its capability to produce. For example, a boiler feedwater pump failure or partial failure in the fuel or combustion air systems may cause load demand or firing rate demand to be limited.

Figure 9-8 is a block diagram of the development of the unit load demand control signal. Note that this shows an automatic system with initiating signals from the power system dispatcher. Items (a) and (b) are a means for the control board operator to switch the automatic control to manual. On manual the operator initiates the turbine governor speed adjustment. Items (c) and (d) permit switching the rate of change limit to manual if the operator desires to cut this limit out of the control strategy. The operator is then responsible for raising or lowering load at a rate that the boiler can be operated with pressures and temperatures at stable levels.

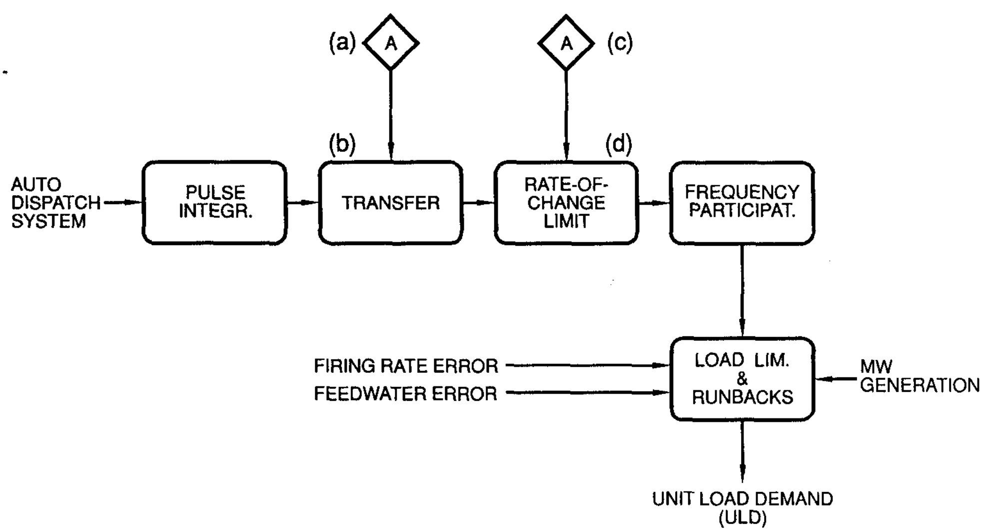


Figure 9-8 Unit Load Demand Development

The signal is then subjected to a frequency participation bias, since the frequency represesents a shift in the relationship of the proportional control governor valve set point and the required valve opening. This adjustment may not be required if the system positions the valves directly instead of through the turbine governor. The block labelled "load limit and runbacks" provides all the operation limits necessary to prevent the unit load demand signal from asking for an unrealistic load under all the various circumstances.

The logic interpretation of this block diagram can have many forms. It may of necessity have many interlocks, feedback, and tracking circuits to obtain compatibility of all the various elements and provide all the desired failure modes. One particular variation may be in the type of information sent and requested by the electric generation control signals that are received from the system dispatch center. For example, if pulses are used, what is their form or type? Is the desired generation value that is received already processed into digital engineering values? One such skeleton interpretation is shown in Figure 9-9.

In this case the electric generation control is in the form of pulses that have a specific length, frequency, and megawatt value. These are converted to a value within a 0-100% range. Item (a) provides a means of switching to manual and setting this value at a fixed point. Item (b) is a control center manual means of setting the ramp rate or rate of change. Item (c) selects this value or that of a high or low limit that has been reached. Item (d) or (e) limits the maximum or minimum of this value, depending on the number of circulating pumps or boiler feed pumps in service or other limits. Item (f) allows miscellaneous runbacks and rundowns to limit the maximum value of the load demand signal.

The basic unit load demand output is then an input into any or all of the boiler firing rate demand subsystems. Detail implementation of this logic would also include internal system interlocking and tracking functions. These are required to assure that all system values recognize blocked input values, provide for the proper feedbacks when a portion or all of the susbsystem is on manual control, provide for any desired switching functions, and generally provide safe action for all failure modes.

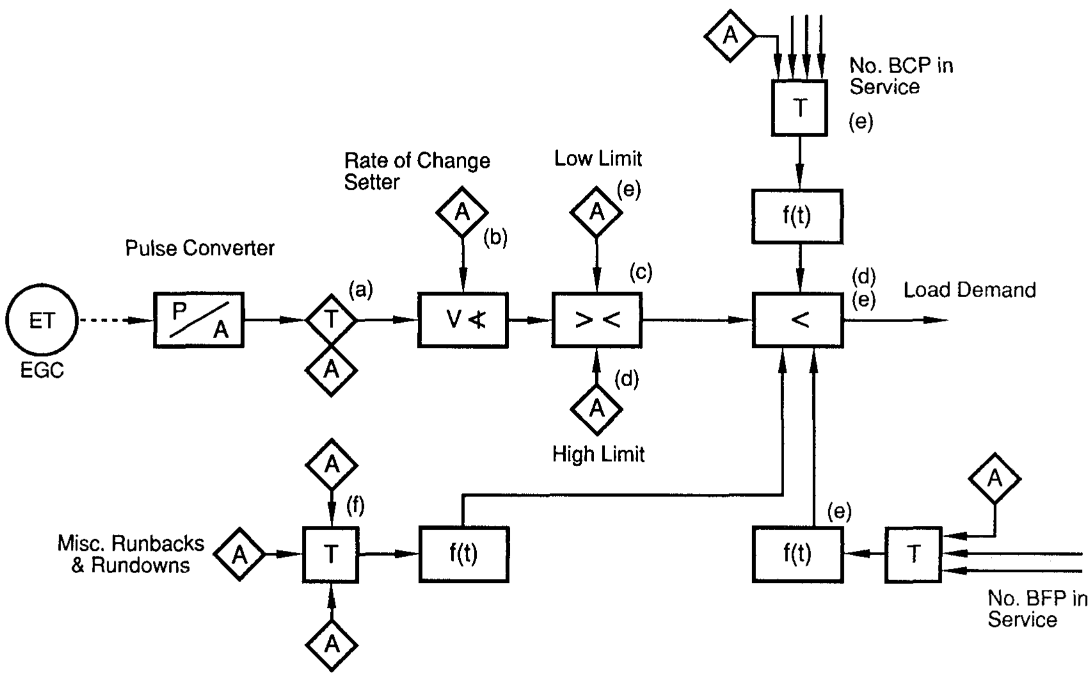


Figure 9-9 Unit Load Demand Development

9-4 Boiler Following — Firing Rate Demand Development

In boiler following control, the control systems for the boiler and turbine are separate and uncoupled. Starting with steady-state loading, any control system demand for more electric power is applied only to the turbogenerator. Figure 9-10 shows a block diagram of the control arrangement for boiler following control.

Either from additional load on the electrical system or from a remote demand signal, the

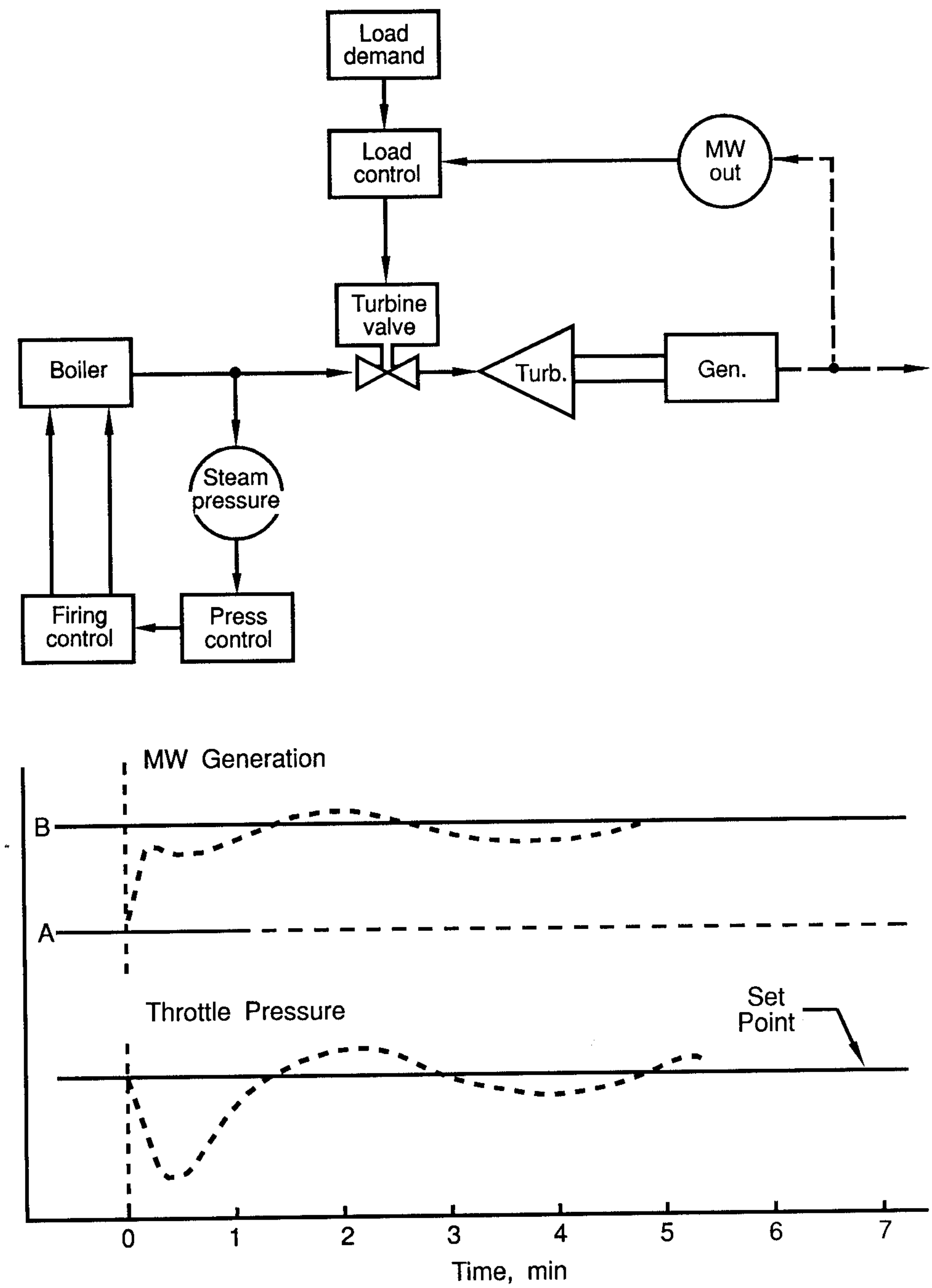


Figure 9-10 Boiler Following Firing Rate Control

turbine governor valves open. The result is that the turbine asks for additional energy input in the form of superheated steam.

Since the boiler was previously producing an amount of steam with a lower total energy level, the pressure will begin to drop. As the pressure drops, some steam will be produced due to the release of energy from boiler energy storage. The electrical megawatt response from this type of system is shown in the time-based curve of Figure 9-10. For a pulverized coal-fired unit, most of the additional MW generation in the first 30 seconds after the change in load demand is probably due to withdrawal of energy from boiler energy storage.

The drop in throttle pressure and the change in steam flow requirement activate the combustion control system to increase the firing rate of the boiler and bring the steam pressure back to its set point. Also from Figure 9-10, the effect of the increased firing rate starts to be effective about 20 to 40 seconds after the demand for additional electric generation. Stability, with pressure at set point and MW generation at the desired level, is not achieved until several minutes have passed. Of the various front-end control arrangements, boiler following is the most responsive to a change in electrical energy demand but also is the most unstable.

The turbine valves open immediately to produce the required additional generation although the pressure may be below set point. As the pressure starts to return to the set point, the electrical generation starts to increase above the desired level, causing the turbine steam valves to begin closing. This causes boiler pressure to increase. The instability of the boiler following scheme arises from this interaction between the turbine valve control and the boiler control. In addition, the steam pressure control may demand a greater or faster change than the boiler can meet.

The skeleton logic for the utility boiler "boiler following" is similar to the industrial boiler logic of Figures 8-7, 8-8, 8-9, and 8-10. Basic differences are the need for greater feedforward precision in a utility boiler control system and that there can be a positive feedback potential in the utility boiler system. The reason for greater feedforward precision is the critical nature of maintaining steam temperatures close to set point at all times. Unlike an industrial boiler, in a high-pressure utility boiler over half of the total energy added to the steam is in the superheat and reheat.

As shown in Figure 9-11, first-stage pressure is commonly used as a feedforward index.

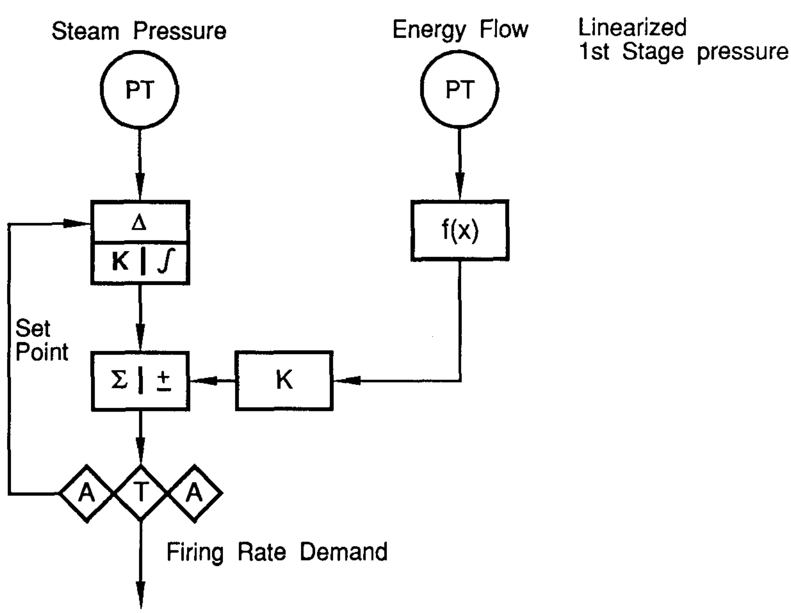


Figure 9-11 Boiler Following Control Logic Using First-Stage Pressure Feedforward

The first-stage pressure signal is approximately proportional to energy added to the turbine, and, therefore, is approximately proportional to the desired energy input of the boiler. It can be made more precise by adjusting the energy value of this signal for changes in feedwater temperature. In calibrating the system, this feedforward signal should be made equal to or greater than the firing rate demand needed for the steady-state steam energy requirement.

As noted, this signal is added to another signal developed from the steam pressure error. This signal modifies the basic feedforward signal to account for adjustments in energy storage of the boiler unit. Thus, on a load increase, first-stage pressure adjusts the firing rate for the new steady-state requirement. As throttle pressure drops and energy is withdrawn from storage, the firing rate is adjusted to a value that, within the pressure recovery time, pays back this energy withdrawal and fills the energy storage to the new required level.

The pressure feedback to this subsystem is pressure at the turbine throttle. Since a fixed set point reference at the turbine throttle is needed, the controller requires both proportional and integral actions. The integral action should be very slow and essentially should function

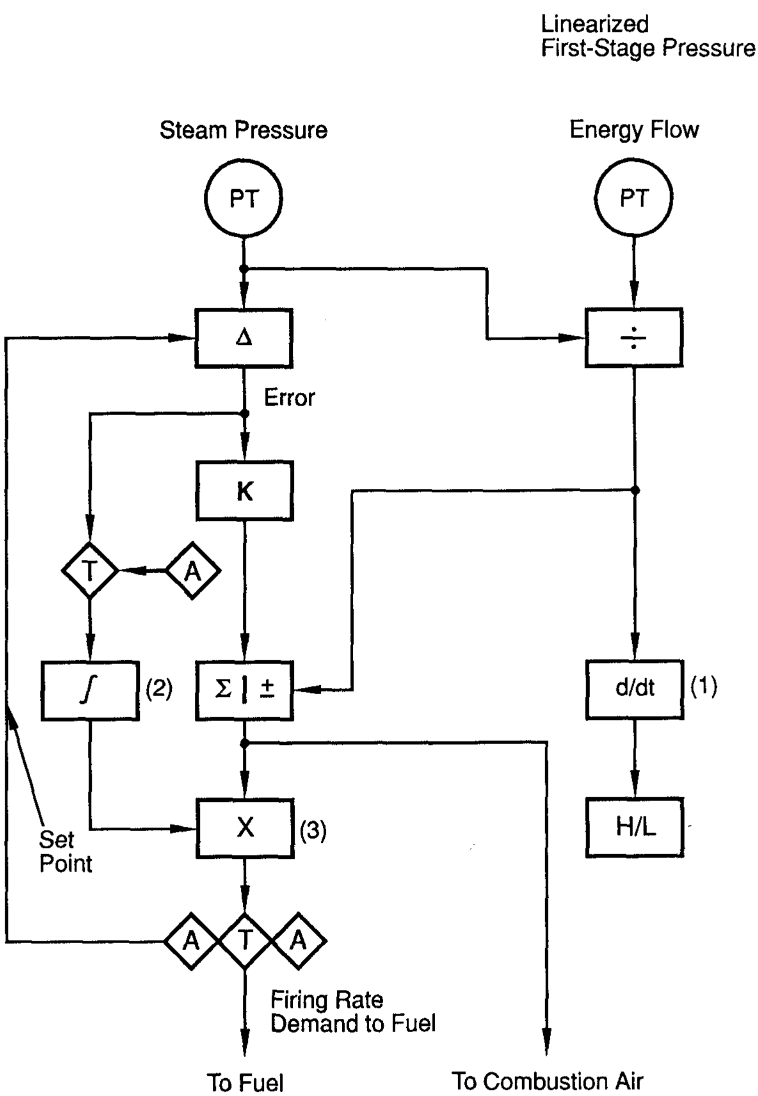


Figure 9-12 Boiler Following with Off-Line Fuel Calibrator

in theory only during steady-state operation in order to adjust for any mismatch between the feedforward demand and the fuel flow.

One arrangement of the boiler following subsystem breaks apart the proportional-plus-integral steam pressure controller as shown in Fig. 9-12. By having the integral portion drive a multiplier in the output, the boiler following subsystem can be used to compensate for fuel Btu variation. To ensure that the integral action functions only when the unit load is at a steady-state condition, the derivative action (1) notes when load is changing and blocks integral (2) action during a load change. This arrangement is based on an assumption that fuel quality changes gradually.

The combustion air requirement should not be affected by this continual calibration since combustion air should be added according to the Btu flow of the fuel. The portion of the firing rate demand that acts as the air flow demand should be taken off before the calibrating multiplier (3). This figure also shows a ratio of first-stage pressure to throttle pressure, which is discussed next.

As in other feedforward arrangements, the feedforward portion of the subsystem is providing the permanent output signal change. The integral value of the controller is necessary only for long-term deviation from set point and should be set very slow (less than 0.1 repeat per minute). Integral is slightly increased when feedforward gain is greater than 1.0.

The description above describes commonly used methods that produce satisfactory performance if the unit is on automatic dispatch control. There is, however, an inherent weakness in this and other such feedforward schemes. If the turbine is not on automatic dispatch control and the governor valves are in a fixed position, fuel supply variations can cause a temporary rise or fall in throttle pressure. This causes the first-stage pressure to change and the unit to pick up a proportionate amount of load. This increases the first-stage pressure a proportionate amount, which introduces positive feedback into the control loop. In order to eliminate this effect, a ratio of first-stage pressure to throttle pressure can be used instead of the first-stage pressure alone. In this way a pseudo turbine valve position signal is developed. Using this arrangement, no such positive feedback will result (see Figures 9-12 and 9-13).

With the computing resources of modern distributed control systems, the precision of the feedforward portion of the control can and should be further improved. The methods described

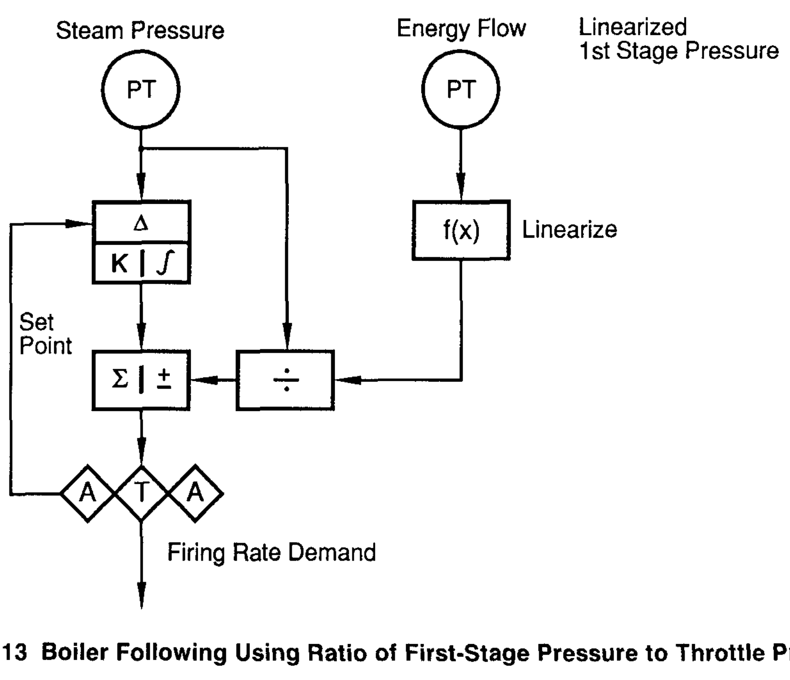


Figure 9-13 Boiler Following Using Ratio of First-Stage Pressure to Throttle Pressure

above, even though perfectly executed, may provide a feedforward signal one or two percent in error. Assume that the automatic fuel calibration described above, or other methods that will be described later, matches the firing rate demand signal to the feedforward signal. If the pressure and temperature are off set point when the load change demand occurs, the feedforward signal will provide fuel and air energy for those same off set point conditions after the change is completed. This then causes a greater than necessary corrective signal from the throttle pressure controller and, thus, greater throttle pressure deviations from set point. The goal should be a feedforward signal that will, if perfectly calibrated on an open-loop basis, fire the boiler at set point conditions after the load change.

An open-loop error from a one percent mismatch from the desired feedforward signal will produce 15 to 20°F open-loop error in steam temperature. The feedforward value should, therefore, be calculated as precisely as possible and be dynamically adjusted so that its open-loop application will result in steam conditions at the pressure and temperature set points.

The performance of the top feedwater heater, or the bypassing of the top heater, will cause feedwater temperature to vary. A drop in the feedwater temperature from the design value causes a greater firing rate on the boiler for the same electrical load.

A corrected feedforward signal can be developed by adding adjustments to the basic signal so that after change, conditions will be closer to the set point. To correct for steam temperature and reheat temperature error, a weighted effect can be obtained by a weighted summation of the two multiplied by turbine input energy. This is shown by items (f), (g), and (h) of Figure 9-14.

The feedforward signal should also be adjusted for the effect of changes in economizer inlet temperature. A change in feedwater temperature requires a change in firing rate because the total heat added per pound of steam changes. This effect can be properly accounted for by the development of feedwater temperature error in items (c) and (d), multiplying by the turbine input energy, and adding to the feedforward signal in item (b). The turbine energy input, (first-stage pressure), should be ratioed to the throttle pressure to avoid the positive feedback described above.

While this arrangement may appear complex, it is very straightforward and logical, and once installed in the software it can be continuously computed in the typical digital distributed control system. The compensations can also be performed in an analog system. Its use in analog systems may be of marginal benefit. An order of magnitude of the adjustment from off set point error of 15 degrees in superheat or reheat is 1 percent. The order of magnitude of the adjustment for an error in calibrated economizer inlet temperature is 2 percent for each 20 degrees. The degradation of the string accuracy for the several elements in series may use up the benefit of performing these calculations. This is not a factor in a digital system.

The MW production or MW demand rate can also be used to develop the feedforward of the firing rate demand signal. A review of Figure 9-7 shows that the steady-state relationship between energy input to the turbine and MW ouput of the turbine is not linear. Accordingly, if the MW rate is used for the feedforward, a function generator (1) should be used for the purpose of linearizing the MW signal relative to the boiler firing rate. Note that the use of the MW signal does not require adjustment for changes in feedwater temperature. The feedwater temperature changes as a result of a change in the withdrawal of turbine extraction steam with a simultaneous change in the MW generation rate for the same first-stage energy flow. Using MW rate as the feedforward signal for firing rate has the same positive feedback potential as using first-stage pressure. By ratioing the function of MW to throttle pressure as shown in Figure 9-15, the positive feedback can be eliminated. It may be necessary to periodically recalibrate the MW signal if turbine or condenser performance deteriorates.

A somewhat different boiler following control strategy is shown in Figure 9-16. In this arrangement the firing rate demand is a computed value based on boiler-turbine relationships. While the author has had no experience in using this type of control, it appears to offer the "boiler following" responsiveness advantage of other boiler following arrangements but with-

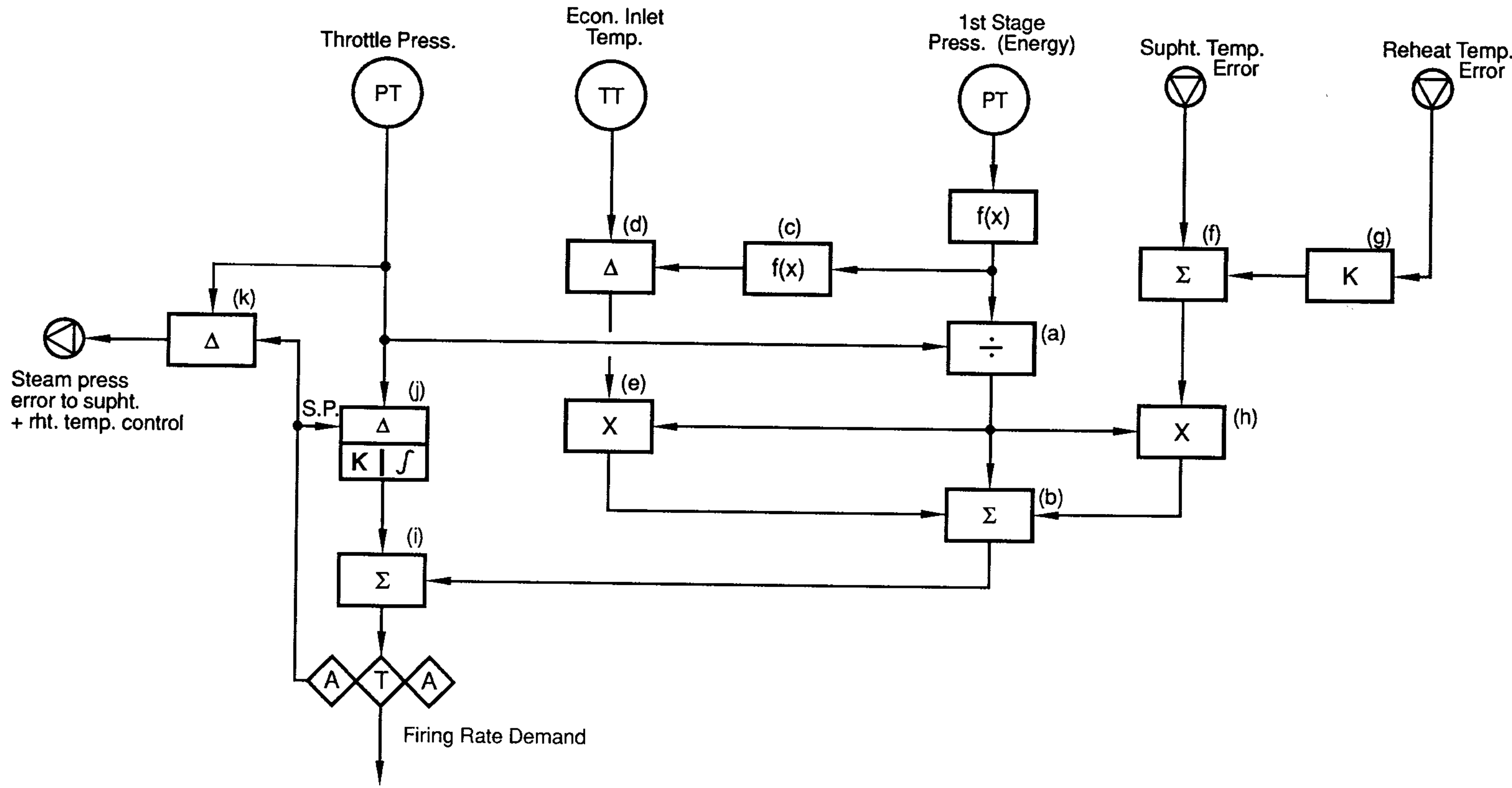


Figure 9-14 Boiler Following with Compensated Feedforward

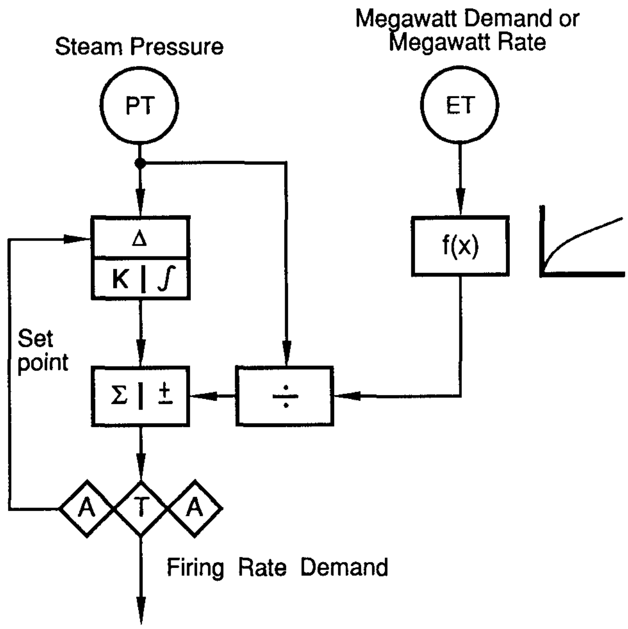


Figure 9-15 Megawatt Rate as Feedforward for Firing Rate Demand

out the instability factor inherent with other boiler following arrangements. If this can be documented, a time-based generation curve should show stability after approximately 2-3 minutes.

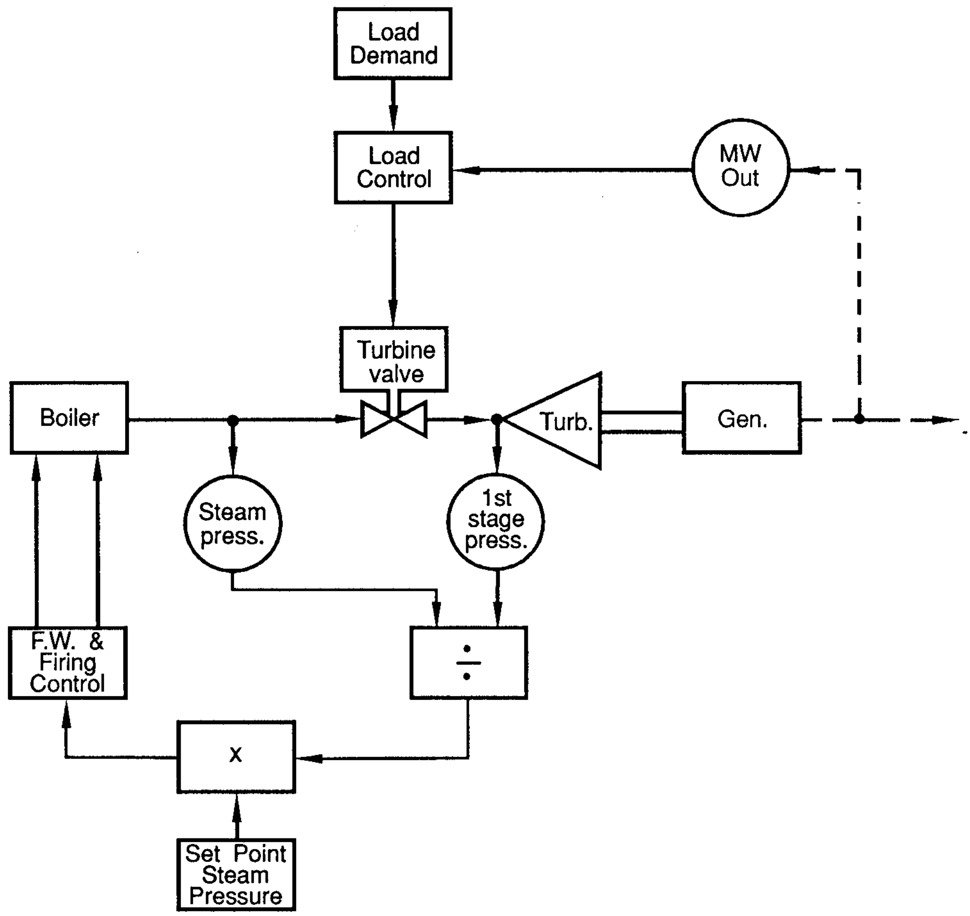
As shown in Figure 9-16(A), the turbine first-stage pressure (input energy) is divided by the actual turbine throttle pressure to develop a linear pseudo turbine valve position signal. This signal is then multiplied by the turbine throttle set point to develop the linear basic firing rate demand signal. If the actual pressure is lower than set point, the calculation produces a firing rate demand that is higher than the calculated pseudo turbine valve position. The opposite is true if the actual pressure is higher than the set point.

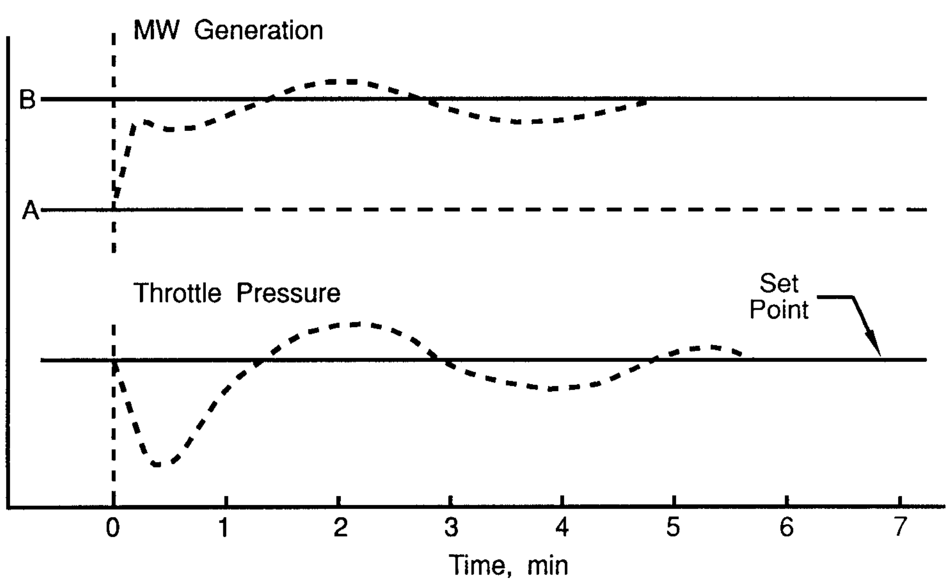
As originally conceived, there was no steam pressure controller with this scheme. The author believes that the system would be improved if a steam pressure controller were added so that turbine throttle pressure would always return precisely to set point. Such a controller should be tuned for a relatively low gain and low integral value to avoid adding any instability to the system. The basic scheme is represented in the block diagram of Figure 9-16(A) and the control logic diagram of Figure 9-16(B). Also as originally conceived, derivative action was added to the firing rate demand to account for any additional over- and underfiring that might be required.

The firing rate demand signal is sometimes developed with single-element feedback control. A variation that has been successfully used in some cases is shown in Figure 9-17. In this arrangement the duty of maintaining the steady-state load is entirely allocated to the turbine throttle pressure control. For the overfiring and underfiring on load increase and decrease, a derivative of the steam flow rate is used. As the load increases and decreases, the derivative block and the proportional block shown are used to calibrate the amount of, and time duration of, the required over- and underfiring.

9-5 Turbine Following — Throttle Pressure Control with the Turbine Valves

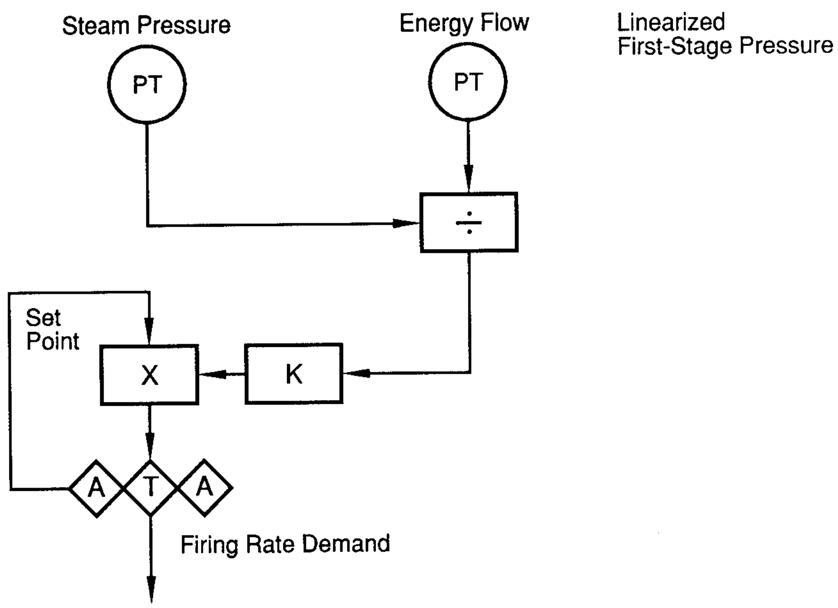
As previously stated, the slowest but most stable arrangement is "turbine following" development of the firing rate demand. The block diagram in Figure 9-18 shows the turbine





(A) Computed Firing Rate Demand, Block Diagram

Figure 9-16 Variation of Boiler Following



Feedforward-plus-Feedback Master Control

(B) Computed Firing Rate Demand, Control Logic Diagram

Figure 9-16 (*Continued*)

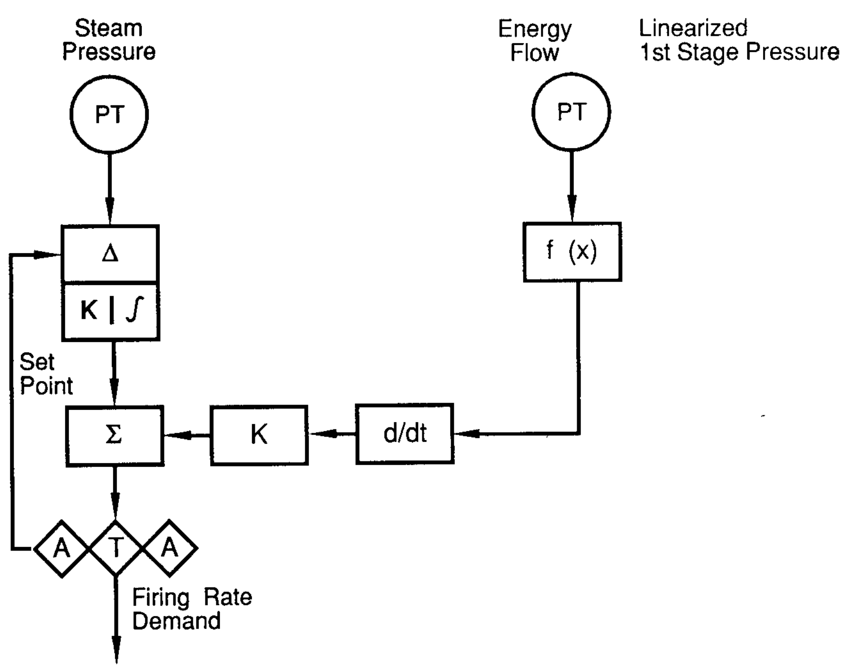


Figure 9-17 Boiler Following Using Feedback Control with Load Change-Based Overfiring and Underfiring

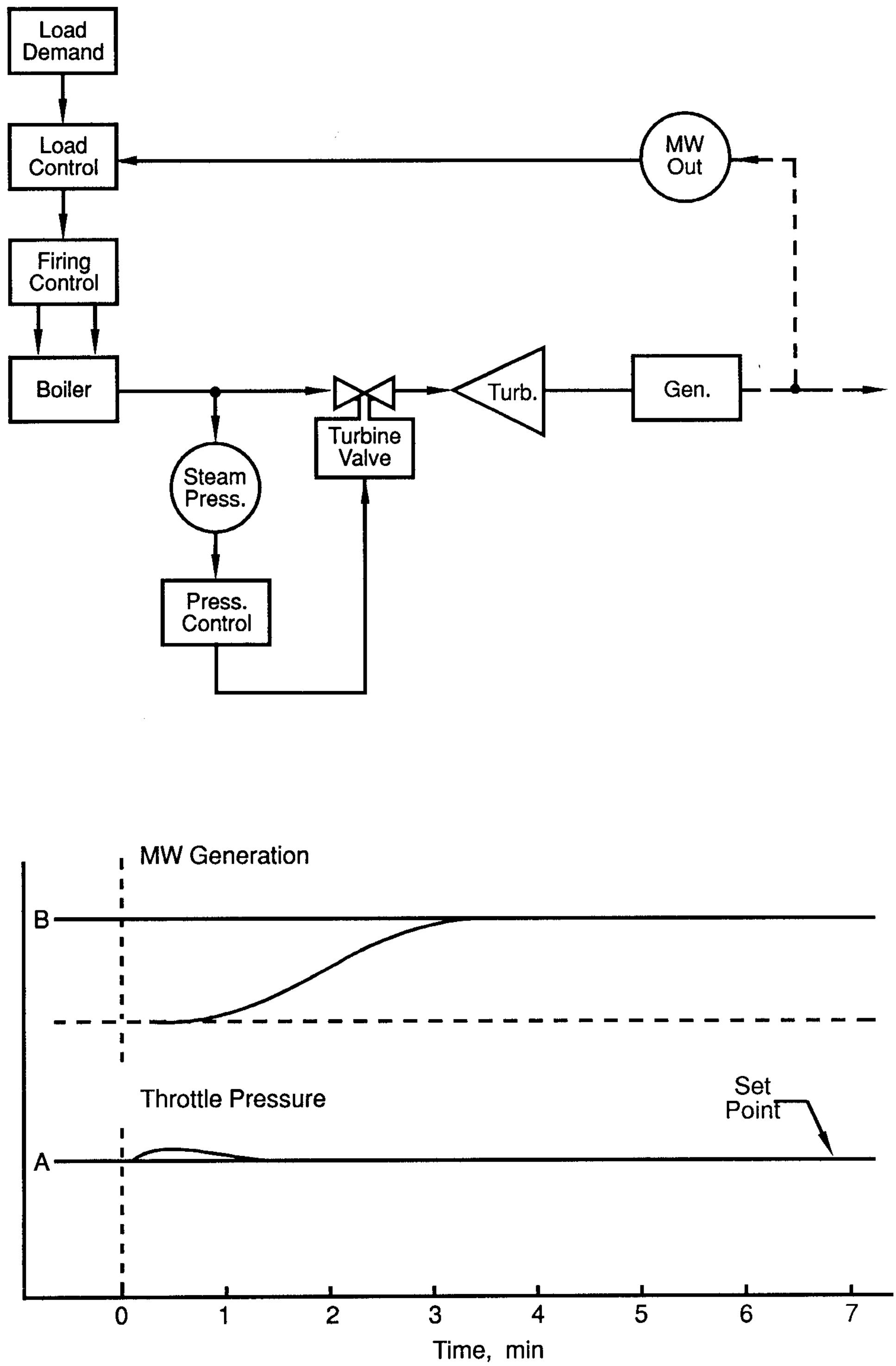


Figure 9-18 Turbine Following Firing Rate Control

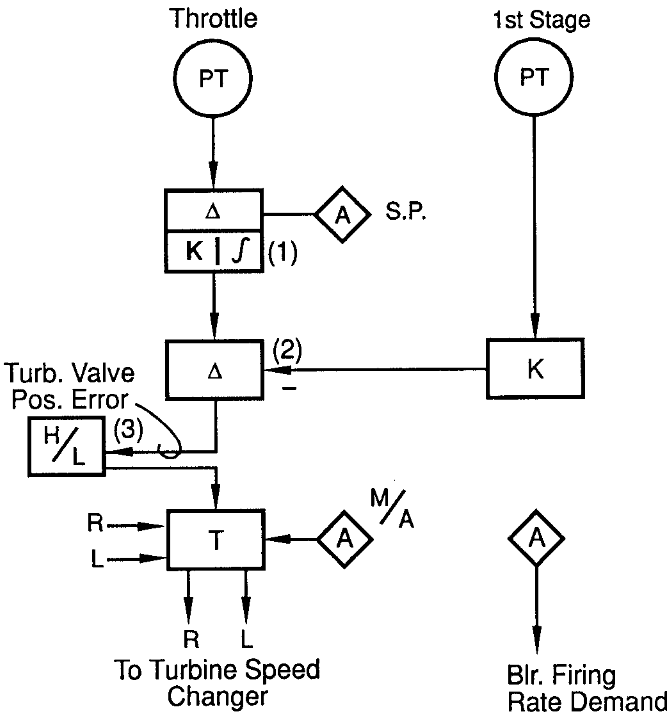


Figure 9-19 Turbine Following Firing Rate Control

following arrangement. The demand for an increased MW rate is used to increase the firing rate. As the additional steam energy is obtained from the increased firing rate, the throttle steam pressure tends to rise. This causes the turbine throttle backpressure control to open the turbine valves. Only then is there any addition to the MW generation rate. This system does an excellent job of pressure control but does not allow the borrowing of energy from, or depositing of energy to, boiler energy storage during a load change. The system is thus denied the benefit of using boiler energy storage to assist in making load changes.

A basic implementation of turbine following control is shown in the control logic diagram of Figure 9-19. The turbine valves are controlled through the ''raise'' or ''lower'' switches of the turbine speed changer motor. An increase in throttle pressure causes the output of controller (1) to increase. The result is that the output of the difference block (2) increases, causing the (H) switch in the monitor (3) to make. The speed changer motor will operate, opening the turbine valves until the first-stage pressure into the difference block causes the output of that block to return to the neutral value, stopping the operation of the speed changer motor. The reverse occurs on a decrease in load. To provide time for the loop to be completed, it may be necessary to add a pulser before the speed changer motor.

9-6 Boiler — Turbine Coordinated Control

By coordinating the action of the boiler firing and the turbine valve action into a single subsystem, the firing rate demand control can be improved. As shown in the block diagram of Figure 9-20, the major portion of the responsiveness of the boiler following method and the additional stability of the turbine following scheme are thus combined.

The implementation of this is shown in the control logic diagram of Figure 9-21. In this arrangement the MW demand is applied to both the turbine valves and the firing rate demand as a feedforward signal. Item (1) straightens the relationship between the MW rate and the

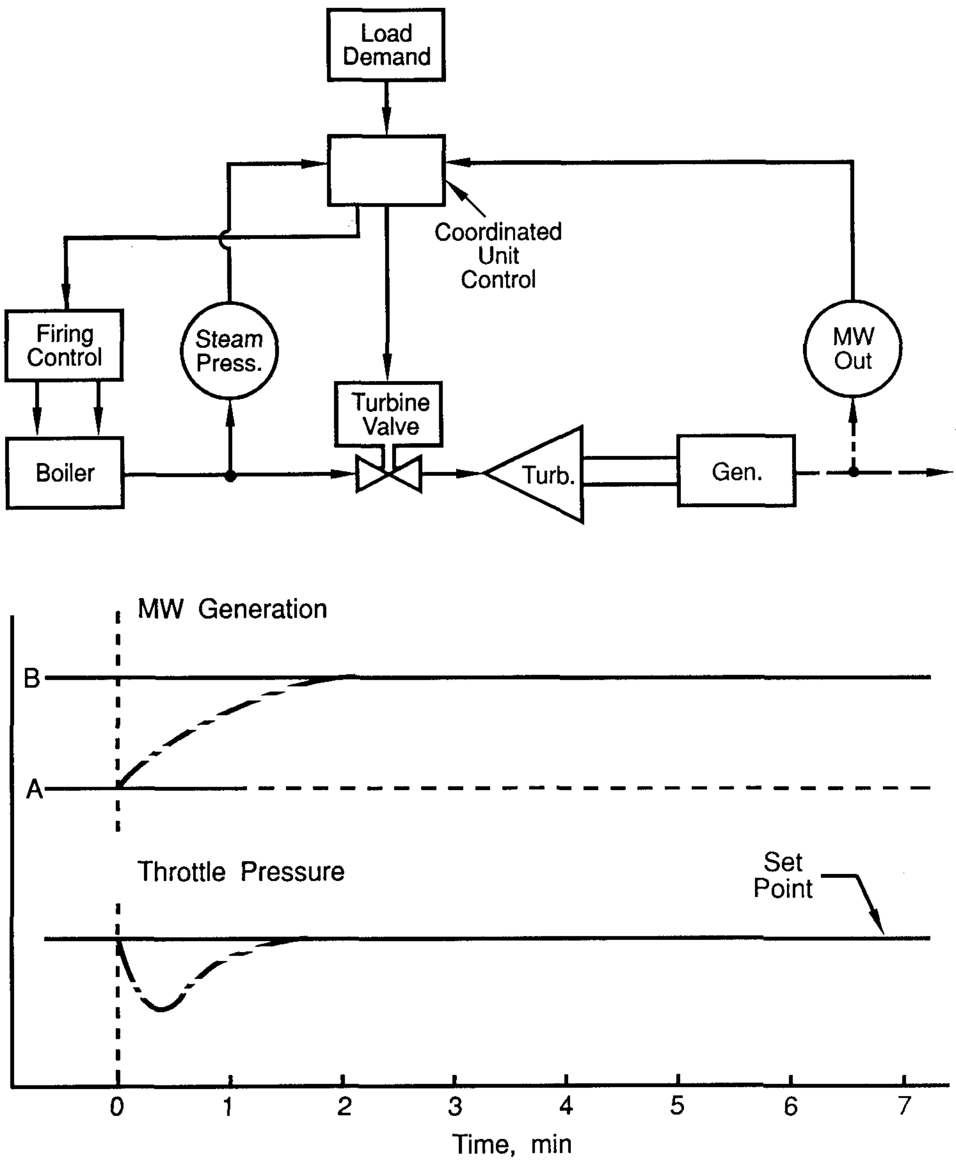


Figure 9-20 Boiler-Turbine Coordinated Firing Rate Control

energy flow to the turbine. Item (2) modifies the feedforward signal to account for any off-normal economizer inlet temperature that would cause the normal firing rate to be modified. Such a change would come from a change in the top heater performance or the removal of the heater from service. The sum of the MW error plus the steam pressure error is the total error input to the firing rate demand proportional-plus-integral controller (5), which is the final control of the firing rate demand. Firing rate demand is used to drive the total error to zero. The dfference between the steam pressure and MW errors is driven to zero by the turbine valve final controller (6). This system can be tuned to make a load increase more stable but less responsive, or vice versa.

Load can be picked up or dropped without stretching the boiler stability. In the end, a sustained load change in either direction can be handled more smoothly and at a faster rate. Typically, it stretches the capability of a boiler following system to pick up load faster than

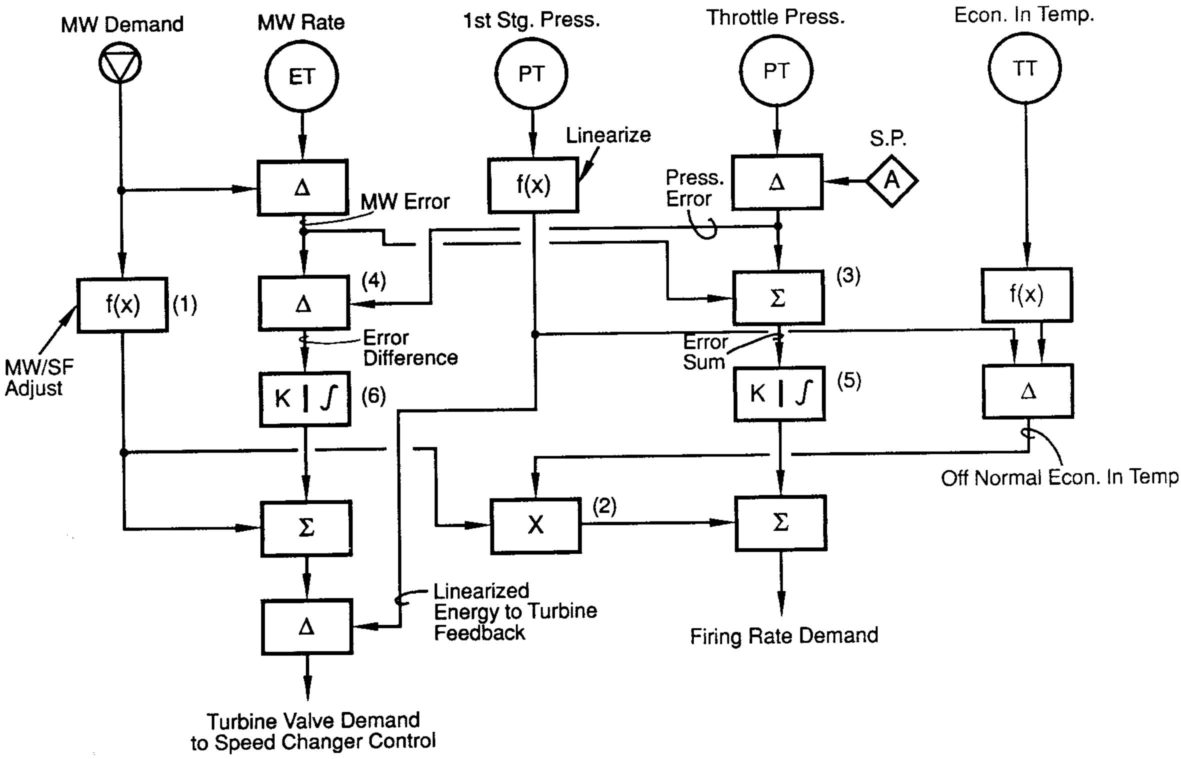


Figure 9-21 Boiler-Turbine Coordinated Firing Rate Control

approximately 2.5 percent per minute. Double this rate is not uncommon with coordinated control of the boiler and turbine.

9-7 Sliding or Variable Pressure Control

Units that are used for peaking loads or that may be subject to a daily wide variation in load are sometimes operated with a variable (sliding) pressure. In practice, the unit is operated at a considerably lower pressure when the load is lower and at a higher pressure at higher loads. This may be desirable for several reasons.

A particular boiler usually can attain full design steam temperature at lower loads if the pressure is reduced. The attainment of full steam temperature at lower loads results in a more constant overall temperature over a wider load range. This results in better temperature matching between the rotating and stationary metal parts in the turbine during load level changes. The result is an improvement in the ability of the turbine to change load more rapidly without turbine damage.

Another reason for variable pressure operation is improved thermal efficiency or a reduction in the heat rate from higher partial load steam temperatures. Lower pressure at partial load operation includes the ability to pack more energy into each pound of steam. Combined with this is a thermodynamic gain in energy conversion potential due to the elimination of a significant portion of the throttling losses across the turbine throttle valves. This is referred to in section 9-2. The result is more energy converted to work per pound of steam.

In addition to the above, the auxiliary power for pumping the feedwater is reduced. It is assumed that such a unit, along with the other present day units, would vary the feedwater flow by varying the speed of the boiler feed pump. As the boiler pressure is reduced, the required feedwater pressure is less and the pump speed can be reduced. This is important because the boiler feedwater pump is the largest single user of auxiliary power.

On older units with feedwater controlled by a valve and a constant speed pump, the auxiliary power is not reduced, and feedwater control valve duty is much more severe with variable pressure. When retrofitting such units to variable pressure operation, consideration should be given to changing to a variable speed pump.

Ideally, the maximum benefit of variable pressure operation would be obtained if the turbine were operated with all throttle valves wide open, full arc admission, and with the pressure adjusted to obtain the desired load output. This manner of operation would result in poor responsiveness to load change demands, even less so than with the turbine following mode.

The typical practice is to operate with all valves open at full load. As the load is reduced with the sequential throttle valve arrangement, the last valve opened begins to close. When that valve is closed, the turbine has all the regulation capability it needs. In addition, it does not have the throttling losses that it would have with that valve partially open. To obtain lower loads, the pressure is reduced only enough in each case to keep the one valve closed and the others open.

For mechanical reasons, there is a lower limit to load reduction with all valves except one open. At this point, the pressure is held constant and the load is further reduced by closing the next valve. A study of the thermodynamics of that unit can determine whether to close that valve and then reduce pressure further or to continue to hold pressure constant and close the next valves in line.

While there are these specific benefits, the control of the unit is more complex than that of a fixed pressure unit. If the unit is a drum-type boiler, the drum level measurement must be pressure compensated to obtain correct drum level measurements over a wide range of pressure. The control should normally be operated in the coordinated control mode. The throttle pressure set point is the output of a throttle pressure program that will provide the desired pressure profile, fixed set point, and ramp set point related to load.

9-8 Heat Rate Optimization with Sliding Pressure Control

During the 1970s a patent was issued that covered a special variation of sliding pressure control. This arrangement combined the turbine valve program with the pressure set point program. The purpose of this arrangement was to improve the heat rate as the load was being increased or decreased. It appears that the specific heat rate improvement will vary, depending on all the specific circumstances.

Assume that one or more of the turbine throttle valves is open, the load is low, and the throttle pressure set point is low. As the throttle pressure program increases the pressure set point to obtain more load, a condition will be reached requiring that an additional turbine throttle valve start to open. This produces a thermodynamic throttling loss. To avoid this, the turbine governor throttle valve program tells the throttle pressure set point program that this is starting to occur.

At this point the throttle pressure set point is reduced. In order to maintain the load, the throttle valve opens to the point that most of the throttling loss disappears. Additional load requirement is then obtained by holding the throttle valve in this position and increasing throttle pressure. Eventually, the throttle pressure available will require that the valve resume opening. When a requirement for an additional valve is signalled, pressure is reduced again so that the valve can open enough to eliminate most of the throttling loss.

9-9 Digital Interlock and Tracking Control Modes

In a control system, information flows from the sensors and is used in various computations. The results of these computations direct the operation of valves, dampers, fan or pump speeds, etc. The process measurements that result are measured by the sensors. In an electric utility boiler control system, there are a number of intermediate break points in the computations. Whenever such a break is exercised and a portion of the overall computation is being constrained, it is necessary that all upstream controllers track the values downstream of the breakpoints. Tracking is a form of interlocking within the system so that the overall system will at all times be ready for any manual or automatic switching action.

The first of these is the manual/auto station, for example, item (a) of Figure 9-9. If manual control is selected, the pulse converter output should be forced to the value of the manual signal. In this way the system can at any time be switched back to automatic with the switch input and output at the same value. When any of the limits, runbacks, rundowns, and so forth are limiting the load demand output, the pulse converter should track the load demand signal so that the system may smoothly leave the limit value as the limit-cause is eliminated. In the case of manual/auto switching, the switch operation can initiate the tracking mode operation. When the system output value is at one of the limits or constraints, the need for tracking can be initiated by monitoring a comparison of the output of the rate of change (b) and the load demand output of (e). In a total system there may be a hundred or more actions that cause some form of tracking. For example, any constraint on the fuel control should force all upstream control functions into some form of a tracking mode. Additionally, cross limits should cause contraints and tracking to occur in the combustion air flow system and all cross limits and upstream control from those points.

With an analog system, these signals are air pressures or voltages. The output of the pulse converter or other integrating device may be an air pressure held by a volume or a voltage held by a capacitor. Comprehensive tracking logic can be very complex in such systems. With a distributed digital system, the signals are digital values in memory that are changed to the desired tracking values.

The load demand signal on the right of Figure 9-9 is shown as ''MW Demand'' on Figure 9-21. The coordinated control shown here can be be blocked by assigning either the firing rate demand or the turbine valve demand to manual control. When this is done, the values on the firing rate demand and the turbine valve demand should ''track'' the manual control values.

This is done by forcing the integral values of controllers (5) and (6) on Figure 9-21 to the output of the manual control. In addition, to prevent the MW demand value from getting out of line, item (a) of Figure 9-9 should be put into a pseudomanual or tracking mode to prevent the development of undesirable values in that upstream subsystem.

The coordinated control of Figure 9-21 is only one of the three basic firing rate demand systems described. Many modern systems have all three and variations of these installed and interconnected in the same software system. Each of these is operator-selectable. Should a particular firing rate demand system be selected, the system interlocking and tracking subsystem should disconnect all other firing rate demand subsystems and put all elements in the mode to track the subsystem being used. The above example represents only a small portion of the necessary tracking interlocks of a complete electric utility boiler control system.

Section 10
Main Steam and Reheat Temperature Control

In Section 9, the use of reheat steam for electric utility boilers and turbogenerators was introduced. For the past 30 years, practically all installations of new electric utility boilers have included both superheat and reheat steam control. A basic control system would control only main steam temperature and use a single control mechanism. It would probably be applied to a smaller or older electric utility boiler or a large industrial boiler. Such boilers would probably not include steam reheaters. The following is a basic discussion of the nature of boiler superheat and the mechanisms used to control its temperature.

10-1 Temperature vs. Boiler Load

Unless a boiler making superheated steam is equipped with a control mechanism, the temperature of the superheated steam will vary, depending on various operating factors. Most industrial boilers that generate superheated steam do not include control mechanisms. Operating factors that may cause the temperature of superheated steam to vary are boiler load, steam pressure, excess combustion air, specific fuel burned, cleanliness of the heat transfer surface, and others.

In the characteristic of steam temperature vs. boiler load, the design of the superheater is particularly significant. The steam leaves the drum and enters the superheater. If this heat exchanger is located so that it can directly ''see'' the flame, it receives radiant heat from the flame and is called a radiant superheater. If the location is such that the superheater cannot ''see'' the flame and receives all its heat by convection, it is called a convection superheater.

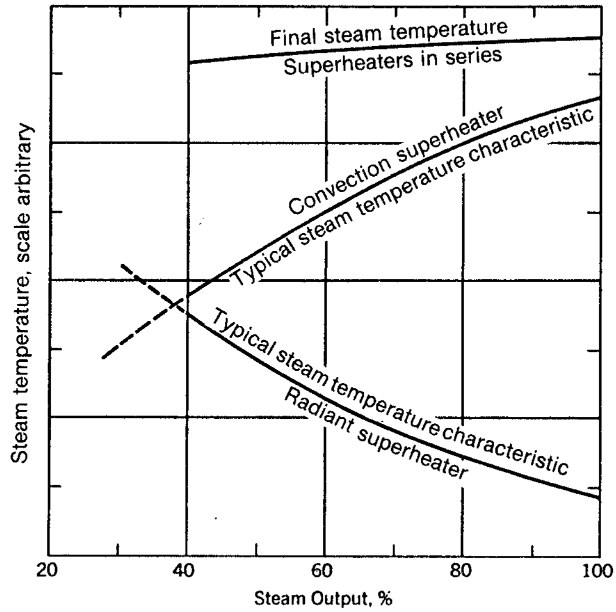
The opposite characteristics of steam temperature vs. firing rate for these two different types of superheaters are shown in Figure 10-1. Note that with a radiant superheater the temperature rises as steam output is decreased. This results from a relatively constant radiant input with a reduction of the ''cooling'' effect of the steam as load is decreased. With a convection superheater the opposite is true. The heat transferred rises as steam flow increases, but the rate of temperature increase becomes less as steam flow continues to increase.

By combining radiant and convection superheater elements in series, steam temperature changes a smaller amount as the steam load changes. This method has been used considerably by one boiler company in particular to achieve a more constant steam temperature. The advantage results from the more constant temperature that is obtained without the complexity and expense of resorting to control mechanisms.

When temperature is not controlled, the normal practice is to design for maximum temperature at full load, as shown in Figure 10-2, with temperature reducing as load is reduced. Since no control is involved, steam temperature is affected by the operational factors mentioned above. The type of fuel changes the flame temperature, producing a change in furnace heat absorption and thus a change in the temperature of flue gases entering the superheater. The cleanliness of the heat transfer surfaces changes their heat transfer coefficient, resulting in a change in heat absorption. A change in the amount of excess combustion air changes flue gas mass flow, resulting in a change in the heat transfer coefficient. An overall result is also a change in furnace outlet flue gas temperature.

10-2 Mechanisms for Control of Superheat Temperature

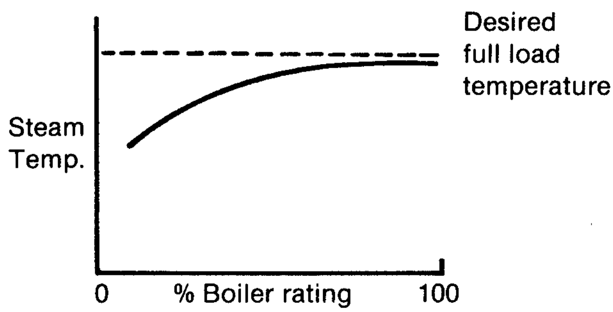
Typical industrial or electric utility boilers designed for the control of steam temperature may be capable of achieving full design steam temperature at 30 percent to 50 percent of full load steam output. Above this load point, a control mechanism is used to reduce the steam



A substantially uniform final steam temperature over a range of output can be attained by a series arrangement of radiant and convection superheater components.

Figure 10-1 Superheater Characteristics

(From *Steam, Its Generation and Use*, © Babcock & Wilcox)



- Temperature increases with higher gas temperatures across superheater for given steam flow (e.g., dirty boiler).
- Temperature increases with higher flue gas mass flow across superheater for given steam flow (e.g., higher excess air).
- Temperature entering superheater is a function of furnace temperature and furnace heat absorbed.

Figure 10-2 Uncontrolled Superheat (Typical)

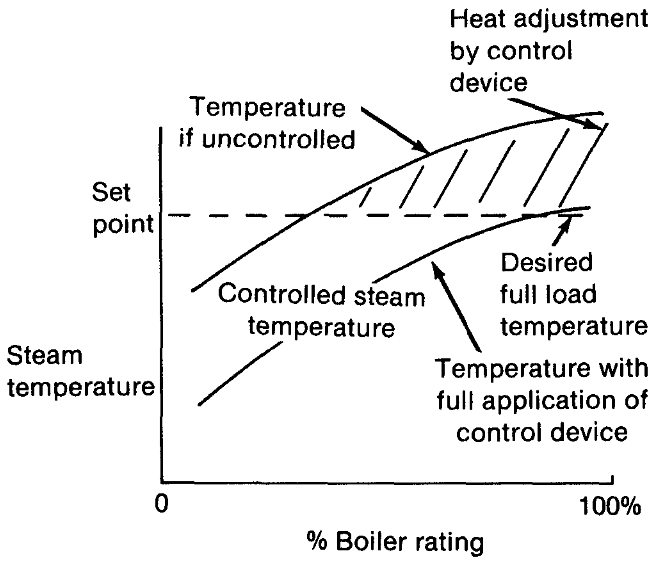


Figure 10-3 Controlled Superheat (Typical)

temperature as the load is increased. The result is that there is a possible family of curves of steam temperature vs. load for such a boiler, as shown in Figure 10-3.

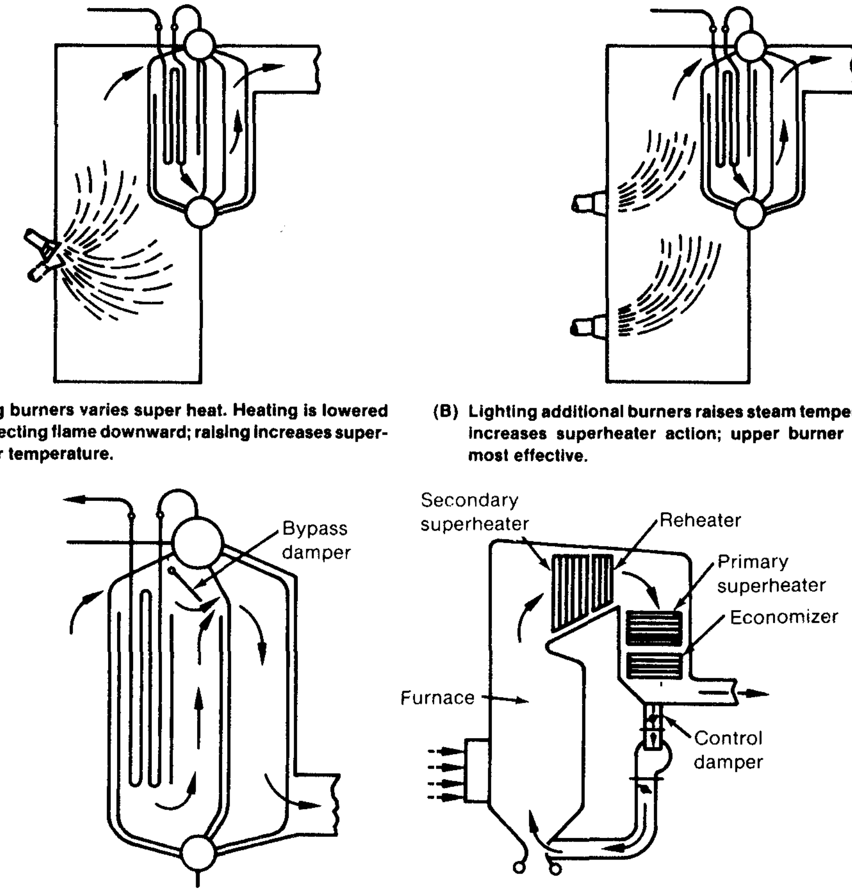
The primary purpose of the control mechanism is to adjust the superheating capacity as steam output changes. The boiler steam temperature is also affected, however, by the cleanliness factor, the fuel being fired, the imbalance between fuel Btu input and steam Btu output, and excess combustion air. The control mechanism must also have the capability of adjusting for these secondary influences in order that the boiler may be controlled at a constant, or other desired steam temperature.

The purpose of steam temperature control is usually to obtain as nearly as possible a constant superheat temperature at all boiler loads. The primary benefit in constant steam temperature is in improving the economy of conversion of heat to mechanical power. Control capability increases the lower load temperature, resulting in the potential for higher thermal efficiency of the power generation process.

In addition, maintaining a constant temperature minimizes unequal expansion or contraction due to unequal mass or material between the static and various rotating parts of power generation machines. This makes possible the use of smaller clearances and also results in higher thermal efficiency in the energy conversion process.

The control mechanisms that are operated by the steam temperature control equipment may involve the fire side or the water side of the boiler. The basic fire side mechanisms change either the temperature of the flue gases entering the superheater or the mass of the flue gases entering the superheater, or both. Different boiler manufacturers may use different methods.

Figure 10-4(A) demonstrates a method of changing the temperature of the flue gases entering the superheater. In this method, used by Combustion Engineering, Inc., burners mounted in the furnace corners are arranged so that the flame can be tilted up or down from horizontal. The burner flame is aimed at a tangent to an imaginary circle in the center of the furnace, and the burners in the four corners of the furnace are all tilted at the same angle. The result is a "fireball" in the center of the furnace, which rotates and which can be raised or lowered in the furnace by changing the tilt of the burners. Lowering the fireball increases furnace heat absorption, which lowers the flue gas temperature as it enters the superheater. Raising the



(A) Tilting burners varies super heat. Heating is lowered by directing flame downward; raising increases superheater temperature.

(B) Lighting additional burners raises steam temperature, increases superheater action; upper burner row is most effective.

(C) Dampers bypass portion of combustion gases around superheater and reheater; are most effective in upper steaming range.

(D) Partial gas recirculation affects the overall furnace temperature and, as a result, influences the total heat absorption.

From *Power* magazine Special Report, "Steam Generation," by Rene J. Bender, Associate Editor, McGraw Hill, NY.

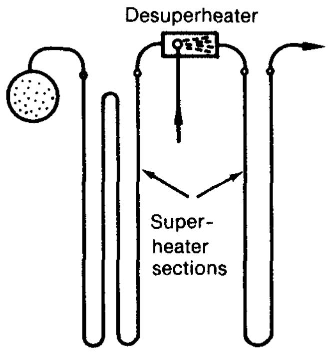
Figure 10-4 Fire Side Control Mechanisms

fireball decreases the furnace heat absorption and thus raises the temperature of the flue gases entering the superheater.

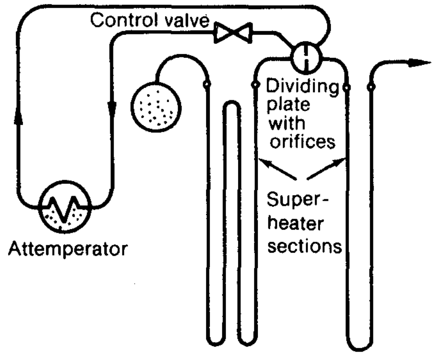
Figure 10-4(B) shows a furnace with burners mounted in fixed furnace wall positions. Note that the burners are mounted at higher and lower elevations. By varying the ratio of the fuel fired in the upper row of burners to that in the lower row of burners, the furnace heat absorption can be modified, thereby changing the temperature of the flue gases entering the superheater.

The flow stream of the flue gas passing the superheater can be split so that the mass of flue gas in contact with the superheater can be varied. Such a mechanism is called a superheater bypass damper and is shown in Figure 10-4(C). The opposite of this is shown in Figure 10-4(D). In this mechanism a flue gas recirculating fan adds a variable amount of flue gas mass flow to the stream in contact with the superheater. An additional fireside method is that of raising or lowering of the percentage of excess air in order to control steam temperature. This method tends to improve the overall heat rate of power generation equipment, although the thermal efficiency of the boiler itself may be lowered.

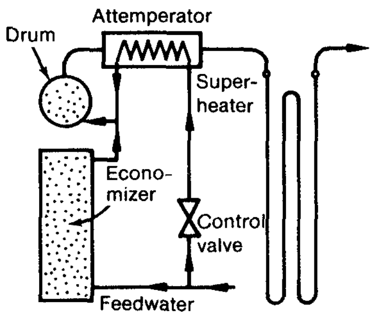
In addition to the fire side methods described above, there are three basic types of water



(A) Pure water sprayed into the superheated steam vaporizes; degree of superheat is reduced by heat of vaporization.



(B) In an effort to avoid steam contamination, desuperheating can also be done through use of shell-and-tube heat eachanger.



(C) In this hookup the saturated boiler steam is partially condensed by feedwater, controlling the steam temperature

From *Power* magazine Special Report, "Steam Generation," by Rene J. Bender, Associate Editor, McGraw Hill, NY.

Figure 10-5 Water Side Control Mechanisms

side methods. Figure 10-5(A) shows the use of a spray mechanism to spray water into the superheated steam. Varying the water flow raises or lowers the steam temperature. Figure 10-5(B) demonstrates a mechanism using a control valve to divert part of the steam to a shell-and-tube heat exchanger. The steam is cooled in the heat exchanger and then mixed again with the rest of the steam, raising or lowering its temperature. The heat exchanger is located in either the steam drum or the mud drum. Figure 10-5(C) uses a shell-and-tube heat exchanger in the saturated steam line between the boiler and the superheater. A controlled portion of the feedwater to the boiler is diverted to the heat exchanger to remove a variable amount of the latent heat, thus raising or lowering the final steam temperature.

These various methods can be used singly or in combination to control the final steam temperature. Which method or combination of methods is used depends upon a number of factors. The particular boiler manufacturer and that company's design philosophy and best competitive offering are important considerations in their selection of the control means and its application.

From a control standpoint, the strategy must be based on the particular mechanisms used and the manufacturer's philosophy for controlling steam temperature. The control characteristics of the different methods may be quite different. The time constant for this process is in minutes. The most rapid response is from spray water, and the slowest response is the method that extracts a part of the latent heat as shown in Figure 10-5(C). Another characteristic of the steam temperature control process is that response time is often a variable function of the steam flow rate.

10-3 Basic Steam Temperature Control Strategies

The strategy used for control of steam temperature for any particular boiler is normally recommended by the manufacturer of that boiler. In a few installations of industrial boilers, and with the steam flow rate reasonably constant, a single-element feedback system may operate satisfactorily. In the typical installation of steam temperature control for all sorts of boilers, some form of feedfoward control, cascade control, or a combination of these is required.

The normal control requirement is to control the temperature within plus or minus 10°F. Figures 10-6 and 10-7 demonstrate two methods of controlling superheat temperature using a water spray as shown in Figure 10-5(A).

Figure 10-6 shows the application of a feedforward-plus-feedback strategy. Since air flow rate is an index of firing rate and excess combustion air, the air flow measurement is used as the anticipatory or feedforward signal. In the summer (x), this signal is combined with the output of the signal from the steam temperature feedback controller (y). The output of this summer provides a signal for the spray water flow control valve. Note that the feedback controller is supplied with an override controller (w).

Controller (w) is a form of override controller that provides a minimum output value track-

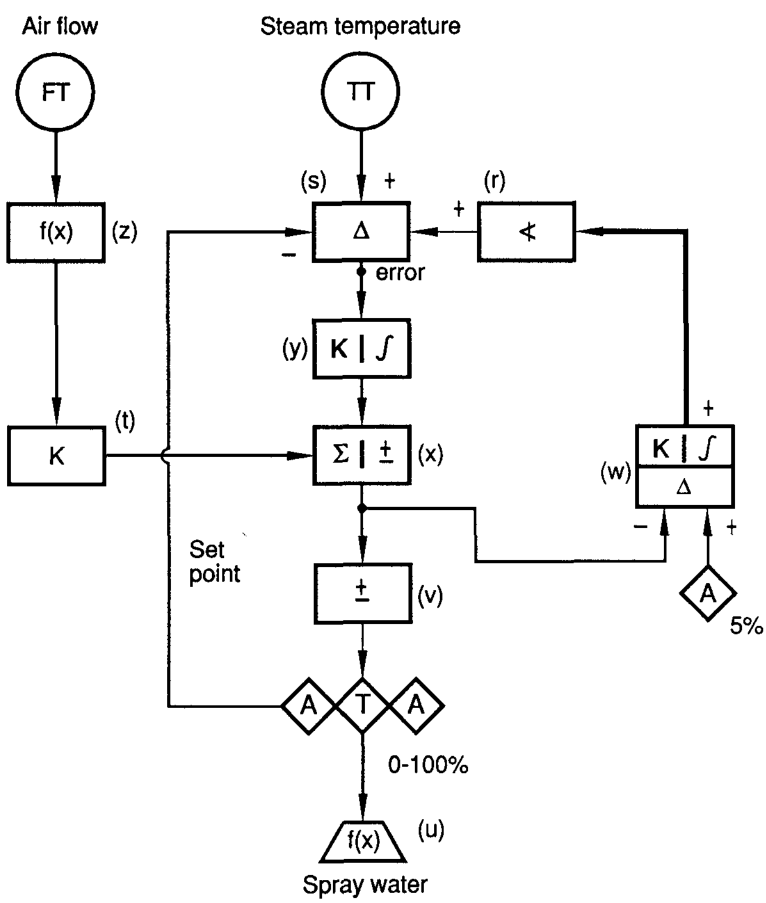


Figure 10-6 Feedforward-plus-Feedback Control of Superheat Spray

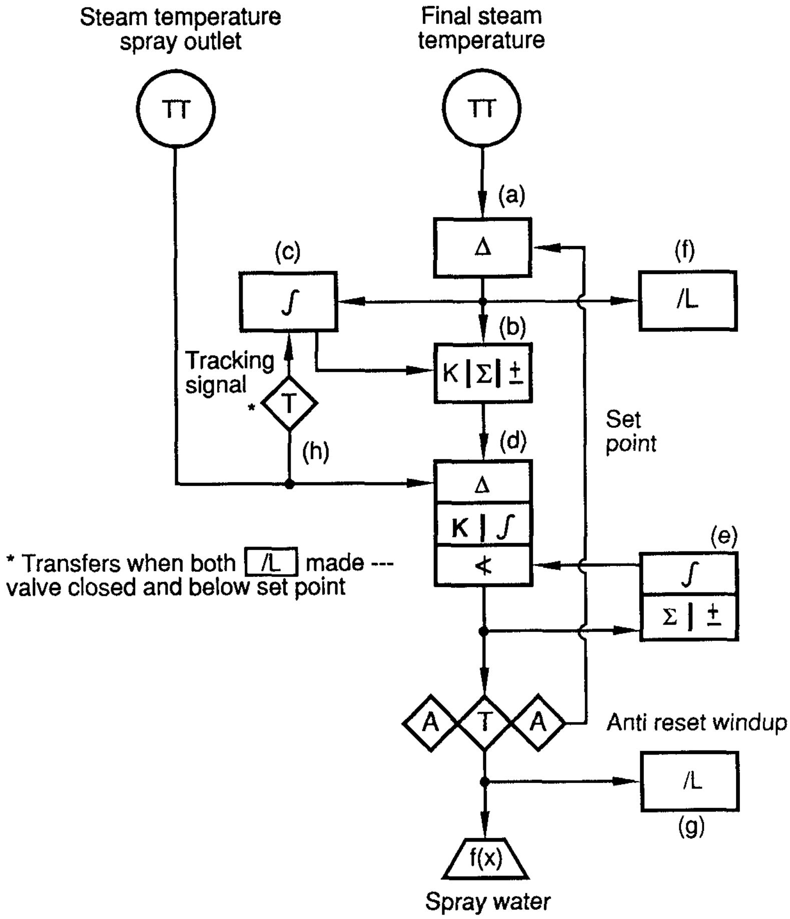


Figure 10-7 Cascade Control of Superheat Spray

ing signal for controller (y). This is included so that when the boiler load is below that of the steam temperature control range, the output of controller (y) will be the signal necessary so that the output signal of the summer (x) will provide a ''just closed'' position of control valve (u). This function is necessary for good control, since on increasing or decreasing steam flow rates, the steam tamperature may be at the design temperature level with different firing and air flow rates. This relationship is also affected by the rate of load change.

The proportional (t) is shown to indicate that scaling of the air flow signal is necessary. In practice this block would probably not physically exist since input scaling capability is an integral part of most summer hardware or software blocks. The bias logic (v) is provided to obtain a positive value signal of the output of summer (x) when the signal to the control valve is reduced to 0 percent. As shown this is a 5 percent bias. This allows a set point of the override controller to be a value of 5 percent with control action above and below this signal level. The f(x) logic (z) allows for a nonlinear relationship between the measured air flow signal and the position demand signal to the spray water control valve.

In calibrating and tuning this system, the relationship between air flow rate and spray water control valve input signal is determined by steady-state testing at the design steam temperature and while operating at different boiler steam flow rates. When this relationship is known, the air flow signal is scaled and functionally adjusted by modifying the f(x) function and by changing the input gain of the summer (x).

With the air flow signal at the level where full steam temperature is obtained without spray water, and with the input gain of that signal adjusted, the second signal of summer (x) is adjusted to a 50 percent value with a summer input gain of 1.0. The output bias of summer (x) is adjusted so that the summer output will be at a plus 5 percent value and bias (v) adjusted to a minus 5 percent value. The output of the bias (v) will then be a 0 percent signal to the control valve.

An examination of the loop of controller (w) shows that, since there is practically no process time constant, its action can be very fast. The gain and integral settings of this controller can usually be arbitrarily set at high values with precise tuning unnecessary. Some adjustment may enhance the ability to come in and out of control range with less overshoot. Should the steam load of the boiler drop with the spray valve closed, the control value will attempt to close the valve further. This causes controller (w) to generate an output signal necessary to drive the set point delta (s) output to zero. This prevents the integral in the controller (y) from acting after the spray valve is closed. The low limit is set at 0 or at some value where its sum to the set point equals the controlled temperature output signal.

With the system calibrated in this manner, on-line tuning of the steam temperature controller (y) is the only remaining action. Preliminary tuning should be under steady-state operation followed by testing under dynamic boiler load conditions. Small tuning adjustments may be necessary due to imperfect or variable relationships in the feedforward portion of the system. Since the feedforward signal does most of the duty that would otherwise be ascribed to the integral action, the integral action can probably be set quite slow at approximately 0.1 repeat per minute. The tuning of controller (w) is available to help the system come into the normal control range smoothly without a large overshoot of steam temperature.

The above description demonstrates again that a multivariable system must be calibrated in addition to the normal proportional, integral, and derivative tuning of the measured variable controller. This calibration should always be done before any tuning of the controller. The description also demonstrates the necessity for the control designer to write a complete functional description of the control system to communicate the system features and the designer's intent. The SAMA diagram displays the functions of the system but does not explain why they have been included by the control system designer.

The system above relies on a predictable relationship between the spray water control valve position and the spray water flow. In some cases, due to different combinations of pumps and heaters, the pressure drop across the spray water valve is not repeatable for each water flow capacity. In such cases a cascade spray water flow control should be added. The valve position demand signal would then become the set point of a spray water flow controller.

Figure 10-7 shows an alternate cascade temperature control system. In this case the primary control is a feedback controller (a, b, c) from final steam temperature. The control logic of this controller is split into three parts to allow the tracking logic to be implemented. Transfer switch (h) circuit is closed when both logic switches (f) and (g) are closed. This occurs when the final steam temperature is below the set point and the spray water valve is closed, indicating that the boiler load is below that of the steam temperature control range.

With switch (h) circuit closed, the integral action of controller (c) is stopped and the output of controller (c) tracks the signal from the steam temperature measurement downstream of the spray nozzle. The error in main steam temperature now exists as an error at controller (d), causing the integral value of this controller to go to a 5 percent value limited by the low limit (e). The output of controller (d) is at 0 due to the proportional error times gain. This keeps controller (a, b, c) ready to immediately assume control when the boiler load again enters the range in which the spray water can be used to control steam temperature.

When the load increases to bring the boiler back into control range, the error in main steam temperature will be reduced, causing switch (f) to have an open circuit, thus blocking the tracking circuit (h) and putting the controller (a, b, c) back to work. With this arrangement this action may occur at different loads depending on the amount of overfiring, boiler conditions, etc.

The secondary controller (d) is a much faster control loop that uses feedback from steam temperature immediately downstream from the spray nozzle. As in the feedforward arrangement, the anti-windup feature is needed when the boiler load is such that no spray water is required.

In tuning the above system, as in any cascade control loop, the secondary controller that operates the spray control valve is tuned first. The gain and integral of controllers (a) and (c) are set at very low values to stabilize the set point input to controller (d). The boiler is then operated at a stable steam flow rate within the steam temperature control range.

The gain and integral settings of controller (d) are adjusted to low values and with a resulting steady steam temperature at the spray nozzle outlet. A heat balance around the spray nozzle can be calculated from the final temperature, spray flow, and temperature before and after the spray to help guide the initial gain setting. The gain is then increased until the control action becomes unstable. The gain is then reduced as the integral setting is increased until the optimum control pattern for the temperature downstream from the spray nozzle is obtained. Analytical techniques as described in the literature or tuning aid devices can also be used. The primary controller (a, b, c) from final steam temperature is then tuned in a similar fashion.

For general guidance, the feedforward strategy is used when there is a good, repeatable relationship between airflow and spray water flow. On some boilers such a relationship is not available, and the spray valve may change position only a small amount over the load range. In these cases the control valve may stay near one position, with the spray water control valve pressure drop accounting for the necessary change in spray water flow. Such installations are candidates for the use of a cascade control strategy.

The systems described are for use with a water spray valve. If burner tilts, bypass dampers, or steam condensers are used, the cascade control system could probably not be used. A change in the type of mechanism used would change the calibration and tuning constants and would probably require the addition of derivative action to the controller.

10-4 Steam Temperature and Reheat Temperature Control Strategies

Most electric utility boilers include a reheater in the boiler along with a superheater. They are controlled to separate and independent temperature set points. This adds orders of magnitude of additional complexity to the systems.

In almost 100 percent of the cases there is a single firing system that affects both the superheater and the reheater. This requires that the control mechanisms be independent of each other. If one of the temperatures is controlled by tilting the burners, the other temperature will be affected and will need spray water or some other control mechanism to control it. The control range of the two temperatures may not be the same, with one temperature reaching full set point at 35 percent of full load and the other at 55 per cent of full load.

All of this creates several basic control problems. Some of these are:

(1) Interaction between superheat and reheat. The single firing system and any control action causes both superheat and reheat temperatures to change.

(2) Interaction between the steam and/or reheat temperature control system and the combustion control system. A change in firing rate causes the steam temperature to change, and a change in spray water flow causes a disturbance to steam pressure and results in firing rate changes.

(3) The heat is added as a firing rate change to change the pressure, but at high pressure 40 to 50 percent or more of the total heat added becomes superheat or reheat.

(4) The design of the control system can affect the plant heat rate or thermal performance. Spraying water into the reheat system causes that additional amount of reheat steam to also

bypass the high pressure turbine. All hot reheat steam goes directly to the intermediate pressure turbine.

(5) Increasing the excess combustion air flow causes superheat and reheat temperatures to increase, improving the turbine thermal performance while degrading boiler performance. At the same time, furnace temperature is reduced, which affects the control of both steam temperatures.

(6) In the event that spray water is not being used for normal control, an override spray control must be in place to prevent either of the temperatures from exceeding allowable levels.

The control system for each unit is a unique system for that particular unit. The manufacturer of the boiler sets the basic requirements on how the boiler is to be controlled. In a large percentage of the cases, the steam temperature control system is furnished in the boiler contract. In this way the boiler manufacturer can be held responsible for the steam temperature control performance.

Figure 10-8 demonstrates the flow path of and heat balance equations for a typical boiler with both superheat and reheat. The hot flue gas temperature T(o) first passes the secondary superheater and then the secondary reheater. In some cases there may be only the secondary reheater. The amount of heat absorbed from the flue gas is a function of flue gas mass flow

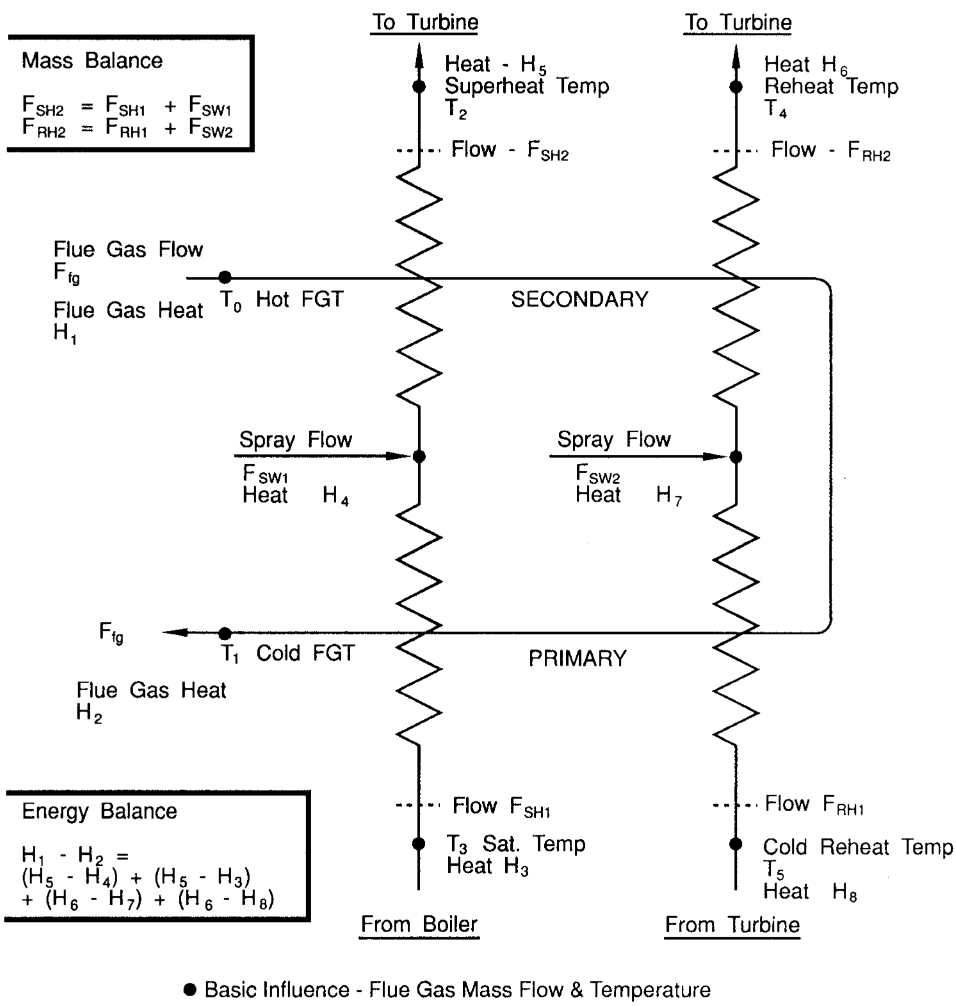


Figure 10-8 Heat Balance Diagram for the Superheater and Reheater

F(fg) the furnace outlet flue gas temperature T(o), and the amount of water sprayed and its enthalpy.

The amount of heat absorbed in the furnace governs the temperature of the flue gases entering the primary superheater and reheater. The amount of heat absorbed in these primary devices then governs the cold flue gas temperature T(1). The T(1) is usually the flue gas temperature entering the economizer.

If at a point in time the steam temperature is 15°F high and the steam flow is 2,000,000 lbs/hr, that represents a need for X lbs of spray water rate in order that the temperature may be at the proper level. A typical value of the specific heat of superheated steam at 1000°F and 1500 psig is 0.6. The 15°F over temperature is (0.6 * 15) or 9 Btu/lb. The steam enthalpy is approximately 1500 Btu/lb, and the enthalpy of 400°F spray water is approximately 375 Btu/lb. The equation to find the amount of spray water is:

$$375 * X = ((2000000 + X) * 1500) - (2000000 * 1509)$$

The result is that X = 16,000 lbs/hr of water at 400°F. This information, that 15°F in steam temperature is equivalent to 16,000 lbs/hr water, is useful information when tuning a water spray control loop. Other information used in tuning includes the ranges of the temperature transducers and the flow capacities and installed flow characteristics of the spray water control valves.

10-5 A Reheat Temperature Control Arrangement for a Combustion Engineering Boiler

Combustion Engineering is the only boiler company to date that has used the tilting burner device to control superheat or reheat temperature. Since they have been consistent in using this device for a period of 50 years, there are a large number of boilers controlled in that manner. Powerful control drives are used to tilt the burners at angles between approximately ± 30 degrees. This is potentially a total angle of 60 degrees, though it is often restricted by the boiler service engineer when the boiler is initially tested. It is quite important to have the tilt angle the same at all four corners to avoid distorting the "fireball" and compromising the controllability.

For a boiler that includes a reheater, the burner tilt control would be used to control reheat temperature. If this can be done successfully, there would, in normal operation, be no water spray to the reheat section, and the unit thermal performance would not be affected. This also means that water spray would be needed for superheat temperature control and a water spray system would also be needed as a reheat temperature override. Abnormal operating conditions of low feedwater temperature from loss of the high pressure heater is one case in which reheat spray might be required.

Figure 10-9 demonstrates the skeleton logic of a feedforward application of a reheat temperature control system. The reheat steam temperature measurement (a) is compared to a set point in controller (b). The normal value of the output of this controller would be approximately 50 percent for plus or minus control. This output is summed in the feedforward summer (c) with the feedforward air flow input. The relationship of the air flow to tilt position is established in the function generation and scaling (f).

The set point is established as a function of steam flow (g). This becomes a fixed set point through the manual function (j) and the low select (h) as the steam flow function increases to that level. In this way the temperature set point is gradually reduced, keeping controller (b) within range as the load drops below that of normal control range. By doing this the reheat temperature can be made to follow a predictable curve when operating below the normal control set point.

The functions (k) and (l) comprise an adjustable bias, so that should the control signal to

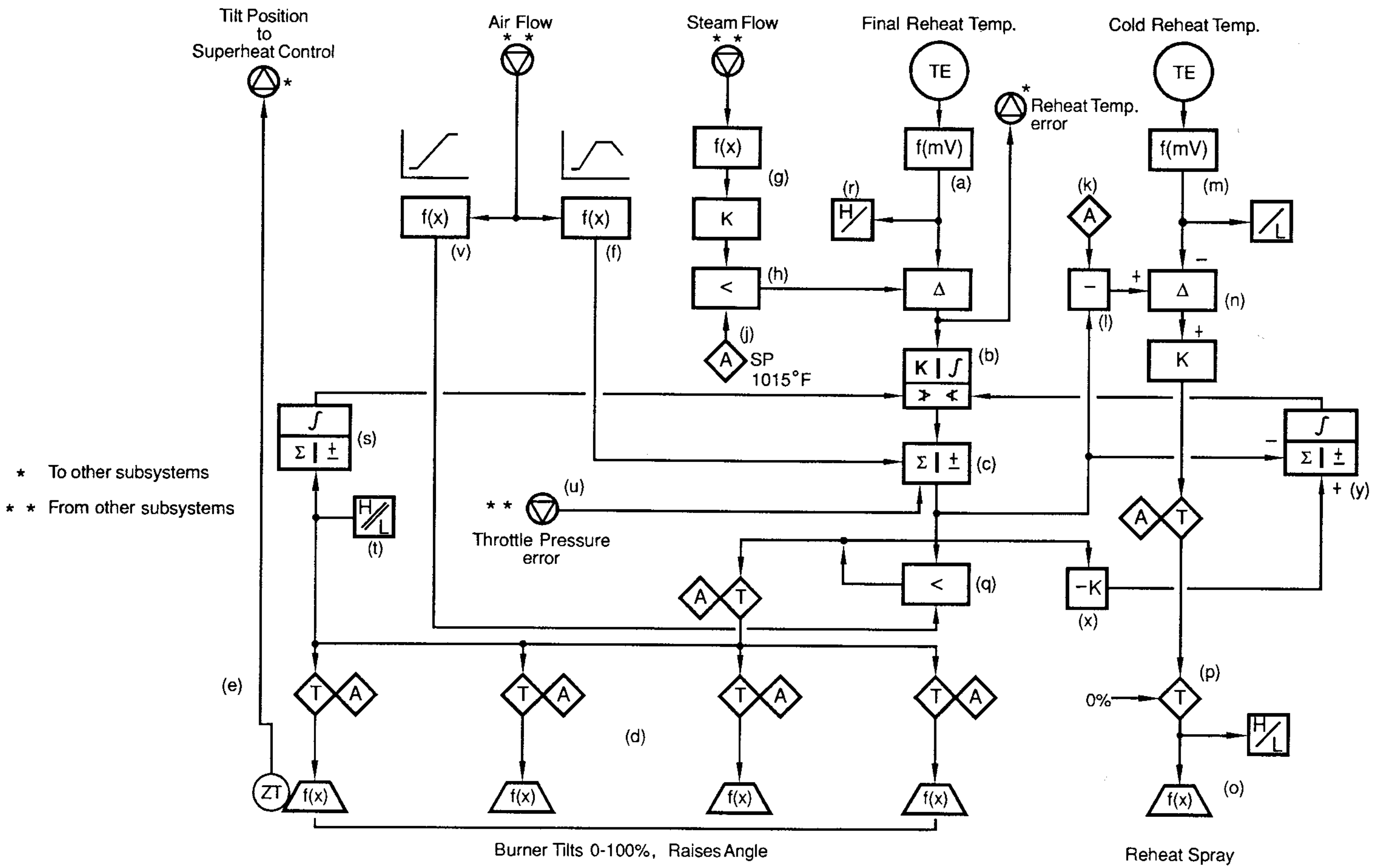


Figure 10-9 Reheat Temperature Control for Combustion Engineering Boiler

the burner tilts reach maximum, a given amount adjusts the demand for burner tilt action and becomes the set point of controller (n) with feedback from a temperature measurement downstream of the reheat spray. The reheat spray valve will be positioned as called for by the cold reheat temperature and the gain of the proportional controller (n).

At all times during normal operation, the interlock (p) will switch the 0 percent signal to the spray valve and allow its operation only when the reheat temperature control calls for spray water as signalled from monitor (r) and at the same time the burner tilts are at their maximum angle as indicated by monitor (t). Another input to the digital logic controlling the spray water block (p) is whether or not the turbine is on line. Should the turbine be tripped, resulting in no reheat steam flow, the absence of such a block could allow spray water to flow back into the turbine with the potential for considerable turbine damage.

Position transmitter (e) sends a tilt position signal to the superheat temperature control system. If superheat temperature is controlled with spray water, this signal is used as a feedforward to automatically change the superheat spray as the burner tilt position is changed. This is intended to prevent the burner tilt change necessitated by the reheat temperature control from causing a change in the superheat temperature.

Two controllers require tuning. All the rest of the adjustments to the system are calibrations that are determined by boiler testing or expert knowledge. A relationship is established between the burner tilt angle and air flow rate. Function generator (f) is calibrated to match this relationship. Function generator (g) is calibrated to match the natural fall off in temperature when operating below the control range. Functions (a) and (m) are millivolt-to-temperature converters for the particular type of thermocouple used.

The bias value of summer (c) is set at minus 50 percent so that the input and output will track at a gain of 1.0. The second input to the summer will then operate at a normal 50 percent value. The main steam temperature controller (b) can be set at a low integral setting (neat 0.1 repeat per minute) since this is a feedforward system with the steady-state burner tilt change coming from the change in air flow signal. The gain of controller (b) is related to the relationship between a change in burner tilt angle and a corresponding change in reheat temperature.

The gain of controller (n) is related to the effect of reheat spray water on a change in temperature downstream of the spray. This can be determined by the heat balance calculations described earlier. The input (u) of throttle pressure error recognizes that, when the error is positive (low steam pressure) and overfiring is taking place, this input automatically lowers the burner tilt angle to compensate. Boiler testing can determine the magnitude of this effect. The information is used to set the gain on this input into summer (c). Signal monitor (t) is set at the maximum and minimum burner tilt angles and ends of the control range. This may be a signal level that represents an approaching excessive temperature or reheat temperature nearing saturation temperature. It is necessary that operation of the water spray does not reduce the temperature lower than a limited number of degrees above this level.

The function (s) sums the tilt control signal with a bias equivalent to the maximum tilt signal. When the tilt signal reaches the maximum desired value, the output adjusts the maximum limit of controller (b) to a value that will hold the tilt signal at the maximum desired value. This will continue until the controller (b) signal is reduced, lowering the tilt signal and causing the output of (s) to release the maximum limit of controller (b). The functions (v), (q), (x), and (y) act in a similar manner to adjust the low limit of controller (b) whenever the tilt signal drops lower than a value programmed as a function of air flow. These limits (high and low) of controller (b) prevent integral windup and keep the system in line and controlling at all loads and temperatures.

The system for a very large boiler may have two furnaces, which would require at least twice the system shown. The air flow and steam flow functions would not be duplicated in such a system. This system shows one burner tilt mechanism position feedback and all eight for the two furnaces are supposed to be the same with the same control signal. Many systems do not rely on the "supposed to" and have individual position signals. These are averaged for the input to the superheat control. The individual position signal can also be used in individual drive unit control to adjust the individual control signals so that all positions will, in fact, be equal.

When the system is placed in operation, the boiler and the interactions may not act exactly according to plan. At that time, modifications in the control systems can often be made to accommodate the actual operating characteristics of the boiler. The choice is between modifying the control arrangement and modification to the boiler. The choice is obvious.

The systems for other types of boilers must be developed individually based on the manufacturer's control philosophy and the devices furnished for sensing and adjusting the temperature. This system encompasses the typical considerations encountered in designing such system applications.

10-6 The Corresponding Superheat Temperature Control for the Combustion Engineering Boiler

Figure 10-10 demonstrates the superheat temperature control system that would complement the reheat temperature control system of Figure 10-9. Signals that are common to the two systems are steam flow (f), air flow (c), tilt position (d), and throttle pressure error (r). The system shown can be the complete system for a smaller boiler or a small portion of a the system for a large boiler that might have two furnaces and eight lanes.

This system is recognized as a feedforward cascade system in which air flow (c), tilt position (d), and throttle pressure error (r) are combined in summer (k) to adjust the set point in controller (l) for controlling temperature (b). Temperature (b) is the temperature just downstream of the water spray control valve. Theoretically, the set point of controller will be reduced as the load on the boiler is increased. The temperature rise in the secondary superheater would be expected to increase as load is increased. Since temperature (a) is constant, then the input to the secondary superheater temperature (b) would decrease.

The bias value of summer (k) is adjusted so that the output of the main steam temperature controller (j) will under normal steady-state operation be at a value of approximately 50 percent. The function generator (f) and gain (k) are used to program the steam temperature control set point when the boiler is operating at a load below the normal control range. Controller (m) is used to set the low limit of controller (j) to a value that will prevent this controller from integrating when the water spray control valve is closed.

Two control valves are shown. The valve (q) is sized for all normal operation. During abnormal operation, such as when the high pressure stage heater is out of service, the feedwater temperature may be 100°F lower than design. This results in a considerably greater firing rate, higher flue gas temperatures in contact with the superheater, and a need for a greater spray water capacity. An additional control valve (p), which doesn't start to open until the normal operation valve is open, furnishes the needed extra spray water flow. The interlock (o) prevents a control signal reaching the control valves until the temperature is within the normal control range.

As in the reheat control system, there are two controllers to tune, (j) and (l). The other control adjustments and alignments can rightly be called calibration. The inputs of the air flow function generator to summer (e) has an initial gain of 1.0, as does the output of summer (e), which adjusts the input gain of summer (k).

At loads below the control range the combined inputs of summer (k) produce an output as set point for the spray outlet temperature that is below the operating temperature of the steam at the spray inlet. This holds the output of controller (l) at 0 on its fixed limit value. As the output of summer (k) rises above the steam temperature at the inlet to the spray, controller (l) will cause spray water to commence.

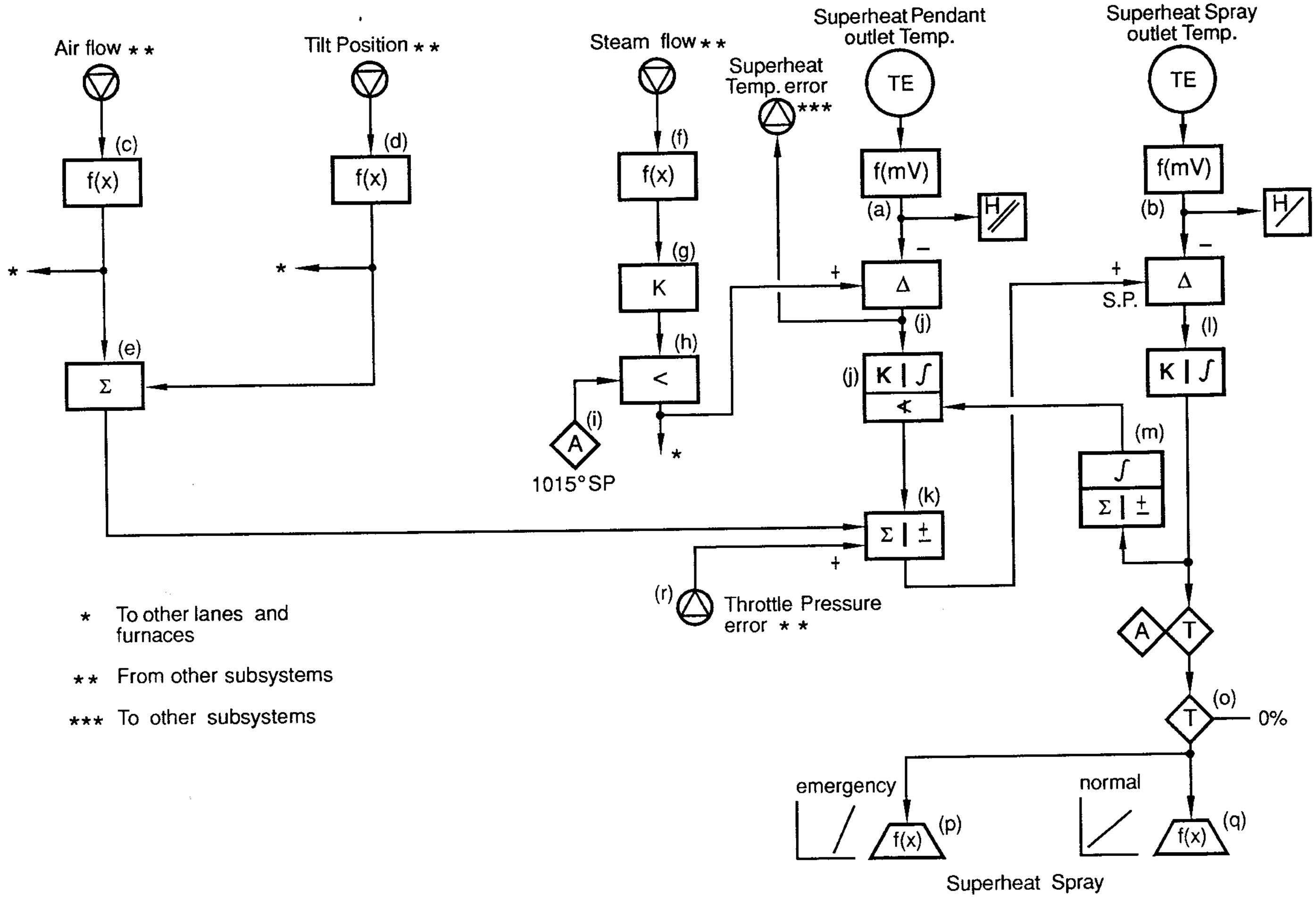


Figure 10-10 Superheat Temperature Control for Combustion Engineering Boiler

By testing the boiler, the amount of increase in the spray water outlet set point over the load range can be determined. This relationship is calibrated into the air flow input gain of summer (k). The throttle pressure effect and its gain into summer (k) are obtained the same way. The input gain of the burner tilt position is related to the change in the air flow input of summer (e) and its gain adjusted according.

A heat balance study of water, steam flows, and temperatures are used in determining relationships between temperature change at (b) and spray water flow for use in tuning controller (l). After this controller is tuned, then the data can be applied to the tuning of controller (j). Controller (m) holds the low limit of controller (j) at the proper value for keeping the system aligned when outside of normal control range. A fixed limiter on controller (l) performs the same function for that controller.

The system arrangements shown in Figures 10-9 and 10-10 represent actual successful systems on the boiler of a 400 MW unit.There are installations where such systems would not be as successful. If no predictable and repeatable relationship can be found between the air flow input and the set point for temperature at the spray outlet, the air flow signal feedforward should be eliminated. In that event, cascade control using only spray outlet temperature for the secondary control should be investigated. In either case, summer (k) should be available for inserting the influences of changes in tilt position or throttle pressure error.

Another successful system on a similar size boiler eliminates the spray water outlet temperature control loop. In this case, the outlet of summer (k) would proceed directly to the spray water control valves. The calibration would then be to establish a relationship between air flow and spray water control valve position. The effect of a change in burner tilt position on a change in spray water valve position would be used to set the gain of the tilt position input into summer (e).

This control arrangement depends on a predictable and repeatable relationship between the spray water valve position and actual water flow. If this is not true, it may be necessary to use the valve control signal as the set point of a spray water flow control loop. A flow control of this sort must control well all the way down to a set point of 0 flow. This is a difficult flow control application due to the poor performance of the flow measurement input near 0 flow. To solve this problem, a wide range flow measurement and control installation, as shown in Figure 10-11, would probably be necessary.

Two differential transmitters, (a) and (b), with differential ranges of approximately 10:1 are used. The output of transmitter (b) is scaled by proportional (gain of 0.1) so that the output when in the range of transmitter (b) should be equal to the output of transmitter (a). Monitor (d), set at approximately 80 percent of full range value, operates the switch (e). In this way the system will be connected to transmitter (b) at low flows. In the example shown, the system should be responsive down to 0.1″ of H_20. This would be 3 percent of full range flow, approximately equal to the control range of the normal control valve.

10-7 Spray Water Sources — Steam and Water Flow Measurements

The source of the spray water supply for reheat temperature control is often at the discharge of the boiler feed pump. Since the reheat steam is at a much lower pressure, the pressure drop across the control valve will be very high, contributing to high maintenance on reheat spray water control valves. To help solve this problem, a feedwater pump interstage connection is often supplied so that some of the boiler feedwater can be supplied at a lower pressure than the normal discharge pressure. Using this water as a source for reheat spray minimizes the maintenance on reheat spray water control valves.

While this water is unmeasured feedwater, it bypasses the high pressure heater and does not come into the feedwater control equation. This is not often the case with the superheat spray water. All superheat spray water shows up as steam at the turbine throttle. If the water source is the discharge of the boiler feed pump and the feedwater flow is measured after the

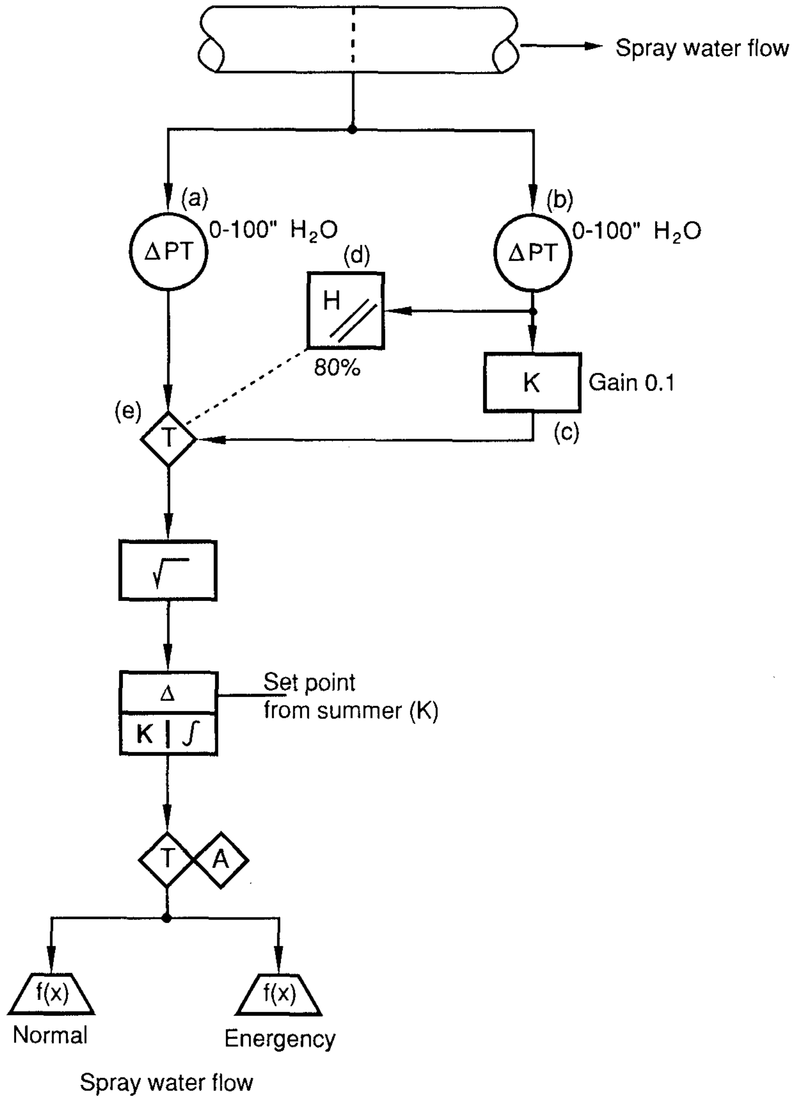


Figure 10-11 Wide Range Flow Contol

high pressure heater, then a direct comparison cannot be made between the feedwater flow and steam flow. Total feedwater flow is the total of measured feedwater flow and spray water flow. For the purpose of comparison between feedwater flow and steam flow in a feedwater control system, spray water flow must be added to feedwater flow or subtracted from steam flow.

The total pressure drop available for the superheat spray system is the summation of pressure drops across piping, the two high pressure heaters to the economizer inlet, the economizer, and the primary superheater up to the spray location. This must match the pressure drop across the spray nozzle, the spray water control valves, unrecovered pressure drop across the flow measurement primary element, friction in the spray water piping, and the elevation difference in head at spray water conditions between the source and the steam line. The control system has only the control valve pressure drop to work with, and this is often a relatively small percentage of the total. This contributes to the problem of non-repeatability between control valve position and flow. In some cases, it may be necessary to install an auxiliary spray water pump to obtain enough valve pressure drop for good control.

10-8 Interactions

Interactions to contend with are those between the steam temperature control and reheat temperature control, between the steam temperature control and combustion control, and between the steam temperature control and feedwater control. The source of these interactions is typically the failure to recognize them in designing the control systems.

For high pressure boilers, pressure adds or subtracts very few Btu. The change in Btu content, to a very large extent, is due to a a change in temperature. A change in the total Btu capacity requirement changes the pressure. Consider a unit with no requirement for a change in total Btu capacity. The temperature is high, which would call for spray water. The pressure is also high, which would call for a reduction in firing rate. If spray water were added, it would make more steam and add to the pressure. This would call for a greater reduction in firing rate. The subsequent change in steam temperature would again change the spray water. This is a temporarry condition but does cause the process disturbance described.

The correct action would be to reduce only the firing rate. This would reduce both pressure and temperature with no change in spray water. The throttle pressure error input to the steam temperature and reheat temperature control systems of Figures 10-9 and 10-10 is one part of the solution to this interaction problem. The other part is to add steam temperature error as an input to the firing rate control system.

The control systems of Figures 10-9 and 10-10 show a solution to interaction between the reheat temperature and the superheat temperature. The reheat temperature controls the tilt angle. By using the tilt angle as an input to the steam temperature control system, its impact on the steam temperature is cancelled.

10-9 Pumping and Firing Rate for Once-Through Boilers

Most once-through boilers operate in the supercritical pressure range (above 3206 psia). The typical throttle pressure of such units is 3500 psig. One unit in the United States (and the world) was designed for operation at 5500 psig, the first of two supercritical units at the Eddystone Station of Philadelphia Electric Co. This unit is over twenty years old, and no more units have been designed for such a high pressure and temperature. A number of supercritical units have been designed for double reheating capability. Initial superheat and reheat design is typically 1050°F, but the initial temperature of the Eddystone unit was over 1100°F, the highest design temperature to date.

A "once-through" boiler can be likened to a long tube. Feedwater is pumped into one end, the tube is heated along its length, and superheated steam emerges from the other end. A simple representation of such a boiler is shown in Figure 10-12.

The steam temperature control systems described above are for drum-type boilers in which the feedwater and steam flows are decoupled by the boiler drum. The saturated steam leaves the drum, and all superheat is added in superheaters having fixed heat transfer areas. With such boilers, separate systems for the control of start up, combustion, feedwater, and steam temperature are required.

With once-through boilers there is one coupled water-saturated steam-superheated steam circuit. Effectively, the total boiler heat transfer surface is divided into variable area boiler and superheater heat transfer surfaces that are constantly shifting in heating surface area during operation. Consequently, the steam temperature control and the other boiler control functions must be melded together.

All such units employ some form of flash tank for start-up and low load operation. Bypass valving systems isolate the turbine from the boiler during the initial part of the start-up process; otherwise, cold water instead of steam could flow to the turbine. Each boiler manufacturer has

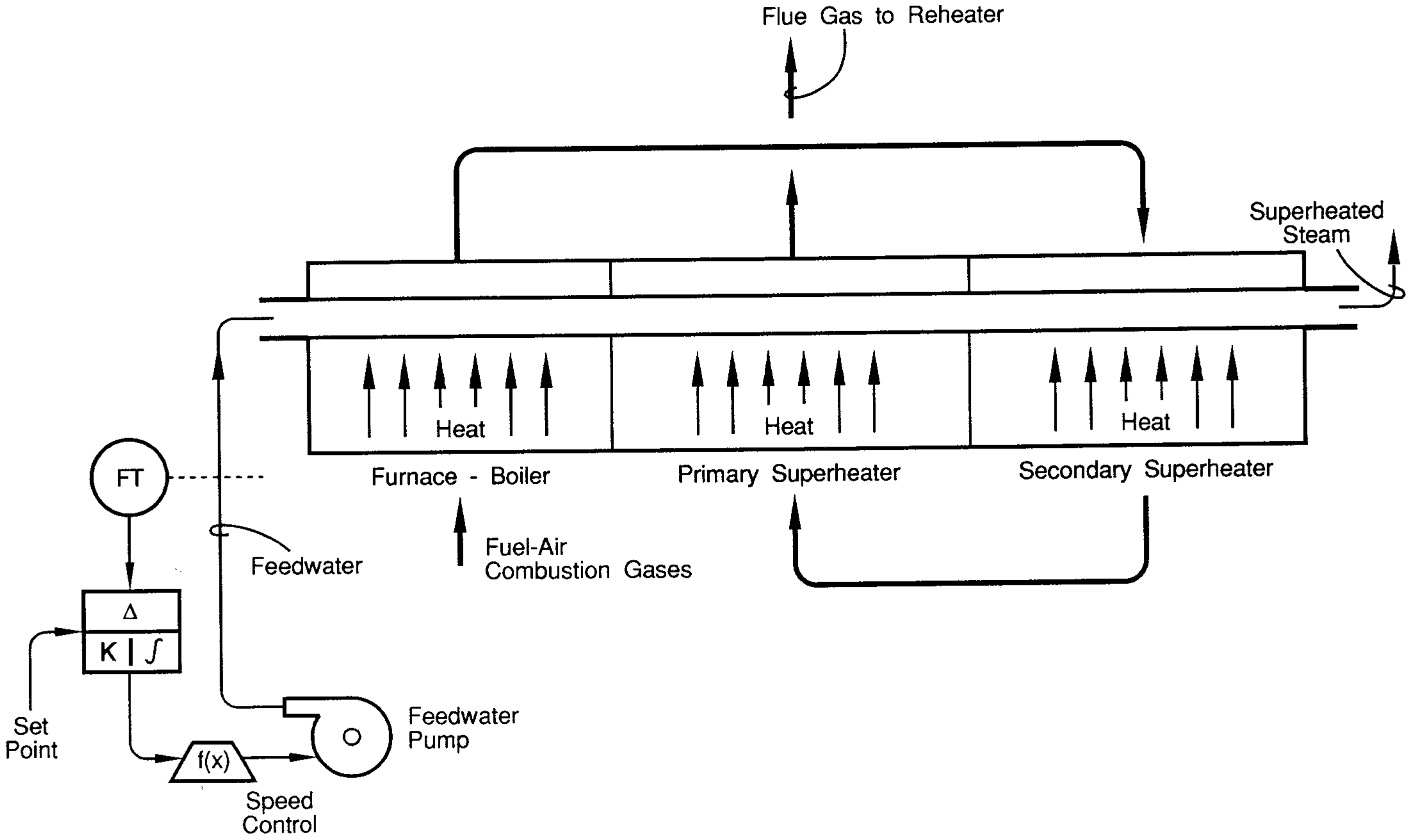


Figure 10-12 Simple Representation of Once-Through Boiler

a proprietary arrangement of such flash tank and bypass systems. A typical piping and valving arrangement for a Babcock and Wilcox "UP" boiler is shown in Figure 10-13. All such systems for the same manufacturer are not the same because of design evolution or because of differences in the desired unit operatin mode.

The "start-up" control for such a unit consists of a considerable block of digital and modulating control logic for operation of this valve system. The following are the three operation modes in the process of initial start-up and transferring from "flash tank operation" with the turbine isolated to "partial to full load operation" with the flash tank isolated.

(1) Cold Cleanup—In this mode the boiler is unfired. Valves 202 and 241 are open. All other valves are closed. The feedwater flow is at the minimum flow set point and flowing through the resistor tubes to the flash tank. Under this condition the water will heat up over a period of time due to the work applied to the water in the boiler feed pump. This operation will flood the flash tank. To prevent any of this water from getting into the steam system, interlocks keep valves 207, 230, 240, and 242 closed. High flash tank level activates an interlock that closes the 205 valve.

The purpose of this mode is to clean up the water to a conductivity of less than one micromho. The open 241 valve causes all flow to be routed through the condenser and the full flow condensate polishing demineralizer. This operation mode continues until chemical tests indicate the water is sufficiently pure enough to proceed.

(2) Hot Cleanup—The boiler can now be fired at a minimum firing rate. The convection pass flue gas temperature is monitored for control actions. This firing causes flash tank pressure to rise so that flow can be maintained without flash tank flooding. With the flash tank steam, a flash tank level at set point is maintained by modulating the 241 valve. As flash tank level is reduced, the 205 valve could be opened but is maintained closed by an interlock until the flash tank pressure has risen to approximately 300 psig. Hot cleanup is continued until the iron in suspension is reduced below 100 ppb.

At 300°F convection pass flue gas temperature, valve 207 opens and flow is maintained through the resistor tubes and through the primary superheater and 207 valve. At about 400°F gas temperature, the expected valve wear on the 203 valve can be tolerated and the 203 valve can be opened. At this time the temperature should not exceed 550°F for water purity reasons. At some temperature lower than 550°F, the temperature can be placed on automatic to control firing rate.

At 120 psig flash tank pressure, flash tank steam can go to the deaerator if it requires steam. At 300 psig the 205 valve opens to admit flash tank steam to warm the secondary superheater and steam lines through drain valve 210. At 500 psig the turbine can be rolled with extra steam going through the 220 valve to the high pressure heater and through the 240 valve to the condenser. The steam is flash tank steam from water through the 202 valve plus steam from the 207 valve at the outlet of the primary superheater. The mixture flows through the 205 valve and secondary superheater to the turbine.

(3) If the water is sufficiently cleaned, the start-up mode can begin. The turbine throttle valve is opened enough to heat up and roll the turbine and bring it up to speed. The firing rate is adjusted to maintain the convection pass temperature and for the flash tank maintaining the pressure. The unit is synchonized and gradually loaded, through the 201 PRV, to the load that can be maintained at the reduced pressure of the flash tank steam through the 205 valve and the 201 PRV valve. Care must be taken match enthalpies for a smooth enthalpy rise for the mixture of 205 valve and 201 PRV steam as the load is increased.

Valve 207 is closed and the 201 PRV valve opened to replace the 207-valve part of the 205-valve steam as flash tank steam from the 205 valve is gradually replaced with steam through the PRV 201. This is a sensitive part of the operation to avoid enthalpy swings (showing up as temperature swings) at the inlet of the secondary superheater and turbine valves. The steam entering the secondary superheater is a mixture of flash tank steam and steam from

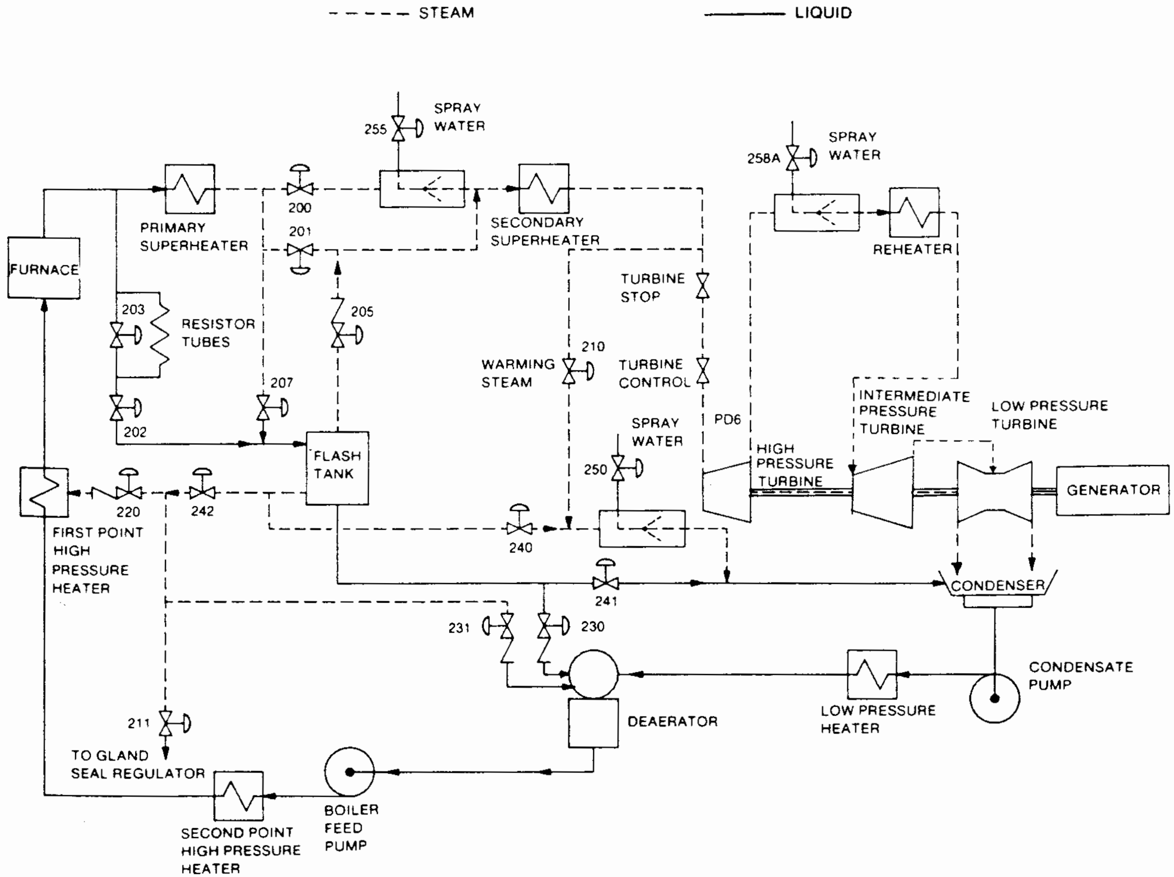


Figure 10-13 Once-Through Boiler Cycle Diagram

(From ANSI/IEEE Std 502-1985)

the 201 PRV. Flash tank pressure is raised to a set point of approximately 1000 psig using the 240 valve. A throttle pressure program adjusts the feedwater flow demand and from that the set point pressure for the 201 PRV.

As the unit is loaded by the 201 PRV set point to a throttle pressure higher than flash tank pressure, the check valve closes, blocking the 205 valve steam. All turbine steam flow is going through the 201 valve, and with 207 valve and the 205 valve closed the flash tank is isolated. At approximately 100 psi higher than the flash tank pressure, the interlock closes the 205 valve.

The 201 PRV continues to increase flow to the turbine until the PRV becomes wide open. The 200 valve is opened on a load demand signal. Since there will be no pressure drop across the 201 PRV with the 200 valve open, the flow through the 201 PRV drops to a very low value. The system is calibrated so that the superheat and reheat spray valves are approximately 50 percent open so that they can trim the temperatures quickly in both directions.

As load is increased by increasing water flow and firing rate, pressure and temperature must now be ramped up to their design values, with particular care that the steam temperature does not get out of line and change turbine temperatures too rapidly. The several modes of operation during this overall procedure are automatically accomplished by the switching of valve combinations and control of valve positions. The enthalpy matching and the ramping of pressure and temperature from flash tank operation is, from a control standpoint, the trickiest part of the "on-line" operation. The unit may be required to operate over a wide load range with the flash tank again involved in the reverse order as load is reduced.

Generally the control loops are in accordance with Figures 10-14(A) and 10-14(B). They are described as follows:

(1) Feedwater flow demand sets the PRV program for the 201 valve.

(2) The 200 valve is opened with a digital interlock from a load index.

(3) The 207 valve valve opening is controlled from primary superheater outlet temperature with a feedforward from convection pass temperature and an override from primary superheater outlet pressure.

(4) The 202 valve is controlled from primary superheater outlet pressure.

(5) Flash tank level controls the 241 valve drains to the condenser.

(6) Deaerator pressure is controlled by the 230 valve using a feedforward from flash tank level and an override from deaerator level.

(7) Flash tank pressure is controlled by valve 240 to a set point from a throttle pressure program. A maximum flash tank limit sets the maximum pressure.

(8) Deaerator pressure is controlled by the 231 valve with a feedforward from flash tank pressure.

(9) First point heater pressure is controlled by the 220 valve.

10-10 Steam Temperature Control for Once-Through Boilers

With once-through boilers there is one coupled water-saturated steam-superheated steam circuit. Effectively, the variable area boiler and superheater heat transfer surfaces constantly shift in heating surface area during operation. Consequently, the steam temperature control and the boiler control must be melded together.

The demand for boiler load changes fuel, combustion air, and feedwater through the use of parallel flow control loops. This demand signal is developed by the coordinated control, turbine following control, or boiler following control. The signal for a once-through boiler can be called "boiler demand" in the same way the basic demand signal for a drum-type boiler is called "firing rate demand."

Combustion is controlled by ratioing the air flow to the fuel flow. Steam temperature is controlled by ratioing fuel and combustion air to feedwater. A factor that must be taken into

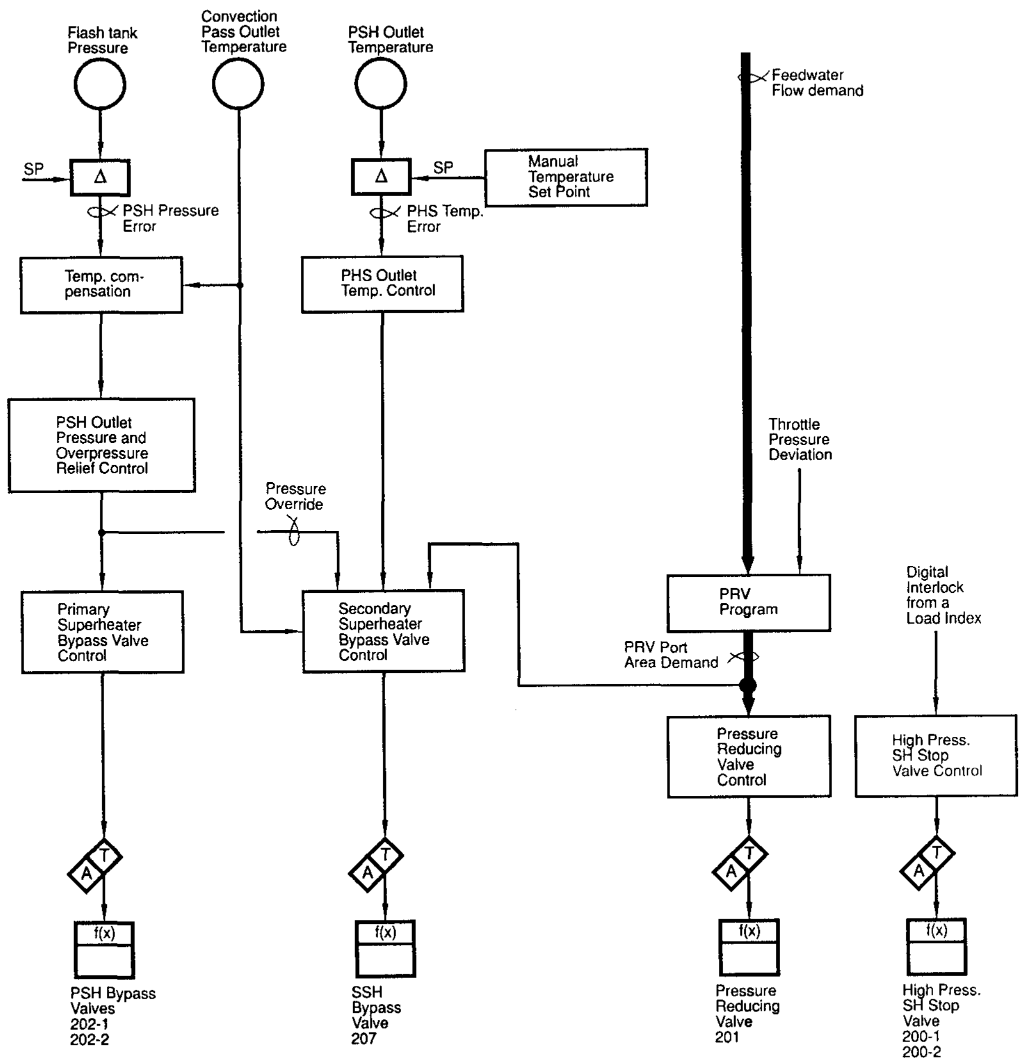


Figure 10-14 (A) Control Loops for "UP" Boiler Start-up and Bypass System

(From © Babcock & Wilcox Co. Control Specification)

consideration is the loop response. Because of the need to obtain more responsive control of steam temperature during transients, a parallel proportional control, using spray water, is incorporated.

The system is calibrated so that in normal load steady-state operation the spray water control valves will be 50 percent open. During a disturbance or a load change, they either open further or close depending on the need. This proportional control is shown in items (a) and (c) of Figure 10-15. The basic steady-state set point temperature is obtained by multiplying the fuel flow relative to the feedwater flow. This is shown in controller (a), (b), and multiplier (d). The multiplier modifies the boiler demand signal, which is the set point for feedwater flow. The proper combustion conditions are obtained with the final percent oxygen trimming signal that is developed in controller (i). The output of this controller and multiplier (h) modify the boiler demand signal to obtain the correct combustion air flow set point. Function generator (j) modifies the boiler demand signal to develop the curve for the correct percent oxygen set point that relates to each particular boiler load.

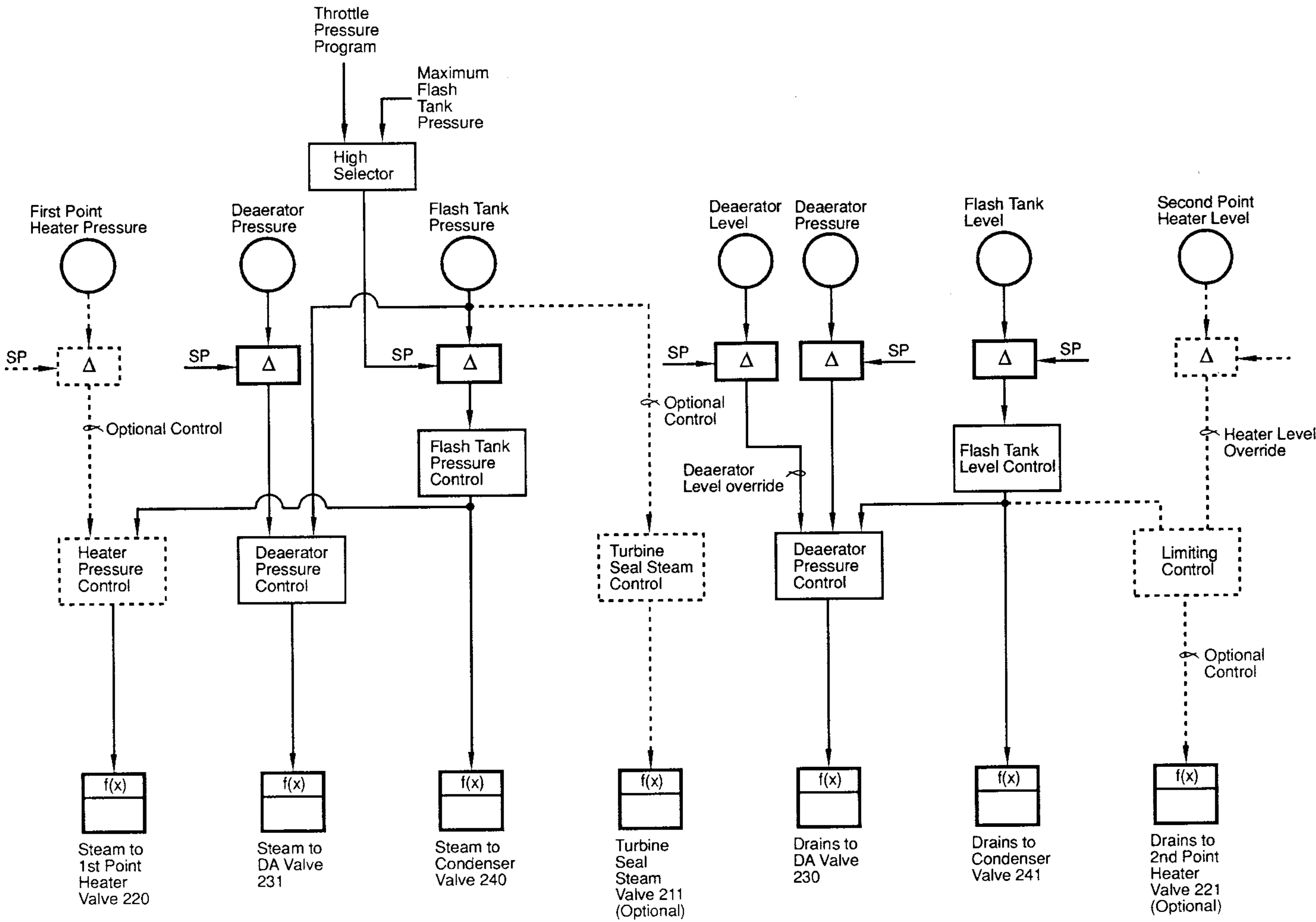


Figure 10-14 (B) Control Loops for "UP" Boiler Start-up and Bypass System

(From © Babcock & Wilcox Co. Control Specification)

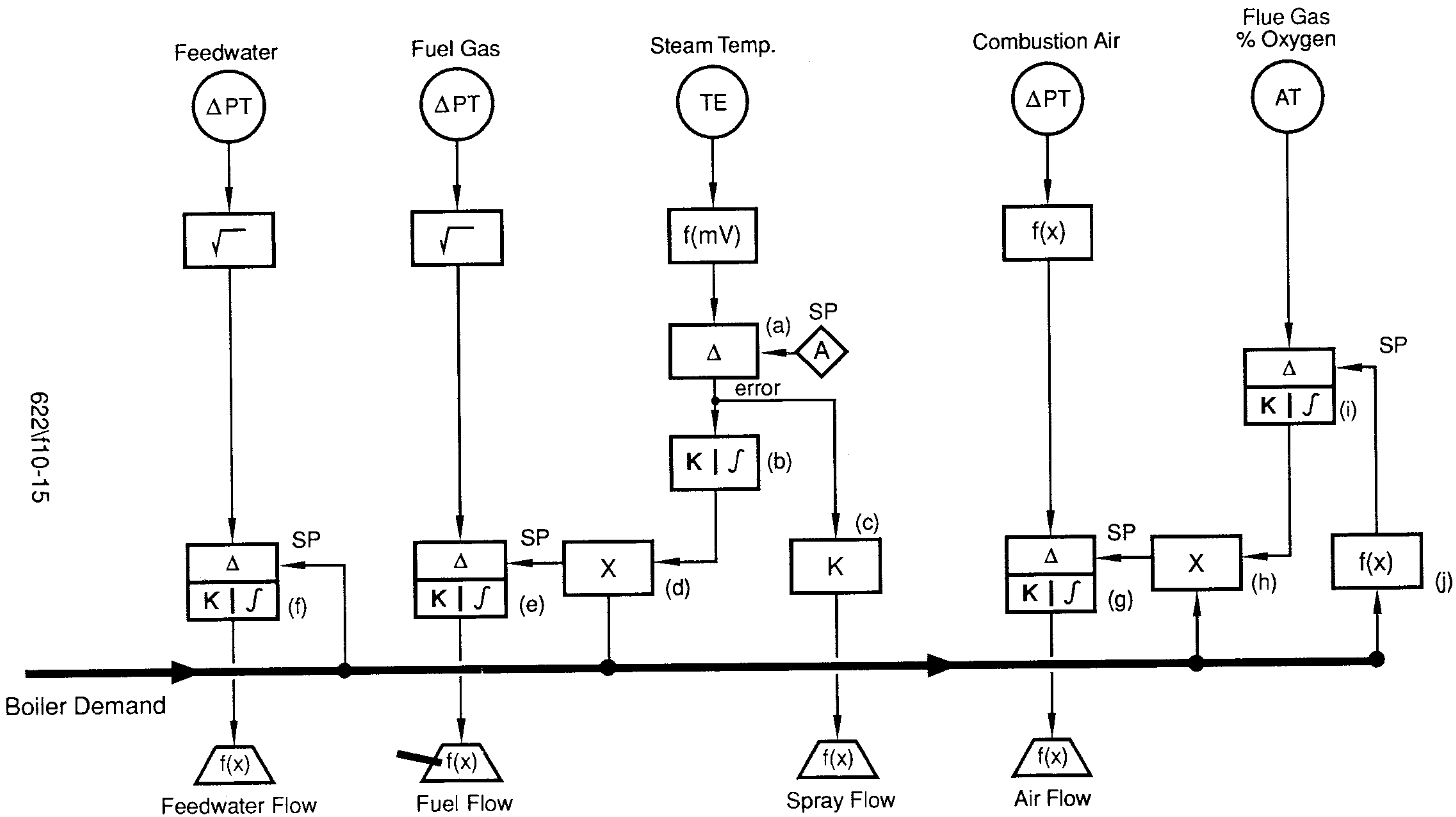


Figure 10-15 Skeleton Control Logic for Once-Through Boiler Control (Gas-Fired)

In practice the fleshed out control logic would include interlocks and tracking circuits, fixed limits, cross limits, manual/auto stations, duplicate circuits for complex boilers, and all other requirements for a complete control logic system. Cross limits are intended to limit other flows whenever something happens to restrict or magnify one of the flows. For example, blockage of, or limits on the air flow should constrain the fuel flow and feedwater flow. Blockage of or limits on the water flow should constrain the fuel and air flow, etc.

The logic shown is for the normal operating range. The complete system would include the controls for the start-up valving system and controls for the transition from the flash tank start-up system to the normal range control.

The "boiler demand" originates in the "boiler following," "turbine following," or "coordinated control" discussed in Section 9.

Section 11
Boiler and Unit Interlocks

The modulating controls that have been discussed previously, and will be discussed again later, can generally be referred to as "on-line" control. Simply put, these controls are in place and operating during all normal operation, after start-up of the equipment and within the load range of the process being controlled. These control systems follow the demands of the process load changes. Such control systems are described by the arrangement of SAMA symbols as discussed in Section 1. There should also be a written system description that conveys the intent of the control application designer.

Another aspect of boiler and process unit control is the interlocking function. The interlocking function is digital (as opposed to modulating) and covers all those "yes-no" decisions involved in the equipment operation. This requirement is primarily one of safety and protection for man and machine. One factor includes the decisions that protect the operators, the equipment, and the ultimate process integrity during the start-up of the process. Another is the safe management of the process during normal operation to assure the continuity of operation.

Some of these interlocks are embedded into the modulating control system to assure proper sequential actions and to take such control blocking actions of the modulating control that may be necessary. The manual/auto transfer is a part of this interlocking scheme. There will be discussion of this later in this section.

11-1 Applicable Codes

A portion of this digital system consists of the burner management and flame safety system, which will be discussed in a later section, along with the respective operating controls for the different types of boilers. The code authority for this portion of the systems is the National Fire Protection Association (NFPA) 85 series.

The remainder of such systems are guided by the IEEE Standard (ANSI/IEEE Std 502-1985) "IEEE Guide for Protection, Interlocking and Control of Fossil Fueled Unit-Connected Steam Stations." This is a skeleton guide to be amplified and fleshed out based on the detail of the units being controlled. The purpose of IEEE 502 is to guide but not dictate practice on any particular boiler or generating unit. The multitude of boiler arrangements make such a goal completely impractical.

Another code now in development by the Chemical Manufacturers Association (CMA) will apply directly to the industrial boilers used in that industry but not necessarily to the electric utility industry. The ideas expressed will, however, provide excellent additional guidance to designers of electric utility control systems.

11-2 Logic Diagramming for Motor Starting and Trip Protection

The examples to be shown and discussed are typical of those needed for protection of personnel, to safeguard equipment against catastrophic damage that might lead to failure, and to protect against inadvertent misoperation that could produce catastrophic results. This part of the overall system is diagrammed with a set of symbols different from those used in the modulating portion of the overall system. A written description of the operation of this part of the system should be integrated with the description of the normal operation modulating system.

Methods of Diagramming

There are different methods for demonstrating and diagramming the logic. Figure 11-1 shows three different ways to show the same basic logic action. These might be called "and-or-not" logic diagramming, ladder logic diagramming, and "yes-no" logic diagramming. Of these, ladder logic diagramming is the oldest and has been used for more than fifty years. It was originally based on the use of electric relays for implementing the logic. The "yes" and "no" diagrams have been used to a large degree in flow charting for computer programming and for implementing hardware-based solid-state logic systems. The "and-or-not" has more recently gained wide acceptance.

The AND function means that if A and B are "yes" then C will be "yes." If either A or B is "no" then C will be "no." This is recognized as the ladder diagram logic of a "series" electrical circuit in which closed switches A and B mean that voltage will be applied to C, and that if either switch A or B is open then the voltage will be blocked. The "yes" and "no" logic also shows that to get a "yes" answer both A and B must be "yes."

The logic of a parallel circuit is an OR circuit. If either switch A or B is closed then voltage

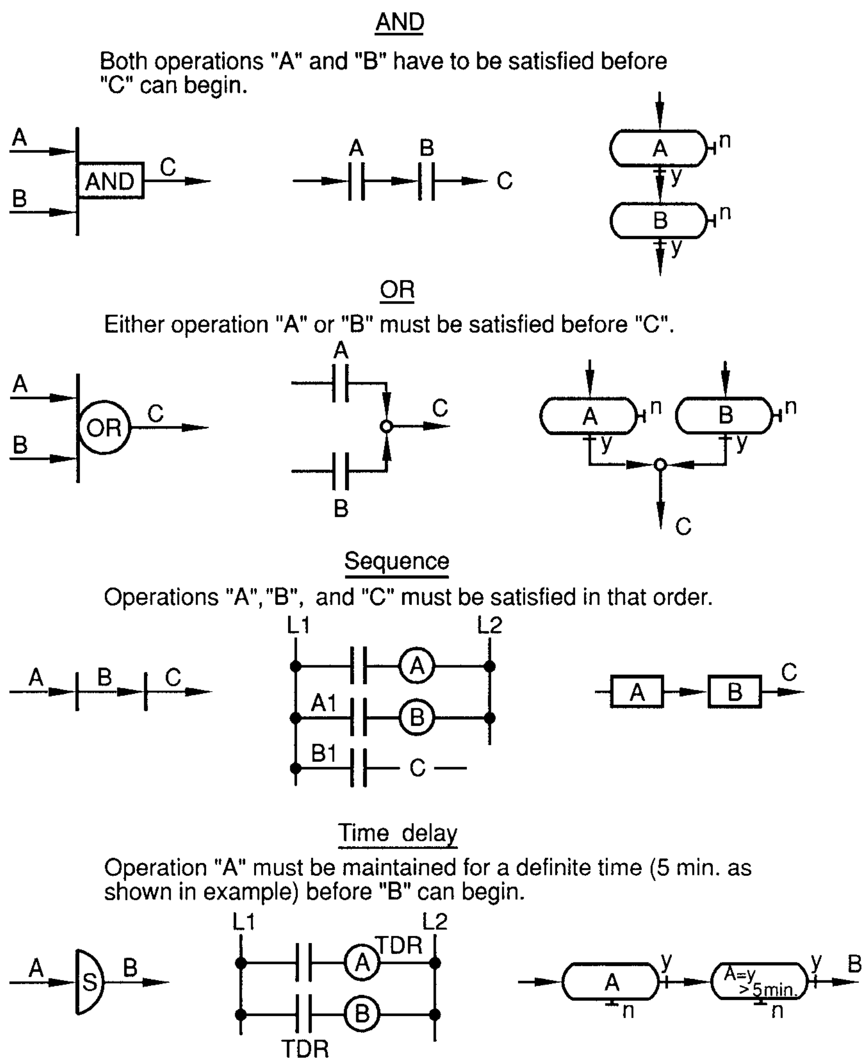


Figure 11-1 Basic Logic Symbols

will be applied to C. It is sometimes desirable to express logic in terms of something that is not true (e.g. a damper not closed) by inserting the word "NOT" in a line. If both switches are open, then the voltage is blocked. This can also be expressed in the "yes" and "no" logic shown with a "yes" output from either A or B showing a "yes."

A sequence can be shown by the method of lining A, B, and C up in a line with a sequence mark between. In the ladder diagram logic, sequence is shown with a switch contact from the first action initiating the second action and a switch contact from the second initiating the third. In "yes" and "no" logic, the rectangular "action" boxes are shown being implemented in series and in a particular direction.

The "time delay" function allows voltage to be applied from A to C an adjustable time after voltage is applied to A. In the "memory" function (not shown), voltage applied to A is applied to C and continues after the the voltage is removed from A. This condition continues until the memory is reset with voltage applied from B. At this time if the voltage has been removed from A, then no voltage will be applied to C. In the ladder logic, TDR represents the coil and contact of a time delay relay.

Typical Logic Presentation

In the example shown in Figures 11-2, 11-3, and 11-4, the implementation of one of the important boiler interlocks is presented. Any boiler that has both an induced draft fan and a forced draft fan requires that these fans be interlocked. To avoid pressurizing the furnace as the forced draft fan is started, there must be an induced draft flow path. This requires that the induced draft fan be started before starting the forced draft fan. In addition, to avoid the possibility of overloading the motors as the fans are started, the fans should be started with the flow control dampers closed. These dampers can be released to control after the fans are started.

The ladder diagram of Figure 11-2 shows this logic. In such circuits, the "normal" condition is with all contacts shown with the relay coils deenergized. If pressure switches were part of the logic, the "normal" condition of their output contact would be with "no pressure" available to the switch. When implementation is by using relays, it should be noted that there can be a very short time period of a total open circuit after the relay coil is energized with the "normally closed" contacts opened but before the "normally open" contacts have closed. If this creates a problem in a particular application, "make before break" relays are available.

With power available and both fans shut down, assume that all permissives of the ID Fan are satisfied. If this is so, contact IDP is closed. Closing the damper closes contact DCID, thus energizing the coil of time delay relay TDR1. This closes contacts of that relay labelled TDR1. One of these (not connected in this circuit) prevents the damper from leaving the closed position. The other TDR1 contact, which is in series with switch PB1, also closes. A momentary contact from switch PB1 can now energize relay R1. A normally open contact of this relay creates a closed contact in the ID fan motor breaker circuit (not shown), which causes the motor to start. As the motor breaker is closed, contact B1 is closed, which seals in the circuit through contacts B1, PB2, and IDP. Any loss of a permissive will open contact IDP and cause the motor to trip, as will also momentarily opening switch PB2. When relay R1 is energized, contact R1 is opened. This deenergizes the coil of TDR1 and starts the timer. After an adjustable time delay, the TDR1 contacts open. This releases the damper for control.

This same logic is expressed in "AND-OR" logic in Figure 11-3. This diagram also shows the preceding logic that starts with tripping action of both fans and the main fuel, after which both the forced and induced draft fans are not running.

The logic as expressed in "yes-no" logic is shown in Figure 11-4. This logic is based on questions with a "yes" or "no" answer. Any action, either automatic or manual, is shown in a rectangular box. Assume that this sequence begins with the "start required" block. This particular logic also shows that having the FD ready and all of the FD permissives OK is necessary for starting the ID fan. With the ID fan running, its control damper can be released,

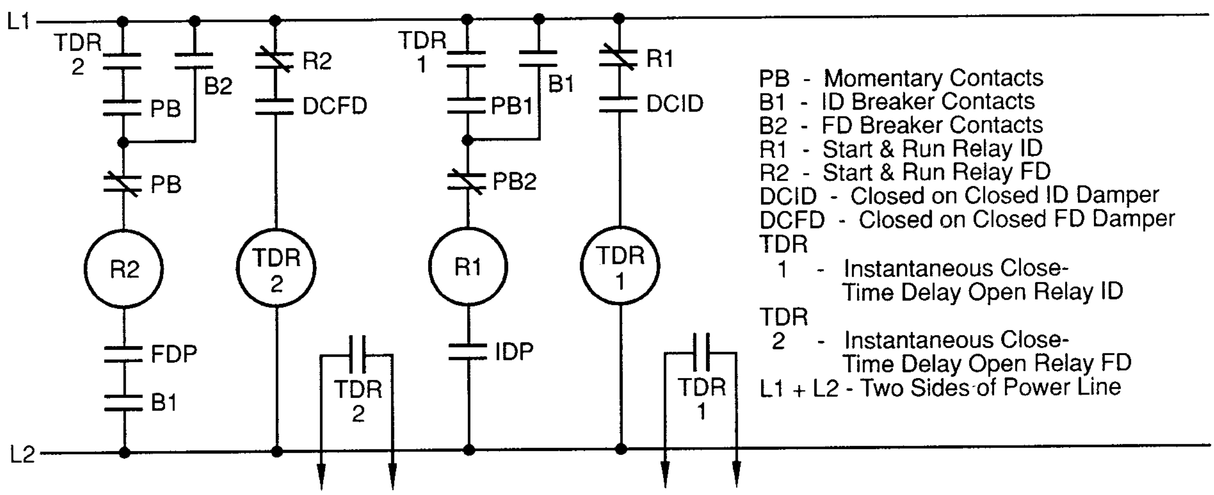


Figure 11-2 Ladder Diagram Logic for Single Forced and Induced Draft Fans

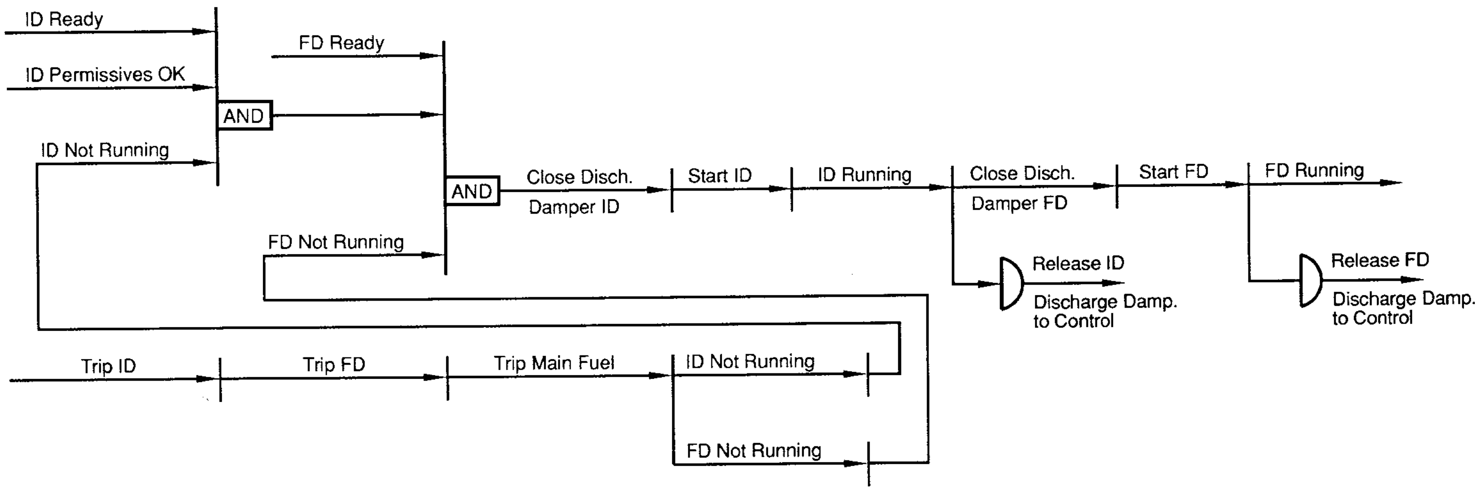


Figure 11-3 AND-OR-NOT Sequential Logic for Single FD and ID Fans

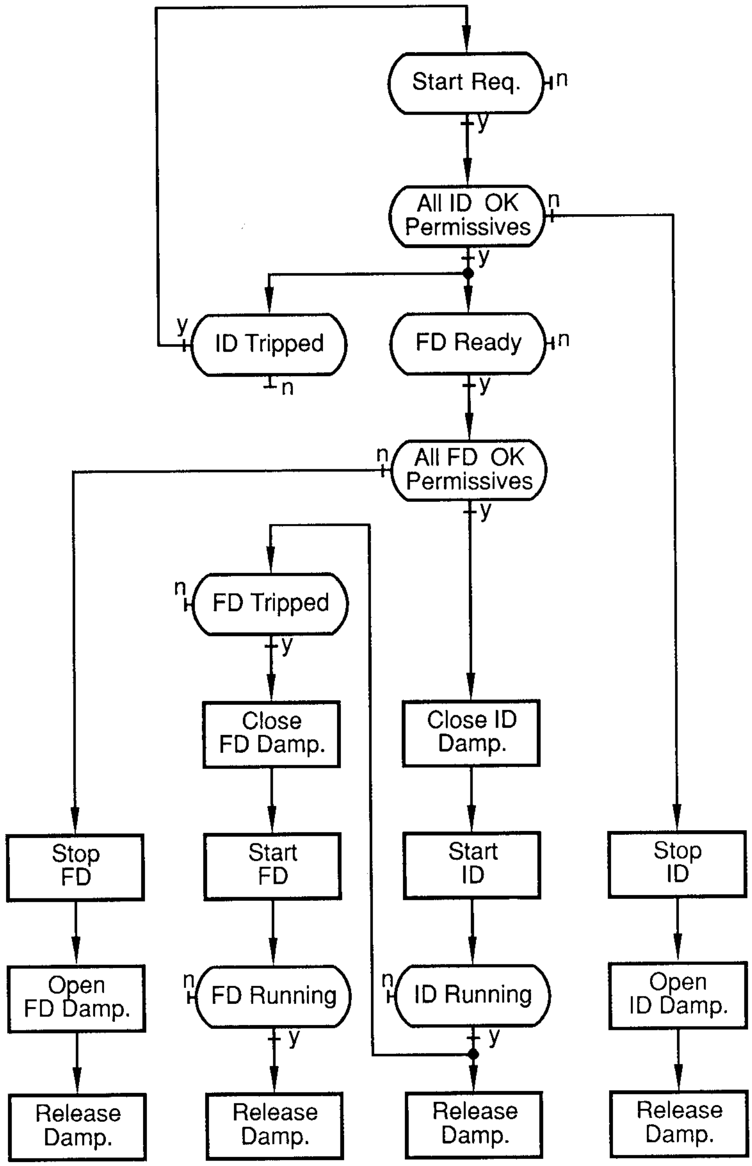


Figure 11-4 Yes-No Logic for Single ID and FD Fans

and the sequence for starting the FD fan can begin. A trip of the ID fan at any time causes the "ID Running" permissive to be "no." With this "no," the block "All FD Permissives OK" is "no" and the FD Fan will be stopped.

When there is more than one set of FD fans and ID fans, additional interlocks are required between the two sets of fans and their shutoff and control dampers. One example of such an interlock network is shown in Figure 11-5. This is an interlock diagram permitting the closing of a forced draft fan motor breaker. The breaker can be closed with a flow path involving either of two ID fans running, their inlet vanes "not" closed or their inlet vane selector stations on "auto" and their discharge dampers open. As the FD fan starts and the air flow reaches the furnace, the pressure increases and the furnace pressure control automatically adjusts the position of the ID inlet vanes.

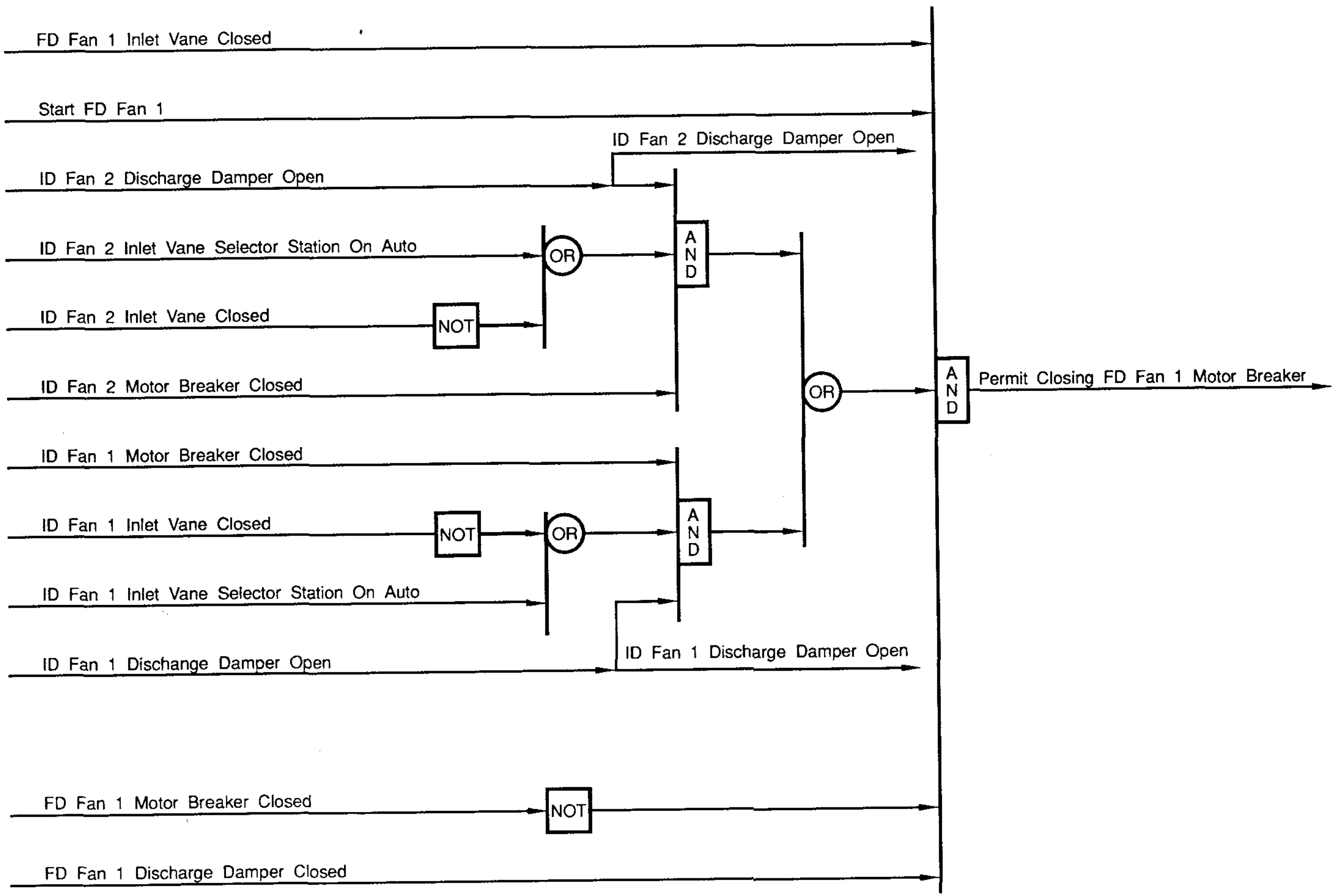


Figure 11-5 Logic Diagram Permissives for Closing Forced Draft Fan Motor Breaker

The induced draft portion of the logic demonstrates that if the induced draft vanes are "not" closed "or" the manual/auto selector station is on "auto," the discharge damper is open "and" the induced draft breaker is closed, then the induced draft permissive is complete. The disruption of any of the inputs to the "and" circuits for both induced draft fans will trip the forced draft fan if it is running. It will continue to run if any input to the "and" circuit is disrupted for only one of the two induced draft fans shown.

An important interlock between each fan of a pair assures that should a fan trip, its dampers will immediately close and the dampers of the other fan will automatically open to pass the additional air flow. The closing of the dampers on the fan that has tripped prevents air flow from the running fan being lost by reverse flow through the fan that has tripped.

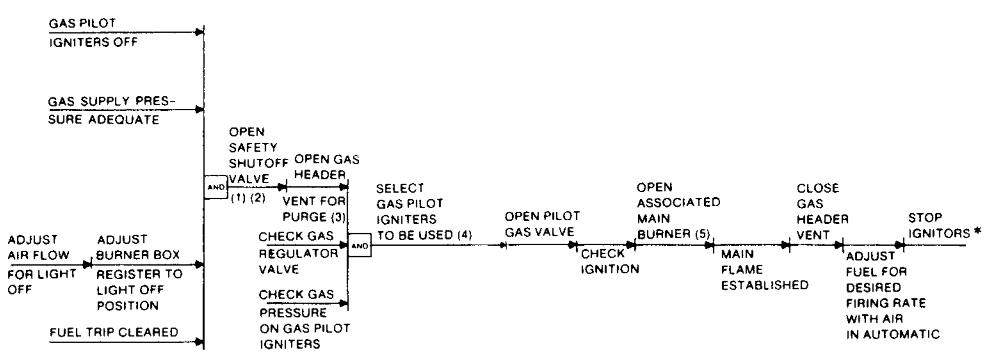
A very important safety interlock deals with starting and stopping burners and the tripping of the main fuel supply. Should the air flow system be tripped, the main fuel must be tripped. If a burner flame goes out, fuel to that burner and the air registers on that burner must be closed. The skeleton of that interlock, as shown in Figure 20 of the IEEE Guide, ANSI/IEEE Std 502-1985, is shown in Figure 11-6. Such interlocks are covered in the NFPA 85 code series to be discussed later in this book. A number of other interlocks that are important for operational safety are also shown in ANSI/IEEE Std 502-1985.

Implementation

Several different approaches and types of equipment can be used in implementing interlock systems. Before the advent of solid state electonics, almost 100 percent of such systems were implemented with mechanical relays. In hazardous atmospheres, explosion-proof boxes connected by solid conduit can be used to house the electrical elements. Alternately, the electrically operated mechanical relays can be replaced with networks of pneumatically operated three-way valves and pneumatic instead of electrical push buttons. Fluidic logic has been used to a very limited extent.

After the advent of solid-state electronics, solid-state relays were developed as a replacement for mechanical relays. The solid-state advantage over mechanical relays was in the lack of maintainence of relay coils and the quality of, or wear of, their contact surfaces. Cabinet systems comprising many solid-state logic cards with discrete logic elements, diodes, transistors, capacitors, etc., were used in the 1960s. In the late 1960s and 1970s, this led to such systems that used integrated circuits.

During this period such logic was also implemented in digital computers, and in the 1970s



*NOTE: Interrupted igniters (small capacity or Class 3) shall be shut down at this time to avoid supporting main flame ignition. Continuous or intermittent igniters (large capacity Class 1 or 2) should be left in service until stable main flame is established. See ANSI/NFPA 85A [1], ANSI/NFPA 85B [2], ANSI/NFPA 85D [3], ANSI/NFPA 85E [4], and ANSI/NFPA 85G [5] for definition of igniter classification.

Figure 11-6 Logic Diagram for Natural Gas Firing

(From ANSI/IEEE Std 502-1985)

specialized devices called programmable logic controllers (PLCs) were developed along with distributed digital control. Today, PLCs of some sort or capabilities built into distibuted control systems are used in a large majority of all installations. The PLCs and distributed control can be programmed using programming languages based on the use of ladder logic or other diagramming methods.

A particular benefit of the use of the systems of today, such as a PLC, distributed control system, or digital computer, is the diagnostic capability. Such systems can include diagnostic programs that output specific action if ill health of the system is detected. Fail-over capability can be included so that a failure can cause switching to built-in redundant systems.

11-3 Digital Interlocks within the Control System

Each section of the modulating or load-following control system must be examined in detail to determine all of the automatic switching, manual switching, and tracking requirements.

Interlocking is required to prevent misoperation of the control system. It is not appropriate to have the air flow control on automatic while the fuel flow is on manual. Should the automatic control reduce the air flow to too low a value for the fuel being fired, a dangerous and potentially catastrophic situation would exist. A similar situation would exist if the fuel were on automatic and the air flow were on manual.

Suppose the induced draft is being controlled by the furnace draft and the controller is on manual while the forced draft is on automatic. An increase in the forced draft could cause excessive furnace pressure—a dangerous condition. To avoid this possibility, a control system interlock system should be installed to prevent such misoperation.

Another aspect of the problem above is protection against the failure of control system power supplies. For that reason, the system may be compartmented so that a power supply failure that would affect one controller in that compartment or one control device regulated by such controls would cause all controllers in that compartment to revert to manual control. An example is to have all the fuel and air flow controls in one compartment, all of the steam temperature and reheat temperature controls in one compartment, and all of the feedwater controls in one compartment.

Protection against maintenance error while the process system is in operation and on automatic may also require interlock protection. A large electric utility control system is usually composed of a number of cabinets and many plug-in electronic cards that look alike. In such systems it is possible to place part of the overall system on manual control for maintenance purposes. If this is done, it is also possible to inadvertently unplug a controller card that is furnishing control action for a part of the system that is still on automatic. A suitable interlock can provide protection by having the system take security action if a card holding an automatic mode controller is withdrawn from its connector. An example of an action could be for all controls in that compartment to revert to manual and hold their output signals.

It is also possible to plug in the wrong module and cause catastrophic damage should a control loop on manual then be placed on automatic. To protect against that problem, the system can be designed to recognize the correct card for the slot and reject or alarm if the wrong card is inserted.

Interlocks can also be designed into the system to enforce a particular order of placing control loops on automatic or going from automatic to manual. Typically, the first control subsystem to go on automatic is furnace draft control since it is independent of other loops. Next is the feedwater control, which can be placed on automatic as soon as reliable signals of drum level, steam flow, and water flow are obtained.

It is in this condition, with feedwater and furnace draft on automatic, that boiler testing is usually made. Only after the testing is completed and the rest of the control system is calibrated based on the test data should steam temperature control and fuel and air flow control be placed on automatic. For an industrial boiler, the last control loop to be placed on automatic is the steam pressure control. For an electric utility boiler this is accomplished by the boiler-turbine coordinated control or boiler or turbine following front end control. The last to be placed on automatic is the load demand control. The reverse of the order described is to be followed when transferring controls from automatic to manual. The control sequence above can be designed into an interlock system that will inhibit any such operator action that is incorrect.

11-4 Classification of Trip Interlocks Relative to Potential Consequence

Two separate classifications involving trip interlock action should be made. One relates to the "hazard" involved in the action. Such an action can occur through a healthy interlock circuit that is operating properly based on the inputs to the system. The action can also occur due to a failure or fault within the system. The hazard can be potential catastropic damage to the operators, to the process equipment, to the process plant, and/or to the environment. The second classification relates to the interlock circuitry and how its integrity is affected by faults within the system. The classification of the interlock system circuits provides a means of matching the reliability of a particular classification of interlock circuit to that of the degree of the particular hazard involved.

Interlock Circuitry

The classification of the interlock circuitry can be made relative to the action that occurs upon a single point failure or fault. This may be called the degree of redundancy or fault tolerance. The goal is keeping the protection in force at all times but recognizing that failures will occur. The classifications described below are named on the basis of the number of unflawed interlock circuits necessary in order to retain interlock protection:

(1) 1-o-o-1—One out of one. A single circuit in which a single point failure in the system will cause an output action. The circuit itself should be designed so that the any output action is always safe (e.g., shut down the equipment).

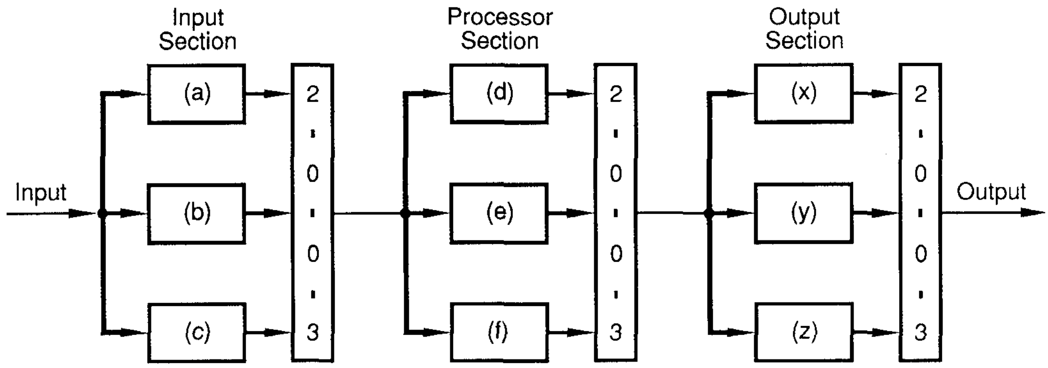
(2) 1-o-o-2—One out of two. This is called redundancy. Two protective circuits are operating essentially in parallel. A single point failure will disable one of the two circuits while the redundant circuit continues to provide the needed protection.

(3) 2-o-o-2—Two out of two. This is not redundancy because a single point failure in either circuit will cause an output tripping action. This allows no fault tolerance yet has two circuits required to hold. Where 1-o-o-2 might be a parallel circuit (normal operation energized with deenergize to trip), 2-o-o-2 would be a series circuit. If 2 flame detectors are required to see flame or the boiler will trip, this is 2-o-o-2. This demonstrates that it is possible to have a 1-o-o-1 circuit with a 2-o-o-2 portion, a 1-o-o-2 circuit with a 2-o-o-2 portion, or a 2-o-o-3 circuit in which a critical portion is 2-o-o-2.

(4) 2-o-o-3 Voting—This could be called triple redundancy. In this type of interlock system, the output of two out of three individual interlock circuits must agree to hold in a circuit in monitoring normal operation. If two out of three agree to trip, tripping action will trip the process equipment device. Any single point failure involving the interlock devices will not trip the operating equipment. A maximum of two such failures will trip, just as a minimum of two good circuits will allow continued operation.

(5) 2-o-o-3 Voting with Fault Tolerance—While this type of system requires two out of three voting to keep process equipment operating, more than two single point failures in separate circuits can be tolerated. For example, if the total circuit is made triple redundant and each circuit broken into three sequential parts, 27 separate potential pathways in 9 circuit segments exist.

Assume the three sequential parts are named "input," "processing," and "output." Input 1 can connect to processing 2 and output 3. With 2-o-o-3 voting in each of the like parts, a single point failure in two out of three of such a circuit part (e.g., "input") will cause the safety action to take place by tripping the piece of process equipment. With the three sections



1. Any single point failure (fault) will not cause a trip but will cause failure of section circuit.
2. Two faults will trip if both faults are in same section, but different circuits.
3. Three faults will cause a trip if the 3rd fault is the second in any section, but different circuit.
4. Three faults can be tolerated if faults 1-3 are in different sections.
5. Four faults will trip if the 4th fault is the second in any section, and in different circuit.
6. All second failures in any section circuit cannot be tolerated.

Figure 11-7 Two out of Three Voting with Fault Tolerance

of the circuit and 2-o-o-3 voting in each like circuit part, a total of two, three, or four single point failure possibilities, out of the total of 9 circuit sections, will cause tripping to occur. One, two, or three single point failures can occur and the equipment will continue to operate with full 2-o-o-3 interlock protection. This is shown in Figure 11-7. Figure 11-7 does not necessarily directly represent any existing interlock device but was developed to demonstrate the concept involved.

If only total circuit output 2-o-o-3 voting is used, up to five single point failures could occur and 2-o-o-3 voting on the outputs would be satisfied. A sixth single point failure would cause tripping to occur. However, such an arrangement would be composed of the circuit of one "input" segment, one "processor" segment, and two "output" segments. This arrangement might satisfy the 2-o-o-3 output voting requirement but would lose its integrity since two thirds of the total circuit would depend on 1-o-o-3.

It is usually not possible to achieve total redundancy that would be immune from a single point failure in the total system. Such situations might require three total flow measurement systems on the same flow, three primary elements and transducers, three sets of shutoff valves in series, or three motor circuit breakers with associated input contacts, or three separate sources of electric power that didn't originate in the same place.

Alarms should advise the operator or maintenance personnel when any "non-tripping" fault occurs that would cause the interlock system quality to degrade, even though full safety protection continues to be maintained.

Boiler Hazards

Some of the more serious hazards related to boilers are as follows:

(1) Loss of flame in a single burner should shut down a burner. Loss of all flame should shut off all fuel immediately.

(2) Operation of forced draft without induced draft. An interlock should prevent this.

(3) Loss of a single forced or induced draft fan of a pair. An interlock should close the dampers on the failed fan immediately.

(4) Operating the boiler with the drum water level low enough to cause damage to the boiler. Such a water level should shut off the fuel immediately.

(5) Operating a balanced draft boiler with excessive pressure in the boiler furnace. While some pressure might be tolerated very temporarily, pressure above a set point level should immediately trip all fuel to the boiler.

(6) Feedwater flow below a set point flow should immediately trip the fuel to a once-through boiler.

(7) Any reheat spray water should be shut off immediately if a turbine trips to avoid water damage to the turbine by back flow through the cold reheat steam line.

11-5 Limits and Runbacks

Limits and runbacks may be applied in essentially two different ways. Either way, control tracking is involved. One form of limiting action is through the use of high or low select functions or high or low limiter functions within the modulating control logic. In this way the signal controlling a loop may be transferred from one signal to another (e.g., fuel flow may be limited to an appropriate value for the available air flow). Other limiting functions involve the use of interlocks.

If two one-half size boiler feed pumps are being used, there will be feedwater for approximately 60 percent of full load should one of the pumps fail. An interlock switch can immediately limit the load demand to a value under 60 percent. At the same time, tracking is initiated so that the value of the load desired from the dispatcher will not exceed the limit demand value. If the load at the time is above the limit value, the reduced load demand runs the load back to the limit value.

Low limits are also needed. These limits are usually implemented either mechanically in the process equipment or through fixed low limiters. The low limits can be applied through a low limit selector with an adjustable low limit input or through a low limiter in which the low limit adjustment is within the low limiter. If a low select is being used, a manually adjusted low limit signal is one of two or more inputs with the low limit overriding all other inputs. If a fixed limiter is being used, there would be one input to the limiter with the output adjusted to the limit set value.

Some examples of necessary low limit applications are as follows:

(1) Limiting air flow to a 25 percent of full load value: mechanically, by blocking partly open a control damper or adjusting the damper linkage; in the control system with a high select and low limit signal.

(2) Blocking control action when fuel burner pressure is at a minimum stable value. This can be done mechanically with a bypass fuel pressure regulator around the fuel control valve.

(3) Blocking pulverizer demand action when the pulverizer control is reduced to its low level stable turndown condition. This can be accomplished by a low limiter in the pulverizer demand signal.

(4) Tripping fuel when an unsafe furnace or boiler condition exists because of low drum level or unsafe combustion due to low fuel pressures.

(5) Blocking the firing rate demand signal when the boiler is at the minimum turndown rate. This can be accomplished with a low limiter applied to the firing rate demand signal.

(6) Protection against low steam pressure may be used individually or as a backup to other limiting action. Assume the firing rate demand is on manual but turbine valves are manually controlled and too far open for the firing rate. The steam pressure will begin to drop, and at some point the initial pressure regulator in the turbine governor will begin to reject electrical load by starting to close the turbine valves. This will continue until the electrical load and the firing rate (boiler load) are in balance. This is a form of turbine following control.

In all cases, whether the limits applied are "high" or "low," tracking action must take place to maintain the rest of the control system in an aligned condition for smoothly moving away from the limit as load demand changes.

Section 12
Feedwater Supply and Boiler Water Circulation Systems

A typical boiler feedwater supply system consists of three basic parts: (1) a set of boiler feed pumps, (2) valves, feedwater piping, and headers, and (3) feedwater heaters. This system is supplied with condensate or chemically treated water at a relatively low temperature. The heaters and boiler feedwater pumps condition the water for proper admission to the boiler. The piping and headers connect the feedwater supply, heaters, valves, and pumps to the boiler.

12-1 The Basic System

Figure 12-1 shows a general arrangement of a basic feedwater supply system. The relatively cool water is admitted to the deaerating heater. Water leaving the heater is deposited into an integral heated-water storage tank. The storage tank is connected to the suction of the boiler feed pumps. At the discharge of the boiler feed pumps, a recirculation line containing a control or shutoff valve is connected. Downstream from this connection is a check valve between the pump and the feedwater supply header so that pressure from the supply header cannot return to the pump. With more than one pump connected, the check valve would be closed on any pump that is not in operation.

The recirculation line is open at low flows and is sized for approximately 15 to 20 percent of pump capacity. The necessity for this recirculation can be deduced from Figure 12-2, which shows a typical set of characteristic curves for a constant speed boiler feed pump. Note that the power consumption is 60 to 70 percent of full load power at a 0 flow condition. With this condition of power input and no flow to dissipate the heat, the temperature would rise very rapidly and damage the pump. To avoid this potential damage, approximately 15 to 20 percent of the water is recirculated to keep the pump from cavitating and overheating.

If the recirculation valve were operated manually, safe operation would dictate that the valve always be in the open position. Under such operation, only 80 to 85 percent of the pump capacity would be available for use. The recirculation valve can be automated with either a proportional or an on-off control. In either case, as the water flow demand is increased above the set point of the control, the recirculation valve is closed. The relationship of these flows as capacity is reduced is shown in Figure 12-3. The recirculation capacity is then available for useful work whenever the flow is above the set point range of the recirculation control.

12-2 Heating and Deaeration

If the boiler feedwater can be heated with steam that would otherwise be wasted, the increased feedwater temperature results in less fuel required to generate the steam. Most industrial plants have enough low pressure exhaust steam for this purpose. Another reason to heat the feedwater is the necessary process of deaeration.

Before the water is put into the boiler, entrained or dissolved gases such as carbon dioxide and oxygen should be eliminated. If allowed to remain in the water, carbon dioxide will pass into the steam and turn into corrosive carbonic acid in the heat exchangers that use the steam. This corrodes the heat exchangers and the condensate return piping system. At the temperature of the boiler water, oxygen, if allowed to reach the boiler, can seriously corrode the boiler. Any remainder that leaves the boiler with the steam can corrode the heat exchangers and return lines.

Eliminating these entrained or dissolved gases before they can enter the boiler is called

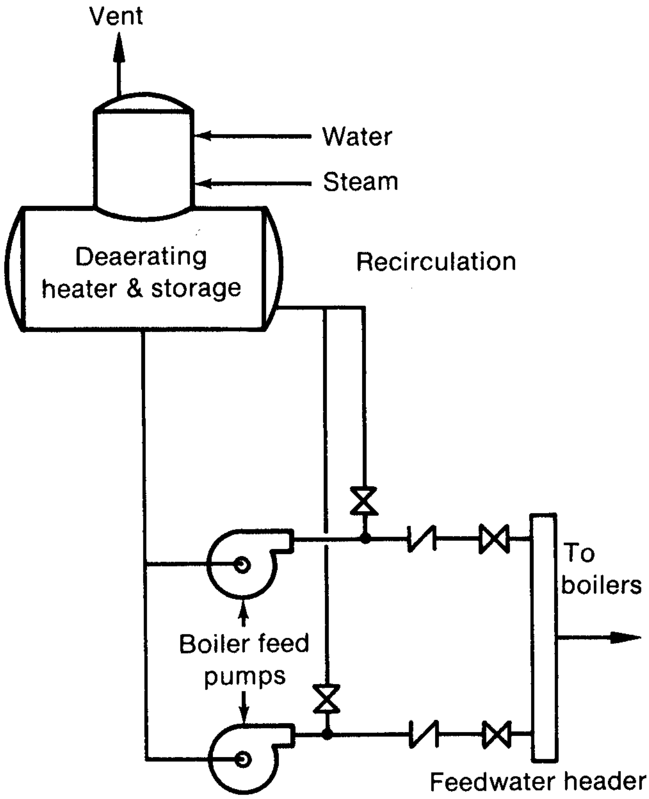


Figure 12-1 Boiler Feedwater Pumping and Heating System

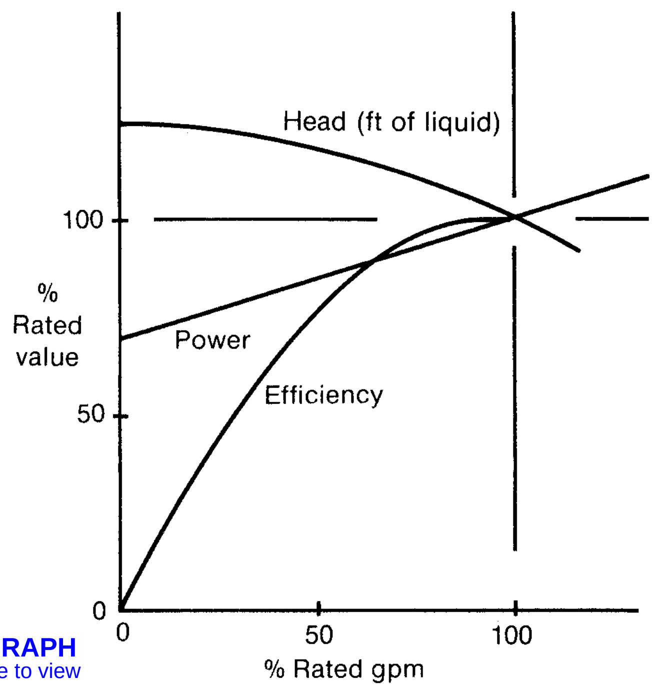
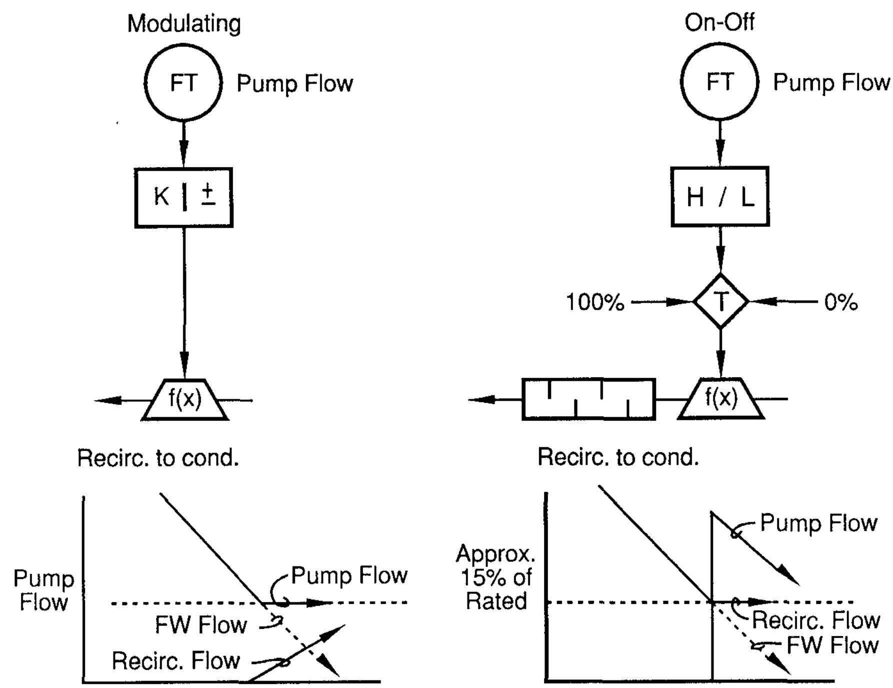


Figure 12-2 Characteristics of a Constant Speed Pump



Considerations

- Valve Pressure Drop
- Regulating vs. Shutoff Valve
- System Disturbance, Interactions
- Thermal Efficiency
- Cycling or Base Load

Figure 12-3 Relationship between Feedwater Flow and Recirculation Flow with Automatic Control

deaeration and is done by a deaerating heater. At colder temperatures, water can hold greater amounts of air, oxygen, carbon dioxide, or other gases in solution. These gases can be removed by vigorous boiling and a procedure for venting the gases to the atmosphere.

In the deaerating heater the water is heated by direct contact with steam by cascading the water-steam mixture over metal surfaces. In this way the water and steam become an intimate mixture at the boiling point. The entrained or dissolved gases are released from the water through this boiling and agitation. Because the pressure in the heater is above atmospheric pressure, the gases can then be vented to atmosphere.

Deaeration removes the oxygen and carbon dioxide as efficiently as mechanically possible. Scavenging chemicals are introduced to the boiler water to eliminate the remaining traces.

This process must be carried out with live steam direct from the boiler if exhaust steam is not available. As shown previously in Section 9, several additional feedwater heaters in series are used with electric utility units. In these, the steam for heating the feedwater between the condenser and the economizer is furnished by extracting steam from the turbine.

12-3 The Boiler Feedwater Pump

The basic characteristic curves of a typical boiler feedwater pump, the power input curve, the head-capacity curve, and the efficiency curve, are shown in Figure 12-2. In all cases the "y" coordinate shows the percent of the rated load value. Rated load is shown as 100 percent on the "x" coordinate value. The units for the discharge pressure or "head" are in feet of the particular fluid being pumped. Since cold water has a higher density than hot water, the discharge pressure in psi will be higher when pumping colder water. The capacity on the "x" coordinate is a volumetric capacity, usually (gpm) for a boiler feed pump.

Boiler feed pumps may also be operated in a variable speed manner. In this case the speed would be varied with a variable speed motor, a magnetic or hydraulic coupling, or a steam turbine. If the pump is operated in a variable speed mode, the head-capacity curves are as shown in Figure 12-4.

The dotted line shows the characteristic of pressure necessary to get water into the boiler. Note that because of the pressure/speed correalation, a significant pump speed is necessary to obtain enough pressure even though there is no boiler load. The upward curve of the dotted line represents the system flow friction as the flow is increased from 0 to 100 percent. Regulation of the water is shown to consist of moving the pump speed through the family of head/capacity curves. Slowing the pump speed as flow is reduced also significantly improves part load efficiency and reduces pump power consumption.

In Figure 12-1 the boiler feed pump suction is shown connected to the tank of boiling water in the deaerating heater. To avoid the flashing of this boiling water into steam at the pump suction, there must be pressure at the pump suction in excess of the saturation pressure of the boiling water. This is accomplished by locating the pump at a great enough difference in elevation below the heater so that there will always be this positive pressure difference. This is called net positive suction head (NPSH), which must also take into account the friction loss due to the flow from the heater to the pump.

In some installations a measured comparison is made between the temperature of the water at the pump suction and the saturation temperature of the water at the suction pressure. When

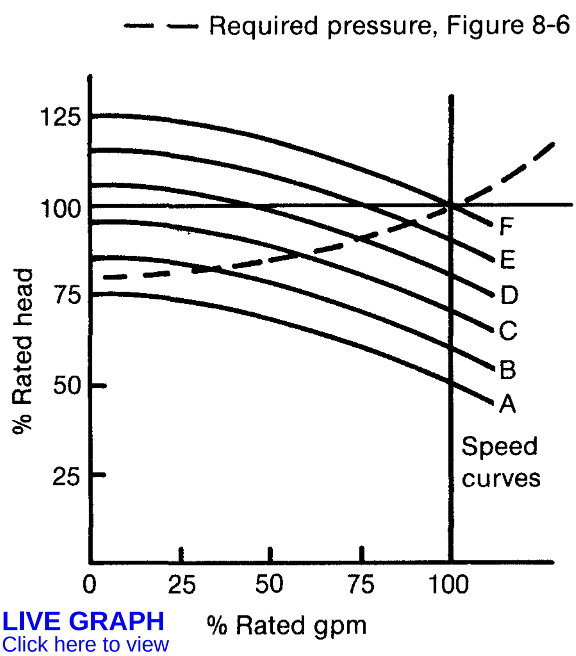


Figure 12-4 Speed Curves, Variable Speed Pump

these come together, a small cold water line to the pump suction opens automatically to assist in pump protection.

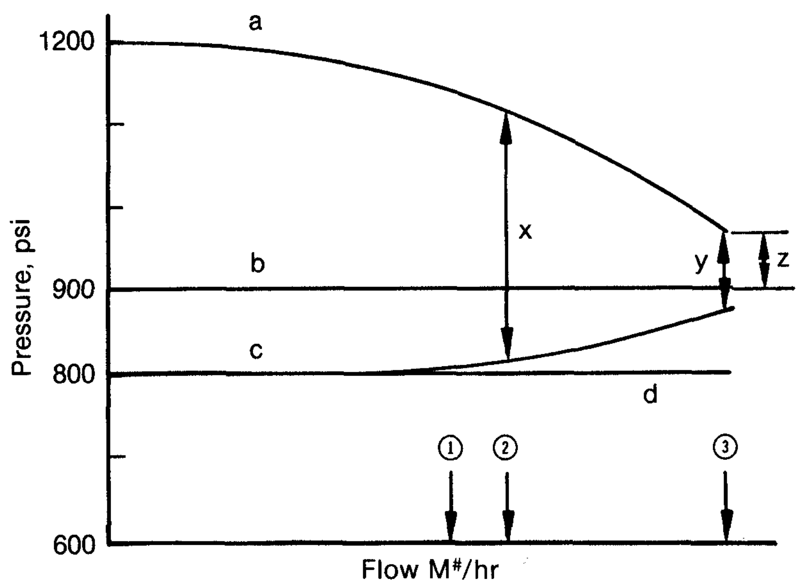
12-4 The Flow Regulation System

As previously shown in Figure 2-14, for drum-type boilers, feedwater is continuously added to the boiler through piping to the steam drum. For once-through boilers, the feedwater continuously flows through the unit to become the steam flow at the boiler outlet.

At the entrance point, it is necessary that the pressure in the feedwater system be slightly higher than the boiler pressures so that the water will flow to the boiler. The feedwater regulating system controls the flow and dissipates the pressure difference between the boiler and the supply from the feedwater pump.

If the pump or pumps are driven by a constant-speed electric motor or turbine, the feedwater supply pressure, except for the piping friction losses, will follow the pump characteristic curves. For drum-type boilers, if the steam header pressure, boiler outlet pressure, or turbine throttle pressure is maintained at a constant value, the drum pressure will be at a higher pressure as determined by the friction losses through the superheater (if the steam is superheated) and the valves and piping. In this case the feedwater flow is controlled by a standard heavy-duty control valve. The pressure drop dissipated by the valve is determined from a system head curve as shown in Figure 12-5.

While the actual flow is based on the pressure drops shown, other considerations are necessary in sizing the control valve. The lower pressure used in sizing is the pressure setting of the drum pressure relief valve, since the system must be capable of adding water with the relief valve blowing.



a. Feedwater pressure
b. Drum safety valve setting
c. Drum pressure
d. Steam header pressure

① Capacity at full firing capability
② Capacity above plus excess for control action
③ Capacity for all normal requirements plus safety valve requirements

Figure 12-5 Typical System Head Curve — Constant Speed

Additional capacity is often designed into the control valve to accommodate the additional water that is lost from the system with the relief valve blowing. In some cases, designing the control valve for the lower differential pressure and the added flow may result in an extremely oversized control valve. In this event it is better to size the control valve for good control and take care of the emergency conditions with manual or motor-operated bypass valves.

Some general guidelines for the control valve are as follows:

(1) Valve pressure drop should be not less than approximately 50 psi in order that the pressure drop will be a sufficient percentage of total system pressure drop to assure responsive control.

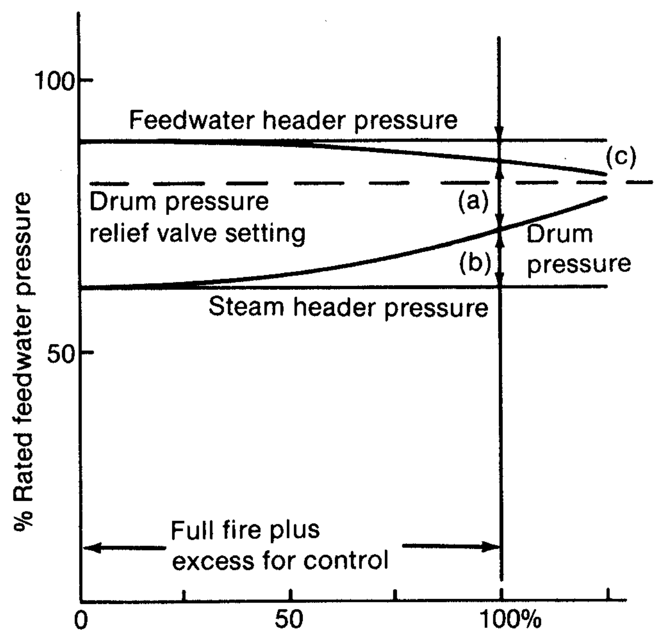
(2) Valve pressure standard should be based on the ASME boiler code if it is the first valve in the feedwater system away from the boiler drum. Otherwise, the valve pressure standard can be based on the less stringent piping code.

(3) The valve body materials should be carefully selected, keeping in mind that very pure water is "metal hungry."

(4) As noted on the sytem head curve, the pressure drop varies with flow and tends to be considerably higher at low flows.

If the boiler feedwater pumps are variable-speed and more than one boiler is used with a feedwater pumping system, the feedwater pressure is normally controlled to a set point by changing the pump speed. In this case the system head curve is similar to that shown in Figure 12-6. With this arrangement pump power is saved by reducing the pump discharge pressure. In addition, the duty of the control valve is not as stringent since the valve pressure drop is lower tends to be somewhat constant for all boiler loads. For design of the control valve, a minimum pressure drop of 50 psi is recommended.

Utility boilers generally are unit-type operations with a single boiler, a single turbine, and



(a) Feedwater valve differential pressure
(b) Superheater differential pressure
(c) Feedwater piping differential pressure

Figure 12-6 Feedwater System Head Curve, Variable-Speed Pumps

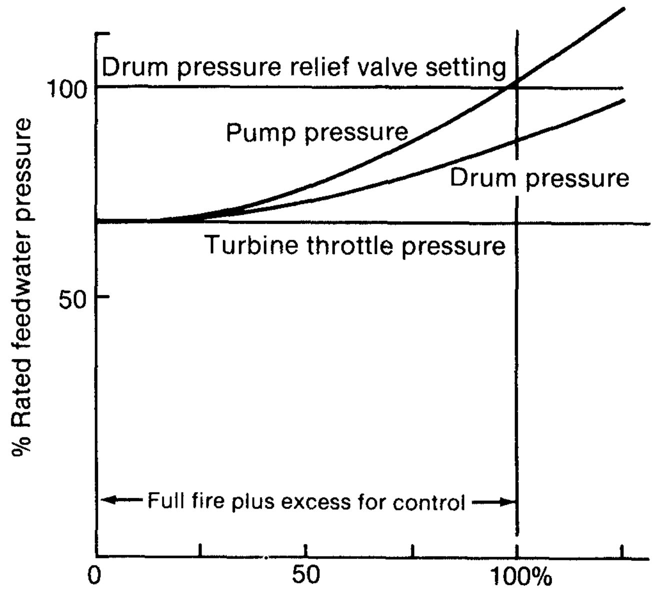


Figure 12-7 Feedwater System Head Curve — Single Boiler Variable-Speed Pump, No Control Valve

a single set of boiler feed pumps. In this case the system head curve is as shown in Figure 12-7. The boiler feed pump speed is regulated to provide just enough pressure to force water into the pressurized boiler drum or through the tubes of a once-through boiler with no extra pressure for a control valve pressure drop. With this lower boiler feed pump discharge pressure, additional pumping power is saved. This depends to some extent, however, upon the method of steam temperature control. If spray water is used, then the system must be designed to provide the necessary pressure drop for the spray water control valve and the spray nozzle.

12-5 Shrink and Swell and Boiler Water Circulation

When the steam load on a drum type boiler is increased, steam bubbles rise through the riser tubes of the boiler at a faster rate. The circulation of the water from the steam drum to the mud drum in the "downcomer" tubes and then as a mixture of steam and water up through the "riser" tubes has previously been discussed in Section 2. By the application of heat to the riser tubes, more steam bubbles are generated as more steam is demanded from the boiler. Similarly, reducing the heat applied to the riser tubes reduces steam bubble formation to satisfy the condition of reduced steam demand.

In the discussion of steam drum internal devices, it was demonstrated that there are two sections of space within the steam drum. One of these receives the steam-water mixture from the riser tubes, separates the water from the steam, and returns the remaining water to the water space. The feedwater is admitted into this relatively quiet water space. The boiler drum water level is measured also in this relatively quiet water space. From the level of the water in this space through the downcomers to the mud drum, there should be a very small number of rising steam bubbles.

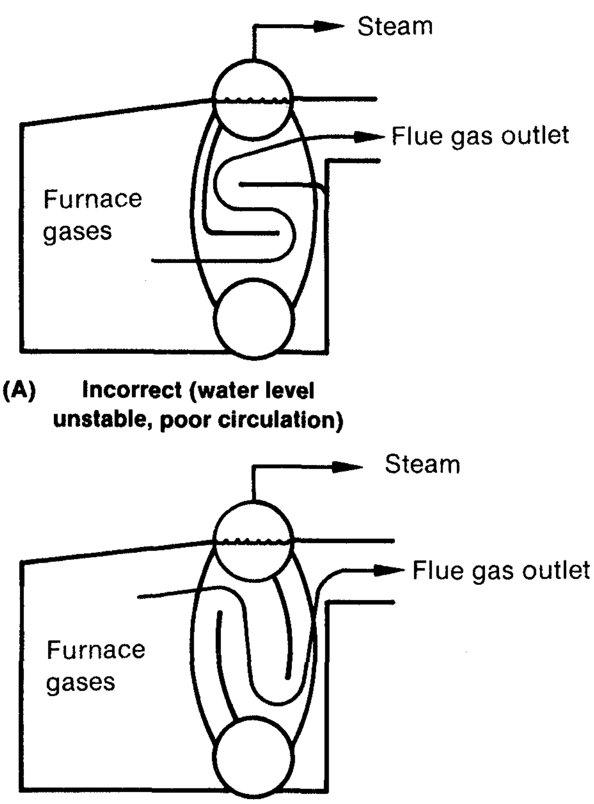
If steam bubbles rise in boiler tubes, these tubes are acting as risers. If they are connected into the steam drum water space, the rising steam bubbles may cause the measured water level to appear unstable. Whether a boiler tube acts as a riser or a downcomer is dependent upon the amount of heat received by the tube. The amount of heat received is dependent upon the temperature of the flue gases that pass around the tube and the circulation through the tube.

The flue gas baffle configuration affects the flue gas temperature in contact with specific tubes. Incorrect flue gas baffle design may cause hot flue gases to be applied to downcomer tubes, turning them incorrectly into risers and resulting in an unstable water level.

Figure 12-8 is based on an actual case of very poor gas baffle design and shows how the riser-downcomer identification can be confused by incorrect flue gas baffling. If this effect is major, as in this case, the flue gas baffling should probably be redesigned. If it is minor, it may be modified by a minor change in either the flue gas baffles or the steam drum internal baffles that separate the water space from the steam-water mixture space.

Assume that a boiler is being operated under steady-state conditions. At any point in time the boiler contains a certain mass of water and steam bubbles below the surface of the water in the steam drum. For this mass of water and steam there is an average mixture density. As long as the boiler steaming rate is constant, the steam-water mixture has the same volumetric proportions, and the average mixture density is constant.

Should the boiler load be increased, the concentration of steam bubbles under the water surface must increase. The result is that the volumetric proportions in the water-steam mixture change and the average density of the mixture, with some adjustment for the temporary change in boiler pressure, decreases. Since the mass of water and steam at this point has changed very insignificantly but the average density has decreased, the result is an immediate increase in the volume of the steam-water mixture. The only place the volume can expand is in the steam drum. This causes an immediate increase in the drum water level even though additional water has not been added. This effect of a sudden increase in drum water level as the steaming rate is increased is known as swell.



(A) Incorrect (water level unstable, poor circulation)

(B) Correct

Figure 12-8 Flue Gas Baffles Affect Riser-Downcomer Identity

When the steam load is reduced, there are fewer steam bubbles in the mixture, the average density of the mixture increases, and the volume of the steam-water mass decreases. The effect is an immediate reduction in the drum water level, although the mass of water and steam has not changed. This sudden reduction of drum water level on a decrease in steaming rate is called shrink.

The amount of water in the boiler at any given time can be called the water inventory of the boiler. If the water has "swelled" due to an increase in steaming rate, the water inventory must be reduced to bring the drum water level down to the normal water level (NWL). If the water has "shrunk" due to a decrease in steaming rate, the water inventory must be increased to return the water level to the NWL. Thus under steady steaming conditions there is less water in the boiler when steaming rate is high and more water in the boiler at a low steaming rate with the drum water level at the normal set point. This accounts for the fact that energy storage in the boiler water is higher at lower loads and lower at higher loads, as shown in Figure 7-4.

Figures 12-9 and 12-10 demonstrate the effect of changing load and inventory state. If steaming rate or load were increased and water flow to the boiler were immediately increased the same amount, the water inventory would remain constant. With this condition the drum level would be forced to remain in the swell condition. Only by delaying the water flow change can some of the excess inventory be converted to steam so that the drum water level can be returned to the set point. The reverse action that occurs as load is reduced requires an addition to inventory by delaying the reduction in water flow rate.

Several factors might change the apparent magnitude of the swell or shrink with a given load change. One of these is the size of the boiler steam drum as related to the water inventory and the change in steaming capacity. Because of greater drum volume, the swell or shrink will be less with a larger drum and no change in any of the other factors. With higher boiler pressure, steam density is greater, and the effect on mixture density, and thus swell and shrink, is less.

The question sometimes raised is why the level in the water space changes since there should be no steam bubbles under the water level in this space. The answer is that the overall effect is felt in this space due to the increased water circulation rate. With increased flow through the downcomer tubes, the frictional pressure drop increases, causing a rise in the water

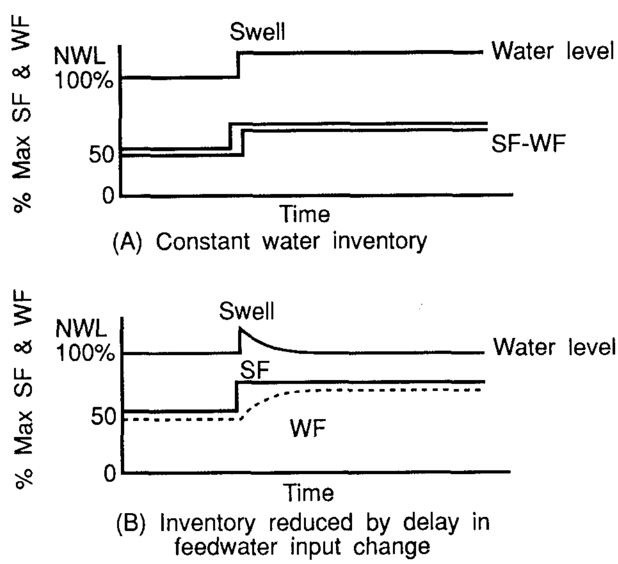


Figure 12-9 The Effect of Swell

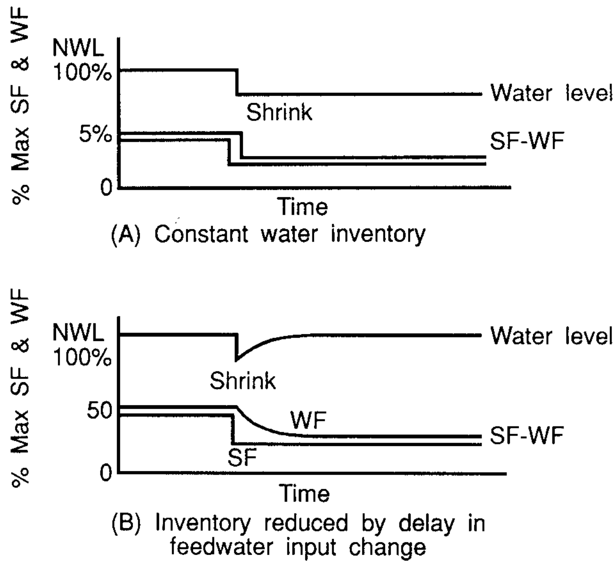
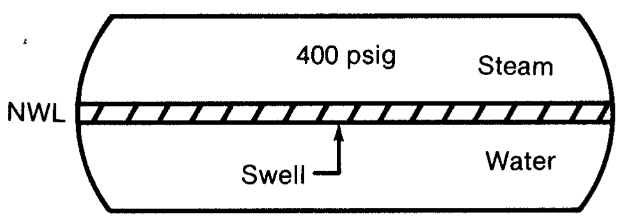


Figure 12-10 The Effect of Shrink

level in the water space to balance the effect of changing mixture density in the steam-water mixture space.

If desired, the change in water inventory related to changes in steaming rate can be calculated from tests on the boiler or can be estimated based on the dimensions of the boiler drum. Referring again to Figures 12-9 and 12-10, the roughly triangular area between the steam flow and water flow curves represents a particular mass of water. A boiler test can relate the time change and the change in steaming rate. The calculated amount should be approximately the same as that obtained using the observed swell and the drum dimensions.

This method is shown in Figure 12-11. From the pressure of the boiler, the boiler water density is determined. The observed swell and the drum dimensions can be used to calculate a volume of water. This value multiplied by boiler water density should be approximately equal to the change in boiler water inventory for the change in boiler steaming rate. These values of the change in boiler water inventory related to changes in boiler firing rate are useful in analytical tuning procedures for feedwater control systems.



Assume steam drum, 20 ft long, 4 ft diam.
Swell = 2 in. (0.166 ft)
Volume of water (cu ft) = $20 \times 4 \times 0.166 = 13.28$
Density @ 400 psig = $1/0.0194 = 51.55$ lb/cu ft
Mass of water = $13.28 \times 51.55 = 684.6$ lbs

Figure 12-11 Change in Water Inventory

12-6 Feedwater Chemical Balance and Control of Boiler Blowdown

In all boilers a proper chemical balance must be maintained. The manner in which this is done can interact with the feedwater control system. Figure 12-12 demonstrates the chemical balance. The chemical control involves the chemical content of the boiler feedwater plus the much smaller quantity of boiler water conditioning chemicals.

The boiler steam scrubbers are intended to prevent any carryover of boiler water chemical content into the steam. All chemicals that enter the boiler through injection or in the feedwater must ultimately be removed in the boiler blowdown. Assuming a constant level of chemical concentration in the boiler water and also a constant concentration in the feedwater, the chemical concentration of the water in the boiler is determined by the ratio of blowdown flow to feedwater flow. If the average blowdown flow is 10 percent of feedwater flow, then the chemical concentration of the boiler water is 10 times that of the feedwater.

As the feedwater flow varies, the blowdown flow should also vary if the boiler water concentration ratio is to remain constant. In normal practice, the blowdown flow rate is adjusted periodically. During the interval between adjustments of blowdown flow rate, the chemical concentration in the boiler water may slowly rise or fall.

If the concentration in the boiler becomes too high, the boiler drum water level becomes unstable. The design and operating conditions of a boiler determine the maximum desired chemical concentration of the boiler water. Operation of the boiler at or near the maximum desired boiler water concentration results in minimum blowdown flow and reduced fuel loss. Assuming a saturated steam boiler, the relationship between percent blowdown and percent heat loss for different boiler pressures is shown in Figure 12-13. As the pressure of the boiler is increased, the blowdown heat loss increases due to the increased saturation temperature and resultant heat content of the boiler water.

Two methods are used to remove the blowdown water from the boiler. A blowoff valve is connected to the lowest part of the mud drum. Periodic opening of this valve for a short time period removes chemical sludge that has collected in the lowest part of the boiler. Blowdown water containing dissolved solids is removed through a continuous blowdown connection into the steam drum. This piping connection removes boiler water a short distance below the surface of the water in the steam drum.

Controlling the continuous blowdown is the normal method for controlling the chemical concentration. This can be done automatically using a boiler water conductivity measurement to control the blowdown flow rate and thus the chemical concentration. Another approach is

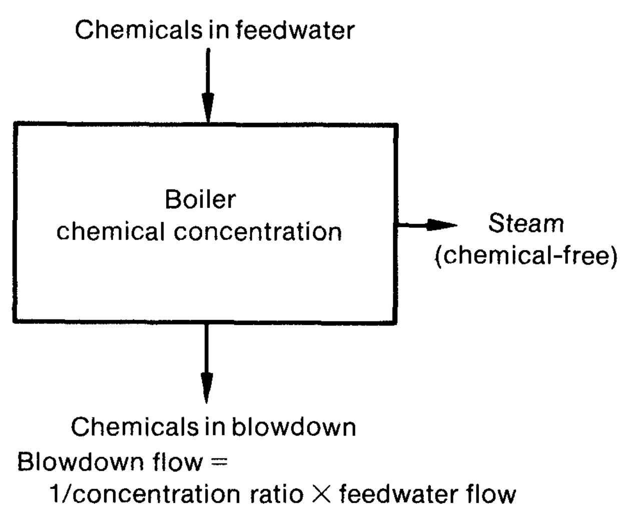
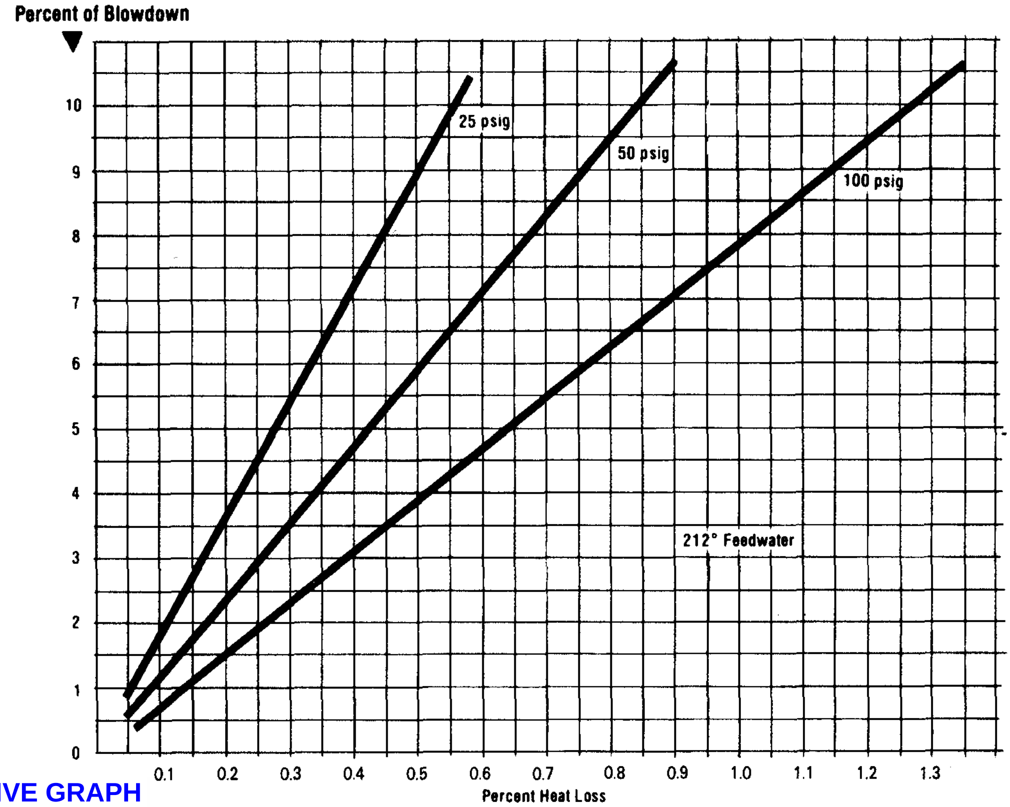


Figure 12-12 Blowdown and Chemical Balance



LIVE GRAPH
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Figure 12-13 Effect of Boiler Pressure and % Blowdown on Blowdown Heat Loss

to combine conductivity with ratio control of blowdown, ratioing blowdown flow to feedwater flow. The automatic control method saves fuel by holding the average concentration closer to the maximum concentration and thus minimizing blowdown flow.

Since the limit is on maximum chemical concentration in the boiler water, lower chemical content in the feedwater allows a greater concentration ratio. The result is reduced blowdown and lower fuel loss due to blowdown. If the chemical content of the feedwater is high and this situation cannot be avoided through use of better water, then the blowdown will be relatively high. In some such cases it may be necessary to measure blowdown flow and add it to the steam output as part of the feedwater control strategy.

If the blowdown is not measured, its percentage of the feedwater flow can be estimated using chemical concentration ratios. A typical method uses the chemical concentration of chlorides in the feedwater and boiler water. Chlorides are selected since they do not change as a result of chemical reactions that may take place in the boiler water solution. Assume that the feedwater contains 20 ppm chlorides and the boiler water contains 600 ppm chlorides. The concentration ratio is 30 and the blowdown percentage is 1/30 or 3.33 percent. In order to obtain correct results with this method, the concentrations should be stable and not changing at the time the measurements are made.

While the point is made above that blowdown flow is a heat loss out of the boiler, a large percentage of this heat is usually recovered by the use of flash tanks. The blowdown flow is taken from the boiler to a low pressure flash tank. The temperature of the blowdown causes boiling to take place and steam is generated. This steam can be used to heat feedwater. Still more heat may be removed by a secondary flash tank. The remaining water contains all the boiler chemicals removed in the blowdown and environmental regulations may require that the chemicals be removed. In this state the water can be reused or discarded as desired.

Section 13
Feedwater Control Systems

In drum-type boilers, the flow of feedwater to the boiler drum is normally controlled in order to hold the level of the water in the steam drum as close as possible to the normal water level (NWL) set point. A typical level control loop would measure level with a level sensor, process this measurement in a proportional or proportional-plus-integral controller, and regulate the flow with a control valve. This typical level control loop usually is inadequate for boiler drum level control.

This inadequacy results from the shrink and swell characteristics of the boiler, which produce level changes during boiler load changes in a direction opposite that to which level would be expected to change with the particular boiler load change. For this reason, control from level alone will produce an incorrect control action any time the boiler load changes.

If, however, the boiler is small, has a relatively large water storage, and the load changes slowly, the simple single element level control may produce control performance that can be tolerated. With larger boilers, relatively less water storage, and faster load changes, the effect of shrink and swell tend to make the simpler systems inadequate.

13-1 Measurement and Indication of Boiler Drum Level

The basic indication of the drum water level is that shown in a sight gage glass connected to the boiler drum. The typical arrangement is shown in Figure 13-1. Since the configuration of the boiler and the distance of the boiler drum from the operator may not provide a useful "line-of-sight" indication, the gage glass image can be projected with a periscope arrangement of mirrors so that the operator may easily view it. In many installations, the use of mirrors to project the water level image to a desired location for viewing may be mechanically complex or practically impossible, and other methods may be necessary. One such method is to use closed-circuit television; yet another is the use of a remote level indicator based on fiber optics.

While the gage glass is the basic measurement, the indication it provides usually is in error to some degree and is not as correct as a properly calibrated level-measuring instrument. The basis for the error can be recognized from Figure 13-1. Condensate from cooling boiler steam circulates through the gage glass. This cooling of the steam and its condensate results in cooler water in the gage glass than in the boiler drum. The greater density of the cool water in the gage glass then shows a lower height water column to balance the column of water in the boiler drum.

Assuming a typical industrial boiler, the gage glass reading often reads 1 to 3 inches of water below the actual level in the boiler drum. The deviation depends on the boiler pressure and the ambient temperature, plus piping and insulation between the boiler drum and the gage glass. For large high pressure electric utility boilers, the difference may be 5 to 7 inches. Some of the newer types of remote drum level-measuring instruments tend to compensate for the difference in readings described above. This may be in the mechanical design or the potential for moving the gage column closer to the drum.

When this fact is sufficiently understood, most of the error can be eliminated by physically lowering the gage glass. This potential for error must be well understood by anyone dealing with boiler drum level measurement and control, or much unproductive work may be performed in trying to make a gage glass and measuring instrument agree.

These types of drum measurement errors should not be confused with drum level measurement differences between the two ends of the drum. Because of water circulation-induced lateral flows inside the boiler drum, such differences are common. On large electric utility

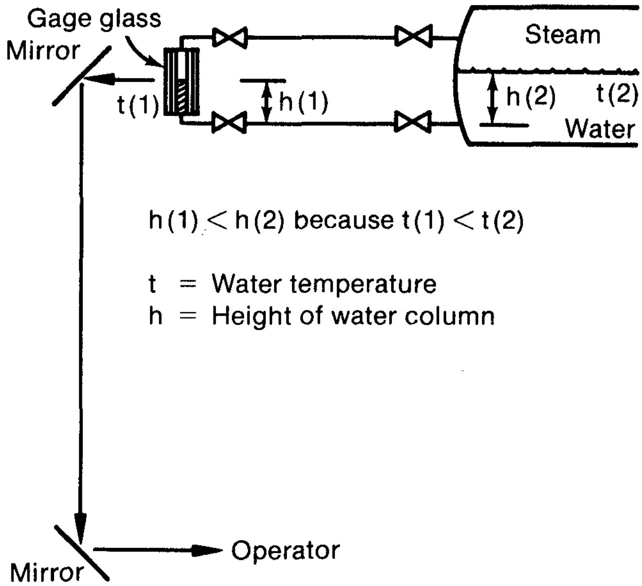


Figure 13-1 Gage Glass Drum Level Indication

units, differences of six or more inches may be observed. In some cases the higher measurement may be at one end of the drum at low boiler loads and at the other end at high flows.

A typical arrangement of a drum level-measuring transmitter is shown in Figure 13-2. The transmitter is a differential pressure device in which the output signal increases as the differential pressure decreases. Typically, the differential pressure range is approximately 30 inches with a zero suppression of several inches.

To determine the measuring instrument calibration, the necessary design data are the location of the upper and lower pressure taps into the boiler drum with respect to the normal water level, the operating pressure of the boiler drum, and the ambient temperature around the external piping. With these data and the desired range span of the tansmitter, the exact

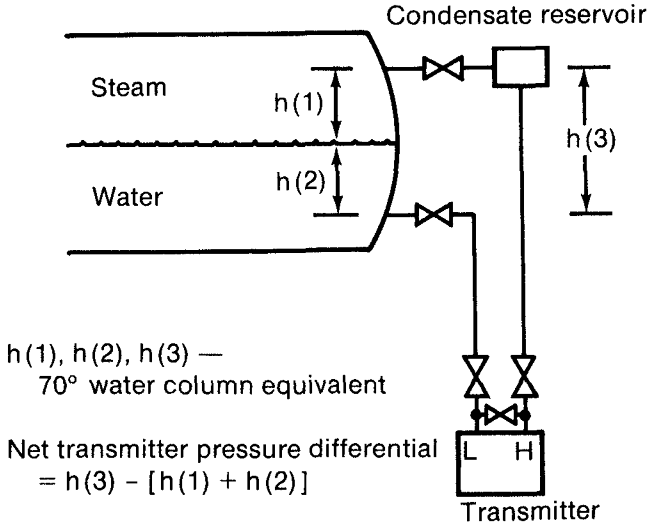


Figure 13-2 Drum Level Transmitter — Connection and Calibration

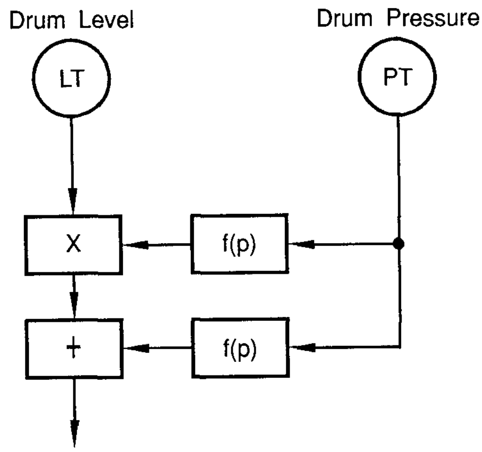


Figure 13-3 Pressure-Compensated Drum Level Measurement

calibration can be calculated by using the standard thermodynamic properties of steam and water.

On the high-pressure side of the measuring device, the effective pressure equals boiler drum pressure plus the weight of a water column at ambient temperature having a length equal to the distance between the two drum pressure connections. On the low-pressure side, the effective pressure equals boiler drum pressure, plus the weight of a column of saturated steam having a length from the upper drum pressure connection to the water level, plus the weight of a column of water at saturation temperature having a length from the water level to the lower drum pressure connection.

Since the instrument measures differential pressure, the boiler drum pressure cancels out, leaving only the water column pressure difference. Since the density of saturated steam and water at saturation temperature changes as drum pressure changes, the level-calibration data will be correct at only a single boiler drum pressure. The signal from the measuring transmitter can be pressure compensated to be correct for all pressures by providing a drum pressure measurement and using it to multiply and bias the basic drum level measurement signal. The computation arrangement is shown in Figure 13-3.

The bias is a calibration value in inches of water that represents the difference in the 100 percent (high) level calibration at a particular pressure compared to the calibration value at the base pressure condition. The multiplication changes the calibration span as the water in the drum changes density due to a change in pressure. The values of bias and multiplication are based on a series of calculations that use different drum pressure data.

Pressure compensation of the boiler drum level measurement is used almost universally on utility boiler applications. Such compensation is usually of little benefit in the normal industrial application where boilers generally operate at a constant pressure throughout the load range.

Because drum level measurement is so important, and because of its measurement uncertainty on large utility boilers, median selecting networks from three separate measurements are often used. This arrangement is shown in Figure 13-4.

13-2 Feedwater Control Objectives

There are several basic objectives of feedwater control systems. Any judgment of performance of these systems should be related to how well these basic objectives are met. The feedwater control system must also cope with external influences or specific drum level characteristics that may tend to degrade the control performance. The major difficulties encountered are shrink and swell and variations in feedwater supply pressure.

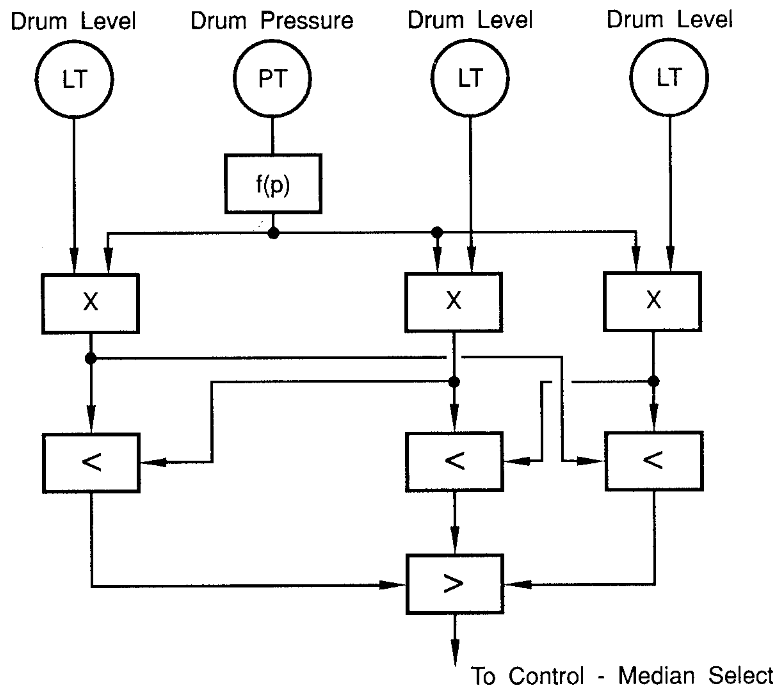


Figure 13-4 Median Select of Drum Level Measurement

In spite of these drawbacks, for drum-type boilers, the system should meet the following objectives:

(1) Control the drum level to a set point.
(2) Minimize the interaction with the combustion control system.
(3) Make smooth changes in the boiler water inventory as boiler load changes.
(4) Properly balance the boiler steam output with the feedwater input.
(5) Compensate for feedwater pressure variation without process upset or set point shift.

Of particular importance is the elimination of interaction with the combustion control system. Such interaction is evidenced by uneven flow of feedwater. Such slugs of feedwater may cause upset to the steam pressure, thus resulting in firing rate changes with no changes in the steam flow rate. The firing rate changes cause shrink or swell and accentuate and continue the problem.

Just as there are basic objectives for good feedwater control, there is a basic pattern to the desired relationships of steam flow, feedwater flow, and boiler drum level that indicate good performance of a feedwater control system. The pattern of these relationships that indicate good feedwater control is shown in Figure 13-5. As steam flow increases, an increase in feedwater would be indicated if the boiler drum level did not swell. The drum level increase should cause a reduction in feedwater flow if the steam flow had not increased.

The proper adjustment of the feedwater control system balances these opposing influences so that the basic control objectives listed above are met. If the influence of drum level is too great, the initial control action will be to reduce feedwater flow. This will ultimately cause drum level to move beyond the control set point to make up for the lost water flow. If the influence of steam flow is too great, the initial control action will be to increase feedwater flow. This action will prolong the time period that the drum level is above set point. These two actions are shown in Figure 13-6.

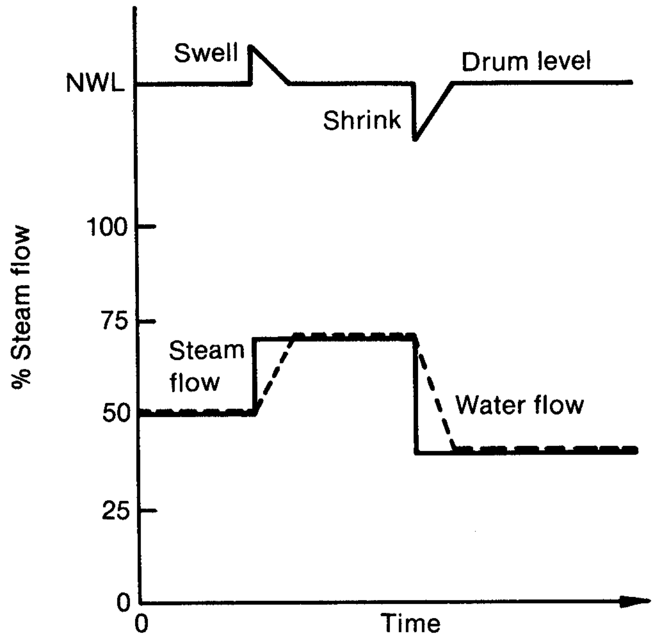


Figure 13-5 Desired Relationships to Meet Control Objectives

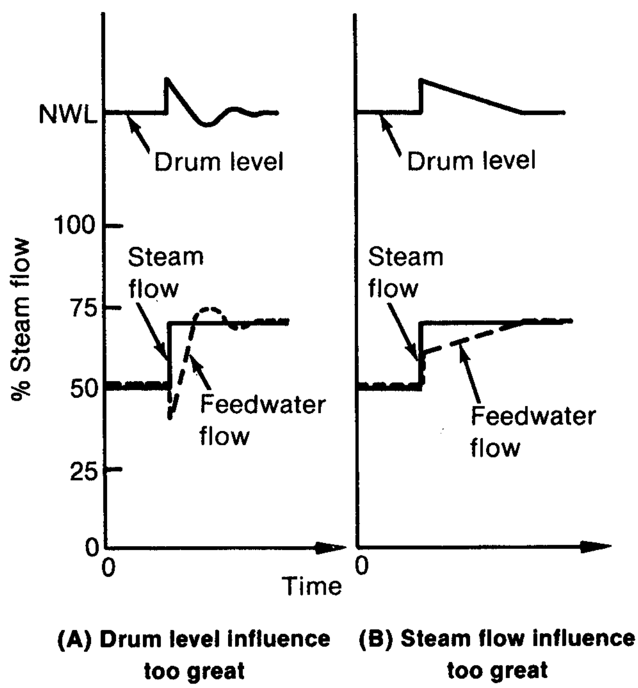


Figure 13-6 Proper Balance of Feedwater Control Influences

The correct action as shown in Figure 13-5 is for feedwater flow to make no immediate change but rather to gradually change the water inventory as indicated by the return of the drum water level to set point.

13-3 Single-Element Feedwater Control

As indicated above, control of feedwater that relies only on a measurement of drum level will probably be adequate only on smaller boilers with a relatively large water volume and with relatively slow changes in load. The most elementary of such control is the "on-off" control normally used with a firetube boiler.

As shown in Figure 13-7, the level is held within about 3/4 inch. In the typical installation, feedwater flow is 0 or 100 percent depending on whether or not the feedwater pump is running. Theoretically, this would cause interaction with the firing rate control. Though the arrangement has this drawback, the water volume is large, shrink and swell account for only a small amount, steam flow changes are usually slow, and this type of feedwater control is usually adequate. This type of control certainly does not meet the control objectives or the desired pattern of the level and flows involved.

Smooth feeding of the feedwater would tend to eliminate the interactions with combustion control and improve boiler efficiency. More power would be needed for the feedwater pump since it would operate continuously. Even though the potential gain may be small, complete operating data on any particular installation can help determine whether control improvement would be economically feasible. If an economizer is used, the feedwater control must be a continuous flow-modulating type to ensure that the feedwater flow continues through the economizer whenever the boiler is being fired. Continuous water flow is necessary to avoid damage to the economizer.

Two general types of mechanical feedwater regulators operate on the single-element (drum level only) basis. These are proportional controls with a permanent level set point offset that is associated with each feedwater control valve position.

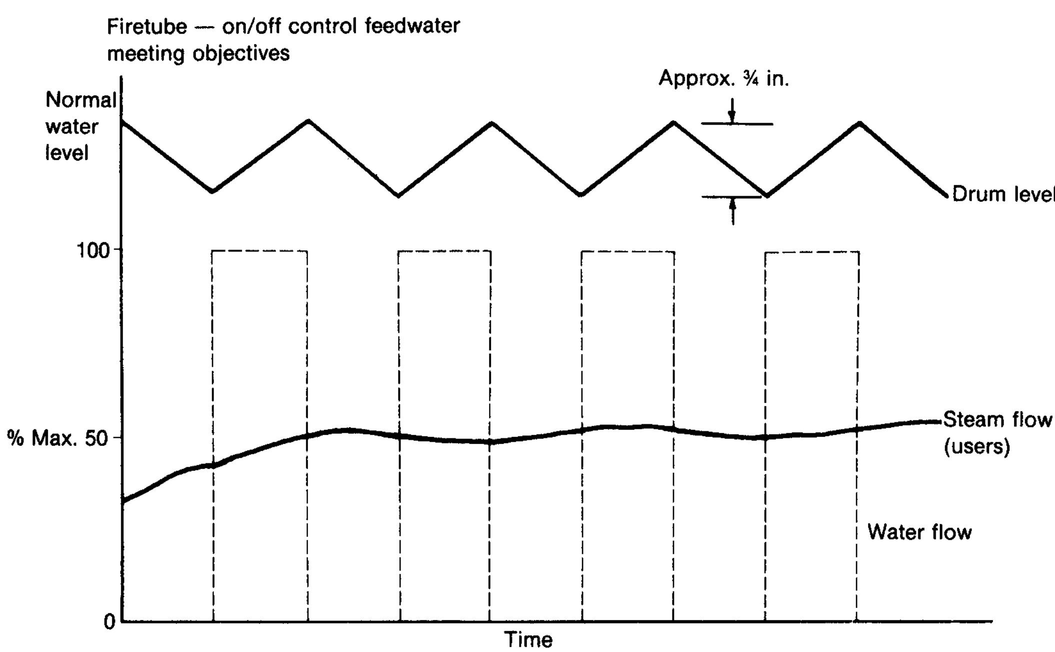


Figure 13-7 On-Off Feedwater Control Action

The first of these, called a thermostatic type, is an inclined tube, the ends of which are connected to the steam and water space in the boiler drum. Condensing steam in the upper part of the tube causes circulation and a corresponding water level to be established inside the tube. As drum water level falls, there is more steam in the tube, which causes it to heat and expand in length. The feedwater control valve is mechanically linked to one end of the tube, causing the valve to open or close as the tube expands or contracts. The elevation of the tube must be carefully located to obtain the proper set point range. The controller gain can be changed only by altering the slope of the inclined tube.

The second of the mechanical feedwater regulators is known as the thermohydraulic type. A cross section of this type of regulator is shown in Figure 13-8. As in the thermostatic type, this regulator also has an inclined tube connected to the steam and water space of the boiler. Around this tube is a closed jacket that contains water or other vaporizing fluid. This outer closed jacket is connected by copper tubing to a bellows-operated control valve. As the drum level falls, the water in the outer jacket is exposed to greater heat transfer, the outer jacket pressure increases, and the bellows-operated control valve opens. The gain of this control is fixed by the slope of the generating tube. The normal shift of drum level set point for these two proportional regulators is approximately 4 inches from low load to high load.

The performance of these two devices is very similar and can be recognized from a chart of their typical results shown in Figure 13-9.

While there are significant inadequacies in these results as compared to the basic objectives

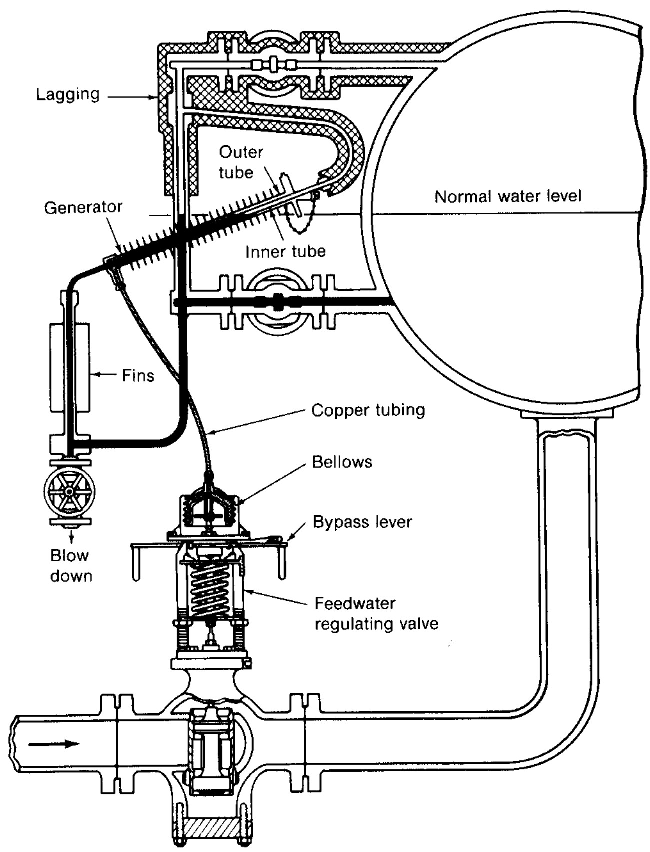


Figure 13-8 Thermohydraulic Feedwater Regulator

(From *Steam, Its Generation and Use*, © Babcock and Wilcox)

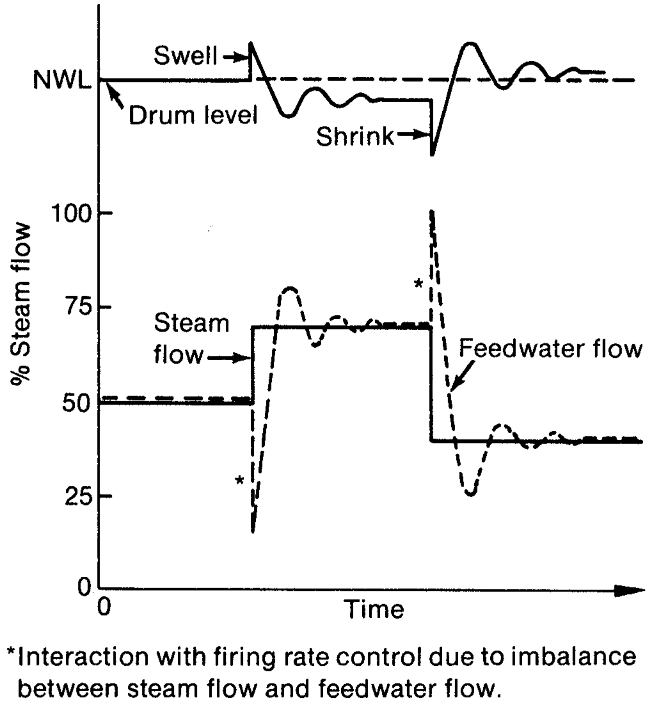


Figure 13-9 Control Action of Mechanical Single-Element Feedwater Regulator

of feedwater control, they can be tolerated in most installations of smaller size boilers with slower load changes. The most serious drawback is the interaction with the firing rate control since this would tend to degrade the boiler efficiency.

An improvement in performance over that of the single-element mechanical system can usually be obtained by using a standard feedback control loop as shown in Figure 13-10. Being able to reduce the controller gain results in less interaction. If the controller were proportional-only as in the mechanical regulator, a greater drum level offset would occur as boiler load changed. Since the low to high load level change is approximately 4 inches, it can be assumed that the 0–100 percent flow change is approximately 5 inches. Assuming a typical measurement range of 30 inches, the gain of this controller would be 6.0. Halving this gain would produce greater stability and less interaction but also twice the proportional offset.

To avoid this unsatisfactory condition, integral control is added. The integral effect must be quite slow since the level signal moves incorrectly at times of boiler drum level swell and shrink. The result as shown in Figure 13-11 is a compromise that is an improvement over the mechanical control but has less than the desired performance. The specific improvement is that the level will return to set point, incorrect action of feedwater flow during load changes is reduced, and the system is generally more stable and less interactive with the firing rate control.

The controlled device, whether a control valve or pump speed control device, should have a linear signal vs. flow characteristic as shown in Figure 13-12. The basic reason for this is that drum level deviations around the set point represent a specific quantity of water over the entire boiler load range. In the case of signal vs. pump speed, this must be nonlinear, as shown, in order that the control signal vs. water flow be as linear as possible. The large increase in pump speed for an initial small increase in control signal brings the pump speed up to the required pressure for admitting water into the boiler.

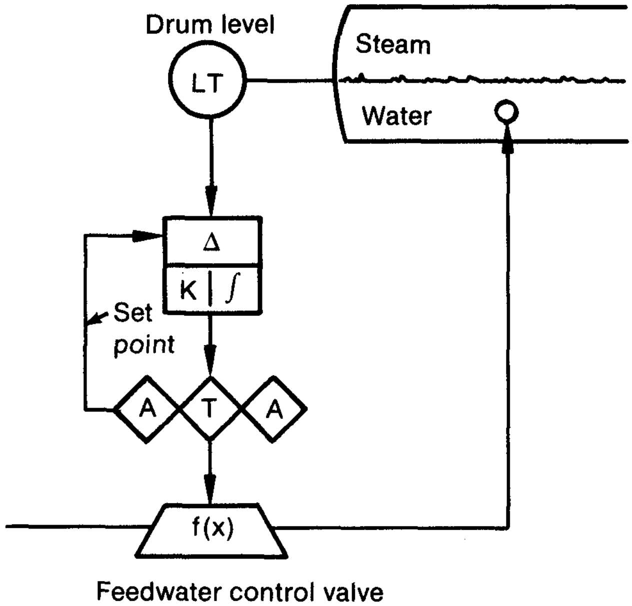


Figure 13-10 Simple Feedback Feedwater Control (Single-Element)

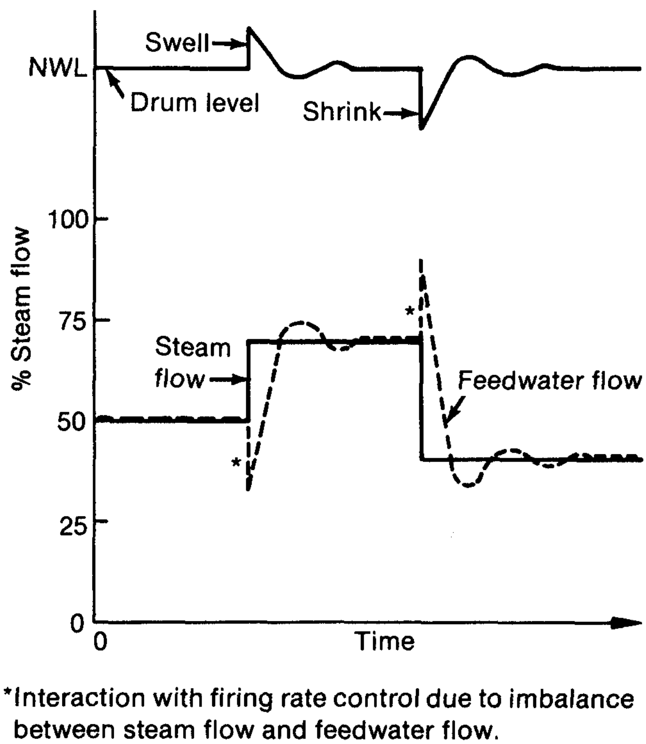
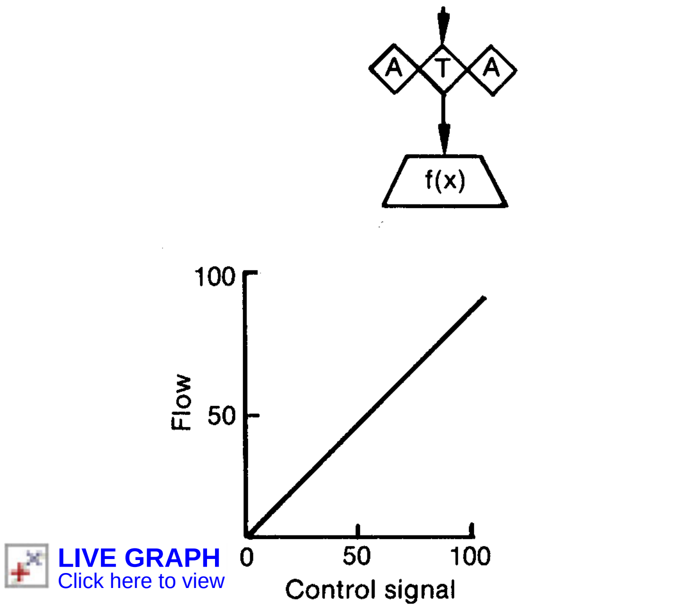
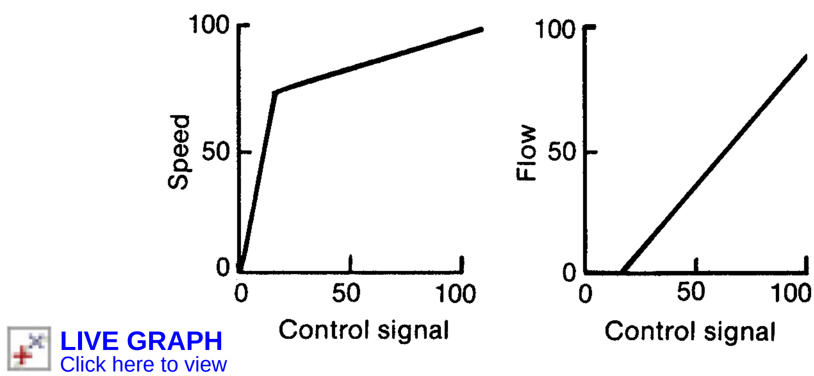


Figure 13-11 Control Action of Simple Feedback Feedwater Control (Single-Element)



(A) Control valve — desired characteristic



(B) Pump speed — desired characteristic

Figure 13-12 Feedwater Control Device Flow Characteristics

13-4 Two-Element Feedwater Control

A two-element feedwater control system is shown in Figure 13-13. This is easily recognized as a standard feedforward-plus-feedback control loop. In this case steam flow is the feedforward signal that anticipates a need for additional feedwater flow. The feedback control from drum water level is shown as proportional-only control. The control valve is characterized so that the control signal vs. feedwater flow is linear. For this system to perform properly and hold the drum level at the set point, it is necessary that the differential pressure across the feedwater control valve be predictable at each flow and that the control valve signal vs. flow relationship does not change.

Under the above conditions the system can be tuned so that performance such as shown in Figure 13-14 can be achieved. This performance is recognized as having the desired pattern of flow and level relationships, and such performance meets the boiler feedwater control ob-

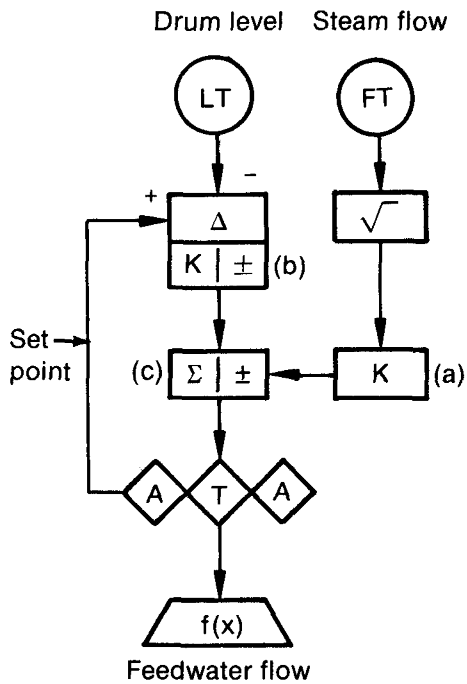


Figure 13-13 Two-Element Feedwater Control System (Feedforward-plus-Feedback)

jectives that have been stated previously, except that of compensating for feedwater pressure variation.

Tuning such a system for proper action during the shrink and swell period requires the correct balance between the effects of steam flow and drum level. As stated before, the desired condition is for water flow to hold its flow rate during a load change and change only as the drum level begins to return to its set point. In this manner water inventory is smoothly adjusted to its new desired value.

Since the drum level control signal calls for a feedwater decrease as the steam flow signal is calling for an increase, the proper gain settings on steam flow and drum level should cause them to offset each other and affect no immediate change in the water flow control valve signal. As the drum level begins to change, the feedwater valve control signal is changed to keep the system in continuous balance until steam flow and water flow are again equal and drum level is at the set point. At this point, since steam flow and water flow are equal, there is no driving force to cause further changes in boiler drum water level.

Assume that the steam flow signal range is 0 to 100 percent for 0 to 200,000 lbs/hr. Assume that the feedwater control valve is sized for a maximum flow of 250,000 lbs/hr and that the control signal of 0 to 100 percent is linear with respect to this 250,000 lbs/hr flow. The correct feedforward gain in item (a) is 0.8. In this way, if steam flow was at 200,000 lbs/hr, the 100 percent signal would be multiplied by 0.8 before going to the control valve. The resulting 80 percent of the 250,000 lbs/hr control valve capacity would provide the correct 200,000 lbs/hr of feedwater to match the steam flow.

Assume also that the range of the drum level transmitter is 30 inches with 0 or normal water level at the center with a 50 percent signal. The 0 percent signal corresponds to minus 15 inches and the 100 percent signal to plus 15 inches. A test of the particular boiler indicates that the boiler swell is 2.5 inches as the steam flow is rapidly increased by 20,000 lbs/hr.

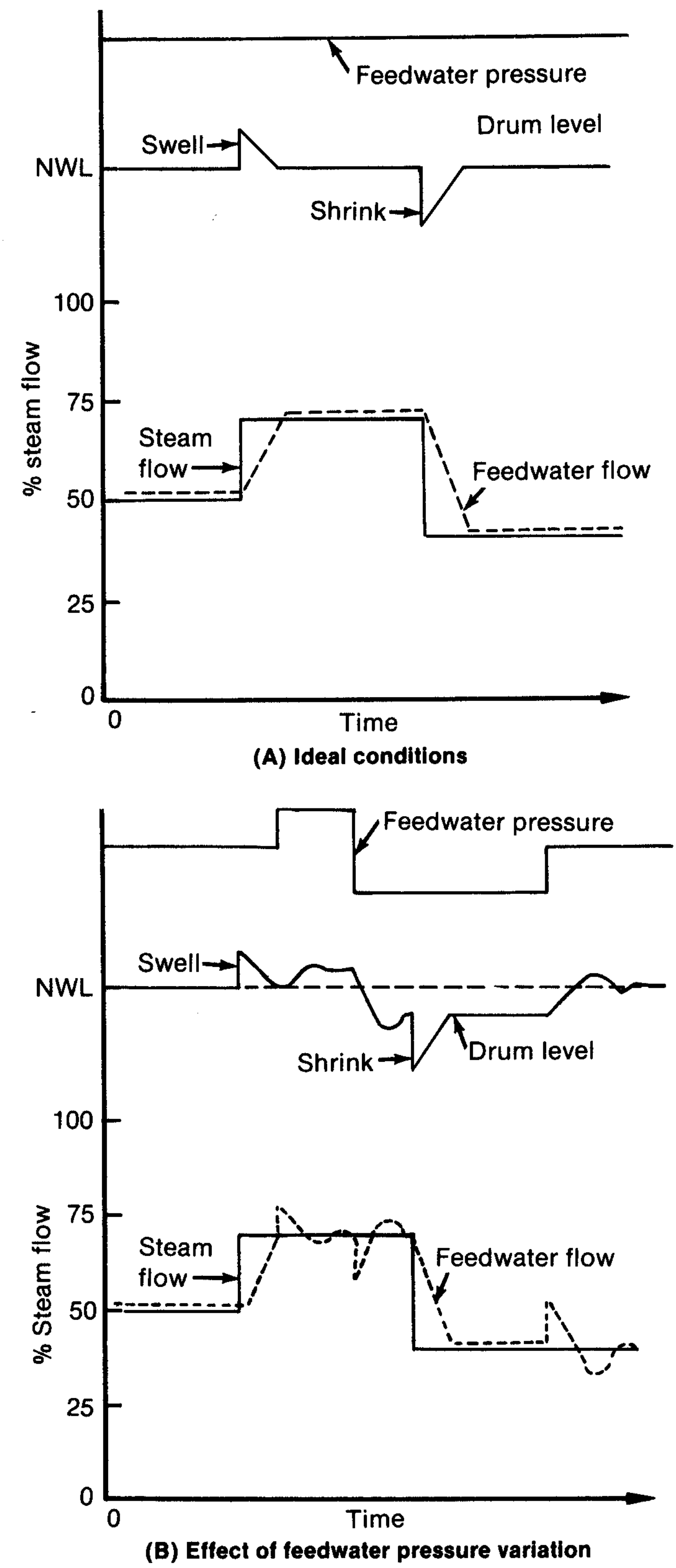


Figure 13-14 Performance of Two-Element Feedwater Control

In determining the proper gains so that the steam flow and boiler drum level signals may exactly offset each other, the steam flow gain of 0.8 must remain fixed to correctly match the steam flow range to the control valve range (Figure 13-13, item (a)). The drum level gain in item (b) must therefore be adjusted to match the effect of the 0.8 steam flow gain. Since the steam flow change is 20,000 lbs/hr or 10 percent of maximum, the net effect from steam flow is 0.8 * 10 percent or 8 percent. The drum level has changed 2.5 inches, which is 8.33 percent (2.5/30) of its range. To match the steam flow effect, the gain on the drum level signal must be 0.96 (8/8.33). This gain is then applied to the transmitted signal in item (b) in Figure 13-13. If the swell had been 1.5 inches, the drum level gain would be 1.6 (0.08/0.05).

The calibration of the system includes a bias adjustment to the output signal of item (b) and item (c). The effect of the output of item (b) should have both positive and negative possibility. If the signals in a particular system can have only positive values, the effective output of item (b) at set point should be 50 percent so that it can change in both upward and downward directions. The 50 percent positive bias of the output signal from item (b) requires the normal water level 0 percent output signal to go to 50 percent. The 50 percent signal combines with the steam flow signal from item (a). This requires a negative 50 percent bias to the output signal of item (c) so that the control valve signal will be correct. These two bias values would be 0 for systems that can work with both positive and negative values.

While the system shown will achieve all of the desired control objectives under the conditions specified, it has a serious drawback if the feedwater control valve pressure differential and thus the control valve flow characteristic are not always the same. Figure 13-14(B) dem-

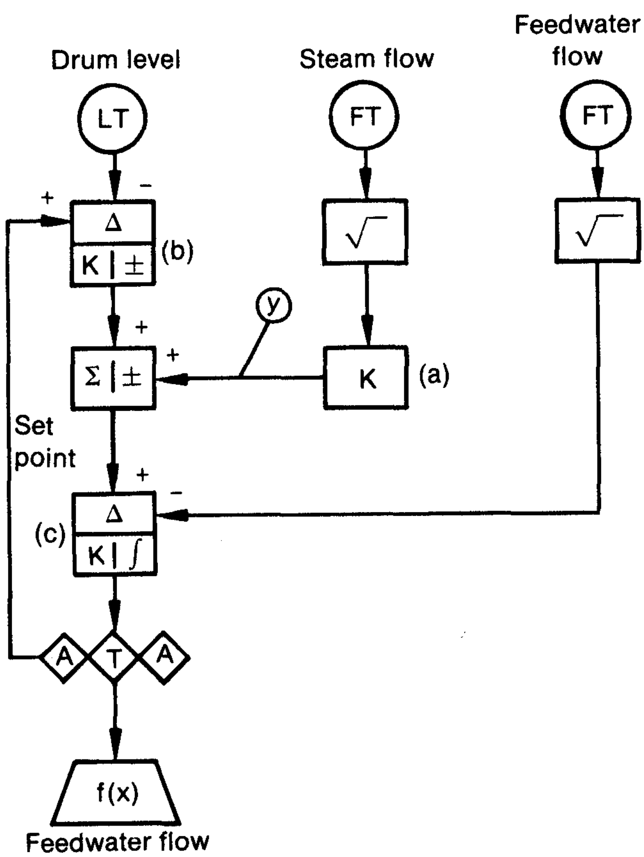


Figure 13-15 Three-Element Feedwater Control (Feedforward, Feedback, plus Cascade)

onstrates how the performance is seriously degraded by variations in feedwater pressure. Such feedwater pressure variations change the relationship between steam flow and feedwater flow. Boiler drum level is then forced to develop an offset from set point in order to bring the steam flow and feedwater flow into balance. Under conditions of unpredictable or variable feedwater pressure, three-element feedwater control is necessary if the desired results are to be achieved.

13-5 Three-Element Feedwater Control

Two-element feedwater control uses the two measurements of steam flow and boiler drum level. Three-element control adds the measurement of feedwater flow into the control strategy. In the preceding paragraphs, it was demonstrated that an unpredictable feedwater valve control signal vs. feedwater flow characteristic seriously degraded control performance. Three-element control assures that the signal vs. feedwater flow will have a constant relationship by replacing the open-loop flow characteristic of the feedwater control valve with a closed-loop feedback control of feedwater flow.

There is more than one way to arrange a three-element feedwater control system, but the most common can be described as a feedforward-plus-feedback cascade control. This arrangement is shown in Figure 13-15. Two other arrangements are shown in Figures 13-16 and 13-17.

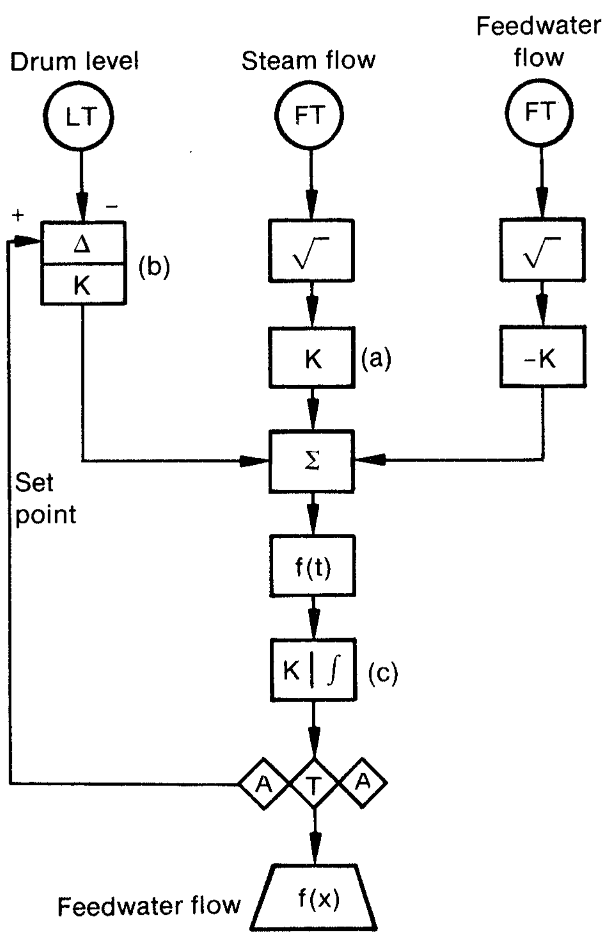


Figure 13-16 Three-Element Feedwater Control (Alternate)

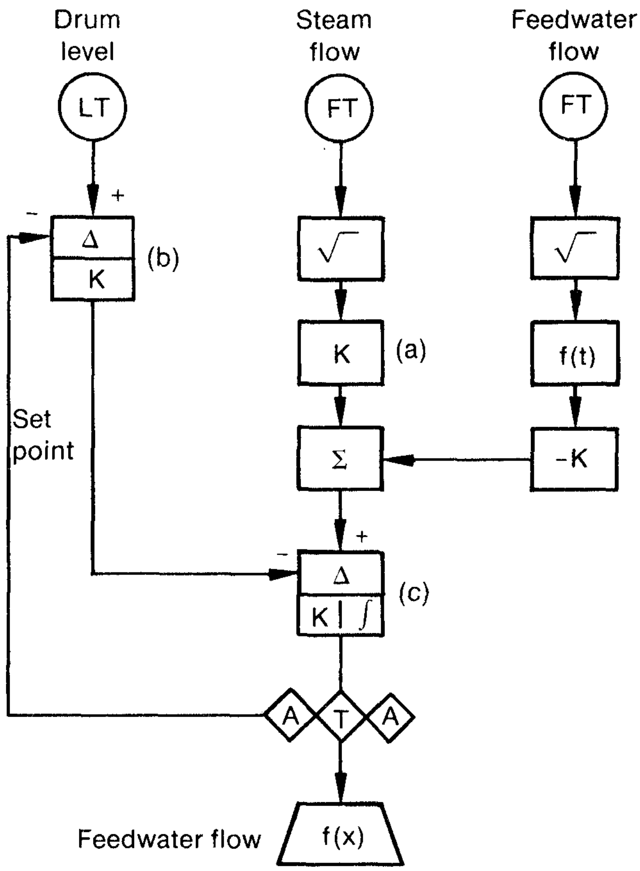


Figure 13-17 Three-Element Feedwater Control (Alternate)

The most common arrangement (shown in Figure 13-15) is tuned using the same thought process that is used with the two-element system except that the feedwater flow control is substituted for the control valve characteristic. If the feedwater flow measurement is 0 to 250,000 lbs/hr, and the steam flow measurement is 0 to 200,000 lbs/hr, the gain of item (a) would be 0.8. With the same drum level transmitter and swell effect of 8.33 percent of drum level span for a 10 percent change in steam flow rate, the gain of the proportional level control (b) would be 0.96.

The tuning of the feedwater control loop would be typical of a flow control loop. A reasonable starting point for the tuning of the flow controller (c) would be a gain of 0.5 and an integral setting of 10 repeats per minute. The tuning and calibration constants of the control arrangements in Figures 13-16 and 13-17 should be identical to those of the arrangement in Figure 13-15.

Properly tuned, the performance of the three-element control should appear approximately as shown in Figure 13-18. Note that this performance, although the feedwater supply pressure may vary, meets all the control objectives and contains the correct pattern of relationships between steam flow, feedwater flow, and boiler drum level.

In some applications, the drum level measurement contains process noise from slight to severe fluctuations in drum level. In addition, the measured steam flow may contain fluctuations or process noise. These noise components may cancel or add depending on their characteristics. If they add, they often create control stability problems. The control arrangement in Figure 13-16 offers more flexibility in eliminating the undesirable process noise effects. In

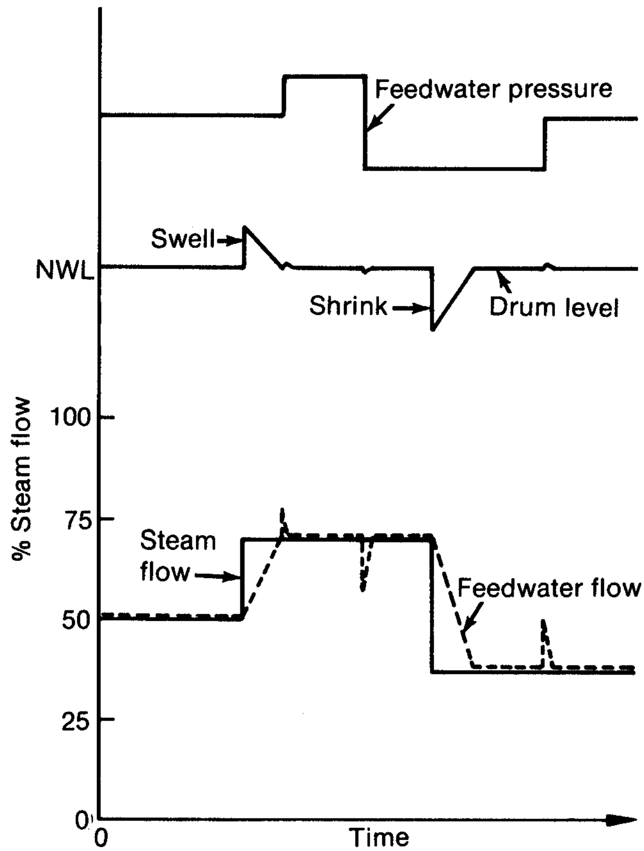


Figure 13-18 Performance of Three-Element Feedwater Control

this arrangement the adjustable time function (f/t) filters the error signal to the controller and thus removes the major share of the problem.

13-6 Control Refinements and Special Control Problems

In Figure 13-18 a control system arranged to cope with a span of potential feedwater pressures is shown. Each change in feedwater pressure causes a change in the control valve differential pressure, effectively changing the control valve maximum capacity. The result is a change to the optimum tuning of the feedwater flow control loop. From a practical standpoint, the control is often detuned to some extent to allow for such variations. Except in extreme cases, three-element feedwater control can be adjusted to achieve satisfactory performance.

Should the differential pressure and, thus, the effective control valve capacity change an extreme amount, it may be necessary to adaptively change the tuning gain of the flow control loop. Figure 13-19 shows a control arrangement that adds a fourth element of feedwater pressure. Such an arrangement is very seldom required but is available to solve such problems or to eliminate the need for compromise tuning. In this arrangement the total controller gain is a result of multiplying the multiplication of the multiplier by the manually adjusted gain of controller (c). The multiplication changes as the feedwater pressure changes and, thus, continually adjusts the overall gain.

In some cases the blowdown flow is variable and of a relatively high percentage that sig-

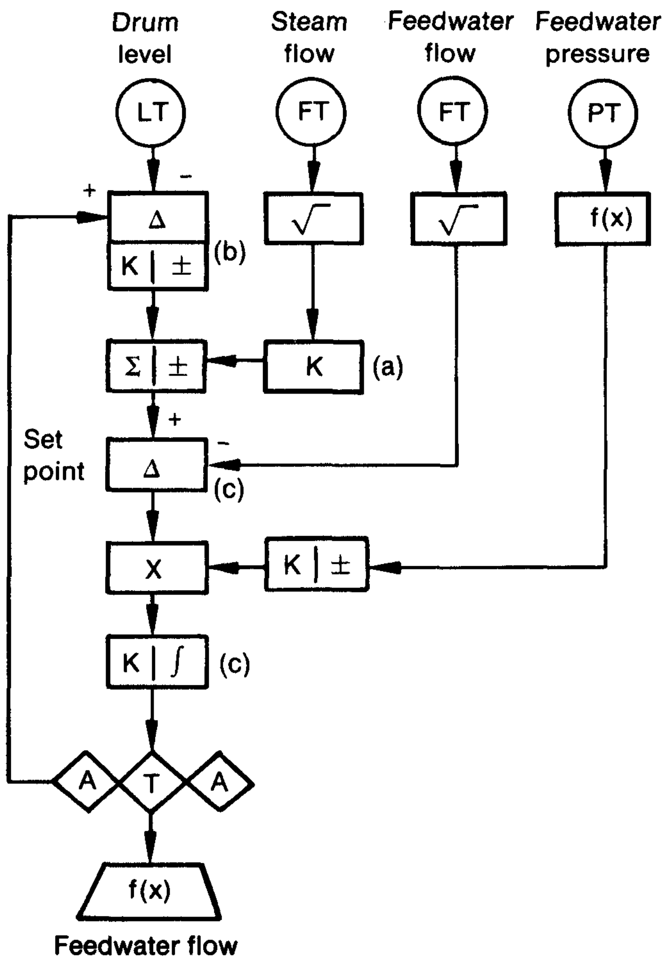


Figure 13-19 Three-Element Feedwater Control (Flow Controller Gain Pressure Adaptive)

nificantly changes the relationship between steam flow and feedwater flow. In this case it may be necessary or desirable to add blowdown flow as a fourth element to steam flow, as shown in Figure 13-20.

On installations where unmeasured saturated steam is taken directly from the boiler drum for soot blowing, an imbalance between measured steam and feedwater flow may cause drum level offsets from set point. A compromise solution to this problem is to add integral control to the level controller and adjust the integral setting so that there is very little integral effect. An integral setting of 0.05 to 0.1 repeat per minute is suggested. Theoretically, because of the "wrong way" action due to swell and shrink, any integral effect adds instability into the system.

A fine point to the proper balance between steam flow and water flow for boilers with rapid load changes is based on the changes in boiler pressure. The steam flow measurement concerns only the steam flowing through the steam line without consideration of the steam flow being added to or subtracted from the energy storage. In fact, the steam actually being generated includes the steam being generated that contributes to raising or reducing boiler pressure. Such steam can be indicated by a derivative (*d/dt*) of the drum pressure. A control arrangement that includes this steam is shown in Figure 13-21.

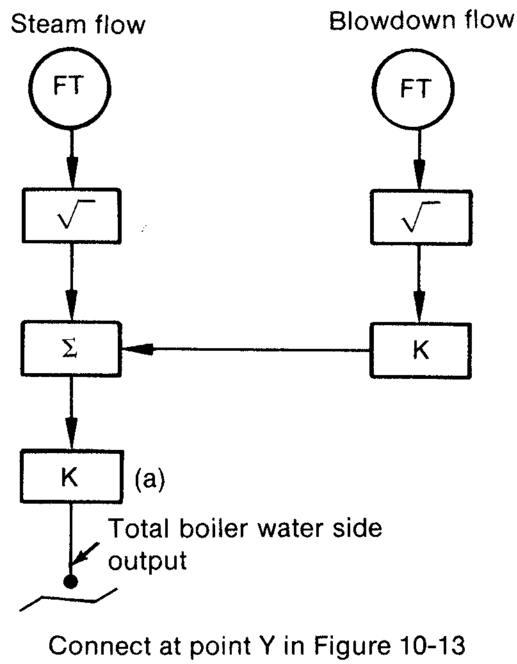


Figure 13-20 Alternate to Steam Flow Input for Three-Element Feedwater Control

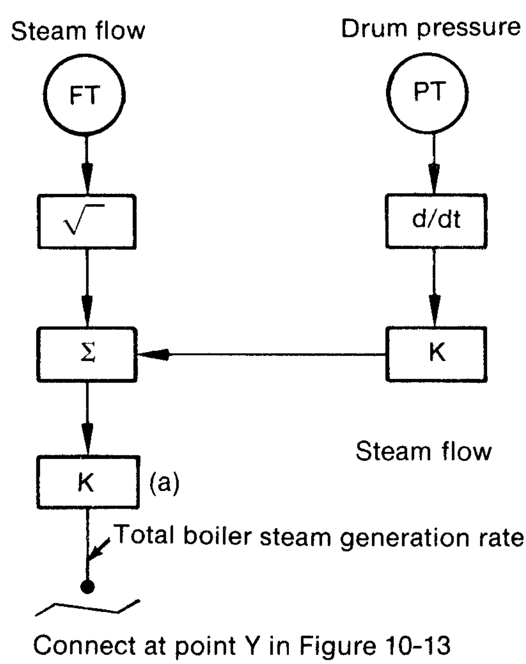


Figure 13-21 Alternate for Steam Flow Input for Three-Element Feedwater Control

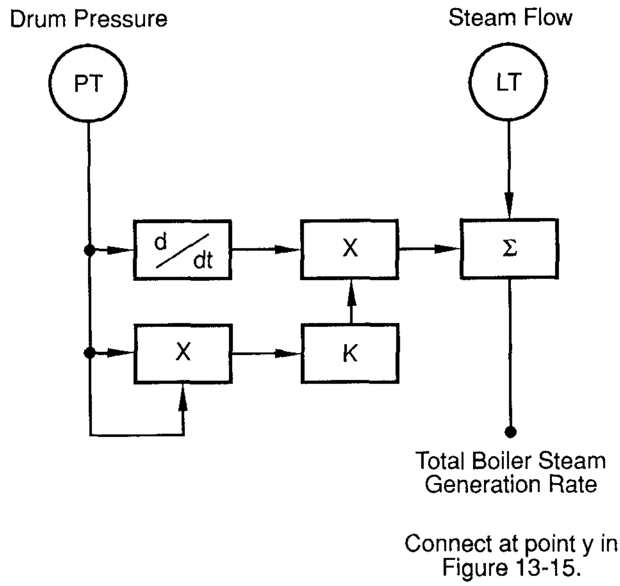


Figure 13-22 Variable Pressure Steam Flow Input for Three Element Feedwater Control

A variation of arrangement is necessary if the boiler is to operated on sliding or variable pressure. As the boiler pressure changes, the specific volume of the drum steam changes. Because of this the derivative of drum pressure relationship to steam flow changes as drum pressure changes. To account for this, a square of the drum pressure can be multiplied by the derivative of drum pressure to preserve the relationship. This arrangement is shown in Figure 13-22.

Variable pressure operation also changes the average density of the boiler water/steam mixture as the pressure varies. This changes the boiler water circulation and may change the boiler swell characteristic. Because of this, optimum calibration of the water level gain may need to change as the pressure on the boiler is changed.

13-7 Control of Feedwater for Once-Through Boilers

As discussed in Section 10, for once-through boilers the feedwater flow control is an integral part of the turbine throttle steam pressure and superheat temperheat control. The overall pumping and firing rate system is a carefully calibrated parallel firing rate and feedwater flow control system with both flow controls operating in the cascade mode.

This type of boiler can be operated in a very stable manner with the firing rate demand on manual and the turbine maintaining the steam pressure with operation in the turbine following mode. If the electrical load is correct, then the pumping is held constant and the firing rate is adjusted to the point of stable set point steam temperature.

It has been found that, due to the extreme interactions while on automatic operation of this type of boiler, the system performance can be improved by using a "control coordinator" approach. In this system arrangement, combinations of inputs trim the firing rate and also trim the feedwater flow pumping rate. A diagram of such a control coordinator system is shown in Figure 13-23.

The decision table in Figure 13-24 demonstrates how this coordinator functions. For example, the control action called for when the steam temperature is "high" should also be

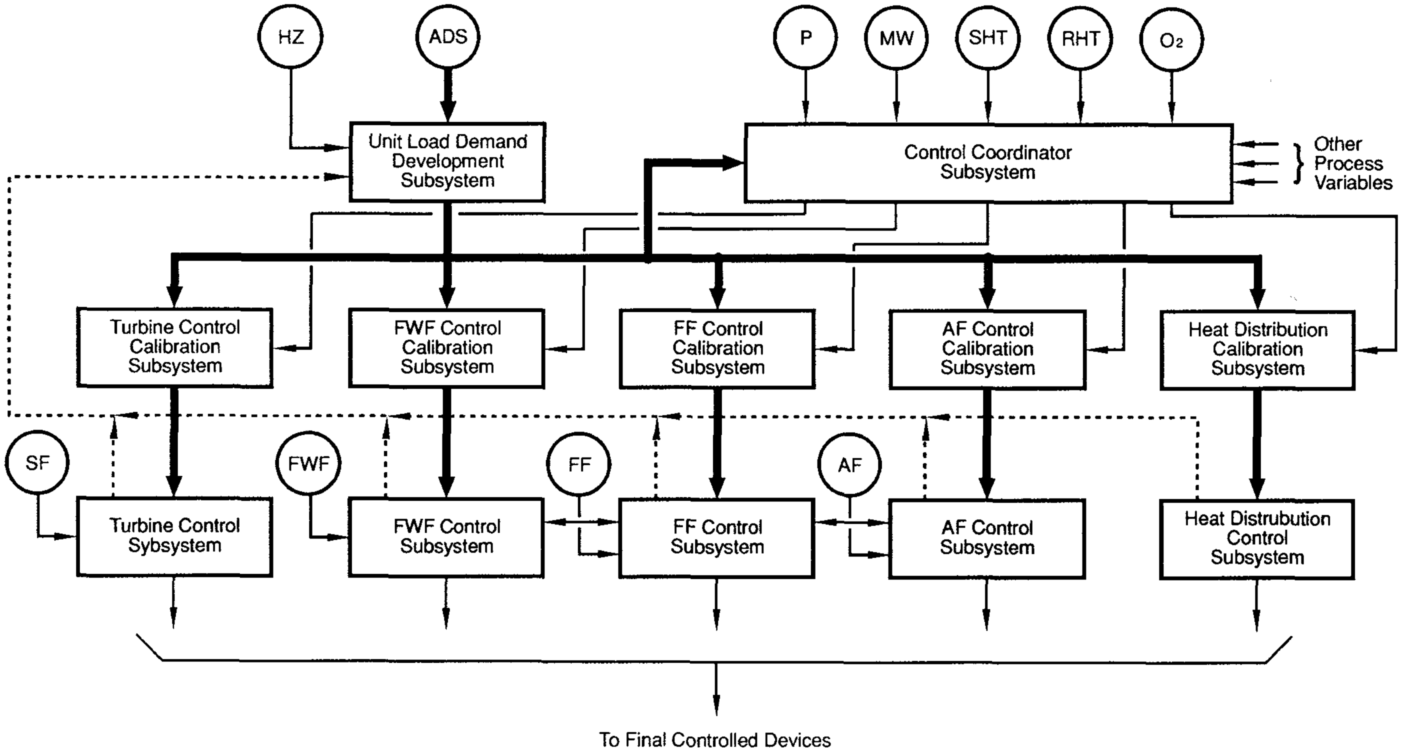


Figure 13-23 Diagram of Control Coordinator Control System

(From Bailey Controls Co. technical paper)

CONDITION				CONTROL ACTION			
PRESS	MW	SH TEMP	RH TEMP	STM. FLOW	FUEL	FW	GAS DIST (RH)
HI	HI	HI	HI	HOLD	DEC	HOLD	HOLD
LO	HI	HI	OK	DEC	HOLD	INC	INC
OK	HI	HI	LO	DEC	DEC	DEC	INC
LO	HI	OK	HI	DEC	HOLD	INC	DEC
HI	OK	HI	HI	INC	DEC	HOLD	HOLD
LO	OK	HI	OK	DEC	INC	INC	INC
OK	OK	OK	HI	HOLD	HOLD	INC	DEC
LO	LO	HI	HI	HOLD	INC	INC	HOLD
HI	LO	HI	LO	INC	HOLD	HOLD	INC
OK	LO	HI	OK	INC	INC	INC	INC

Figure 13-24 Control Coordinator Decision Table

(From Bailey Controls Co. technical paper)

based on the condition of steam pressure error, MW error, and reheat steam temperature error. The values of these other variables and the magnitude of their deviations must be brought into the decision table with the control coordinator output going to all control loops. In essence this is a form of model-based control.

Section 14
Boiler Draft Systems

The term ''draft'' has many different meanings, but in this context it is defined as a ''current of air.'' Associated with this meaning is its definition as an air or flue gas pressure that is slightly negative with respect to atmospheric pressure. When in a conduit connected to atmospheric pressure, such negative pressure would create a current of air or flue gas. The common definition of draft loss is the difference in draft or pressure resulting from the flow of the air or flue gas. These terms are used in connection with pressure and flow measurements of boiler flue gas and combustion air.

14-1 Draft Losses in Boilers

Boiler combustion air and flue gas flow through a system that includes the boiler furnace, ductwork, and various types of heat transfer surface. The driving force for this flow is an air or flue gas pressure or draft. The combustion air originates in the atmosphere; eventually its derivative, the flue gas, is exhausted to the atmosphere. The total draft or pressure is divided up by all those elements in the flow path that tend to resist or obstruct the flow. The amount of the pressure differential for each of these elements is called its draft loss.

The draft loss for these flow restrictors generally follows the same differential pressure vs. flow square root relationship as an orifice in a piping system. The relationship may deviate to some extent from a true square root, due to variations of specific volume over the flow range in addition to innate characteristics of the restriction. Figure 14-1 demonstrates the total and divided draft losses with respect to flow for a simple boiler without heat recovery equipment. Adding heat recovery equipment such as an air preheater requires additional draft loss on both the combustion air side and the flue gas side of the heat exchanger. This addition is shown in Figure 14-2.

Other factors may cause an apparent deviation in draft loss that is entirely unrelated to flow. When air and flue gases are heated, they become lighter and tend to rise. In boilers this is known as the stack effect. If the flow is moving up, this produces an apparent draft loss smaller than that accounted for by flow alone. If the flow is moving down, the effect is an increase in the measured draft loss with respect to that produced by the flow alone. The amount of this stack effect in each case is affected only by the temperature (and thus specific volume) of the flue gases. Figure 14-3 demonstrates this stack effect in boilers.

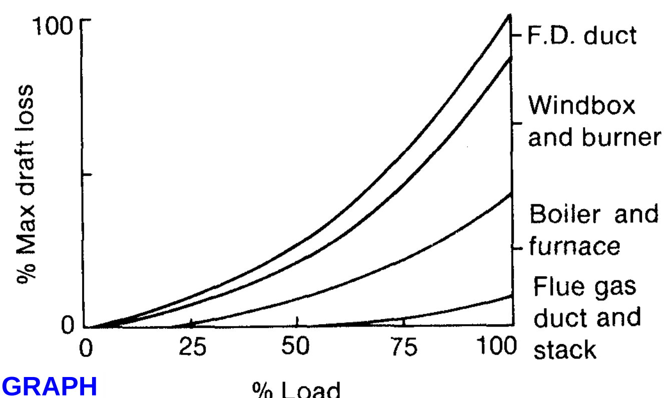


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Figure 14-1 Draft Losses for All Boilers (No Air Preheater)

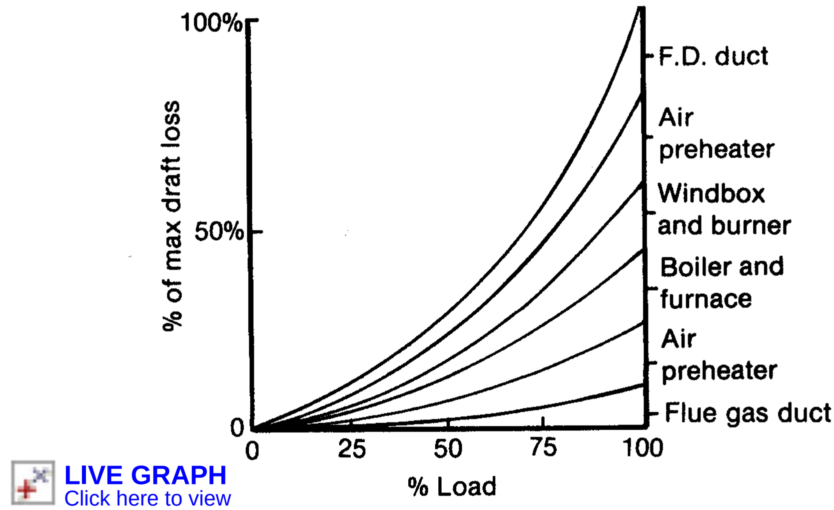


Figure 14-2 Draft Losses for All Boilers (Including Air Preheater)

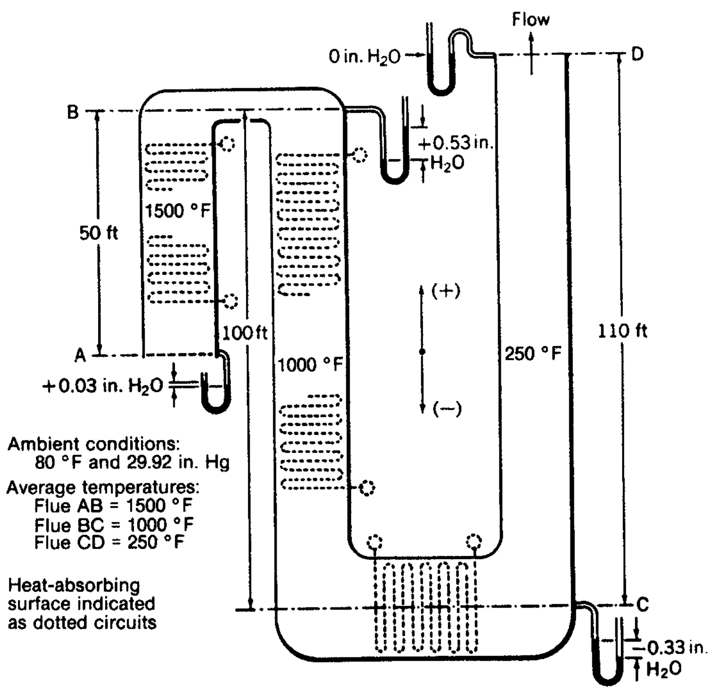
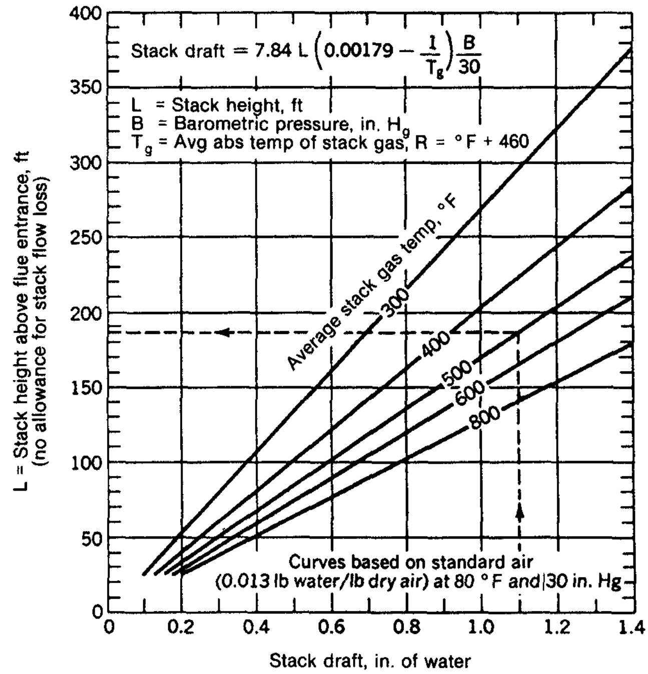


Figure 14-3 Stack Effect in Boilers

(From *Steam, Its Generation and Use,* © Babcock and Wilcox)

14-2 Natural Draft and Forced Draft

Natural draft is a term used when the air flow through the boiler is a result of the stack effect in a chimney or stack. The stack is connected to the flue gas passage of the boiler. If the flue gas specific volume (primarily due to its temperature) is less than that of the outside atmosphere, the flue gases at the top of the stack will rise, creating a suction that will induce combustion air flow through the boiler. The draft that will be produced by a stack is a function



Stack height required for a range of stack drafts and average stack gas temperatures.

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Figure 14-4 Stack Draft

(From *Steam, Its Generation and Use*, © Babcock and Wilcox)

of the height of the stack and the flue gas temperature. This relationship is shown in Figure 14-4.

Theoretically, the draft produced is independent of the stack diameter. In practical terms, however, the stack diameter must be sized so that the draft produced by the stack is not appreciably used by friction from the flow of the flue gases up the stack.

From the above it follows that some stack draft will always be present and can be used alone or in combination with a combustion air fan or fans. Generally, natural draft alone produces much less draft and is available for much lower draft losses than those available with mechanical draft. The result is poorer heat transfer and lowered boiler efficiency. On an economic basis, natural draft should be used as a supplement to, rather than as a substitute for, mechanical draft.

Mechanical draft is that produced by combustion air fans. In the case of boilers, a fan or air blower that takes suction from the atmosphere and forces combustion air through the system is called a forced draft fan. A fan at the end of the boiler flow system path that takes its suction from the boiler flue gas stream and discharges the flue gas to the stack is called an induced draft fan. The static pressure and flow characteristics of fans result from the specific design of the particular fan. The fan combinations available to the boiler system designer are (a) forced draft plus stack, (b) forced draft and induced draft plus stack, and (c) induced draft plus stack.

14-3 Pressure-Fired Boilers

A boiler sytem that contains no induced draft fan and whose furnace may operate under a positive pressure over some portion or all of the load range is called a pressure-fired boiler.

In this type of boiler, the pressure in the furnace varies as the load is changed. This is due to the variation in the various draft losses with respect to boiler load.

Figure 14-5 represents the physical arrangement of such a boiler system. A key point with such boilers is that the furnace must be airtight or flue gastight. This is necessary so that the very hot flue gas of the furnace cannot leak to the atmosphere. A small leak under such circumstances will deteriorate the material around it, eventually destroying the furnace walls and creating an operational hazard. Such furnaces are made pressure-tight with a welded inner casing or a welded seal between the furnace wall steam generating tubes.

A profile of the draft and pressure of a simple pressure-fired boiler without heat recovery equipment is shown in Figure 14-6. Note that at 70 percent boiler load the draft losses are approximately 50 percent of the full load draft losses. The draft or negative pressure at the right side of the profile results from the natural draft of the stack. If the stack draft were less, the entire profile would be raised. In this case the result of the natural draft is that the furnace is under pressure only at the higher loads.

A pressure and draft profile for a boiler with an air preheater type of heat recovery equipment is shown in Figure 14-7. The result of this is that the additional draft losses cause the furnace to be operated under higher pressure at all loads and to be under positive pressure except at the very low boiler loads. If an economizer had been used for heat recovery instead

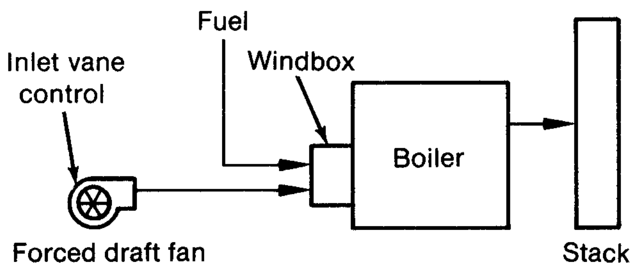


Figure 14-5 Pressure-Fired Boiler

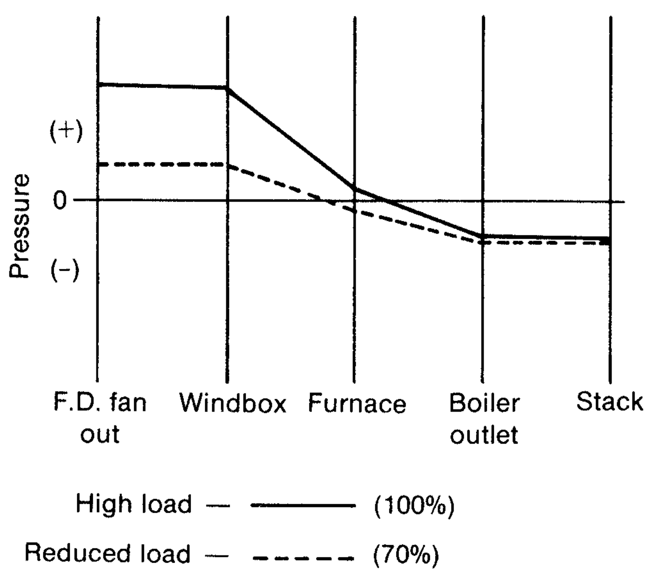


Figure 14-6 Profile of Pressure and Draft of a Pressure-Fired Boiler (Typical — No Air Preheater)

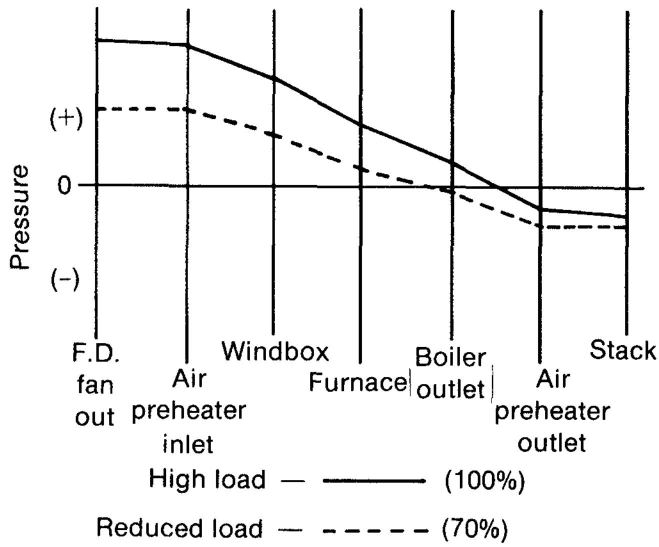


Figure 14-7 Profile of Pressure and Draft of a Pressure-Fired Boiler (Typical — Includes Air Preheater)

of an air preheater, then there would be no additional draft loss on the combustion air side flow path.

14-4 Balanced Draft Boilers

In many cases furnaces cannot be operated under pressure because of leakage around the fuel burning equipment. An example of this is stoker firing of solid fuel. In other cases systems have been designed for negative furnace pressure operation to reduce furnace maintenance, or they were designed and constructed before pressure furnace technology was developed. Such boiler systems usually rely on the use of an induced draft fan in combination with a forced draft fan. In these cases the induced draft fan is used to reduce the furnace pressure and assure that it is always negative with respect to atmospheric pressure.

Such systems are called balanced draft sytems and have an arrangement as shown in Figure 14-8. In the balanced draft system, operating the furnace under a negative pressure assures that any leakage will be relatively cool combustion air leaking into the furnace instead of very hot combustion gases leaking out. In normal practice the furnace pressure or draft is controlled to a very slightly negative pressure set point by regulating either or both the forced and the

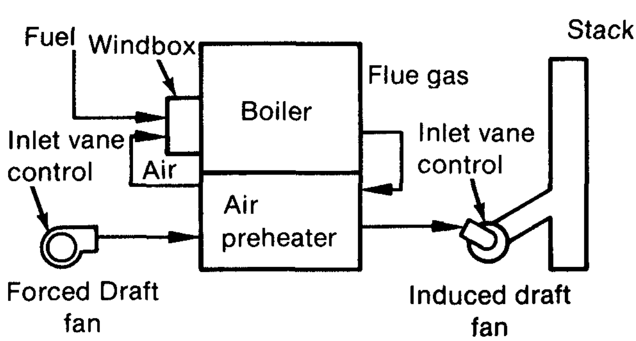


Figure 14-8 Balanced Draft Boiler (With Air Preheater)

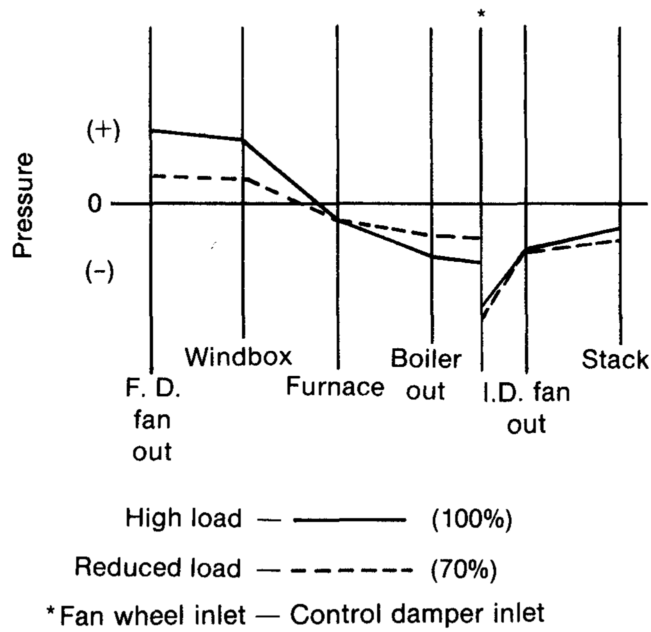


Figure 14-9 Profile of Pressure and Draft of Balanced Draft Boiler (Typical — No Air Preheater)

induced draft fans. In this way any atmospheric air leaking into the furnace or boiler is minimized.

In the balanced draft boiler the forced and the induced draft fans share the load of moving the combustion air and flue gases through the system. The balance point is the pressure or draft in the furnace. This pressure level is determined by the relative amounts of "push" and "pull" of the forced and induced drafts, respectively.

A pressure and draft profile of a balanced draft boiler is shown in Figure 14-9. Note that the furnace draft is slightly negative for all boiler loads. Adding a heat recovery combustion air preheater to the system adds additional draft losses to both the combustion air and the flue gas sides of the furnace but does not change the controlled furnace draft set point.

14-5 Dampers and Damper Control Devices

The most common device for controlling boiler drafts and air flows is some form of damper or vane in the flue gas and air stream. These take numerous forms, ranging from a single-bladed damper than can be rotated to provide a variable flow resistance to a complex multibladed control vane or louver. A simple representation of the damper as a control device is shown in Figure 14-10.

Generally, all dampers have a nonlinear flow vs. damper opening characteristic as shown. A multibladed damper tends to be more linear than a single-bladed damper. Dampers with adjacent blades rotating in a counter direction tend to be more linear that when all blades rotate in the same direction. In both cases the actual flow characteristic is more or less linear depending upon the ratio of the pressure drop across the damper to the total system pressure drop. The flow vs. opening characteristic is more linear with more of the system pressure drop occurring at the damper and less linear with the damper accounting for less of the total system pressure drop.

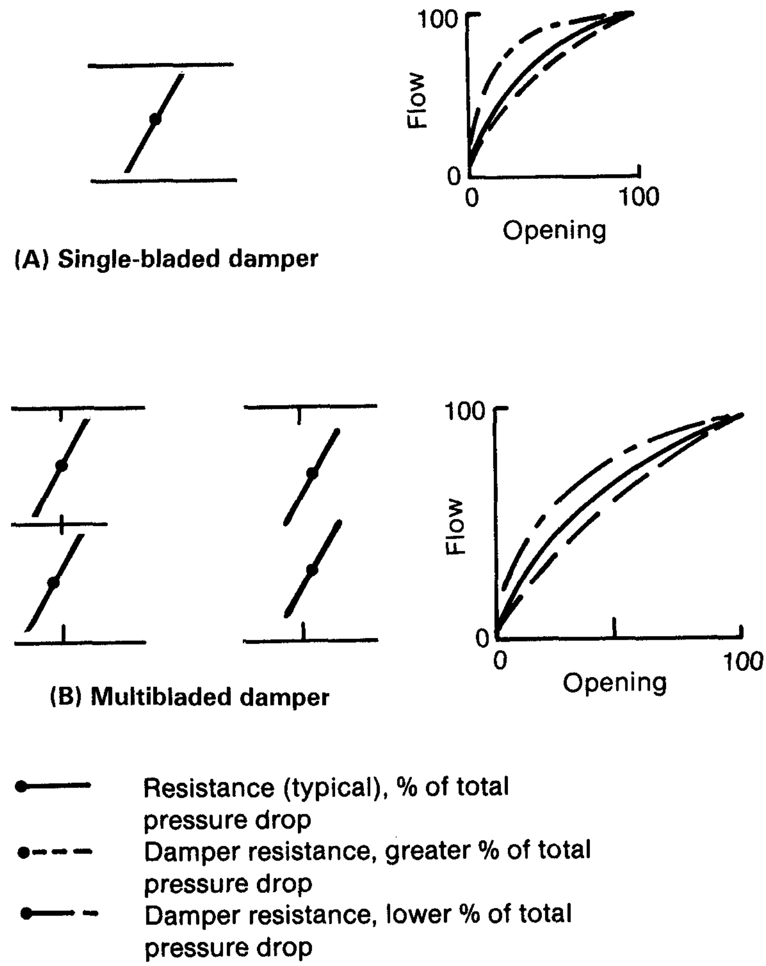
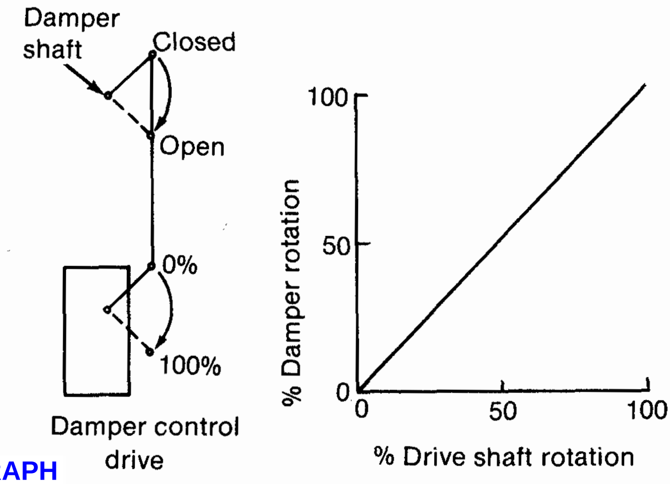


Figure 14-10 Damper Characteristic Flow (Typical)

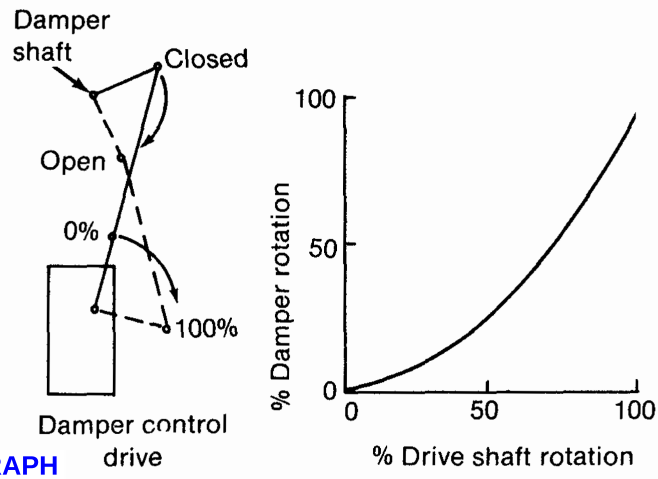
For air flow or draft control it is very desirable to have linear characteristics of flue gas or air flow vs. the control signal. A method of using a linear damper actuator while effectively altering the basic damper flow characteristic is called linkage angularity. This term applies to the modification of the linkage angles and lengths between the damper and its actuator.

The use of linkage angularity is demonstrated in Figure 14-11. In (A) the linkage driving and driven arms are of equal length and parallel. The angular rotation of the actuator shaft and the damper shaft will be equal and thus linear with respect to each other. By altering the lengths of the driven and driving arms and changing their relative angles as shown in (B), the actuator motion will be nonlinear with respect to the damper. The result is a nonlinearity, which, when combined with the opposite nonlinearity of the damper flow characteristic, tends to produce a linear actuator motion vs. flow.

If this mechanism is to be used, the starting angles of the linkage must be fully adjustable in small increments. For this reason, damper actuators are often furnished with a spline on the output shaft and drive arm. It should also be noted that linkage angularity applies a nonlinear torque to the damper shaft and may be used to amplify the torque over a portion of the range while reducing it over another portion of the range.



(A) Linear

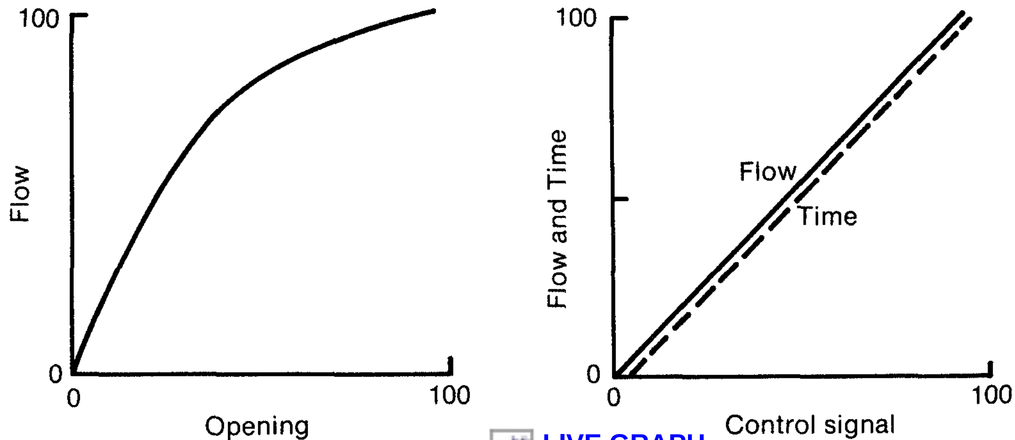


(B) Nonlinear

Figure 14-11 Damper Linkage Angularity

The flow vs. control signal can also be made linear by making the control signal versus actuator position nonlinear. In this way the basic nonlinear damper flow characteristic can be combined with an opposite nonlinear actuator characteristic to provide a linear flow vs. control signal characteristic. This linearization method uses an actuator positioner that incorporates a nonlinear cam. The shape of the cam determines the particular nonlinearity.

On a dynamic basis, the results of the two methods of linearization differ, though they may be the same on a steady-state basis. Figure 14-12 demonstrates that the angularity method produces a flow vs. control signal characteristic that is linear on both a steady-state and a dynamic basis. With a linear stroking time of the actuator, the time (0 to 100 percent) vs. flow (0 to 100 percent) characteristic is linear. In addition, the flow vs. control signal characteristic is also linear.



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(A) Initial

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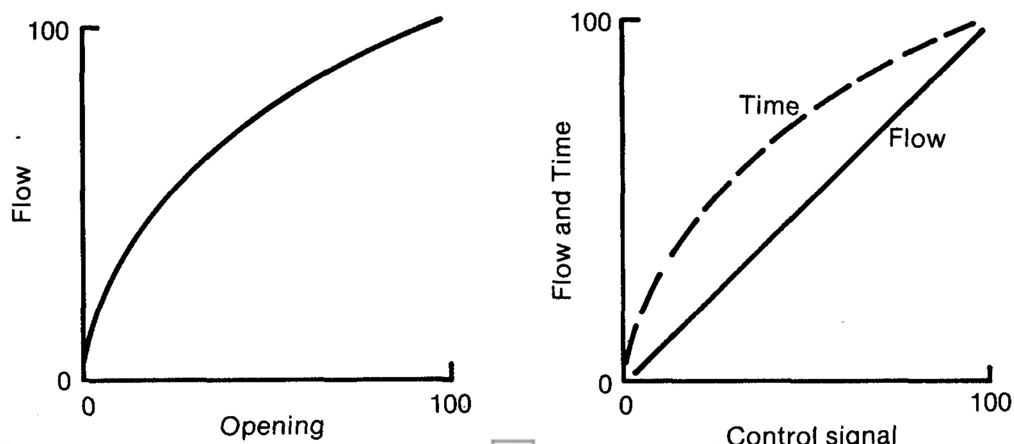
(B) Linkage angularity plus straight-line positioner

Figure 14-12 Desired Damper Characteristics

In Figure 14-13, a cam positioner and parallel linkage are used. The steady-state characteristic of flow vs. control signal is linear, but the time vs. flow characteristic is nonlinear due to the nonlinear position vs. flow characteristic of the actuator. This is important during very rapid changes in load when a proper match between fuel and air is necessary on a second-by-second basis. Time linearity also improves the precision of tuning the control loop over the complete range of load.

The actuators for dampers are commonly called control drives and can be either pneumatic or electric as desired. They need not be matched to the operational medium or signal type of the control system. In Figure 14-14, several different ways of controlling a pneumatic piston actuator are shown.



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(A) Initial

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(B) Straighten characteristic — with positioner only

Figure 14-13 Desired Damper Characteristics

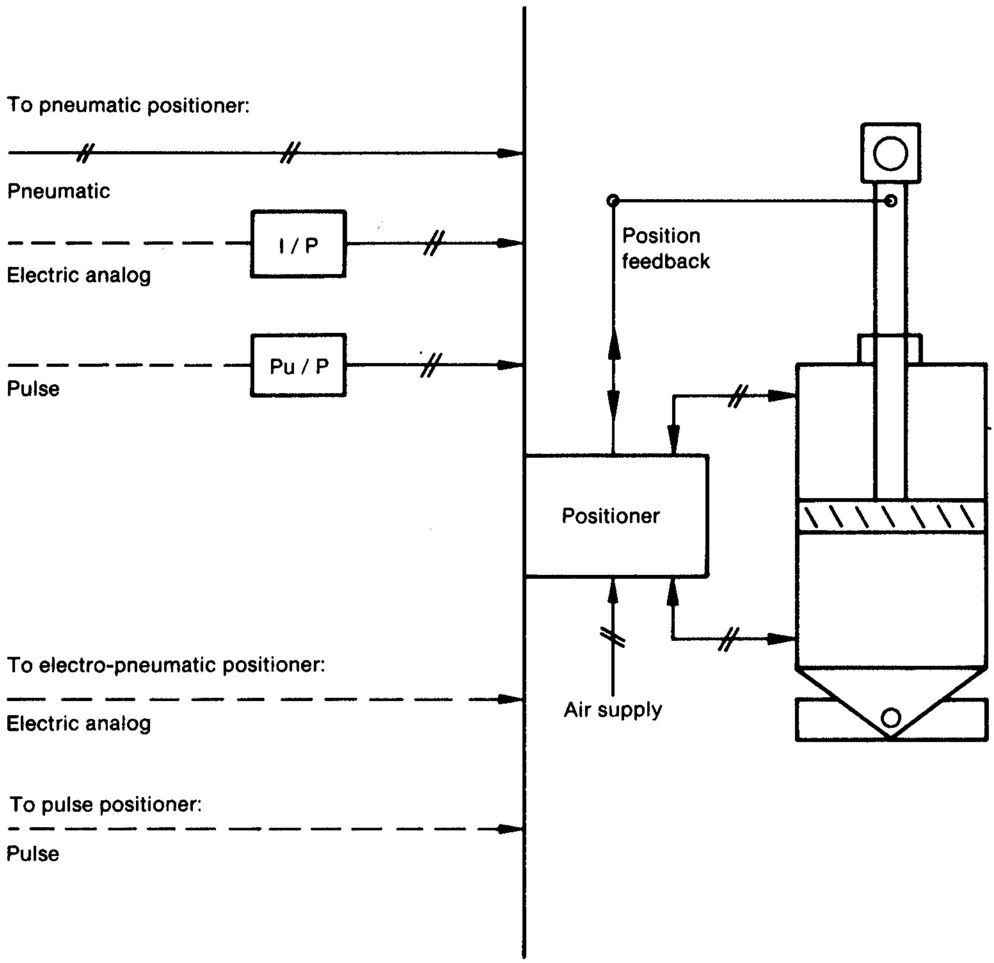


Figure 14-14 Pneumatic Piston with Optional Positioners

If the control system is pneumatic, the signal is connected directly to the positioner of the piston. If the control signal is an electric analog signal such as 4 to 20 mA, then a I/P (current-to-pneumatic) converter is used to convert the current signal to a proportional pneumatic signal. For an electric analog signal, the piston positioner can be changed to one that will receive the electric analog signal directly.

Similarly, electric digital pulse signals can be used to control a pneumatic control drive. One method is the use of a motor-operated pneumatic loader to convert the digital signal to a standard pneumatic signal, with the pneumatic signal connected directly to a standard pneumatic control drive. Another method is the use of a digital positioner that will convert the digital pulses to a corresponding piston motion.

The torque rating of a pneumatic piston operator is usually based on a piston pressure differential of approximately 20 psi less than the full air supply pressure. Additional torque can be developed if the full air supply pressure is applied to one side of the piston with 0 pressure on the other. This is likely, however, to result in erratic or jerky positioning. To use a lower differential reduces the power rating of the device.

Pneumatic piston operators are not inherently self-locking on air supply failure. By using a compressed air receiver of sufficient capacity, they can operate for a period of time after an electric power failure that stops the air compressors. If the design engineer judges it to be

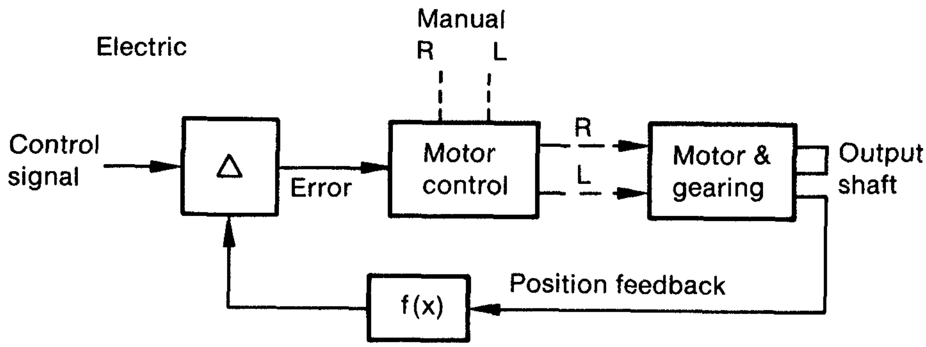


Figure 14-15 Electric Damper Operators

necessary, mechanical locks can be applied to the control drives to lock them in position upon a failure of the air supply. It is normally not sufficient to merely trap the air that is in the piston. Air failure locks are costly, and many engineers feel that the same amount of expenditure to improve the reliability of the supply air is more cost-effective.

Electric motor-operated damper control drives are usually inherently self-locking upon power failure. Many engineers feel that this action is a significant advantage for the use of electric rather than pneumatic operators. In these electric operators the rotary motion is converted to angular motion by various drive screw or worm gear arrangements.

A typical electrical arrangement is shown in Figure 14-15. A difference circuit measures the difference between the control signal and a characterized position feedback signal. Any error that results is amplified and controls the current to the motor. The direction of rotation of the motor is determined by the polarity of the position error. To operate the motor manually, a switch disconnects the motor controller, and a set of raise and lower push buttons is used to increase or decrease the controlled variable.

In the above case the motor follows an analog control signal. A digital controller can also be used to pulse drive the motor directly in either a clockwise or a counterclockwise direction. One of the inputs to the digital controller is a position signal from the motor-operated control drive. The controller uses this signal along with a computed position demand to direct the pulse-driven motor. Characterization can be obtained using a curve-fitting formula in the controller or by the feedback of a precharacterized position signal.

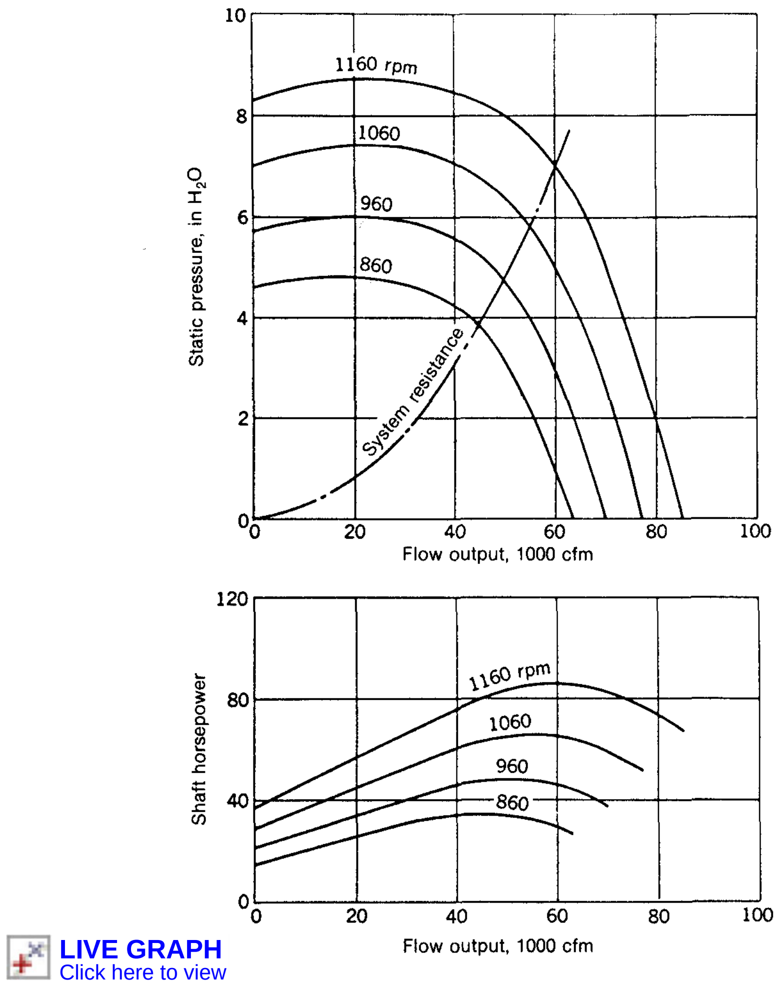
The maximum torque or power rating of an electric damper operator is typically based on approximately 25 percent over the normal full load current of the motor. If the control drive is loaded to exceed this value, a torque limit switch prevents the current from increasing. The 25 percent overload is usually required only for breakaway or to start the controlled device moving. It should not be designed for use in normal operation, since to do so would continually overload the motor and overheating would result.

14-6 Draft or Air Flow Control Using a Variable-Speed Fan

If the only concern were the control of the draft or air flow, the fans would be operated at constant speed. Another concern, however, is the power requirements of the fans.

A typical set of fan characteristics is shown in Figure 14-16. Referring to this figure, if it is assumed that the boiler is 100 percent loaded and the flow requirement is 60,000 cfm, the power requirement is approximately 87 shaft horsepower. If the load is reduced to a flow requirement of 50,000 cfm and fan speed remains at the 1160 rpm level, the power requirement is approximately 85 shaft horsepower. For this approximately 17 percent load reduction, the power requirement has been reduced only approximately 2.3 percent.

If the fan speed is reduced to 960 rpm simultaneously with the load reduction, the power



Graph to show how desired output and static pressure can be obtained economically by varying fan speed to avoid large throttling losses.

Figure 14-16 Variable-Speed Fan Characteristic Curves

requirement is reduced to approximately 48 shaft horsepower. For this 17 percent reduction in flow, the power reduction is approximately 55 percent. Note that both sets of values are on the system resistance curve. In addition, the speed reduction is also approximately 17 percent. In all cases the percentage reduction in speed approximates the percentage reduction in flow capacity.

The following are the laws of fan performance related to speed:

(1) Capacity or cpm varies directly as the fan speed.
(2) Pressure varies as the square of fan speed.
(3) Power varies as the cube of fan speed.

If the fan were sized exactly for full load at full speed, then a normal 4:1 flow control turndown would require a 4:1 speed turndown for the fan. In a normal installation it is not uncommon to find fans designed with considerable excess capacity. In such a case, the full load fan speed may be 50 to 70 percent of the design fan speed. The result is a fan speed turndown requirement of 6 to 8:1 for a capacity turndown of 4:1. The above assumes that the fan speed alone is used to control the capacity.

Fans can be operated under variable speed control to save fan power in several ways. If the fan is steam turbine driven, the speed can be modified by adjusting the speed setting of the speed governor or by simply using a control valve to control the steam flow to a turbine. If the fan is motor-operated, two-speed or variable-speed motors may be used. Fan speed may also be varied, even though a constant speed motor is used, by using a variable-speed magnetic or hydraulic coupling between the motor and the fan. Variable-speed drive motors are also available, and their speed regulation can be used to control flow.

In whatever manner the fan speed is adjusted, the speed response often deteriorates rapidly at speeds below approximately 1/3 rated speed. This affects the air flow control dynamics with respect to fuel flow but may not affect speed positioning accuracy. Since a fan in a typical installation is often designed with excess capacity, the capacity turndown at 1/3 rated speed is approximately 2:1, much less than that required. The net result is that most installations require some damper control participation at lower loads to extend the rangeability of good control response.

Figure 14-17 depicts a satisfactory non-interacting method of combining the damper and

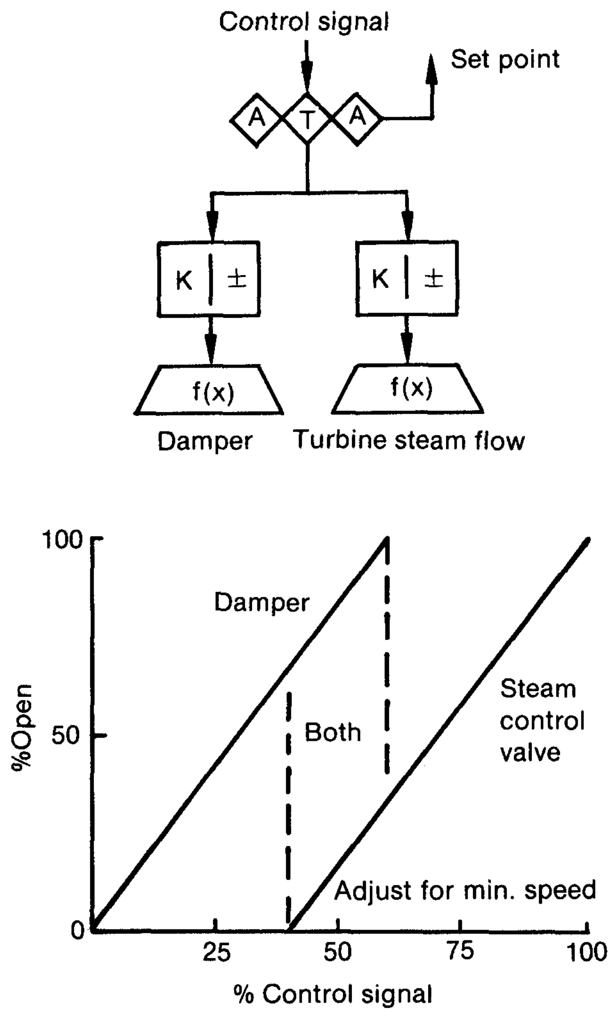


Figure 14-17 Combined Damper and Fan Speed Control

speed control. This is called split range control. The positioners of the actuators are adjusted to split the control signal range with overlap in the center of the range.

The damper control drive is adjusted to open the damper completely as the control signal changes from 0 percent to approximately 60 percent. The speed control device of the fan is adjusted to accelerate the fan speed from 1/3 rated speed to the desired full load speed as the control signal changes from approximately 40 percent to 100 percent. In the overlap area between 40 and 60 percent, the damper is becoming less responsive as it nears the 100 percent open position, and the fan speed is less responsive due to its being near the 1/3 rated speed. Combining the two in this portion of the range results in a somewhat uniform flow response over the entire range.

If the speed control mechanism is a steam turbine, experience has provided some lessons in the application. Some of the methods are listed below in the order of their suitability.

1. A separate conventional steam control valve ahead of the speed governor is non-interacting with the speed governor and is a good control solution. Particular care is necessary in sizing the valve, since the turbine power (analogous to turbine steam flow) may have a turndown of 10 or 15:1 as the control valve pressure drop increases over a range of 10:1. In some cases the clearance area around the valve plug may be too great for the desired minimum flow condition.

2. Using a device to change the speed set point on a high quality hydraulic or electronic speed governor is usually a satisfactory solution for a wide range control. It may be necessary to detune the flow control loop to avoid interaction between the flow and the speed control loops.

3. A control valve in the oil circuit of a hydraulic governor has the same potential interaction problems of method 2 above and, in addition, has a considerably narrower speed control range.

4. The simple application of a diaphragm operator to a mechanical speed governor is usually not satisfactory. The motion vs. steam flow characteristic and the valve stroke of the speed control valve results in poor control if the valve is used as a flow control valve.

The basic rules for fan speed control with the end purpose of reducing the auxiliary power requirements of the fan drives areas follows:

(1) Operate the fans at the lowest speed consistent with good control response.

(2) Operate control dampers as near open as possible consistent with good control response.

(3) Apply the damper control to the fan inlet rather than the discharge.

Figure 14-18 shows the relationship between power requirements for the various methods of motor-driven fan output control. The curve for a turbine drive fan or a variable frequency motor-driven fan would be similar to that shown for a magnetic or a hydraulic coupling. In all cases, as shown by Figure 14-18, inlet vane control consumes less power than discharge damper control.

14-7 Minimum Air Flow

Current National Fire Protection Association (NFPA) regulations tend to limit the air flow control range requirement to 4:1. This results from the stated 25 percent of full load air flow as a minimum flow for many types of boilers. Some general statements can be made concerning this minimum:

(1) It does not apply to stoker-fired boilers or to any boilers that retain a significant fuel storage within the furnace.

(2) For a single-burner gas- or oil-fired boiler with a constant speed combustion air fan, the minimum limit can often be applied as a mechanical stop on the control damper.

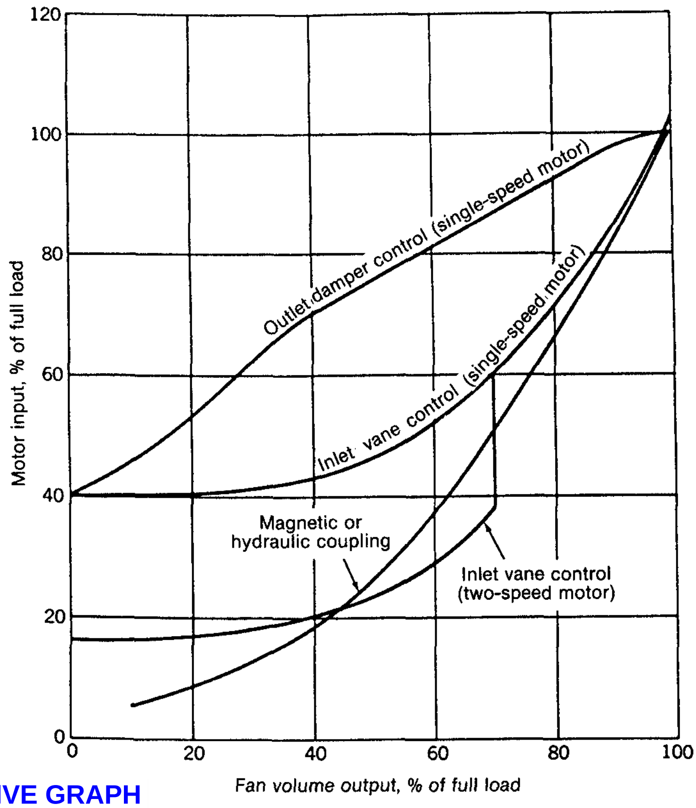


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Figure 14-18 Comparison of Methods for Saving Fan Power

(From *Steam, Its Generation and Use*, © Babcock and Wilcox)

(3) For a multiburner boiler, the minimum flow rate control should be based on flow measurement. This is necessary since changing the number of burners changes the resistance of the flow path.

(4) If the minimum air flow is 25 percent of full load air flow with 10 percent excess air (110 percent total air), reducing the fuel flow below 25 percent will result in a large increase in excess air. For example, reducing fuel to 15 percent of full load to satisfy load demand would result in 83 percent excess air $-$ (110 * (25/15)) = 183 percent total air.

(5) When it is known that the boiler steam flow range will be greater than 4:1, the control designer should take care to include necessary anti-windup and tracking features in the control system design.

Section 15
Measurement and Control of Furnace Draft

The need for the control of furnace draft occurs only in balanced draft boilers. In Section 14 it is stated that the normal practice is to control furnace draft at a very slightly negative pressure set point. When such a pressure measurement is made and used for control purposes, the stack effects must be carefully considered.

15-1 Measurement of Furnace Draft

Figure 15-1 demonstrates the choice of pressure tap locations for measuring furnace draft. The pressure connection on most boilers is located on the front, side, or roof of the furnace. Although the measurements at these three locations would be for the same furnace chamber of a particular boiler, the measurement values would differ due to the differing stack or chimney effects. The measurements at different elevations will differ by approximately 0.01 inch H_2O per foot elevation. The measurement in the roof of the furnace will be the highest value.

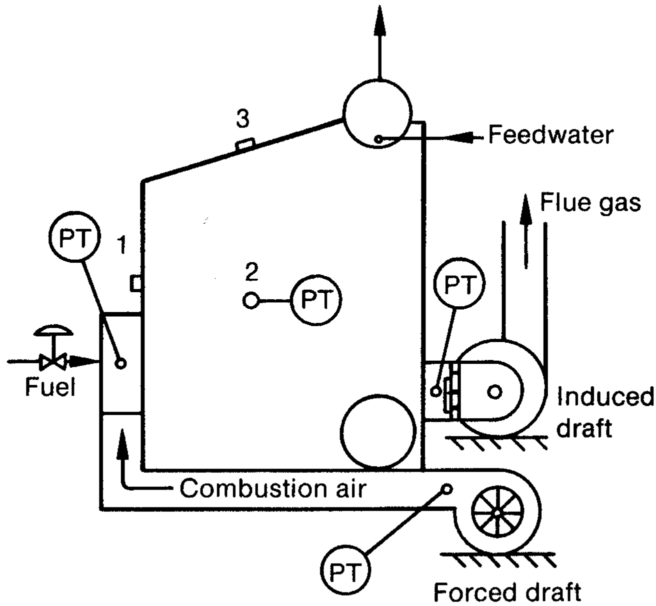
Since it is necessary to have negative pressure at all points, the value at the furnace roof becomes the controlling factor in determining the desired set point for the control of furnace draft. Thus, if the pressure at the furnace roof is to be minus 0.1 inch of H_2O and the connection for measuring furnace draft is located at an elevation 15 feet below the furnace roof, then the set point for this control loop should be approximately -0.25 inch of H_2O. On a large boiler the connection might be as much as 50 feet or more below the roof elevation. In this case, the set point should be approximately -0.6 inch of H_2O or at a lower pressure.

Because of the very low pressure involved, the pressure connection should be large enough so that slight changes in the furnace draft can be very quickly felt by the measuring instrument. General practice is shown by Figure 15-2. The actual connection is a 2-inch pipe size, and the piping to the instrument is often 3/4 to 1 inch in size. The 2-inch connection is provided with a tee and a plug in order that the plug can be removed and the connection easily cleaned. The piping size shown is typical for the older furnace draft transmitters that have significant displacement. Modern transmitters used for furnace draft measurement have very low displacement. Recent response tests with these transmitters have proven that good response can be obtained with tubing as small as 3/8 inch.

In some cases involving balanced draft coal or solid fuel boilers, it is appropriate to drill a small hole (approximately 1/8 inch) in the plug. This allows a small amount of air to be drawn into the furnace at all times to help prevent soot or ash from plugging the connection. This procedure should never be used with pressure-fired boilers. For these boilers it is necessary that the instrument connection systems be free of all leaks in order to avoid the introduction of H_2O vapor, soot, or ash into the connecting piping.

For most boilers a furnace draft or pressure transmitter will operate normally within a pressure range of less than 1 inch of H_2O. For presenting the information to an operator, a normal instrument pressure range of $+0.1$ to -1.0 inch of H_2O is typically used. Such a narrow range is not normally satisfactory for control purposes. On fast changes of flow capacity or under abnormal operating conditions, the actual pressure or draft may exceed this range and thus not provide the controller with all the intelligence necessary during the period of change.

Furnace draft measurement is also subject to considerable process "noise." The use of a narrow range transmitter tends to accentuate the effect of such noise in the measurement. An additional factor is primarily a limitation of analog control. In this case it is quite often impossible to reduce the controller gain to a low enough value. Therefore, the general practice is therefore to use a control transmitter range of approximately $+1.0$ to -5.0 inches of H_2O.



Notes:
1, 2, 3 — alternate furnace draft pressure connections.
Reading changes approx. 0.01 in. of H_2O per ft elevation.

Figure 15-1 Measurement of Furnace Draft

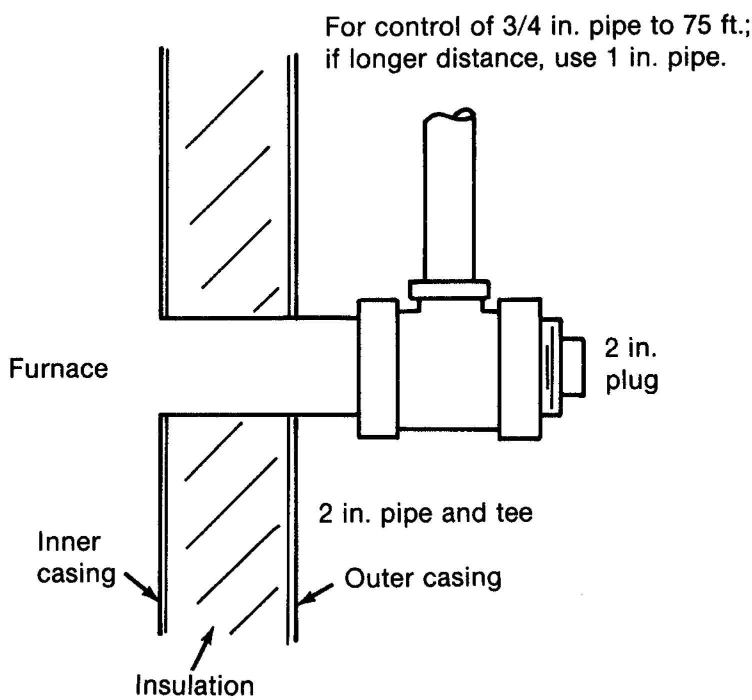


Figure 15-2 Measuring Tap for Furnace Draft

15-2 Furnace Draft Control Using Simple Feedback Control

The simplest form of the furnace draft control loop uses a simple feedback control loop. In this case the control of air flow is usually assigned to the forced draft with the furnace draft control regulating the level of induced draft. Generally, it is most desirable to measure air flow on the forced draft side of the furnace. Assigning the air flow control to forced draft tends to reduce interaction between the air flow and the furnace draft control loops.

The control arrangement is shown in Figure 15-3. The air flow capacity is changed by modulating the forced draft. As shown here, the resulting change in furnace draft feeds back to the controller, causing a series change to the induced draft. It is also possible to assign the air flow change to the induced draft with the series action taking place on the forced draft. In that case the controller action would be reversed.

On many installations a control loop of this type is very difficult to tune for satisfactory results under dynamic load changing situations. The series action of the control allows too much time difference between the changes to the forced and induced drafts. Theoretically, these should be moving in parallel.

In addition, the large amount of process noise as a percentage of the measurement signal tends to require tuning adjustments of lower than desirable gain and slower than desired integral. In some cases, if a standard feedback control alone is used, it may be necessary to remove all proportional action and rely on integral control alone. This tends to accentuate the problem of the series time delay.

One solution to this problem is to use a differential gap controller or a nonlinear controller such as an error squared controller. In the differential gap controller, no control action takes place as long as the furnace draft is within an adjustable band around set point. The gap is adjusted so that the normal process noise does not cause control action. Only outside this band does control action occur. In the nonlinear error-squared control, all proportional and integral

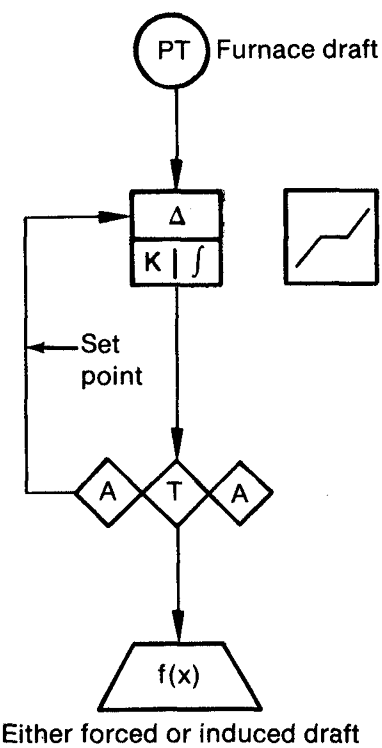


Figure 15-3 Furnace Draft Control (Single-Element Feedback Control)

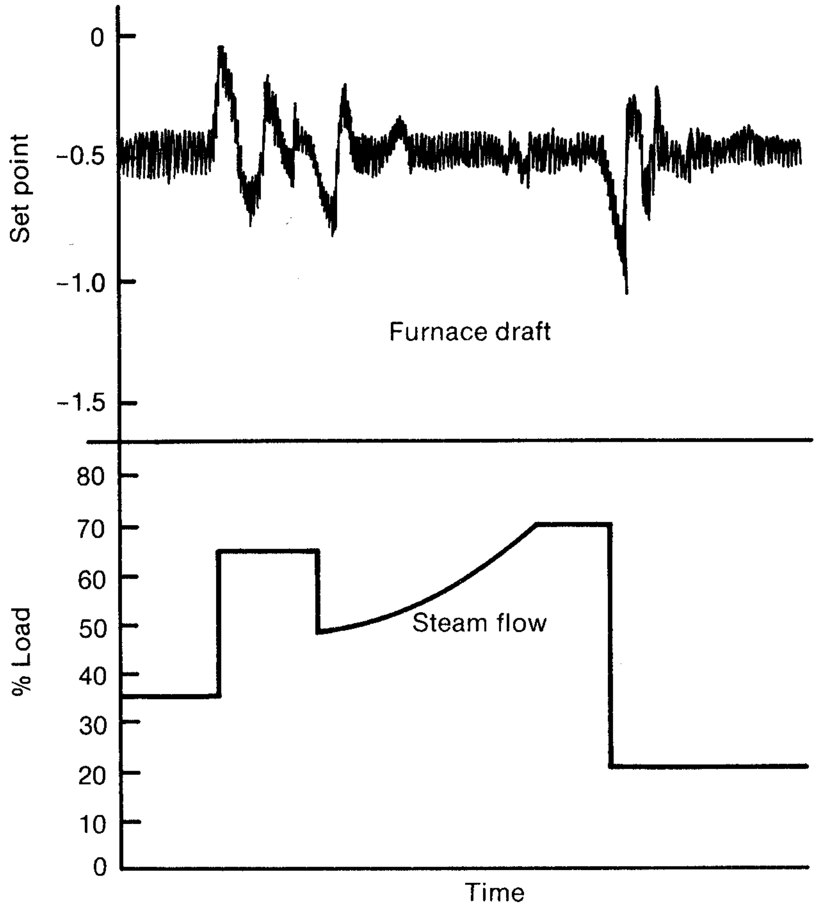


Figure 15-4 Furnace Draft Control (Single-Element Feedback Control of Induced Draft Fan)

control actions are based on the square of the error from set point. The controller is thus quite insensitive to process noise and also to required control action when close to set point. This also tends to accentuate the problem of the series time delay.

For comparison purposes, Figure 15-4 demonstrates the performance of a typical feedback furnace draft control loop. The excursions tend to be large with respect to the set point value, and the control tends to be unstable due to the effects of the process noise.

Such a control loop may be the single most difficult boiler control loop. Assuming a measurement at the furnace roof, the goal is to hold the furnace draft to a set point of -0.1 inch of H_2O with an excursion range of plus or minus 0.05 inch of H_2O, while the process noise is usually a minimum of ± 0.1 inch of H_2O and a typical overall capability of the fans at 6 to 10 inches of H_2O. For large electric utility boilers the fan capability may be 25 inches of H_2O or more, but the control performance described is still required.

15-3 Furnace Draft Control Using Feedforward-plus-Feedback Control

Figure 15-5 demonstrates an improved control loop for the control of furnace draft. In this case the signal to the forced draft control device is added in the summer (a) to the output of the furnace draft feedback controller. In this way the series time lag between forced and induced control action is eliminated. Note that it is necessary to provide a bias function in the

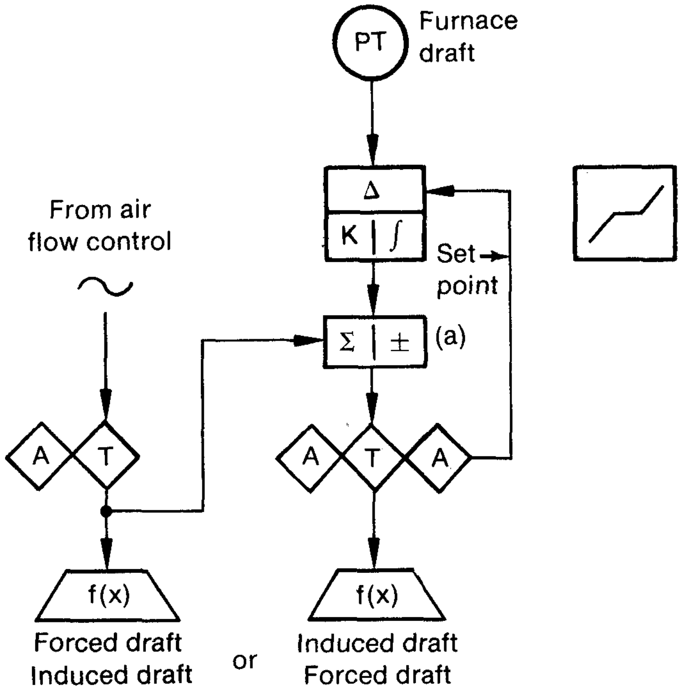


Figure 15-5 Furnace Draft Control (Feedforward-plus-Feedback Control)

summer (a). This is necessary so that the output of the furnace draft controller will operate normally in the middle portion of its output range. This allows the controller to equally add or subtract from the feedforward signal as necessary. A proper control alignment for the summer (a) would show it having gains of 1.0 on both inputs and with bias of −50 percent.

In applying this or other feedforward control it is necessary to parallel the flow characteristics of the two parallel control devices (in this case forced and induced draft). If this is not done, the two will not provide the proper parallel effect, and much of the benefit of the feedforward control may be lost. It is also necessary to select the proper feedforward signal. Measured air flow should not be used as the feedforward signal. A positive feedback effect and a series time lag is introduced into the loop due to interaction between the air flow and the furnace draft measurement.

Figure 15-6 shows performance of the feedforward system on a comparative basis with the feedback arrangement. In this case the capacity changes can be made with much smaller deviations from the furnace draft control set point. Because of the feedforward action, the furnace draft controller can be considerably slower in action without reducing the effectiveness of the control loop. This adds control stability by reducing the gain and integral requirements and thus reducing the effect of process noise. Since the forced and induced drafts operate in parallel, any potential interaction between the forced and the induced draft control is significantly reduced.

15-4 Furnace Draft Control Using Push-Pull Feedforward-plus-Feedback Control

In the diagram shown in Figure 15-7, the feedback portion of the control loop is improved by applying it in a push-pull manner. The feedforward portion of the loop is identical to the feedforward portion of the system described in Figure 15-5. The control signal from the air

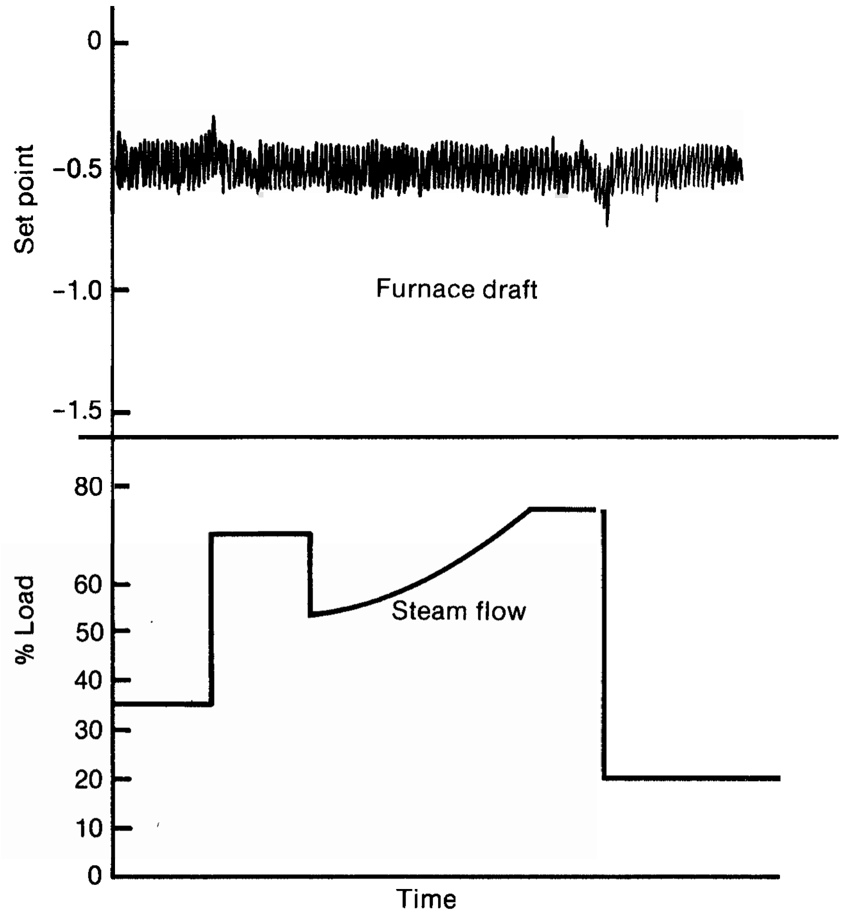


Figure 15-6 Furnace Draft Control (Feedforward-plus-Feedback Control of an Induced Draft Fan)

flow controller is used as an input to the summer (a). The other input to this summer is the output of the furnace draft feedback controller. Properly aligned, both of these inputs would have a gain of 1.0. As before, a −50 percent bias is applied to the output of the summer (a).

An additional function, difference (b), uses the same two inputs as the summer (a). When properly aligned, both inputs to the difference function (b) have a gain of 1.0, and a bias of +50 percent is applied to its output. This arrangement provides improved dynamic performance by allowing the feedback controller to add to the induced fan control signal while simultaneously subtracting from the forced draft control signal. The description above covers the manipulation of the signals only. The feedback controller is direct acting with a more negative furnace pressure input producing a more negative output control signal.

The system can thus adjust on a dynamic basis for any control result that differs from that calibrated into the basic feedforward system. For example, it is more "forgiving" in regard to the paralleling requirements of the calibration of the forced and induced control devices. In the basic feedforward system, the control signal vs. flow characteristics are used to match the forced and induced drafts. If flow resistances change, the matching deteriorates and affects the feedforward performance. This arrangement tends to automatically compensate for these changes in flow resistance.

Improved performance of the feedforward portion of the system reduces further the control demand on the feedback portion of the control loop. The result is improved control stability

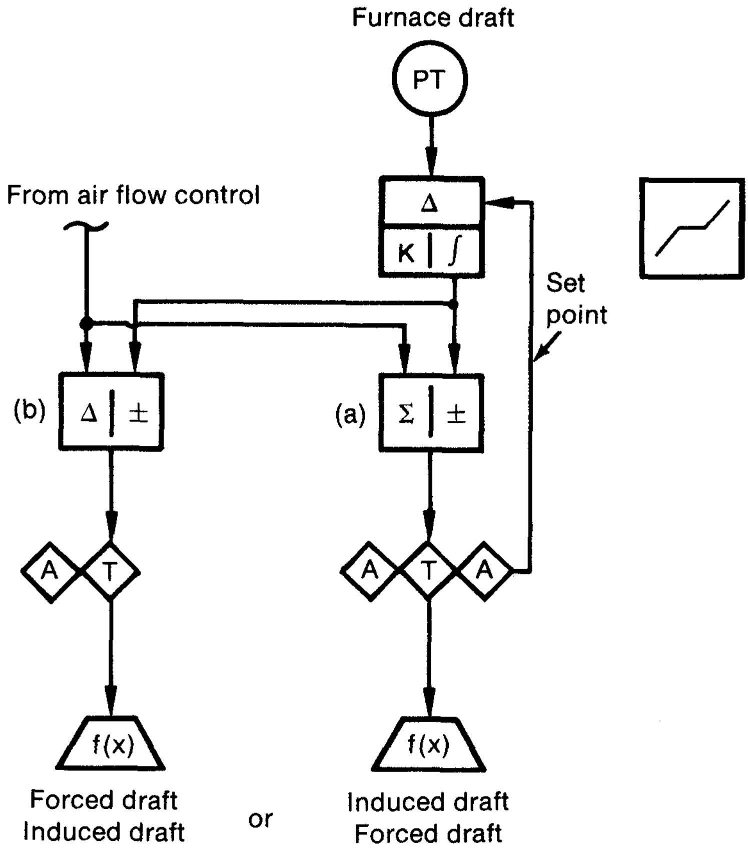


Figure 15-7 Furnace Draft Control (Push-Pull Feedforward-plus-Feedback Control)

through further reduction in the gain and integral requirements of the feedback controller and, thus, lower effects from the process noise.

While the feedforward-plus-feedback arrangements discussed apply equally to industrial and the largest electrical utility boilers, additional controls for implosion protection should be applied to large electric utility boilers.

15-5 Protection against Implosion

A hazard to boiler operation that was rarely experienced before 1970 is furnace implosion. Under certain conditions, the negative pressure on the walls of the furnace, boiler, and duct-work can create forces great enough that these walls cave inward, doing considerable damage and extensive unit outage. This hazard is one usually experienced only on electric utility boilers.

Implosion should not be confused with explosion. The majority of explosions occur while lighting off the boiler. Implosion can occur at any time during normal boiler operation.

As electric utility units have become larger, the wall expanses are greater in area, resulting in the potential force being greater with the same negative pressure. In addition to the potential for greater negative pressure, many of the boilers have been designed in the past without consideration for such large negative forces.

The original design of many pressure-fired boilers did not take into account the possibility of high negative pressures. Many of these boilers have been converted to balanced draft, with induced draft fans having the potential for high negative suction pressures. Some of the conversions have been the result of the addition of pollution control on the discharge of the induced draft fan, with fan head capacity adjusted upward because of this.

The head/flow characteristics of the fans, as shown in Figure 14-16, show that as the flow is reduced to a low value the maximum head design of the fan is approached. If the flow is reduced in an orderly manner with a balanced draft in the furnace, the induced draft inlet louvres and/or fan speed is reduced simultaneously and no problem is encountered.

Suppose there is a sudden drop in the forced draft resulting in a high negative furnace pressure. With the much greater induced than forced draft capacity, the result is a large negative pressure increase at the suction of the induced draft fan. This can occur upon any sudden or rapid loss of forced draft at a faster rate than can be accommodated by an induced draft control reduction. The result may be a large increase in the negative pressure at the induced draft suction because the static pressure of the fan characteristic must adjust for that particular flow level. Such an increase may cause an implosion if the new negative pressure exceeds the design limits of the boiler and ductwork.

A more likely cause is a main fuel trip and loss of flame. Very quickly the furnace heat is sucked up by the waterwall tubes and the furnace gases approach the temperature of the tubes. This sudden reduction of furnace flue gas temperature from 2000^{+} °F to several hundred degrees F causes a collapse of the furnace gas volume, and the other boiler gas volumes, and they immediately shrink to less than half their previous volume. Temporarily there may be a greater negative pressure in the furnace than at the fan, which tends to reverse or halt the flow through the induced draft fan. Due to its flow characteristic, the fan tries to establish flow by raising the negative pressure at the fan suction. If this negative pressure exceeds the negative pressure structural strength of the unit, implosion occurs.

The action described may occur in seconds or fractions of seconds, much faster than the normal on-line furnace draft controls can accommodate. Because of this, a separate high-speed subsystem for implosion protection is required as a part of the furnace draft control system.

Due to the critical nature of such an arrangement, it is installed downstream in the control system of the auto/manual transfer station so that it cannot be overridden by the operator.

Typically, the following should occur should there be a main fuel trip.

(1) The signal from a main fuel trip and wide range furnace draft control transmitter causes a high speed operator on the induced draft control to run back the inlet vanes of the induced draft fan. This runback signal will gradually be dissipated as the furnace draft moves back toward set point.

(2) The main fuel trip signal blocks the action of the forced draft vanes and holds them in a fixed position.

(3) When the furnace draft has returned to near the set point level, the normal furnace draft controls are in command and the forced draft is unblocked and free to move.

The regulations of the code ANSI/NFPA 85G cover this subject and should be adhered to in designing the system. The block diagram in NFPA 85G is shown in Figure 15-8. This block diagram in NFPA 85G and the discussion therein states that for reliability purposes there are to be three furnace draft transmitters, each connected to a different furnace tap. The signal to the control is the result of 2-out-of-3 (2-o-o-3) of these transmitters. The specific control logic may have many variations and differs with the type of boiler, type of fans, arrangement and number of fans, methods and devices for flow control, type of fan drives, etc.

Note that the 2-out-of-3 voting called for in NFPA 85G is different from the selection of the median value of three transmitters. While median select appears to have been used in almost all cases in the past, it is one-out-of-three and therefore a less stringent test than two-out-of-three voting. One version of the logic for the more secure 2 out of 3 voting is shown in Figure 15-9. Median select of three transmitters is shown in Figure 13-2 for selecting the median of three drum level measurements.

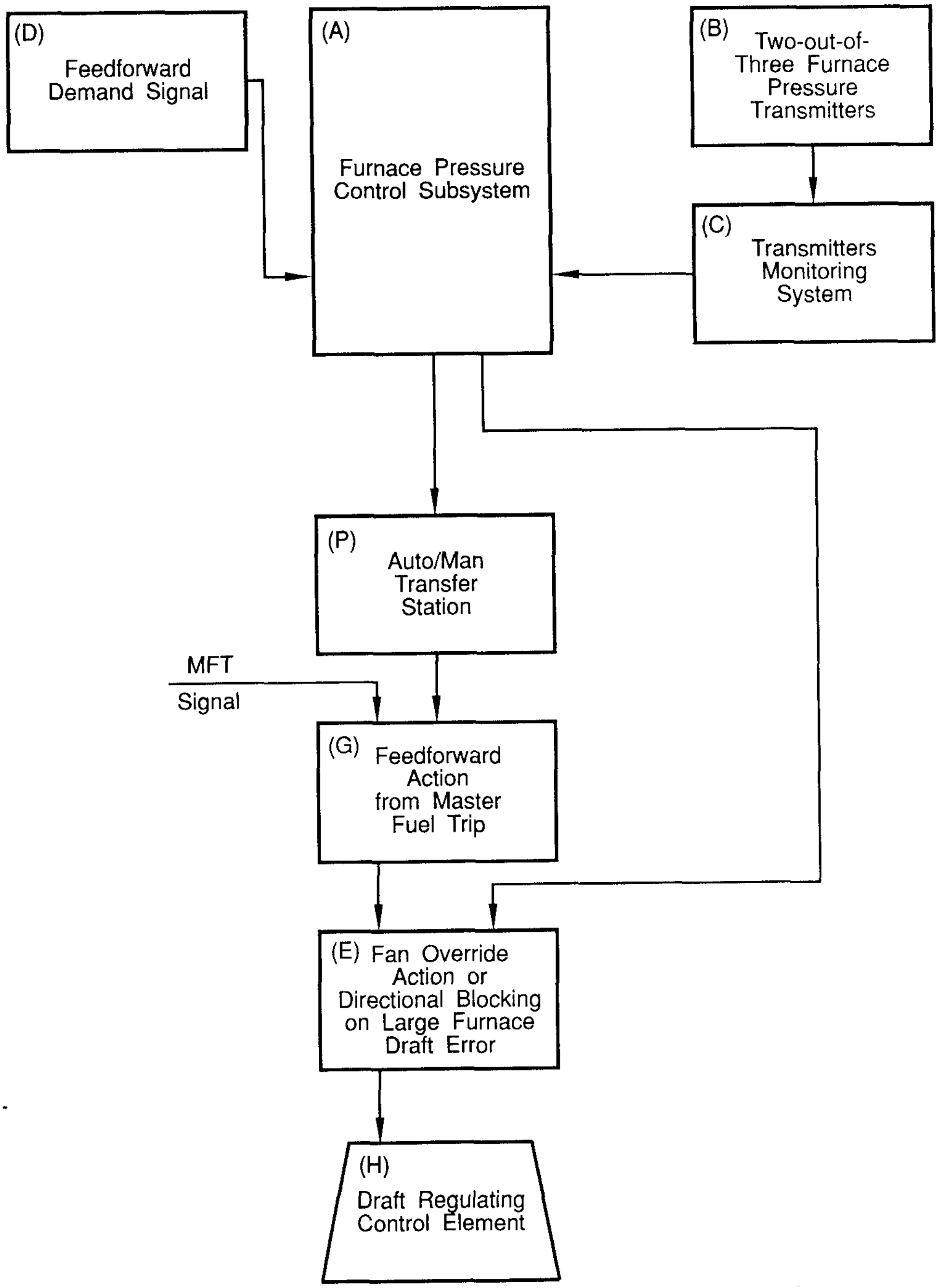


Figure 15-8 Furnace Draft and Implosion Protection Control Logic

(From ANSI/NFPA 85G, Prevention of Furnace Implosions in Multiple Burner Boiler Furnaces — 1987 Edition)

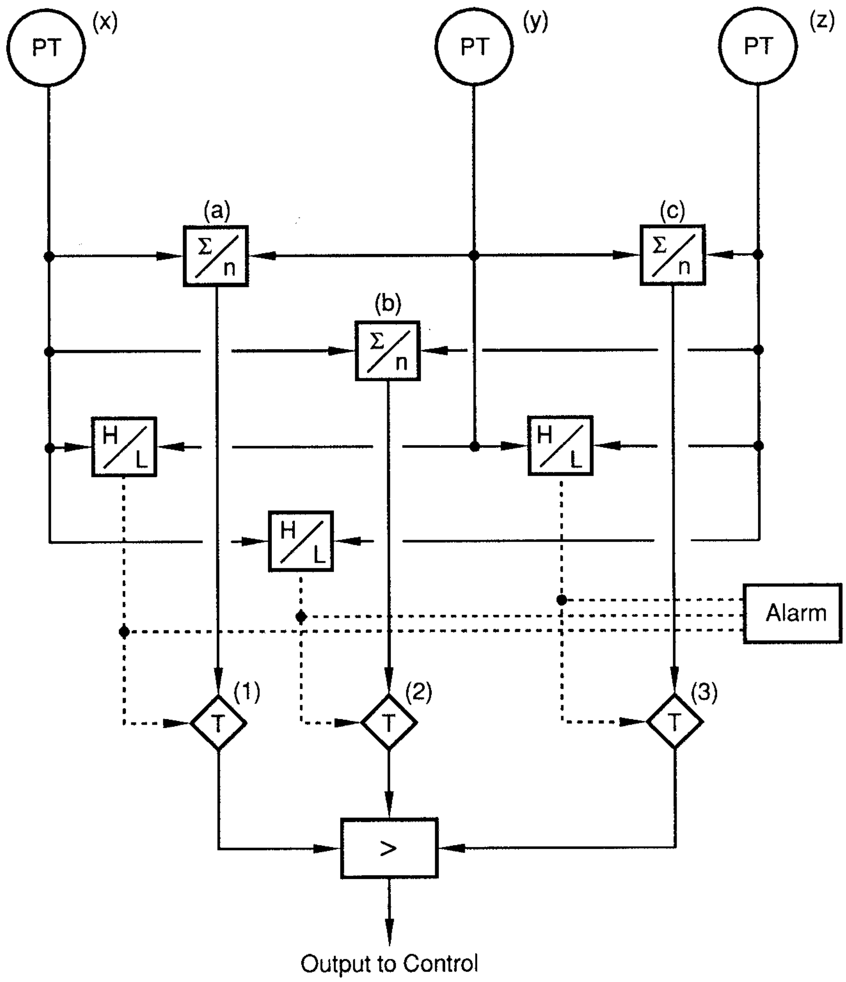


Figure 15-9 Logic for Two-out-of-Three Voting Using Three Transmitters

In the logic arrangement of Figure 15-9, averages of the outputs from each pair are developed in items (a), (b), and (c). Next the magnitude of the error or deviation, plus or minus, is monitored in the signal monitors (1), (2), and (3). The signal monitors are set for a maximum allowable deviation, plus or minus, for example, 5 percent. If there is no such deviation, there is no signal from the signal monitor, and switches (1), (2), or (3) will be open, allowing the average of that pair to enter the high select units (e). If the deviation is greater than 5 percent (for example), it will be alarmed.

Assume (x − y) and (y − z) are each within a 5 percent deviation limit. This means that the (x − z) deviation is not greater than 10 percent. The average of (x − y) and (y − x) will each enter the high select (e). The higher average of each pair will be the output and will be within 2.5 percent of the average of all three inputs. If all three are within the deviation limits, the output will be within 1.25 percent of the average of all three.

If only one pair or 2 out of the three are within deviation limits, then the average of that pair will be the single input to (e) and will be selected as the high value. If there is excess deviation in all three, the output of (e) will be zero and all three will be in alarm. At this time system security can block control action. Alternately, the system can be programmed to trans-

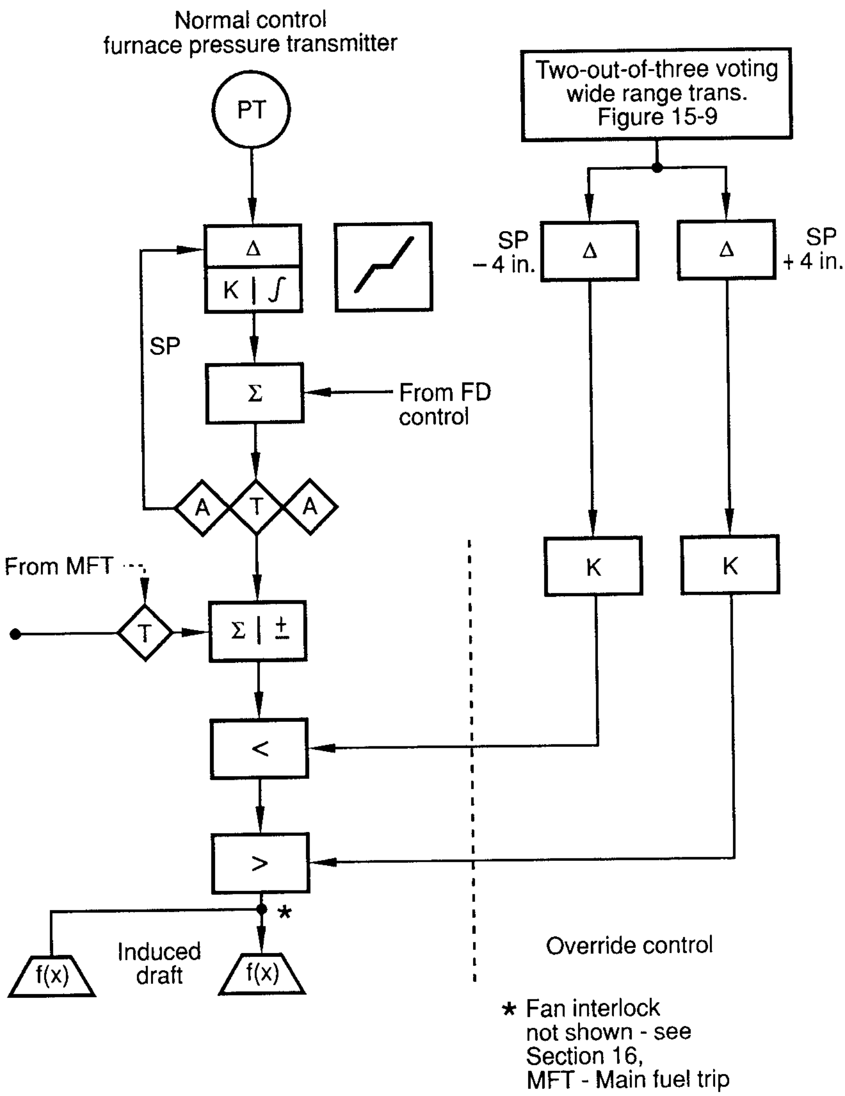


Figure 15-10 SAMA Logic for Furnace Draft Control Including Implosion Protection

fer to the next lower level of security by automatically transferring to median select of the three transmitters.

A SAMA diagram representation of one approach to the entire furnace draft control logic, including implosion protection, is shown in Figure 15-10. This arrangement is consistent with the block diagram in NFPA 85G.

Section 16
Measurement and Control of Combustion Air Flow plus Related Functions

The measurement and control of combustion air flow is key to the proper functioning of any boiler combustion control system. Accurate measurement of the combustion air flow is difficult and often must be improvised into the boiler system design by the control designer. It is important to remember that relative air flow is much more necessary for controlling boiler performance than accuracy of weight or volume of air flow. The important consideration is the proper amount of oxygen to burn the fuel available at any particular point in time.

The amount of oxygen in any particular volume flow or mass flow varies. This variation is due primarily to the variation in air density arising from the temperature and humidity of the air. Air is often measured at elevated temperatures and can easily carry a large percentage of its volume as superheated water vapor. Automatic temperature compensation may be necessary unless there is a predictable relationship between flow rate and temperature. Even within the normal ambient temperature range, the water vapor may occupy 10 percent of the volume. Water vapor, of course, has no oxygen content and does not contribute to burning the fuel. Because of this, even an absolute 100 percent accurate mass flow measurement of air flow would not completely fill the bill.

In addition to the measurement errors described, other factors to be considered are the difference in excess air requirements at different loads and measurement problems due to stratification or poor upstream duct configurations. Because of all the above, the procedure is to calibrate the measurement system as accurately as possible and apply off-line continuous correction of the calibration through flue gas analysis measurement and control. The true air flow measurement is, thus, after such continuous calibration.

In view of this, the correct method for initial calibration of the air flow measurement is to do so by the use of field combustion tests. By using such tests the air flow measurement is calibrated to match, on a relative basis, the fuel flow or other measure of air flow requirement. The field calibration compensates for the variation in required excess combustion air as the boiler load changes, plus any individual characteristic of the air flow primary measuring element.

In making the tests, the boiler is operated at a minimum of four base load points and with correct combustion conditions. Readings of air flow, fuel flow, and various other significant operating parameters are taken. The air flow output signal is then adjusted so that it reads the same as the total fuel signal.

The net result is that if a differential pressure type of measuring device is used, the flow vs. differential pressure calibration of this measurement is rarely a true square root relationship even though it normally approximates a square root.

16-1 Differential Pressure Measurement of Air Flow

Combustion air flow is customarily measured with some form of primary measuring element that is installed as a part of the boiler duct and fan system. This is used with a differential pressure measurement device. The ducts are of various shapes and sizes; they also have numerous 90 degree bends, short straight runs, and other features that are normally considered to be detriments to accurate measurement. These factors have a very significant effect on the actual flow coefficients and their characteristics of flow vs. differential pressure. This is one factor that necessitates field calibration by using the results of boiler combustion tests.

An excellent discussion of this subject and solutions to some of the measurement problems are given in the ISA Technical Paper ''Air Flow Measurement Techniques'' by Lyle F. Martz of the Westinghouse Electric Corporation. This paper appears in the proceedings of the 1984 ISA Power Symposium.

Any permanent pressure drop in the system as a result of the installation of the primary element increases the requirement for power to drive the combustion air fans. For this reason it is desirable that the primary element have a low differential pressure at full boiler capacity. Typically, the secondary differential pressure measuring devices have design differentials of 1 to 2 inches of water at maximum signal output.

Different types of primary elements have different discharge coefficients. The result is a difference in permanent pressure loss. The choice between primary elements based on permanent pressure loss (and, thus, fan power consumption) may be difficult to justify on an economic basis. Consider that the difference might be that of discharge coefficients of 0.6 and 0.85. If the full load differential pressure is 1 inch of H_2O, the permanent pressure loss would differ by 0.25 inch of water at full load. This would, however, be reduced to 0.0625 inch of H_2O at 50 percent load and 0.0156 inch of H_2O at 25 percent load.

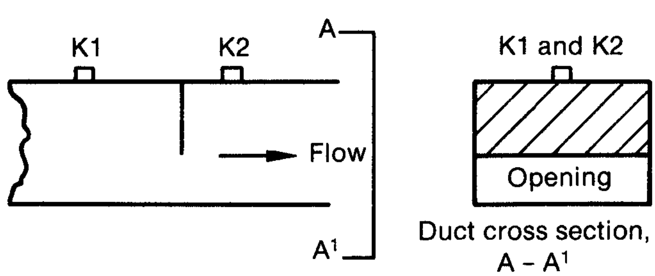
One potential primary device is an orifice segment in the forced draft duct. Figure 16-1 shows this type of device. It is simple to design and install, but its drawback is lower pressure recovery and, thus, greater permanent pressure drop. Considering the individual nature of the ductwork, an accurate design is impossible. An approximate design combined with field calibration can produce good results. The Martz paper mentioned above furnishes valuable insight into this method of measurement.

An approximate design can be made by considering the duct as a round duct and designing an orifice plate in a standard manner. The *d/D* (orifice diameter/pipe diameter) is then converted to an area ratio (*a/A*), which will be the square of the *d/D* ratio. Using the area ratio, the opening area can be determined. This area is subtracted from the duct cross-section area to yield the area of the orifice segment.

In order to reduce the permanent pressure loss of the measuring device, a Venturi-type duct segment, as shown by Figure 16-2, can be installed. The design of such a duct segment should be undertaken only by someone with good design basis information, such as a boiler manufacturer. This does not assure a good design, however, since the author experienced one case in which a design for 2 inches of H_2O differential yielded an actual differential pressure of 8 inches of H_2O. A recalculation confirmed the original design.

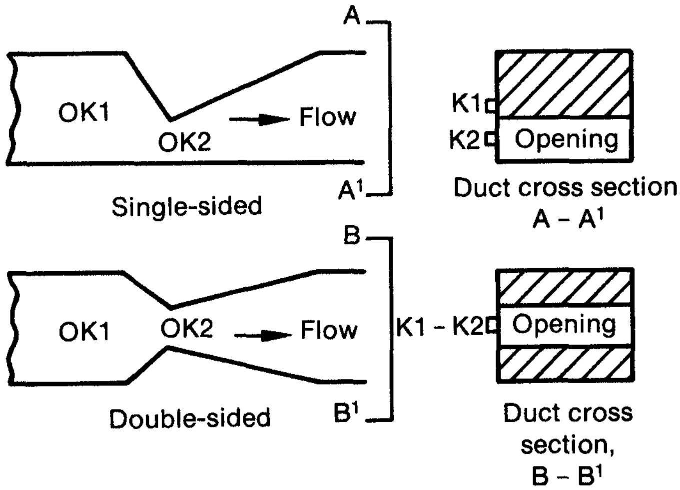
Further reduction in permanent pressure loss can be obtained by using an air foil design, as shown in Figure 16-3. The design of an air foil also requires background of such a design along with empirical data that is based on the actual results of previous air foil designs. Air foil designs are usually made by boiler manufacturers. A primary device of this type is also somewhat less expensive to construct than the venturi duct section.

Another technique that requires no additional power consumption is to the use the pressure drop across the boiler parts. One method is the use of the pressure drop across the air side of



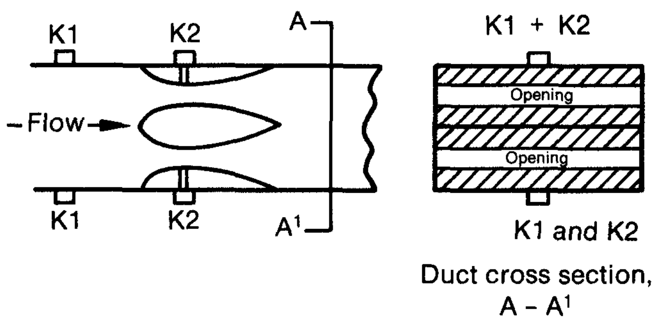
Note: K1 and K2 are pressure connections for ΔP.

Figure 16-1 Measurement of Combustion Air Flow with Orifice Segment in Duct



Note: K1 and K2 are pressure connections for ΔP.

Figure 16-2 Measurement of Combustion Air Flow with Venturi Section in Duct

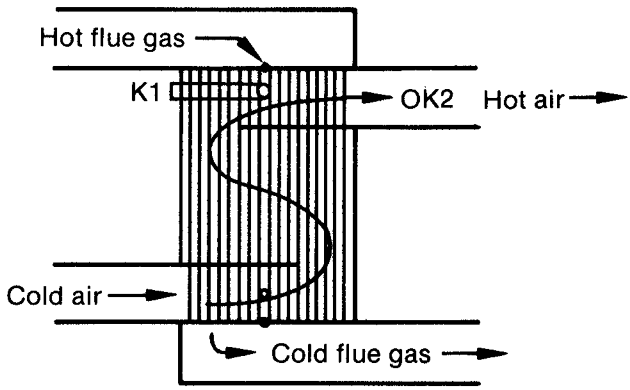


Note: K1 and K2 are pressure connections for ΔP.

Figure 16-3 Measurement of Combustion Air Flow with Air Foil in Duct

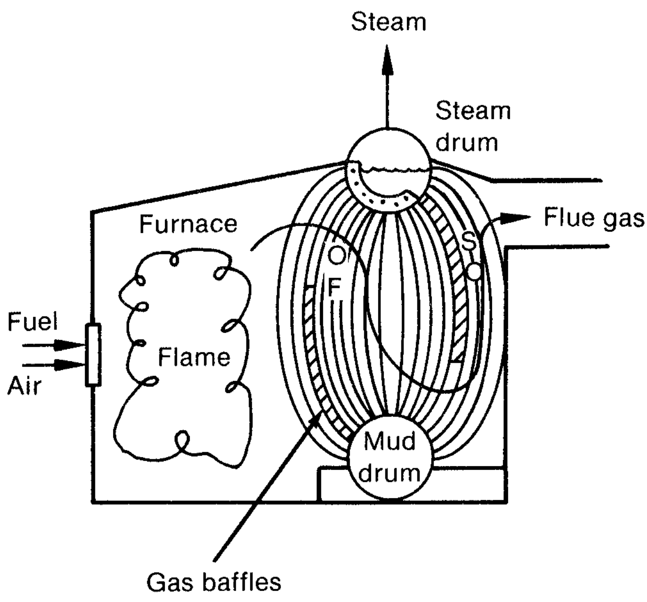
a tubular air preheater, as shown in Figure 16-7. There are usually 2 or more inches of H_2O available at full boiler load. In most such air preheater arrangements, the difference in elevation between the pressure connections requires compensation for the chimney or stack effect due to the difference in temperatures. The method of connection shown in Figure 16-4 will usually provide the necessary compensation. Using the preheater pressure drop is not a satisfactory method with a rotary regenerative air preheater because of variable flow path cleanliness and variable seal leakage.

Since the combustion air accounts for over 90 percent of the mass of the flue gas products of combustion, a measurement of flue gas flow can be used as an inferential measurement of combustion air flow. Figure 16-5 shows this method, which uses the pressure or draft differential across the boiler tube passes. The use of such a measurement tends, however, to produce a greater interaction between the fuel and air flow control loops. A further disadvantage is that such an air flow measurement is affected by soot or other foreign deposits on the boiler tubes. Another disadvantage is the unavailabilty in many cases of sufficient draft loss. As shown here, a difference in elevation of the pressure connections is used to compensate for the ''chim-



Note: K1 and K2 are pressure connections for ΔP.

Figure 16-4 Measurement of Combustion Air Flow Using Differential Pressure across a Tubular Air Preheater



Note: F and S are pressure connections for ΔP.

Figure 16-5 Inferred Measurement of Combustion Air Flow by Pressure Drop across Baffles in Flue Gas Stream

ney'' effect that results from temperature difference of the flue gases at the two measurement points.

Other differential pressure primary element devices that can be used are various devices based on the Pitot principle. In the Pitot tube, the pressure differential is the difference between the static pressure and the velocity head or pressure. Such devices are the Pitot Venturi, the piezometer ring, the ''piccolo'' tube, the Annubar™, and other forms of the Pitot tube. In some cases these are used in multiples in order to obtain averages of different points within the duct. For these devices the permanent pressure loss is very small and, thus, as compared to the restriction devices some power saving results.

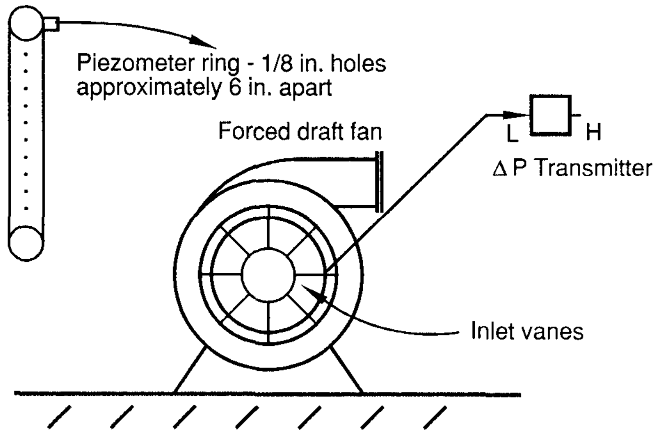


Figure 16-6 Typical Piezometer Ring Installation

The piezometer ring and the "piccolo" tube work on the same principle. They are usually mounted on the inlet to the forced draft fan to measure the velocity of the combustion air as it enters the system. This measurement may gradually deteriorate if there is a variation in the leakage rate of the combustion air preheater. These devices are shown in Figures 16-6 and 16-7. The averaging Pitot tube device, the most common of which is the Annubar™, is shown in Figure 16-8. All of these devices can produce close to 2 inches of H_20 differential at full capacity.

The calibration method for the air flow measurement by combustion testing is the preferred and most precise method when dealing with total combustion air flow. This total combustion air flow may be made up of several streams that are added together. These individual streams also require calibration. The whole air flow measuring system is in correct calibration when the total air flow signal matches the fuel signal and the individual flows add up to the total flow.

Calibration of the individual flows can be accomplished without the boiler operating by taking readings of Pitot tube traverses up and down and across a duct. From these, an average

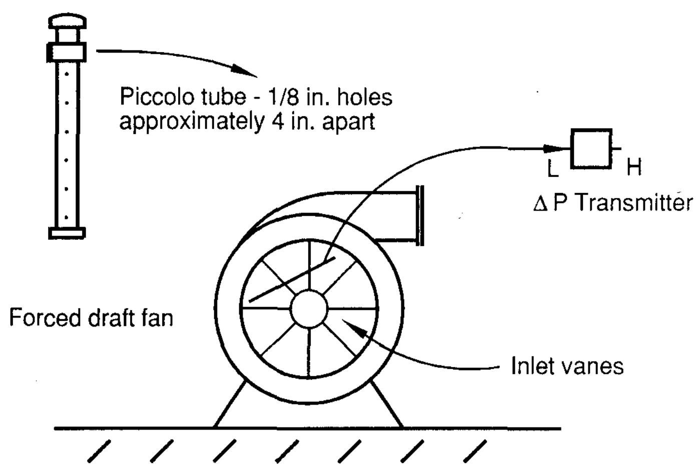


Figure 16-7 Piccolo Tube Installation

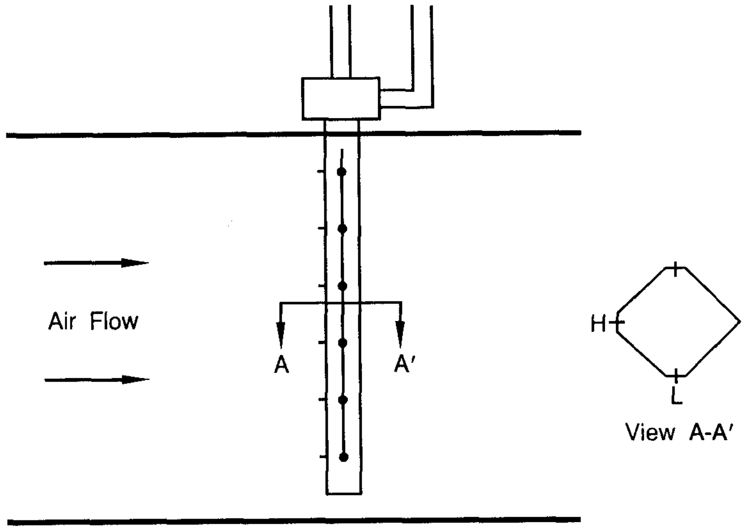


Figure 16-8 Averaging Pitot Installation

flow velocity is determined. By making corrections for air temperature of the normal flowing condition compared to the test condition, the correct calibration can be calculated.

In some cases the flowing temperature can be altered with a steam coil air heater so that similar tests at a different temperature, or a temperature close to the normal operating temperature, can be run. The calibration that is attained, adjusted if necessary to normal operating temperature conditions, should match the calibration achieved by the method of the preceding paragraph. If they do not, at least one of the tests is in error, and sufficient retesting should be done to assure confidence in the calibration.

16-2 Non-Inferential Methods of Air Flow Measurement

The measurement methods described above are methods for inferring air flow from air flow differential pressures with the basic flow velocity formula $V = (2gh)^{0.5}$. In recent years the use of a fundamental measurement of mass flow is being tried.

This method is an enhancement and development stemming from the ''hot wire anemometer.'' This device has been widely used in the HVAC field for flow measurement testing. A heated element is in the path of the air flow. As the flow increases, heat is absorbed, the wire heating element cools, its resistance decreases, and additional electrical current is required to maintain the same heated state. The current can be transformed into Btu and, using the specific heat of the air (approximately 0.24 Btu/lb), the mass flow of air can be determined.

The modern version of this device uses such elements in arrays across a duct in order that total air flow can be accurately measured. Because of the potential for air flow stratification in a duct, the flow must be measured at a number of points. An approximate number of elements is one per square foot. The configuration of this device is shown in Figure 16-9.

This will produce a mass flow measurement of the combustion air but must be further adjusted for variations in excess air. A function generator connected to the output signal produces a signal compensated for the desired variation in excess air as the boiler loading varies. A continous calibration of this signal from a flue gas analysis trim control system is necessary to compensate for the effect on combustion oxygen flow from humidity variations. Only a very small percentage of the installations now use this method and any clearcut advantages or disadvantages have not been fully determined.

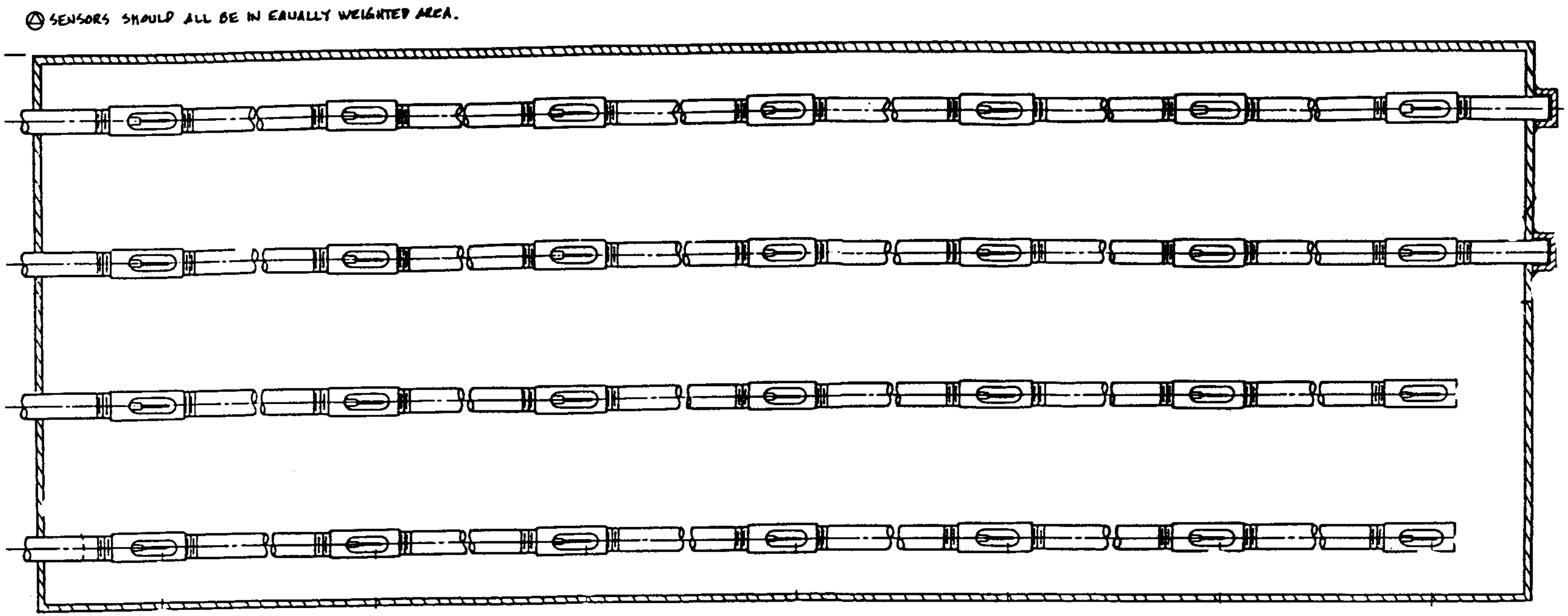


Figure 16-9 Heat Absorption Air Flow Measurement

(Courtesy Kurz Instruments Inc.)

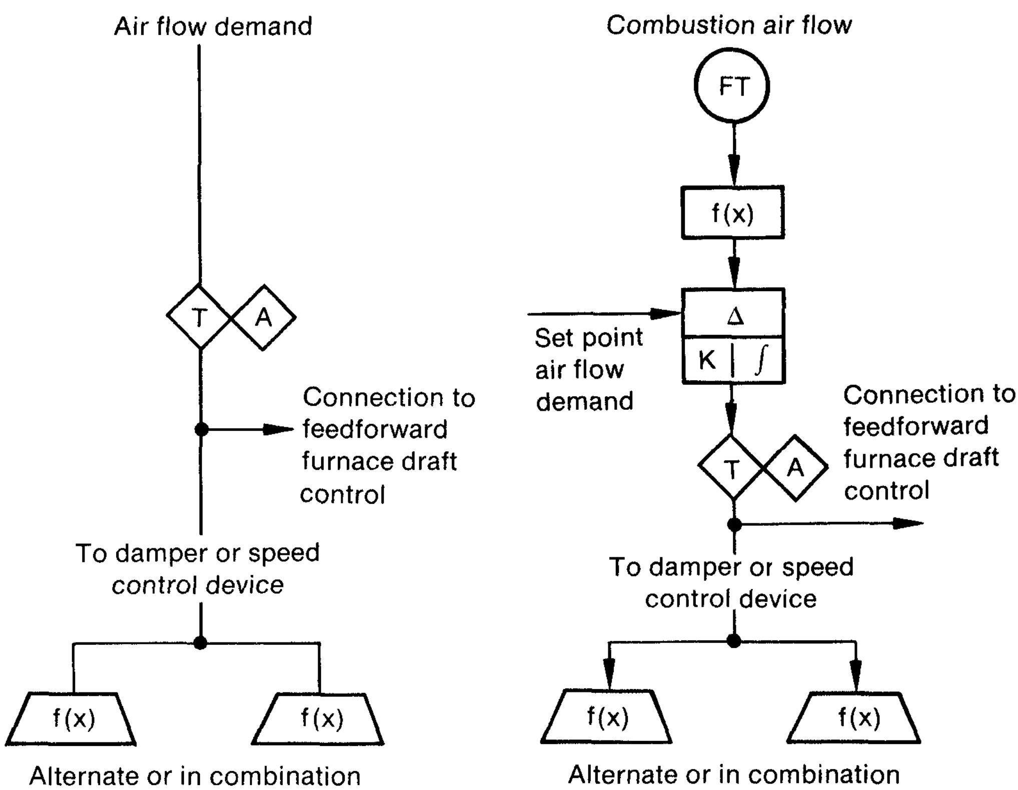
16-3 Control of Air Flow

Either open-loop or closed-loop control can be used for air flow control. An example of each of these two control arrangements is shown in Figure 16-10. In the open-loop arrangement the combustion air flow demand resulting from the boiler steam load is satisfied by positioning the controlled device. The expected result is a certain quantity of air flow as governed by the characteristics of the controlled device and fan speed.

At constant fan speed the position of the controlled device determines a close approximation of the flow rate. This is true only if a high percentage of the total system pressure drop occurs across the controlled device. If this is not true and the upstream or downstream pressure varies, the flow rate will vary.

To compensate for such changes, closed-loop feedback control is used in order that the flow rate and the control signal remain equal. In this case, a deviation from the air flow set point feeds back to reposition the controlled device in order to maintain a given air flow. This is a typical feedback flow controller that utilizes both proportional and integral control functions. If the flow measurement and the controlled device are reasonably well matched in flow capacity, a starting point for the controller tuning is an initial gain (proportional) setting of 0.5.

The correct integral setting is geared to the total feedback time (usually a few seconds) of the flow control loop. The result is typically a starting point for the integral (repeats per minute) setting of 10 rpm. The gain and integral tuning of the loop are also affected by process



(A) Open loop **(B) Closed loop**

Figure 16-10 Combustion Air Flow Control

noise. It should be remembered that the air flow control time response ultimately should be matched with fuel flow response. This may result in one of these loops having less than optimum tuning.

If the boiler uses both forced and induced draft fans, it is desirable to connect the control signal to the controlled device as the feedforward signal in a feedforward-plus-feedback furnace draft control loop. This tends to reduce or eliminate interaction between the air flow and furnace draft control loops.

The arrangement above concerns installation with not more than one forced draft fan or one set of forced and induced draft fans. For the larger boilers used in electric utility installations, two or more sets of fans operating in parallel are almost universally used. Typically, most boilers of over approximately 600,000 lbs/hr steam flow maximum capacity would use two or more sets of fans.

If two or more fans normally operate in parallel to supply combustion air, the single-fan failure mode must be considered. If two or more fans operate in parallel, the failure of a fan would allow the output of the operating fan or fans to be lost through the reverse flow openings to fan suction of the non-operating fan. In general, the requirements for parallel fan systems are:

(1) a change in gain between 1- and 2- or more fan operation;
(2) automatic closing of shutoff dampers on the inoperative fan to avoid air recirculation;
(3) the ability to balance the fan loads;
(4) usually, installation of additional control devices on the fan discharge dampers in order to achieve tight shutoff of air flow; and
(5) opening all dampers with all fans are tripped.

If not more than two fans are operated in parallel, the simplest approach is that shown in Figure 16-11. In the case of a single fan trip, digital interlock logic operates the transfer switch (a) in the control circuit of that fan and also operates the common transfer switch (b). In this way the 0 percent signal (e) is connected to the controlled devices on the fan that has tripped. Should the second of two or the third fan of three trip, the switches (a) will be in their tripped condition, but the common switch (b) will switch, admitting the 100 percent control signal (f) to all sets of control drives.

The shutoff damper control devices are calibrated for quick opening when the control signal is above 0 percent. The key to the operation is the digital interlock logic that operates the switches (a) and (b). This logic must be designed to fit the requirements of the particular installation.

As one or the other fan is tripped and the signal to its control drive goes to 0 at the inputs of (z) and (w), the gain of the control for the remaining fan is doubled. If one or the other is on hand control and its control signal fixed, the gain on the other is doubled. By adding to one and subtracting from the other, the manual signal from (u) allows the operator to balance the fan loads as desired.

If more than two sets of parallel fans are used, the control arrangement shown in Figure 16-11 does not provide a proper solution. One solution to this control problem is shown in Figure 16-12, which could also be used if there were two fans operating in parallel. The damper interlocking is the same as described above, except that there would be an additional item (a) for each additional fan if there were more than two fans in parallel.

The modulating control arrangement in Figure 16-12 acts in the manner previously described in the boiler load distribution of Figure 8-12. The control loop gain is automatically changed by summing the control signals in summer (c) and balancing the sum against the air flow demand signal in the high gain-fast integral controller (d). In Figure 16-12, with two fans of equal size, the input gains of summer (c) would be 0.5. The fans can be balanced manually by using the manual bias controls shown. Automatic balancing can also be added if desired.

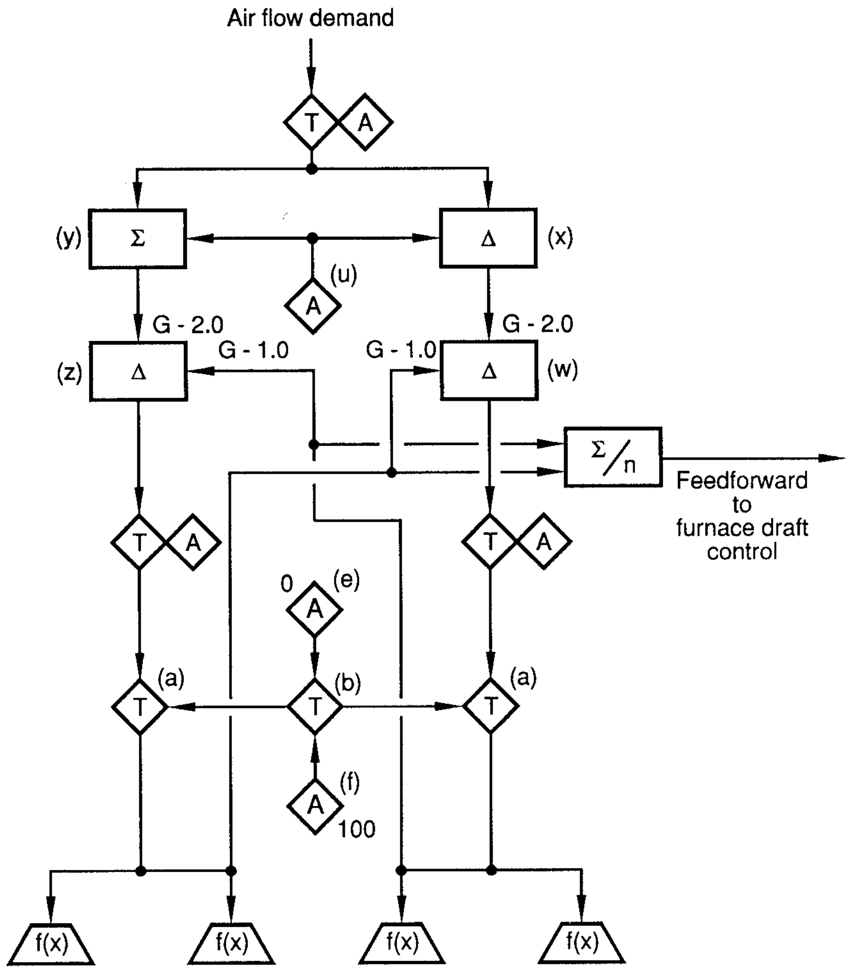


Figure 16-11 Air Flow Control (One or Two Fans in Parallel)

If the air flow control is open loop, this arrangement can be used in almost all analog or digital control loops. If closed loop, then cascade control is being used. The controller (d) is the secondary loop following a relatively fast primary flow control loop. This requires an order of magnitude speed requirement of the secondary loop as compared to the primary loop. Because the feedback and sensing are continuous and almost instantaneous for an analog control system, the controller (d) can be tuned very fast and still have a stable output.

Because of the sampling time interval and data transfer timing for a specific installation, there may be practical tuning limitations for a digital application of this technique. Such limitations might cause a large amount of detuning of the primary air flow control, resulting in unsatisfactory control performance.

In such a case, the alternate approach shown in Figure 16-13 can be used. This is the same technique discussed in Section 8 and shown in Figure 8-13. The fan damper interlock is the same as that of Figures 16-11 and 16-12. If a third fan is added and an additional item (a) is added, a third input with three 0.33 input gains into summer (c) is added. To provide a third gain potential, a third item (j), a third input into summer (d), and three 0.33 input gains into summer (d) are added.

In order to avoid significant changes in air flow when the dampers are changing position due to a fan trip, it is necessary to carefully match the control characteristics of the fans that

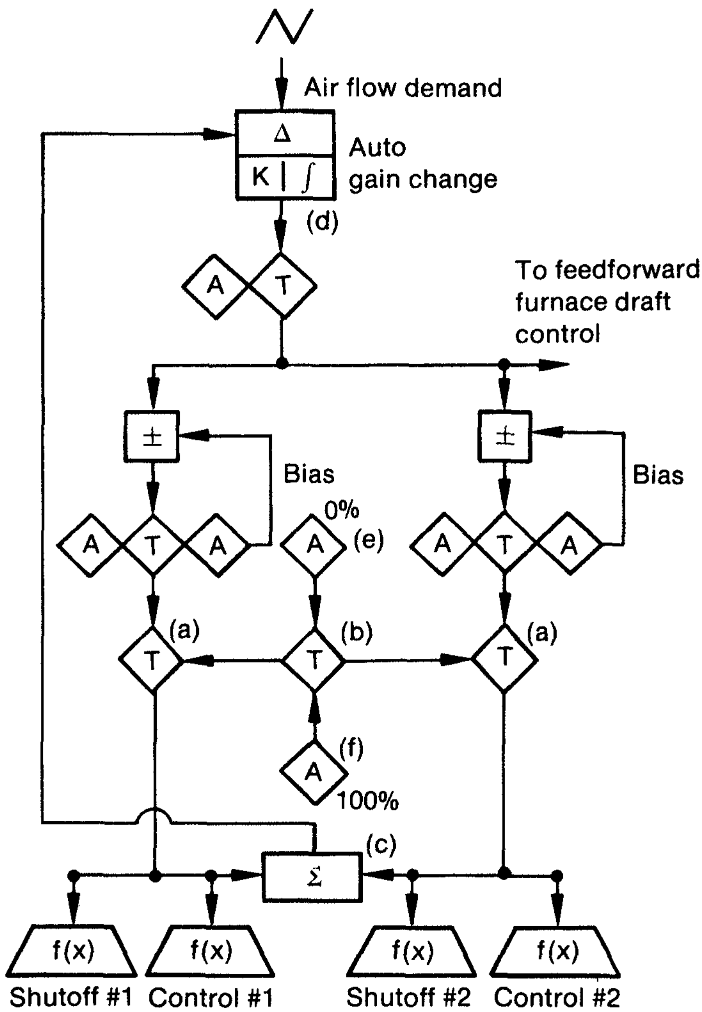


Figure 16-12 One, Two, or More Fans Operating in Parallel (Air Flow Demand Is Open Loop)

operate in parallel. This is accomplished by carefully matching the flow/position calibration and the timing of the controlled devices on the parallel fans. If the boiler uses both forced and induced draft, the time and flow characteristic matching should involve all fans (both forced draft and induced draft). In some cases the size and power of the damper control drives, and thus their stroking speed, may be different for the forced and induced drafts. Matching the characteristics in these cases results in the speeding or slowing of one of the sets of control drives.

16-4 Flue Gas Dew Point Control

Some of the boiler operation functions are closely related to the control of draft and air flow. These are the control of flue gas dew point and the blowing of soot from the boiler tubes.

Flue gas dew point control may be necessary only in cases where heat recovery equipment, such as an air preheater or an economizer, is applied to a boiler. In these cases the flue gases are cooled to a lower temperature than with a simple boiler with no heat recovery. Both of these heat recovery devices can be represented by the diagram, shown in Figure 16-14, of a

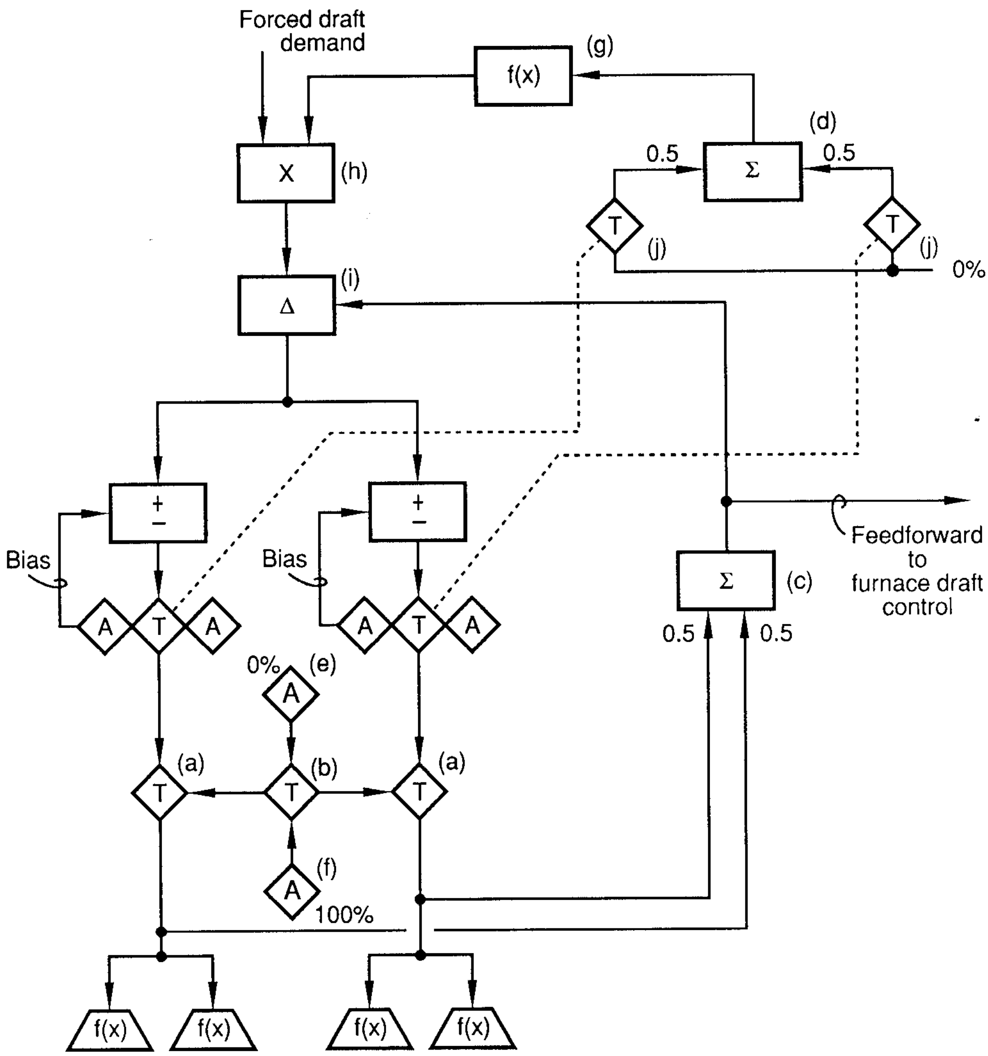


Figure 16-13 Air Flow Control — Closed-Loop, Using One or More Fans in Parallel

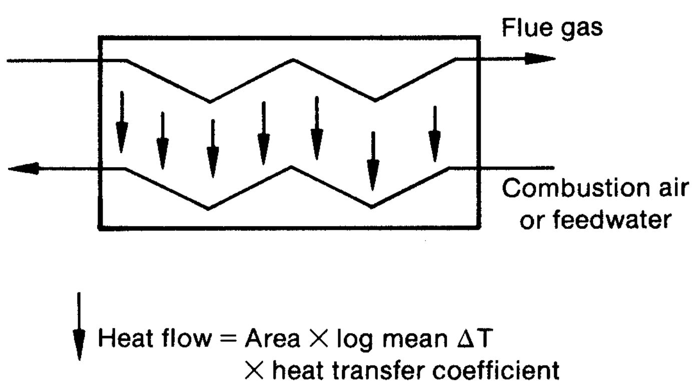


Figure 16-14 Economizer or Air Preheater Counterflow Heat Exchanger

counterflow heat exchanger. In the case of an economizer, the incoming fluid would normally be in excess of 220°F. In the case of an air preheater, the incoming combustion air might normally be less than 100°F.

In order to avoid corrosion at the cold end of an air preheater, it is necessary to maintain the flue gas temperature above the dew point temperature. The flue gas dew point temperature is determined by the moisture content of the flue gas and the presence and percentage of SO_2 and SO_3 in the flue gas. From the standpoint of moisture alone, the lowest expected dew point temperature would be approximately 135°F for flue gas from natural gas with no sulphur content. The addition of even small amounts of sulphur in the fuel, and thus SO_2 and SO_3 in the flue gas, causes a significant shift upward in the dew point temperature. The resulting moisture would be a weak solution of sulphurous and sulphuric acid.

Natural gas is normally free of sulphur, though some, called sour natural gas, contains sulphur in the form of hydrogen sulphide. There is often some small percentage of sulphur in fuel oil whose flue gas has a lower water vapor content than natural gas. Coal may also have a significant sulphur percentage, but its flue gases have a lower moisture percentage. Thus fuel oil, for a given sulphur percentage, may produce flue gas with the highest dew point temperature, often in excess of 200°F.

At the "cold end" of an air preheater, if the air temperature is 80°F and the flue gas temperature is 300°F, the average metal temperature in contact with the flue gas is considered to be 190°F, the average of the flue gas and combustion air temperatures. With a dew point temperature of 200°F, acid moisture would collect on the metal surface and corrosion would take place.

This can be avoided through use of a control method that raises the incoming combustion air temperature so that the average cold end metal temperature is above the dew point. As the incoming air temperature is raised, the flue gas temperature shifts upward a similar amount. In the example above, raising the incoming air temperature to 120°F would cause flue gas temperature to shift upward to approximately 340°F. The average metal temperature would then be 230°F, well above a 200°F dew point temperature. If the dew point temperature were 230°F, the combustion air temperature would have to be raised still higher than the 120°F point.

A control method that accomplishes this is shown in Figure 16-15. An air heater, with steam as the heating medium, is placed in the combustion air stream ahead of the flue gas heat recovery air preheater. The steam is controlled to this heater in order to develop the desired combustion air temperature. The control loop controlling the steam flow is shown in Figure 16-15. A simple feedback control loop as shown is usually adequate. The average of the flue

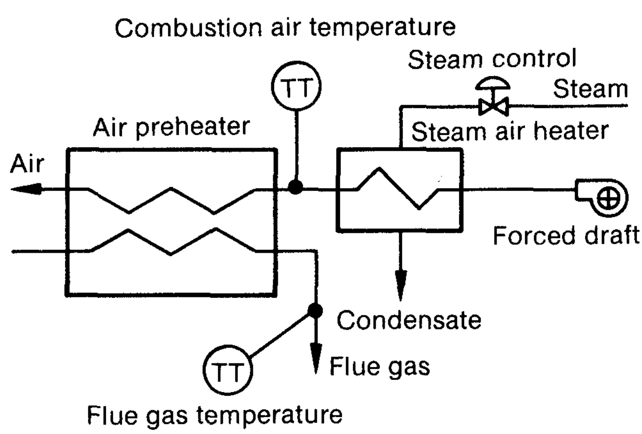
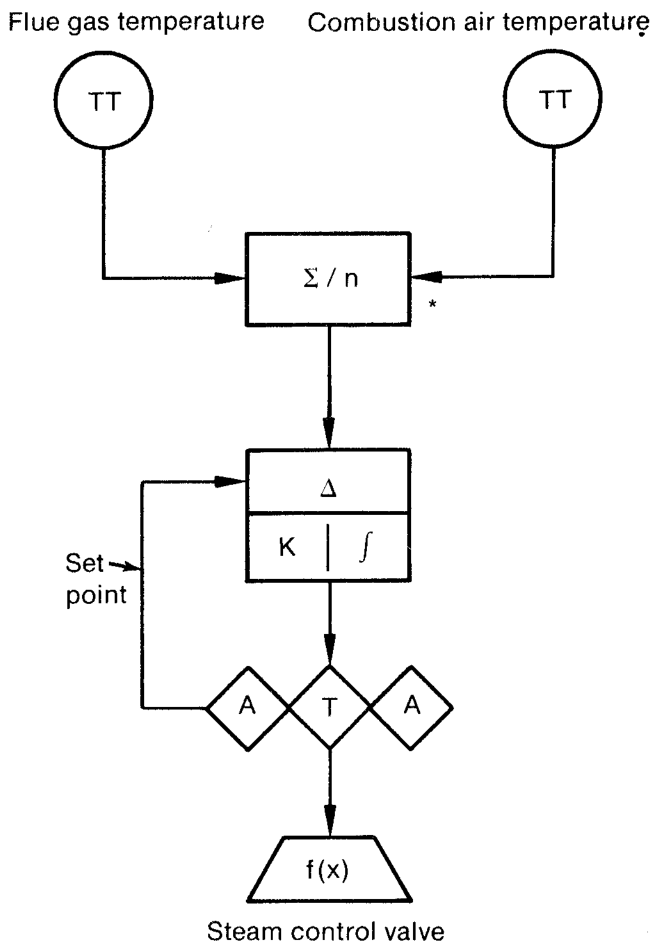


Figure 16-15 Dew Point Control Process Arrangement



*Average of flue gas and air temperatures equals pseudo "cold end" metal temperature.

Figure 16-16 Flue Gas Dew Point Control

gas and air temperatures, shown as the feedback, is considered a pseudo metal surface temperature. An alternate approach is to control the air temperature to provide a minimum flue gas temperature based on an assumtion of the coldest expected inlet combustion air temperature.

If corrosion-resistant materials are used, the allowable metal temperature can be decreased. This basic problem can also affect the corrosion of metal stacks when they are used. In these cases, the problem occurs due to low outside ambient temperatures. The solution to this problem is the use of corrosion-resistant materials for the inside of the stacks or the insulation of the stacks.

The operation of this type of dew point control has a small impact on air flow and draft control by changing flow resistance on both the flue gas and air sides of the boiler. A 40°F rise in air preheater inlet air temperature will change the average specific volume across the air side of the preheater and change its pressure drop by approximately 2.5 percent. The change in draft loss on the flue gas side will be a smaller percentage amount. This change will be only slightly noticeable even though open-loop air flow control is used.

Other types of dew point control rely on variable recirculation of a portion of the preheated air back to the forced draft fan inlet. This method has a greater impact on the air flow control by requiring a variation in the opening of the inlet vanes of the forced draft fan.

16-5 Soot Blowing

Soot buildup on the boiler tubes is a normal occurrence for liquid or solid fuel boilers. This soot accumulation can be reduced to some extent by maintaining the correct combustion conditions. In any event it must be removed periodically in order to avoid a severe loss in heat transfer. Figure 16-17 illustrates the loss in heat conductivity due to soot accumulation. Soot is an excellent insulator, as shown, and a thin layer can significantly reduce heat transfer. The effect is an increase in boiler draft loss and flue gas temperature, with a resultant loss in boiler efficiency.

Soot is normally removed from the tubes with devices called soot blowers. These are devices mounted along the sides of the boiler from which jets of steam or compressed air are used to blast the soot from the tubes. Some fuels have chemical characteristics that cause the soot to adhere to the tubes. In these cases fuel additives are often used to change the characteristics of the soot in order that it may be more easily removed from the tubes.

The normal practice is to start the soot blowing at the furnace and sequentially blow soot into the flue gas stream. Operation of soot blowers near the front of the boiler, in particular, may cause severe pulsations in the furnace draft. For this reason it may be desirable to reduce the set point of the furnace draft control and increase the level of combustion air flow during soot blowing periods.

Since the 1960s there has been continuous development of directing the soot blowing sequence through the use of computer control. The success of such schemes has been shown to depend on the availability in the computer of a relatively sophisticated boiler model along with the input of continuous measurements of boiler excess air, flue gas temperatures, boiler tube temperatures, and other factors.

The model develops values of draft losses and tube and flue gas temperatures based on the boiler load, fuel characteristics, and excess combustion air. In this way local dirty spots in the boiler are identified and may be cleaned in a random sequence as required by the model. In addition to improved heat transfer, a benefit of successful computer-directed soot blowing is a significant reduction in the cost of the compressed air or steam blowing medium. A boiler model sufficiently detailed to be used for this application is expensive and can probably be justified only on utility size boilers. Before embarking on this approach to soot blowing, a careful economic study should be made.

Some more simplistic methods of computer-directed sequencing of soot blowers, based on

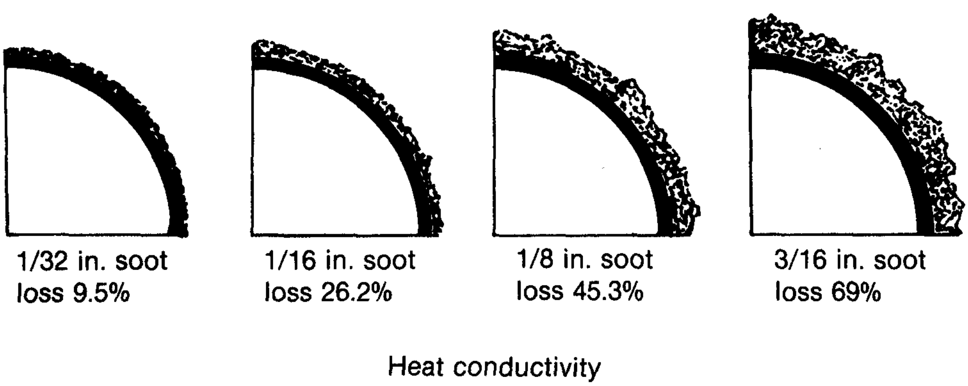


Figure 16-17 The Effect of Soot Buildup

(From a technical paper, © Diamond Power Specialty Company) measurements of only boiler draft losses and flue gas temperatures, have been investigated. The overall problem is quite complex, and such simple solutions may cost significantly more and be less economically effective than the traditional time-based sequencing.

Soot blowing may also impact the feedwater control performance. In some cases saturated steam for the soot blowers is taken directly from the steam drum output and is unmeasured. In these cases the measured steam flow and the measured feedwater flow of a three-element feedwater control system will not match. The effect is to cause a small shift in the boiler drum level set point. If this causes a serious problem in a particular case, integral action can be added to the drum level controller. As stated in the section covering feedwater control, the integral setting should be a very low value, approximately 0.1 rpm (repeat per minute).

Section 17
Flue Gas Analysis Trimming of Combustion Control Systems

In Section 6, the effect of flue gas constituent percentage on boiler efficiency was discussed. It follows that flue gas analysis can be used to more precisely control the ratio of fuel to air.

There is a general perception that flue gas analysis instrumentation is more complex, more costly, and less reliable than other types of measuring instruments normally used in boiler control systems. On the other hand, the analysis of flue gas provides a measurement that is more precise than fuel/air ratio measurements that use simpler methods. Flue gas analysis does, however, consume some time, is after the fact, and the combustion process can change very rapidly. In order to achieve control precision along with timeliness of control action and maximum reliability, the general practice is to design the basic control system using the most reliable basic instrumentation and to use flue gas analysis as a superimposed trimming recalibration control loop.

Flue gas analysis trimming control is applicable to combustion control systems for all types of fuel firing and all types of basic boiler control systems from the simplest to the most complex. Since the application is universal, it is being discussed separately from the discussion of the more individual basic systems.

There are a number of variations of the analyzer-based trimming control loops. These variations are based primarily on the particular flue gas analysis made. There are also degrees of desired fuel/air ratio precision, control sophistication, and costs as a by-product of these variations. In this context the desired fuel/air ratio is that which results in the lowest fuel cost operation of the process.

17-1 Useful Flue Gas Analyses

Several flue gas analyses are potentially useful in combustion control trimming loops. The analyses are used individually or in combination, based on the type of trim control desired. Generally, the following analyses and the combinations in which they may be used are as follows.

Analyses

(1) % Oxygen (% O_2)—Excess combustion air is a function of % oxygen.

(2) % Opacity—A measurement of smoke or particulate matter. Due to environmental standards, this measurement may be needed for use in limiting the control action based upon other analyses.

(3) % Carbon dioxide (% CO_2)—When total combustion air is greater than 100% of that theoretically required, excess combustion air is a function of % carbon dioxide.

(4) Carbon monoxide (CO) or total combustible in the ppm range—This measurement is that of unburned gases. Measurement in the ppm range is necessary if desired control precision is to be obtained.

Uses

(5) % Oxygen as an individual control index. (This is the most tested, with control application since the early 1940's.)

(6) PPM CO or total combustible as an individual control index. (Application of this method began approximately 1973.)

(7) % Oxygen in combination with ppm CO or total combustible. (Applications began approximately 1977.)

(8) % Carbon dioxide in combination with ppm CO. (Applications began approximately 1977.)

(9) % Oxygen in combination with % opacity. (Applications began approximately 1977.)

The time of actual field application is important since the number of installations, the testing and feedback time period, and the potential number of different types of applications on the multitude of boiler variations are important factors in solidifying the application practice.

17-2 Methods of Flue Gas Analysis

Since approximately 1970 new methods have significantly improved the reliability and precision of flue gas analysis. Prior to this time flue gases were analyzed by drawing a sample of the flue gas from the boiler ducts, washing and cooling the sample, and passing it into the flue gas analyzer. This action reduced the sample temperature below the dew point, and the water formed in the combustion process was condensed. Since the sample as analyzed did not contain this water, it was called a ''dry'' sample.

The sampling system also introduced a time lag in the analysis. This time lag was generally in the range of 30 seconds to 2 minutes. In addition, due to the potentential for sample line plugging as the sample reached the dew point, the sampling system was the major source of maintainence in the analyzer system.

The newer generation of analyzers has made a quantum jump in reliability by analyzing the flue gas on the ''wet'' basis. This is done by using methods that do not cool the hot flue gas and, therefore, do not cool it below the dew point. There are three basic methods for accomplishing this.

The ''In Situ'' Point Sample Method

This method uses an analyzer probe, which is inserted into the duct at the point of analysis. An analysis cell on the end of the probe analyzes the hot flue gas flowing past it. With no sampling system time lag, typical response times of these analyzers is 5 to 10 seconds. Figure 17-1 shows the arrangement for this method. Percent oxygen and ppm CO can be measured in this manner.

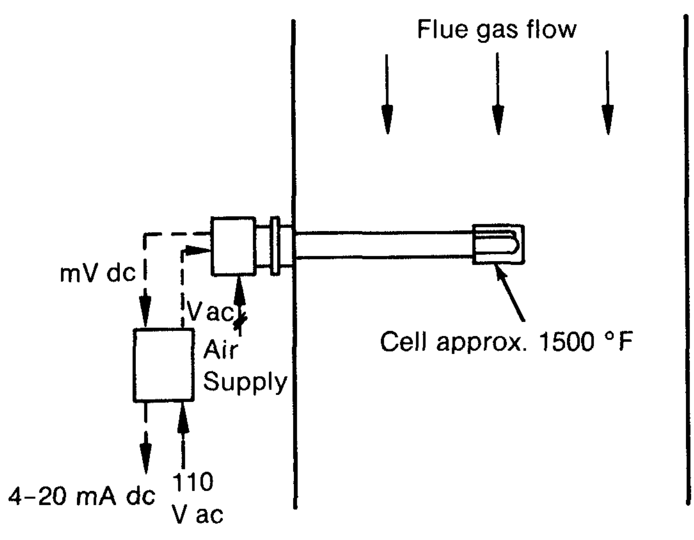


Figure 17-1 In Situ Flue Gas Analysis (Point Sample)

The analysis will be that existing at the point of cell location in the duct. Due to stratification of the flow in large ducts, electric utility size boilers generally use several probes at different locations in the duct cross section. The best location of these probes can be determined by traverse testing. The outputs of the multiple analyzers are often averaged.

The location of this type of probe along the flue gas stream is very flexible, needing only an open area on one side of the boiler duct system for probe insertion. Because of the air seal leakage of regenerative air preheaters, the analyzer probes should be installed in the flue gas stream ahead of such air preheaters.

If % oxygen is being measured, the measuring cell is zirconium oxide. The principle is that of a fuel cell. The cell reacts to the ratio of the partial pressure of oxygen in the flue gas to the partial pressure of oxygen in reference air that is also admitted to the cell. Because of the cell temperature, any residual combustible gas is burned in the cell and absorbs some of the oxygen in the flue gas. The result is, therefore, a measurement of % net oxygen.

The output of the cell is a millivolt signal that is an analog of the logarithm of the partial pressure ratio of the % oxygen in the flue gas to that of the reference air. In order that results will be accurate and repeatable, the cell temperature is closely controlled. The typical controlled temperature set point for such cells is in the 1300 to 1600°F range. The upper temperature limit for this method is that of the cell temperature.

The cell output (approximately 0 to 100 mV) is further processed by inversion, linearization, and amplification to produce a linear signal of % oxygen vs. milliamps (mA). In this form the signal can easily be used by any form of standard control instrumentation.

If ppm CO is being measured, the analysis cell operates on the infrared absorption principle. In this principle carbon monoxide absorbs a part of the infrared energy from an included infrared source. The amount absorbed is within the frequency range specific to carbon monoxide. Measuring the absorption of infrared energy within that range provides a measurement of carbon monoxide.

The resulting electrical signal in some (not all) of the ppm CO analyzers is compensated for % moisture, temperature, and excess air and is processed and amplified to that of a standard instrumentation signal. The upper limit on flue gas temperature for this method is typically in the 600 to 700°F range. Even below this range infrared energy emitted from the flue gas interferes with that emitted by the infrared source. The result is a need for significant and precise temperature compensation. Some of the most recent designs have been compensated to allow an upper limit temperature of 1000°F.

Extractive or "Ex Situ" Method

The extractive or ex situ method is shown in Figure 17-2. In this method a small sample of the flue gas is drawn from the duct to the heated cell housing mounted on the duct wall. Because the sample is admitted back into the duct after analysis and never cooled below the dew point, sampling maintainence is not normally a problem. As with the "in situ" method, the analysis is on the "wet" basis.

For an air aspirated sample, the response time of this method is typically in the range of 10 to 15 seconds. For natural or thermal aspiration, the response time is in the range of 25 to 40 seconds. Both time periods consist only of the sample transport dead time duration from the tip of the sample probe to the cell. Typically, another 15 seconds of dead time should be added for the sample transport time from the burner to the sample point. This applies to all flue gas analyzers.

The measurement is the analysis of the flue gas existing at the tip of the sample probe. As with the in situ method, for installations in the large ducts of electric utility size boilers, averages of several individual duct measurements are needed. Special averaging sample probes can be used or multiple analyzers can be used and their output signals averaged.

As with the in situ analyzers, the location along the flue gas stream is flexible since installation is made from one side of the boiler or flue gas ducts. Also as with the in situ ana-

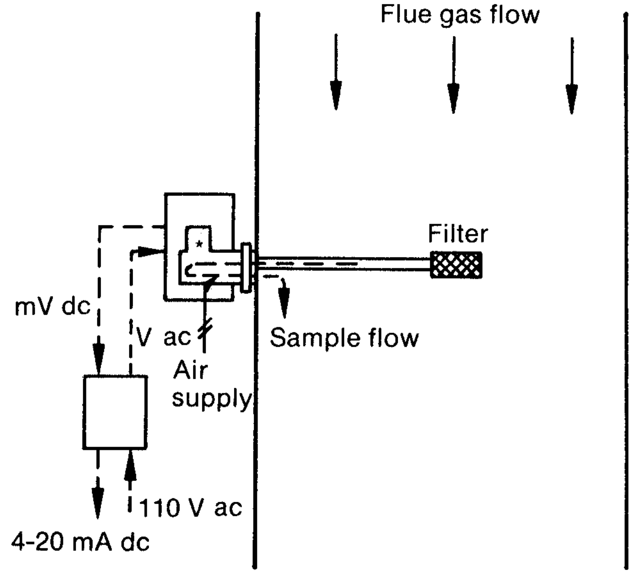


Figure 17-2 Extractive or Ex Situ Flue Gas Analysis (Point Sample)

lyzers, the installation should be ahead of sources of "tramp" air, such as the leakage of the seals of regenerative air preheaters. This method is applicable to the measurement of % oxygen and ppm or % total combustible.

If % oxygen is being measured, the measurement principle is the same as that of the in situ analyzers. In this case the temperature-controlled zirconium oxide cell (approximately 1500°F) is located in the cell housing, mounted on the duct instead of in the flue gas stream. The measurement signal and signal processing functions are also the same for the two methods.

Since the temperature-controlled cell is located outside the flue gas stream, cell temperature does not limit the application of this type of analyzer. In this case, the limit is based on the material of the sample probe. By using sample probes of ceramic material, flue gases can be analyzed with this system up to more than 3000°F.

If ppm total combustible is being measured, a catalytic combustion principle is used. In this case, any remaining combustible gas that is present in the flue gas is "burned" in the measurement cell to produce a signal that is then amplified to that of standard measuring instrumentation. The transport of the flue gas sample to the cell is identical to that of the % oxygen measurement. The cell can also be housed in the same assembly with the cell for measuring % oxygen, and the same flue gas sample can be used. In this case, two output signals are obtained: % oxygen and ppm or % total combustible.

Light or Infrared Beam across the Stack

The third general method is the use of a beam of light or infrared energy across a flue gas duct or stack. This arrangement is shown in Figure 17-3.

If ppm CO is being measured, the beam is infrared energy. The same principle of selective infrared absorption as used in the "in situ" ppm CO is also used in this type of analyzer. Analyzing the infrared absorption in an additional frequency range is sometimes used to measure % carbon dioxide (CO_2). As with the in situ measurement of ppm CO the normal application limitation of 600 to 700°F flue gas temperature applies. As indicated previously, some manufacturers now specify 1000°F as the maximum temperature.

The beam is transmitted through the flue gas from one side of the stack to the other. Both

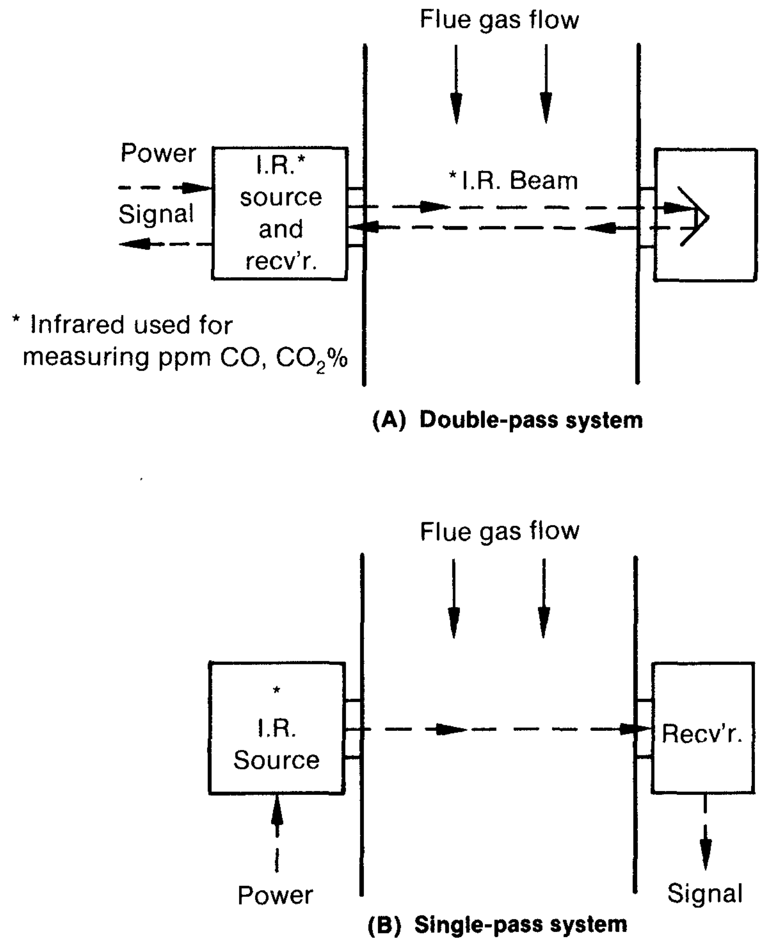


Figure 17-3 "Across the Stack" Flue Gas Measurement

single-pass systems (measurement on the opposite side of the duct or stack from the source) and double-pass (in which the beam is reflected back to the side of the source) systems are used. Carbon monoxide in the ppm range and opacity are generally measured in this manner. If ppm CO is measured, The formula that is used to calculate the signal is known as "Beer's Law." One factor used directly is the length of the infrared path. The result is that a double-pass device has a greater signal and greater sensitivity

If % opacity is to be measured, the essential principle is that a beam of light is shone across the duct, and the measurement of light intensity that is received on the opposite side is a measure of % opacity.

For the applications described above, in general, locations must be found that are free of obstruction on both sides of the duct or flue gas stack. This often dictates that the analyzer be located in the flue gas stream after the air infiltration or leakage at the air preheater. This also tends to make the installation and location of this type of analysis less flexible and more costly than that of the single-side installations. While the above is correct in general, there is one design of this type of ppm CO analyzer that uses a single-side boiler duct penetration.

Comparison Factors

A number of different manufacturers can supply analyzers as described above. Each manufacturer has designed his instrument and based his sales argument on his analysis of all the "for" and "against" factors. Any particular manufacturer is likely to emphasize his strong points and deemphasize his weak points in any comparison. "For" and "against" cases can be made to favor any particular choice of measurement or useful combinations of measurements. For example:

(1) In situ vs. ex situ
(2) Point sample vs. average sample
(3) % Oxygen vs. CO ppm
(4) Total combustible ppm vs. carbon monoxide ppm
(5) % Oxygen-CO vs. % oxygen-opacity
(6) % Carbon dioxide-CO vs. % oxygen-CO
(7) % Oxygen-CO vs. % oxygen-total combustible

17-3 Pros and Cons of Measurement Methods and Gases Selected for Measurement

This boils down to:

(1) the selection of the constituent gas or gases for measurement in relationship to their intended use; and
(2) the quality of measurement of these gases based on the capability of normally used measurement methods.

Selection of the Constituent Gas or Gases

Since the mid 1970s, there has been something of a controversy between the use of % oxygen for trimming control as opposed to CO in the ppm range. Another controversy from the 1940s was the use of % CO_2 vs. % oxygen.

Percent oxygen won the battle with % CO_2 many years ago due to its nonambiguity in the low excess air ranges. The same % CO_2 reading may mean either an excess or deficiency of air. The % oxygen is relatively unaffected by the carbon/hydrogen ratio of the fuel. The percent of span for the same excess air span is greater for % oxygen, and the measurement accuracy of % oxygen analyzers is better than for those measuring % CO_2.

For the use of ppm CO, a theory was advanced that optimum low cost operation is obtained if the CO can be kept at a constant set point somewhere in the range of 250–400 ppm. The second part of this was that only a constant set point ppm CO single-element feedback controller trimming the air flow control was necessary.

It is now generally agreed that CO in the ppm range should not be used alone. This measurement can provide useful information, but if used it should be used in combination with % oxygen or % CO_2. Both of these are valid, but, for the reasons above, the preponderance of the argument favors the use of % oxygen. A constant set point ppm CO has been highly touted as the final arbiter for optimizing the combustion process. We now know that if a final control from ppm CO is used, the set point should be variable with respect to boiler load, just as the % oxygen set point should be vary as the load changes.

Percent oxygen can be used alone for control but should be used with a variable set point related to boiler load. The control should include a small margin of excess air. A margin of 3 to 4 percent excess air (0.6 to 0.8 percent oxygen) is usually suggested.

Because of this margin, the ppm CO control can theoretically be operated closer to the limit. Even the ppm CO control, however, should have some margin. The ppm CO measurement is "noisy" with constant fluctuation above and below the operating set point. Because of the nonlinearity of the ppm CO vs. excess air curve, the loss for each fluctuation above the operating point is greater than the opposite fluctuations below the operating point. To be most economical, the average of the two should not exceed the loss of the indicated optimum point.

Limiting Factors in Reducing Excess Air

There are a number of limiting factors to the reduction of excess air. CO in the flue gas is one of them, but in the majority of cases the lower limit on % excess air is due to other economic factors. Some of these are:

(1) smoke or opacity,
(2) furnace maintenance,
(3) furnace slagging and cleaning problems,
(4) unburned carbon in the refuse,
(5) unburned gases other than CO, and
(6) burning of stoker grates.

More discussion of the limiting factors may be found in subsection 17-7.

The most extensive study, involving 4000 individual boiler tests and 75,000 flue gas analyses combustion tests, was published in an ASME paper in 1926. No study or other material approaching this comprehensive study has since been made. Generally, the results indicated that for gas-fired and stoker-fired boilers CO was the limiting factor in approximately 60 percent of such boilers tested. In oil-fired boilers, carbon monoxide was the limiting factor in approximately 20 percent of such boilers and in less than 10 percent of the pulverized coal fired boilers.

It follows that minimizing the flue gas losses from excess air and unburned gas by minimizing CO is the answer to most economic boiler operation in only some boiler installations. In other installations the limitation must be based on other limiting factors such as smoke or opacity, furnace temperature on slagging or grate maintenance, and unburned carbon in the solid refuse.

Whatever the lower excess air limit, it should be controlled. An excess air control technique can be developed for any boiler using % oxygen control, a combination of % oxygen and ppm CO, or a combination of % oxygen and % opacity. The preponderance of data indicates that a % oxygen measurement must be part of any strategy for optimizing unburned gas losses.

Quality of the Flue Gas Analysis Measurement for the Stated Purpose

Various claims are made for the superiority of one measurement method over another. One comparison pits the ''across the stack'' measurement against the in situ and ex situ methods. In this case, the issue is the need for an ''average'' analysis as obtained with the ''across the stack'' method or the ''point'' analysis of an in situ or ex situ method.

The argument for ''average'' is that the total is measured and that an averaged analysis is a true measurement of all the flue gas. The argument for a ''point'' sample is that any air infiltration tends to flow along the sides of the boiler or duct where the air enters. Therefore, a point near the center of the duct is more indicative of burner performance.

Both can be in error since the weighted average percentage can differ from the point sample. An average of more than one point sample or an ''across the duct'' average is often recommended. These may be average analyses, but they do not take into account the specific flows in each area and are, therefore, not truly weighted averages.

Another comparison is made based on the need for ''one-side'' or ''two-side'' duct penetration. The mounting of the measuring device is much simpler and more flexible with regards to to location for a single-side device. The relative location inflexibility of the ''two-side'' analyzer often requires its location in the duct system downstream from air leakage points, such as those of the seals of regenerative air preheaters. Dilution of the flue gas with air changes all volumetric measurements of the flue gas constituents. In addition, locating the measurement further downstream increases the transport dead time of the flue gas from burner to measurement point.

A "method" comparison is also made between the in situ method and the ex situ method. Response time is slightly shorter with the in situ method. Since the measuring cells of these analyzers have a millisecond range response time, in the in situ device an electronic design time constant of 5 to 7 seconds is used to reduce the process noise of the measuremnt. This is not dead time and, therefore, is less innocuous from a control standpoint, though there is also 10 to 15 seconds dead time for the flue gas to flow from the burner to the measuring cell.

In the ex situ device the sample transport produces a slightly longer time constant of dead time. The ex situ device requires a clean, compressed air source for aspirating the sample. The quantity of compressed air is approximately 0.5 cfm. The in situ device uses a similar amount of clean compressed air in the cell as a reference % oxygen.

Other comparisons are based on ease of and cost of measuring cell replacement. The measuring cells of both devices tend to deteriorate with time. Since the cell is outside the duct in the ex situ device, a claim is made that cell replacement is simpler. The cost of the replacement cell is usually less for the ex situ device.

Opposed to the above, the in situ design appears to have some advantage for "dirty" fuels. For these cases, depending on the installations and characteristics of the soot or particulate matter, sample filters and sample probes for the ex situ device (non-existent in the in situ) may gradually become plugged. If this occurs, then periodic blow back may be required or other forms of periodic maintenance may be required.

A comparison can also be made based on the maximum temperature of the flue gases that can be measured. As previously stated, the normal upper limit for the infrared "across the stack" analyzers is in the 600 to 700°F range. For the in situ % oxygen device, the upper limit is the set point of the controlled cell temperature, usually in the 1300 to 1600°F range. For the ex situ analyzer the upper limit is based on the temperature limit of the sample probe used and is in the range of 3000 to 3200°F.

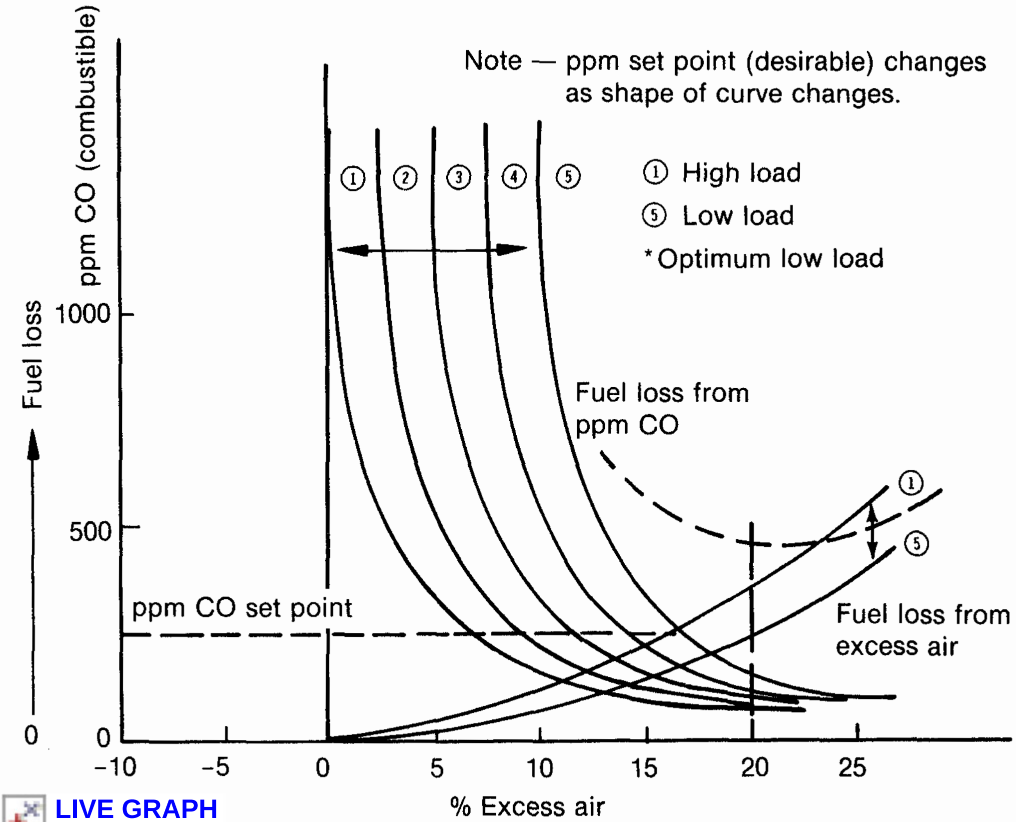
17-4 Flue Gas Analysis vs. Boiler Load

The most desirable set of flue gas analyses for a boiler is determined only by the performance of the fuel burning equipment. Control systems cannot improve the basic performance capability of a boiler or its fuel-burning equipment. A proper control system can, however, operate the boiler very near its "best" performance level for that particular load and other environmental conditions.

In general, all boilers require more excess air for complete combustion of the fuel as boiler load is reduced. Figure 17-4 demonstrates the relationship between fuel losses from excess air and those from combustible gases remaining in the flue gases. As a result of the increase in excess air requirement as boiler firing rate is decreased, the combustible gas curve shifts to the right as boiler load is reduced.

As discussed earlier in this book, flue gas temperature must rise as boiler load increases in order to increase the log mean temperature difference that is necessary for greater heat transfer rates. Because of that fact, the flue gas heat losses shown in Figure 17-4 will be greater at higher loads. In Figure 17-4, the shape of the combustible gas curve remains the same at all loads. This is one possibility. One actual set of boiler tests, as shown in Figure 17-5, shows that not only does the combustibible gas curve shift to the right as load is reduced but also that the shape of the curve changes. Note that this test is for a particular boiler and the curves for particular boilers vary.

The lowest fuel loss occurs when the sum of the two losses (excess air and combustible gas) is at minimum. This point occurs on most boilers when the excess air is such that it produces a combustible gas content of the flue gas at some point in the range of 200 to 2000 ppm. The precise point is a function of several factors that include the fuel burned, the excess air level, the flue gas temperature, and the shape of the combustible gas fuel loss vs. excess air curve.



LIVE GRAPH
Click here to view

Figure 17-4 Excess Air-Combustible Gas Relationships

When reducing excess air, the optimum point is reached when an increment of excess air reduction (e.g., 0.1 percent oxygen) is equal to the loss from the increase in ppm CO. The ppm CO level at which this occurs is at the point where the slope or derivative value of the ppm curve produces that loss. Since the flue gas temperature is higher at higher loads, the gain for 0.1 percent reduction in percent oxygen is greater than at lower loads. This means the optimum ppm CO level is higher at high loads even though the ppm CO curve shape were to remain the same. The change in the shape of the curve as load changes, Figure 17-5, also shifts the CO level for the optimum point, because the slope of the loss value at a particular CO value is different for each of the three curves shown.

The result of an analysis of Figure 17-4 and 17-5 is that both percent oxygen and ppm CO should vary as the load is changed. Comprehensive boiler tests on each particular boiler is necessary to develop the optimum percent oxygen and ppm CO.

The following two basic control application theories are used for flue gas analysis trimming. Additional control approaches are based on combinations of the two.

(1) Percent oxygen (% O_2) control is based on controlling to a fixed percentage of excess air based on a feedback of % O_2. Achieving the minimum fuel loss is accomplished by testing the boiler, determining the capabilities of boiler and fuel-burning equipment, and programming the % O_2 set point to match the test conditions. The desired set point is basically a function of the boiler firing rate. Usually, a small set point increase is added. This excess air "cushion" of approximately 3 to 4% is normally sufficient to allow for variations in the boiler-burner performance from that of the "as tested" condition.

(2) Control from ppm combustible gas was earlier based on the assumption that, for dif-

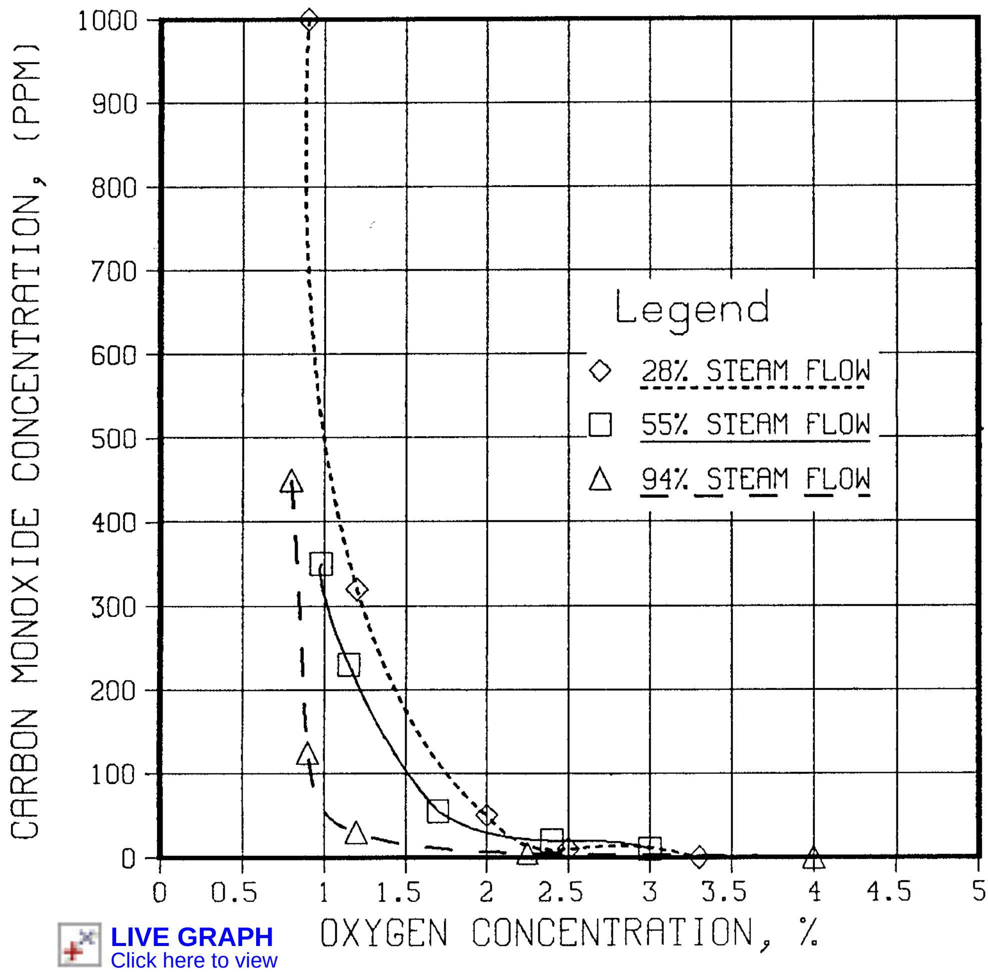


Figure 17-5 Excess Oxygen and Carbon Monoxide Relationship at Various Loads for a Gas-Fired Boiler

ferent boiler loads and "best" economic performance, this measurement is at a relatively constant level. As shown above, this is not correct, and, as Figures 17-4 and 17-5 show, the ppm CO should vary and probably increase as the load is changed. The change will depend on the particular fuel, burner, boiler, and operating conditions. The combustible gas set point for an optimum minimum fuel loss must, therefore, be determined at each load by actual boiler test. The results should then be used to develop a load-programmed set point for % combustible gas, just as is done for % oxygen.

Since any change in boiler-burner performance will automatically produce more or less combustible gas, no excess air cushion is required as with % O_2 control. The result is a theoretical indicated incremental benefit in fuel loss reduction that results from the basic 3 to 4% excess air advantage for this approach.

17-5 PPM CO vs. PPM Total Combustible Gas

Parts per million (ppm) combustible gas and ppm carbon monoxide (ppm CO) are not necessarily synonymous terms. While carbon monoxide is the most widely recognized combustible gas, the particular fuel-burning process and its combustion chemistry progression often produce other combustible residual gases "instead of" or "in addition to" carbon monoxide.

While the fact of other combustible gases has been proven in many boiler tests, there is insufficient data on the subject to state, without any reservation, the precise implication to control design. The result is a lack of total assurance that the governing gas should be carbon monoxide instead of some other combustible gas, such as one of the aldehydes, or vice versa. A simple qualitative test for the presence of aldehydes requires only the bubbling of a small sample of flue gas through an aldehyde detector. The indications are that both CO and aldehydes are present to some degree.

A general guide is that knowledge of the flame characteristics can be useful in helping to define the application. A luminous flame, such as that of coal, fuel oil, and some gaseous fuel flames, indicates that carbon monoxide should probably be the controlling gas. A clear flame, such as that often obtained with rapid combustion of natural gas, may indicate that more aldehydes than carbon monoxide will be produced. Some of the evidence also indicates that, for such flames, preheated combustion air may increase the probability of aldehydes being the predominant combustible gas.

Since a clear or nonluminous flame is obtained in practically all cases only with gas firing, then for all nongaseous fuels carbon monoxide should probably be the controlling residual combustible gas. It follows that for gas firing the correct controlling combustible gas should be carefully determined ahead of time. If this cannot be done or if the aldehyde test confirms its presence, then a total combustible gas sensor (including carbon monoxide and other combustible gases) should be used.

In some cases a measurement of % opacity may be substituted for a measurement of ppm CO. A comparison of the characteristics of % opacity and ppm CO vs. % excess air is shown in Figure 17-6. Since % opacity is a measurement of smoke produced, its use as a control limit may be necessary to avoid noncompliance with environmental regulations. In stoker-fired boiler applications, % opacity may increase when operating above particular excess air levels. This results from the effect of flue gas velocity increasing particulate carryover from the combustion chamber. The effect of increasing opacity as excess air is increased may also occur with oil burners when high excess air results in a white smoke.

17-6 Control Applications Used for Flue Gas Analysis Trimming

Earlier in this section the basic control methods for flue gas analysis trimming control were listed. The specific control logic used in these methods is described in the following paragraphs.

Percent Oxygen as an Individual Control Index

This method has been used for approximately 40 years, so application guidelines are well known and understood. This control application is shown in Figure 17-7. The function generator (a) develops a % oxygen set point signal as a function of boiler load or other index of firing rate. The particular function generated should be based on boiler tests at three or more boiler loads. The boiler operator has the ability, through the manually generated set point bias signal and summer (b), to shift the % O_2 set point curve up or down without changing its shape.

The controller (c) is tuned for a low gain and relatively slow integral response in order to obtain control stability. The low gain results from the relationship between total air flow change and % oxygen change. A change of 0.1 % oxygen is equal to approximately 0.5 % change in toal air flow. If the analyzer has a total range of 0 to 5% oxygen, then at 40% analyzer span (2% O_2) the effect on air flow is 10% of total air flow or of air flow span. If the limiting air flow controller gain is 1.0, at a gain of one for the trimming action, the gain limit of the percent oxygen controller would be [1/(10/40)] or 0.25.

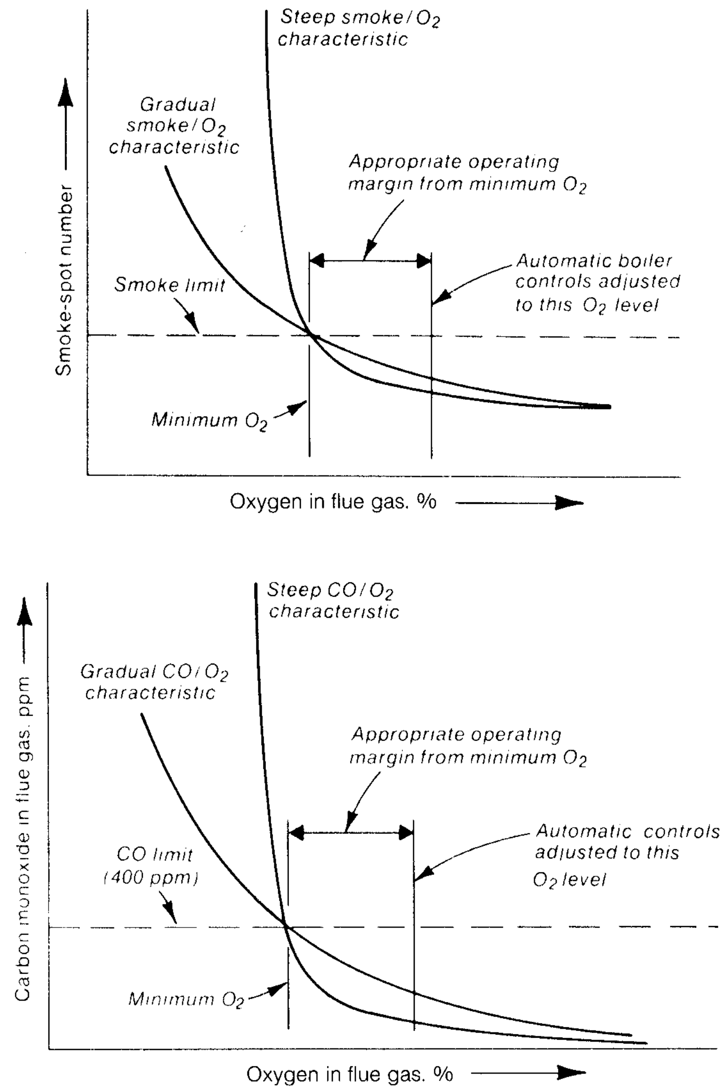


Figure 17-6 Oxygen/CO and Oxygen/Smoke Characteristic Curves

The slow integral tuning of the controller results from the accumulated time constants in the control loop. These consist of controller and control tubing time constants (if pneumatic control), the time constant of the controlled device, the transport time from the control dampers and valves through the combustion process to the analyzer, and the analyzer time constant. Under the best conditions, all of these together will be at least 15 to 20 seconds and probably more. Of these, the transport time of the flue gases is an almost pure delay time constant, which varies as a function of boiler load.

Because of the greater potential for analyzer failure as opposed to the generally used flow measurements, limits (d) are applied at the output of the trimming control. The limits can be applied as limits to the controller output or as a reduced gain at the point the trimming control enters the basic system. If implemented by the "reduced gain" method, manual control has a broader signal range, and the gain at the point of entry to the basic system becomes part of the overall loop gain. This allows an increased gain for the % O_2 controller.

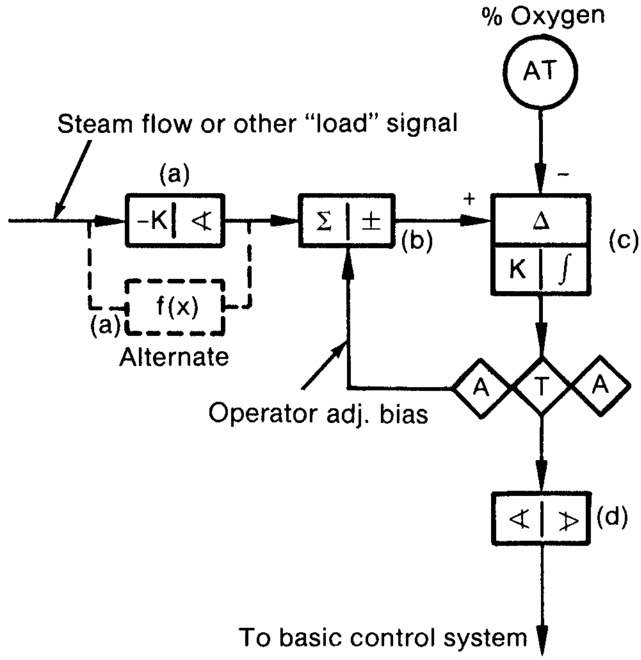


Figure 17-7 Percent Oxygen Trim Control Loop

The function generation for most installations can be a two-slope function generation, as shown in Figure 17-8. On those installations where such a simple method is not sufficient, a more complex function generation based on multiple slopes or a polynomial equation is necessary.

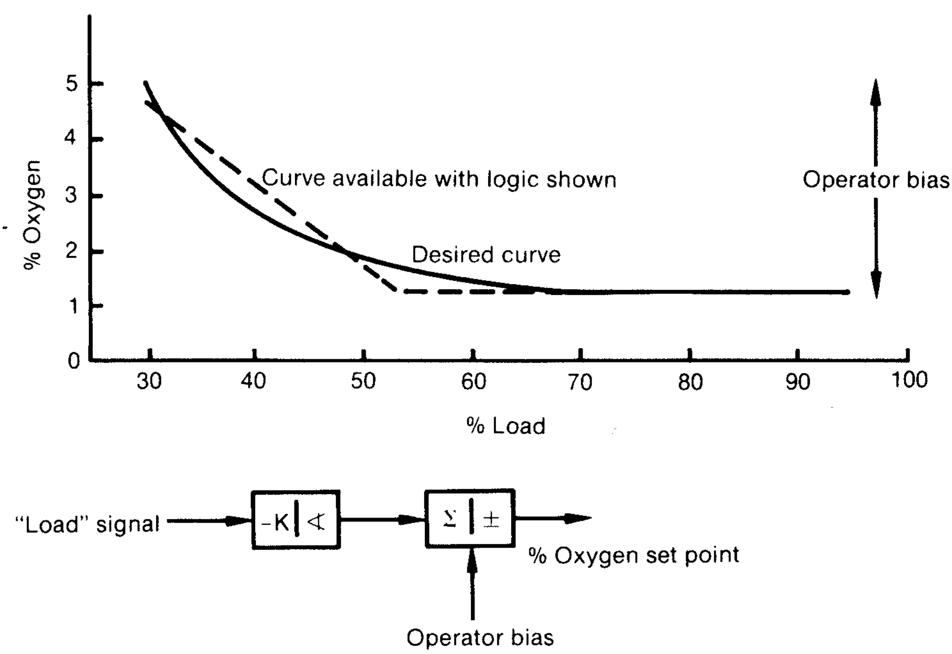


Figure 17-8 Capability of Elementary f(x) Logic

For any trimming control, upon loss of flame or when operating at a boiler load below that of the minimum air flow limit, the controller has no way to control the excess combustion air. If these operating possibilities are not taken into consideration, then unsafe operating conditions can occur when the controller again becomes effective. The controller action in these situations should be carefully analyzed, and security action should be designed into the system. As a minimum, the controller should be automatically transferred to the manual or tracking mode. Additionally, the system may be designed to automatically revert the controller to the automatic mode when boiler operation is again within the range for controller operation and other requirements for automatic operation are satisfied.

Total Combustible or Carbon Monoxide in the PPM Range as an Individual Control Index

Based on the original theory of a constant ppm CO set point at all boiler loads, which has been previously described, the control arrangement is shown by the solid lines of Figure 17-9. The dotted lines show the addition to this control that would be necessary for operation at an optimum minimum fuel loss. The f(x) function would be determined by factors of fuel analysis, excess air, flue gas temperature and actual boiler testing to determine the shape of the ppm CO/excess air curve. Since this control loop should be used only in conjunction with percent CO_2 or percent O_2, the ouput of this control loop is an input to a more comprehensive air flow trimming loop.

In this control application, the control loop uses a simple feedback controller (a) with an adjustable external set point for ppm CO. The feedback measurement (b) of ppm CO or combustible gas usually has a measurement range of 0 to 1000 ppm. For the same reason described

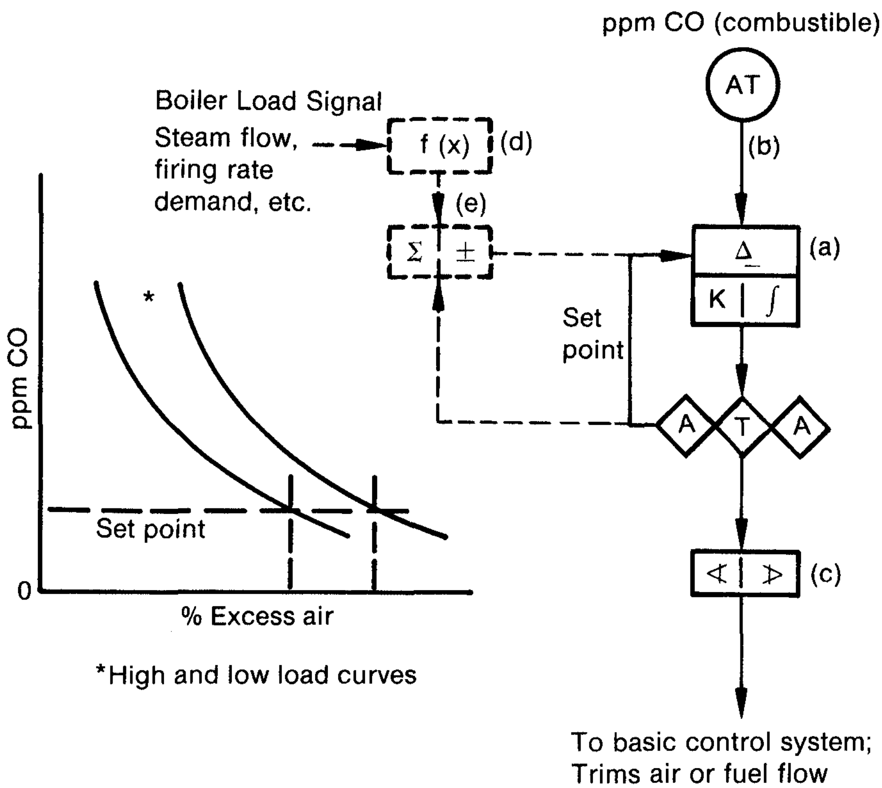


Figure 17-9 PPM CO Trim Control Loop

in the % oxygen control, limits (c) prevent the controller from causing unsafe boiler operating conditions upon a failure of the flue gas analyzing system. Also as in the % oxygen trimming control, the limits can be applied to the controller output or by use of "reduced gain" at the point the trimming control enters the basic control system.

When a control loop as shown is tuned for stable operation, the gain will probably be a very low value with the integral setting also at a low value. A typical relationship is that, when the ppm CO is at set point, a change in ppm CO of 100 ppm (10% of measurement range) is accompanied by a change in % oxygen of approximately 0.1%. Since 0.1% O_2 is equivalent to approximately 0.5% total combustion air, 100 ppm CO change will occur if the total combustion air is changed by approximately 0.5%.

Relating this to controller tuning, an approximate controller gain limit is 0.005/0.1 or 0.05. The values given should be considered as general guidelines only, since the equivalents may be as small as 75 ppm CO or less to 0.1% O_2 or as large as 150 ppm CO or more to 0.1% O_2, depending on boiler load, fuel being fired, and excess air level.

For the integral portion of the controller, the same time constants that exist in the % O_2 control loop must be considered for the ppm CO control loop. Of these, the "pure delay" time constant for transport of the flue gases from the combustion chamber to the measurement point may be slightly greater. This occurs if an "across the stack" analyzer is used, and it must be installed further downstream in the flue gas duct system. A difference in time constant of the measuring analyzer may also affect the relative settings of the intregral (repeats per minute) tuning.

In addition to the security action of limiters (c), the same security actions for operation below the minimum air flow set point or upon flame failure should also be taken with this type of trimming control. As a minimum, the control loop should be reverted to the manual operating mode should either of these situations occur.

Other situations may occur that call for blocking control action from a ppm CO or ppm total combustible controller. If smoke is excessive, the controller must be inhibited from further reducing the excess air. When using ppm CO control, the presence of hydrocarbon combustible gas in excess of that of carbon monoxide should prevent any further reduction in excess air. For stoker-fired boilers, the increased furnace temperature as excess air is reduced may cause excessive grate temperatures.

Trimming Control Based on a Combination of % Oxygen and PPM CO

Early installations of ppm CO trim control (beginning in the early 1970s) demonstrated some of weaknesses of the above approach. In addition to the points above, CO or combustible gas formation may be affected by rapid changes in firing rate or excess air levels. Using the above control, poor burner performance can cause an excessive air flow change until enough combustion air has been added to dilute the flue gas back to the ppm CO set point. With the control set point at the minimum limit of % excess air, relatively small analyzer errors may drive the boiler operation into an excessive unburned fuel situation.

If boiler slagging or some other excess air limiting factor should be the limiting factor, then the ppm CO/% oxygen curve at the indicated set point is relatively flat. Such a relatively flat curve would greatly reduce control effectiveness.

Without other means of monitoring excess air level, improper operation may continue for an unlimited period. In addition, the potential benefit of a few percent less excess air that results from basing the control on ppm CO is lost. To avoid inadvertent fuel losses from such occurrences, the level of excess air in the flue gas should always be monitored by measuring % oxygen.

From measurement to combining % oxygen with ppm CO in a trimming control loop is a simple step. Two relatively simple control arrangements—of the many used or suggested—of a flue gas analysis trimming control loop based on both % oxygen and ppm CO are shown in Figures 17-10 and 17-11.

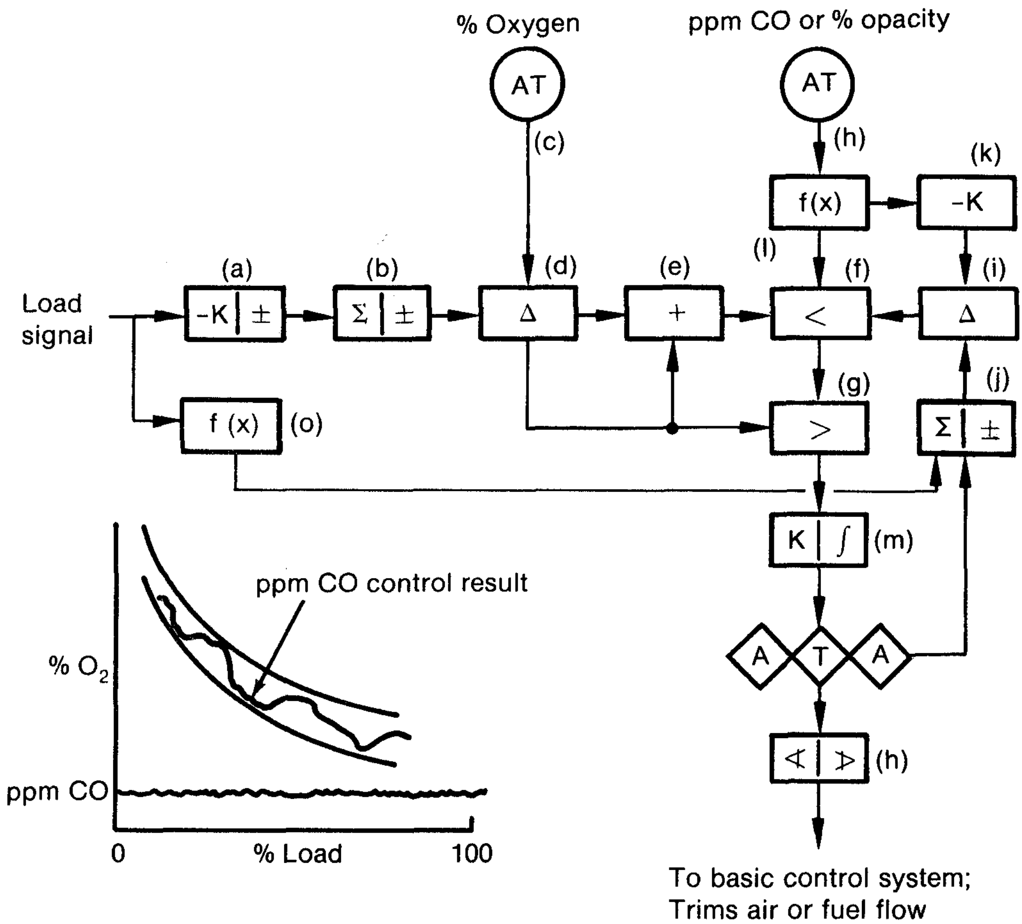


Figure 17-10 Trim Control, % Oxygen plus PPM CO or % Oxygen plus Opacity

In operation, the control normally operates from ppm CO within a band of % O_2. At either a high % O_2 or low % O_2 limit, the control action switches to a % oxygen control. Since the desired excess air is a function of boiler firing rate, these % O_2 limits should be shifted as a function of firing rate. In this control arrangement, the difference between the high and low % O_2 limits is approximately 1% oxygen (the approximate band equivalent to 1000 ppm CO). The low limit should never be below approximately 0.3% O_2 to allow the low % O_2 control loop to function.

The control arrangement shown in Figure 17-10 operates as follows. For any change in either of the two measured variables there will be a change in the other variable. The approach shown is to match the effects of % O_2 and ppm CO so that a single controller can be used. The boiler firing rate vs. % O_2 relationship is developed by the function generation of item (a). This function is the low % O_2 limit and is determined by boiler testing. This function is balanced against the % O_2 analyzer (c) signal to produce an error signal in the difference logic (d). The width of the % O_2 band is determined by the positive bias (e), which produces the upper % O_2 limit. The value of bias (e) is determined by the range of the % oxygen signal and the desired width of the band between the high and low % oxygen set points. If the desired band is 2 percent oxygen and the % oxygen signal range is 0 to 5%, the band would be 40 percent and the positive bias (e) would be 40 percent.

Changes in ppm CO are in the opposite direction from those of % O_2. The ppm CO signal from analyzer (h) is inverted and matched in range to the % O_2 signal in the negative propor-

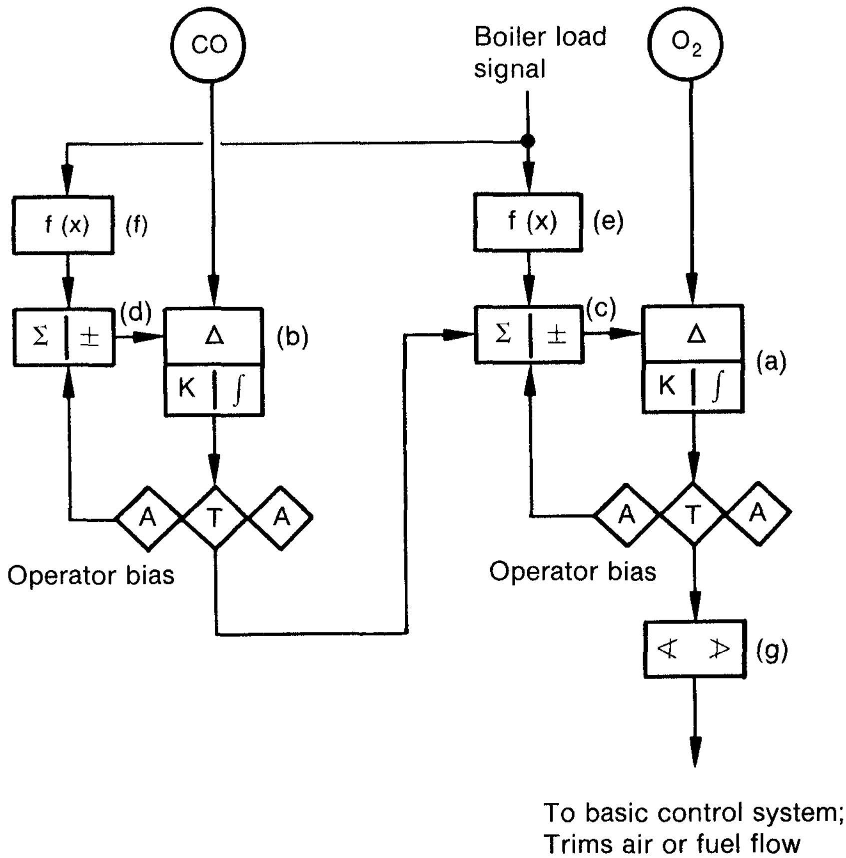


Figure 17-11 Trim Control from % Oxygen plus PPM CO

tional logic (k). An error signal of ppm CO relative to its set point is developed in the difference logic (i). Since ppm CO is highly nonlinear, the ppm CO signal is linearized in function generator (l). The desired ppm CO set point is determined by function generator (o) and potentially biased by the operator in summer (j). The result is a linear ppm CO error signal from the difference logic (i) to match the linear % O_2 error signal.

From this, three potential error signals are produced, each of which are essentially linear and have approximately the same control response and gain characteristics. Bumpless automatic selection of the desired error signal is accomplished by the high and low selector logic, (f) and (g).

The selected error signal enters the controller (m), which then produces the trimming control signal. Using a single controller avoids any control windup problems when the control action is switched from % oxygen control to ppm CO control and vice versa. As in the other trimming control arrangements, the control action should be limited as shown in the high/low limits of limiter (n). Alternately, the limiting approach of "reduced gain" where the signal enters the basic control system can be used. Tuning of the controller is the same as that used with % O_2 control.

The controller can be forced to operate as a % O_2 controller by arbitrarily adjusting the ppm CO set point to a high or low value. Under steady-state operating conditions, the % O_2 control is tuned first. The set point of ppm CO is then adjusted to the desired value, and the operation as a ppm CO controller is checked. If the system has been properly aligned, the only further tuning may be a small decrease in the integral (repeats per minute) setting. Control should be automatically switched to the manual mode when the firing rate is below air flow minimum or upon flame failure.

Other arrangements of this trimming control make use of separate controllers for % O_2 and ppm CO. The arrangement shown in Figure 17-11 uses cascade control from the ppm CO control to trim the % O2 control set point.

As indicated previously, the ppm CO set point should be load-programmed. This is accomplished in function generator (f). The operator has the opportunity to bias the curve up or down in the summer (d). Both inputs to summer (d) have a gain of 1.0. The % oxygen set point is load-programmed in function generator (e). The output of function generator (e) acts as a feedforward signal to the % oxygen set point in summer (c). Summer (c) also provides the opportunity for operator biasing and for trimming the % oxygen set point signal through action of the ppm CO controller (b). All three inputs to summer (d) have a gain of 1.0.

Since the time constants for % oxygen and ppm CO are nearly equal in this cascade loop, the primary ppm CO control must be detuned to obtain overall control stability. The secondary controller is always % oxygen control and has the tuning constraints described earlier.

Another control arrangement uses switching between the two controllers. In designing such control loops, control windup, switching interlock logic, and the necessary tracking action to allow bumpless switching action must be considered.

Trimming Control Based on a Combination of % Oxygen and % Opacity

The previous arrangements should be inhibited from reducing excess air in the presence of excessive smoke (high opacity). All fuels except gas have a smoke potential in an excess air condition. In some installations and or some operating conditions, smoke can become excessive by legal limits while the ppm CO is below the desired value. If it can be determined that this is always true for a particular installation, then % opacity can substitute for ppm CO in the control arrangement in Figure 17-10. This is based on the marked similarity of the curves of ppm CO vs. excess air and % opacity vs. excess air shown in Figure 17-6.

This application has been used particularly for spreader stoker applications. In these applications the low % O_2 limit is usually that required to prevent excessive furnace or grate temperatures. The high % O_2 limit is that which produces a minimum opacity level. Additional excess air may cause particulate carryover due to flue gas velocity. Except for these variations, the calibration, tuning, and alignment procedure of this control arrangement is the same as that using ppm CO as the feedback signal.

17-7 Limiting Factors in Reducing Excess Air

The preceding parts of this section discuss the use of flue gas analysis control as a means of improving boiler efficiency. The boiler efficiency is not always the issue, and the flue gas analysis is not always the limiting factor. It is possible to improve boiler efficiency while causing the performance of the overall system, which includes the boiler, to deteriorate.

If a boiler is used to furnish steam to an electric power generation turbine, the gain in boiler efficiency may be at the expense of turbine performance. It is well known that such installations use superheated steam in practically all fossil fuel installations. It is also well known that reducing the excess air of a boiler may cause steam temperature to be reduced unless some steam temperature control mechanism is in the active control range (refer to Section 11). A reduction in steam temperature reduces the thermodynamically available energy for conversion to power. It has been shown that up to the point of excessive combustible gas formation, if excess air reduction reduces the steam temperature, the overall result is a decrease in boiler-turbine system performance.

Reduction in excess air also results in increased furnace temperature. For coal-fired boilers, this can increase furnace slagging and clinker formation (fusing of the ash). The result is difficulty in ash removal. Another problem related to furnace temperature is excessive grate temperature of stoker-fired boilers. An excess air level above that for minimum fuel consumption may be necessary for installations with this problem.

The percentage of carbon in the refuse is another factor to be considered when the boiler is fired by coal, wood, or other solid fuel. Representatives of boiler manufacturers have stated that they can find no specific relationship between the level of combustible gas in the flue gas and the % carbon in the refuse.

All such factors should be carefully considered before the decision to apply control equipment based on analysis of the flue gases. The control application method, its economic evaluation, and its implementation should be based on knowledge of the particular limiting factors that may apply.

Section 18
Fluid Fuel Burners for Gas, Oil, and Coal

All burners must perform all five of the following functions:

(1) Deliver fuel to the combustion chamber
(2) Deliver air to the combustion chamber
(3) Mix the fuel and air
(4) Ignite and burn the mixture
(5) Remove the products of combustion

In the case of coal or oil, delivering fuel to the combustion chamber also includes preparing the fuel so that it will burn. Section 5 discussed how gas and oil were delivered to the burning system and how oil and coal are prepared for burning. Figure 5-8 shows that the fuel must be gasified, and this is part of the first function. Figure 5-8 also shows that turbulence for thoroughly mixing the fuel and air is one of the three ''T''s of combustion.

The source of the energy that produces this turbulent mixture is the kinetic energy of the fuel and air streams. Restrictions in each of these streams at the burner convert potential energy of the fuel and air, due to their pressures, into kinetic energy. This is done by restricting the flow areas and thus increasing the fuel and air velocities as they travel through the burner. In reducing fuel and air flow rates as the burner is turned down, this ''mixing energy'' is sharply reduced.

The result is that there is a normal requirement for higher excess air as the flow rate of a given burner is reduced. Figure 18-1 shows the typical relationship between excess air and flow rate for gaseous (including pulverized coal) or liquid fuel-burning burning systems. Combustion control systems must automatically compensate for this relationship by shifting the fuel-air ratio as the burners are turned down.

18-1 Burners for Gaseous Fuel

Figure 5-1 shows the simplicity with which gaseous fuel is automatically delivered to the plant as it is used. The characteristics of the burner design must be adjusted to match the characteristics of the fuel delivered to the burner by the firing rate control.

Pressure at the burner is one means of classifying gas burners. This classification can be described as follows:

(1) Low pressure—2 to 8 ounces per square inch
(2) Intermediate pressure—8 ounces per square inch to 2 psig
(3) High pressure—2 to 50 psig

Burners are selected according to the particular gaseous fuel and the pressure available or to fit the flame characteristics desired. The gas issues from the burner through small orifices that vary in size and capacity depending on the maximum pressure provided. The higher pressures usually have a greater turndown capability. Turndown is the firing rate ratio over which the firing rate can be reduced without flame instability.

Combustion air may enter the burner through atmospheric pressure, which is higher than the suction at the burner. Such a burner is called an atmospheric burner. The suction is either developed by natural draft or by inspiration, in which a relatively high velocity jet of gas creates suction as it enters the combustion chamber. Mechanical draft, created by a forced draft fan or an induced draft fan or both, is the other method of delivering air to the burner. In order to control the fuel-air ratio accurately and for burner safety, all the combustion air should enter through or around the burner.

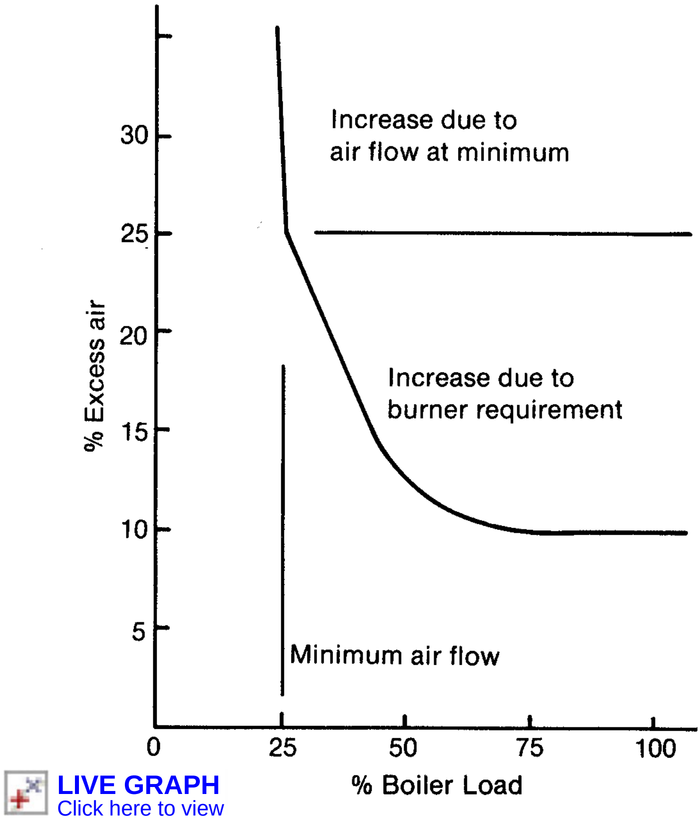


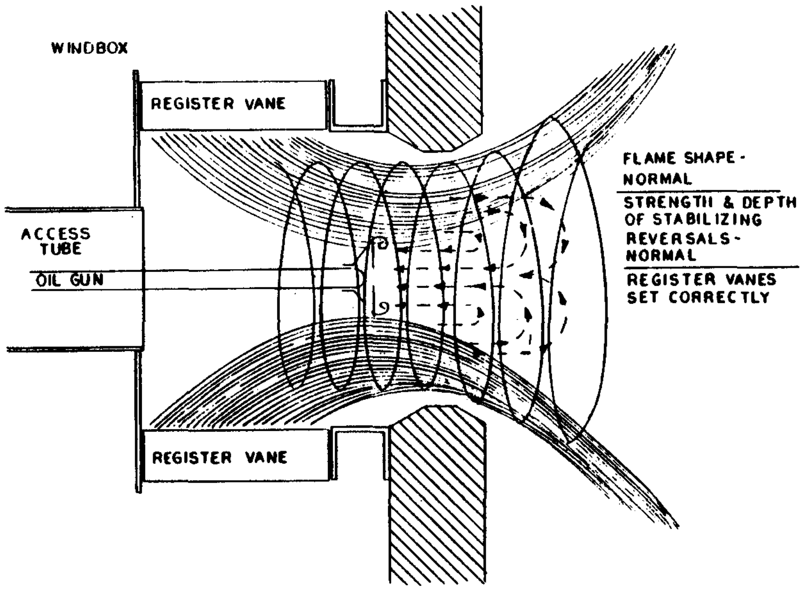
Figure 18-1 Excess Air vs. Boiler Load, Typical Curve

The most common burners in use in boilers are those that are sometimes referred to as nozzle mix burners. The fuel issues from a small orifice and is injected into, and thoroughly mixed with, a whirling vortex of air. A counterrotation whirl is often imparted to the issuing gas in order to provide greater turbulence. This air vortex is shown in Figure 18-2. The basic idea is to mix the fuel and air thoroughly so that complete combustion will take place with a minimum of excess air. Figure 18-2 indicates oil-firing, but the same air register mechanism is used for both gas and oil.

A small pilot burner is usually used to obtain ignition. This pilot can be continuously lit, or it can be ignited each time the burner is started, and shut down after the burner is lit. Ignition characteristics often are affected by the type and shape of the flame and whether the fuel and air are mixed immediately or in stages. Staged combustion is obtained by initiating the flame with primary air and later admitting the rest of the air that is necessary to complete the combustion. Staged combustion produces a longer and lower-temperature flame and fewer nitrous oxides.

The final function of the burner is to remove the products of combustion. This is accomplished by their replacement with new mixtures of fuel and air that are forced into the combustion chamber by fans and fuel pressure.

In Figure 18-3(A), a cross section of a ring-type gas burner is shown. This is an older but common type of high-pressure gas burner for boiler use. The normal pressure operating range is from approximately 1 to 10 psig. As the air passes through the air registers, the velocity is increased and a whirl is imparted to the air, which then is forced toward the burner throat. The gas issues from the small holes shown on the inside of the hollow gas ring and is injected



Side elevation of rotating flow register

Figure 18-2 Burner Air Flow Vortex

(From Forney Engineering Co.)

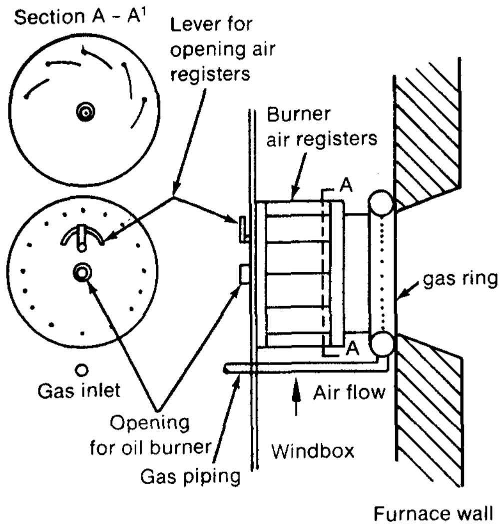
by the fuel pressure perpendicular or nearly so to the air stream. When such burners are properly adjusted, they can be operated at a minimum pressure of approximately 1 psig while maintaining a stable flame. Because of the relatively small gas orifices, typically 1/8 inch, dirt or foreign substances in the gas stream may tend to plug the burners. The boiler must be shut down to clean such burners.

Figure 18-3(B) depicts a "gun-type" gas burner. In this burner the air register, with its function of imparting a whirl to the combustion air, is similar to that of the ring burner. The gas, however, is injected into the air stream from the center instead of the outer periphery. Because space is more limited on the tip of the smaller diameter gun, the gas orifices are larger and fewer in number than on the ring burner. An advantage of this burner over the ring burner is that the gas orifices are larger and need less cleaning; they can, however, be cleaned by removing the gun while the boiler is in operation. When properly adjusted, burners of this type can be operated at pressures in excess of 20 psig and with a minimum-pressure stable flame at approximately 1 psig.

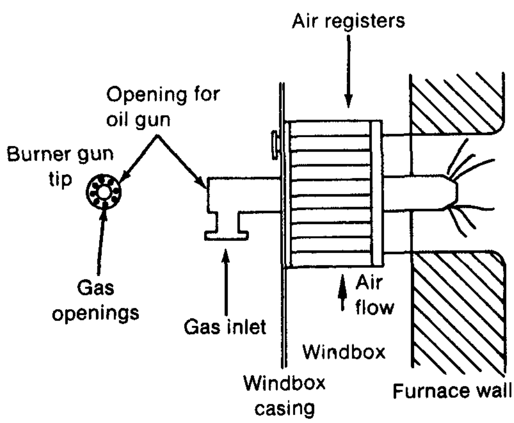
Figure 18-4 represents a spud-type burner. In this burner the gas ring is external, with a number of smaller guns or spuds connected to the ring. This burner is a design for obtaining the claimed benefits of both the ring and the gun burners. The spuds can be removed for cleaning while the boiler is in operation, and the gas orifices are greater in number and smaller for better dispersal of the gas into the air stream.

The burners above are all of the nozzle mix type. Other gas burners for boilers are designed so that the gas and air issue from the burner in somewhat parallel streams. The fuel-air mixing takes place in the furnace instead of at the burner.

Figure 18-5 shows a burner of this type. Although only oil and coal nozzles are shown, gas nozzles can be added to or can replace the oil and coal nozzles that are shown. As shown in Figure 18-5 these burners are mounted at the corners of a furnace and directed so that the



(A) Ring-Type Gas Burners



(B) Gun-Type Gas Burner

Figure 18-3 Gas Burners

gas and air streams are tangent to a vortex or "fireball" in the center of the furnace. Since the fuel and air mix gradually, the combustion process is slower and at lower temperature. The result is that such burners naturally produce lower NOx (nitrous oxides) in the flue gases.

The turndown of a gas burner is related to its maximum pressure and the minimum pressure that will support a stable flame. The turndown ratio can be calculated if the minimum and maximum pressures are known. If the maximum pressure produces flow in the subcritical (below Mach 1.0) range, the burner can be considered as a flow orifice with flow varying in accordance with the square root of the differential pressure. The burner pressure is the upstream pressure, and the downstream pressure is atmospheric pressure. The pressure at the burner changes the flowing density, which affects the flow in accordance with the square root of the absolute pressure ratio. If 14.7 psia is considered the base, the approximate turndown

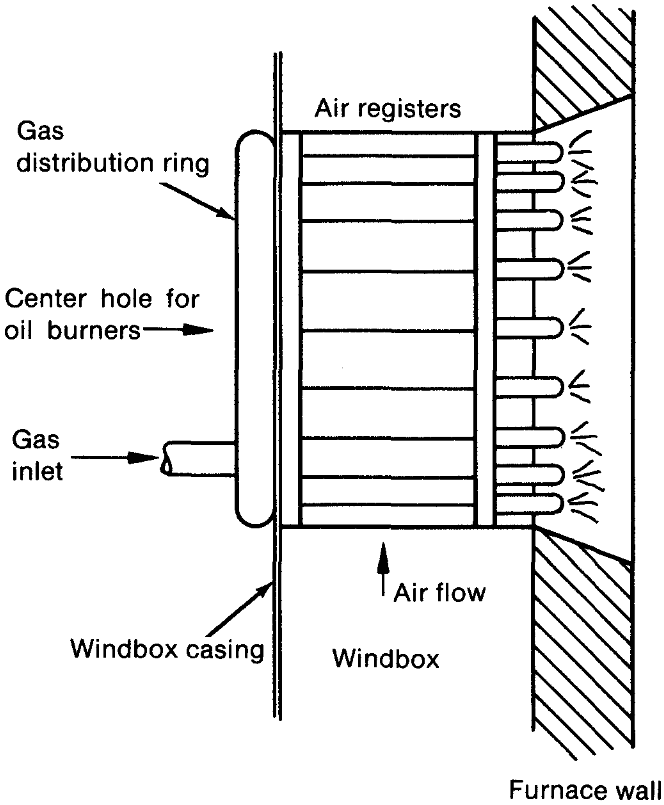


Figure 18-4 Spud-Type Gas Burner

can be calculated in accordance with Figure 18-6(A). Note that a very slight reduction in the minimum pressure significantly increases the turndown ratio.

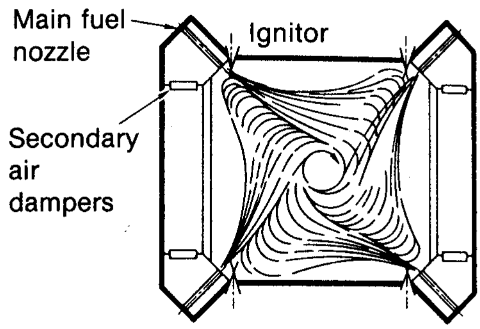
For higher pressures producing flow in the critical velocity range (greater than Mach 1.0), the flow varies as a relationship with a combination of differential pressure and with the burner absolute pressure. Figure 18-6(B) is an example of the calculation of burner turndown for a maximum burner pressure of 25 psig. In the critical velocity range, flow change is directly proportional to change in the absolute inlet pressure. For this burner, above approximately 70 percent of maximum flow the velocity is critical, and below this point it is subcritical.

For all the above burners, the firing rate is adjusted by regulating the fuel and combustion air in parallel. The control device for the fuel is a control valve in the fuel supply line as shown in Figure 5-1. All of the above burners also have openings for insertion of a fuel oil gun so that the burner may be used as a combination oil and gas burner.

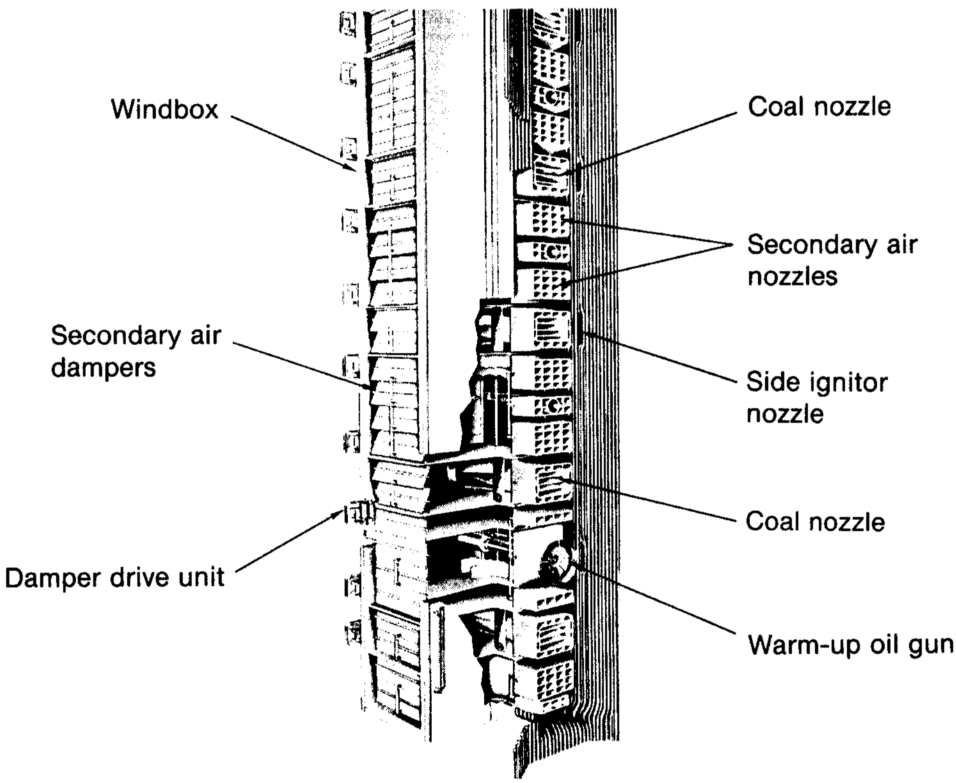
18-2 Pulverized Coal Burners

Though coal is a solid fuel, in pulverized coal systems where coal is finely ground and transported to the furnace in a primary air stream, some burners are similar to the gun-type gas burners. They have similar air registers, often with provision for auxiliary oil or gas fuel to be burned in the same burner. Figure 18-7 is a drawing of such a combination burner.

Figure 18-8 shows the detail of two modern pulverized coal burners. One of these is a staged combustion burner with secondary and tertiary air registers that allow the shaping of the flame and introduction and mixing of the fuel and air in stages. The benefit is a reduction of NOx in the flue gases.



(A) Tangential firing pattern



(B) Tilting-type burner

Figure 18-5 Tilting Tangential-Type Burner for Gas, Oil, and Coal

(From *Fossil Power Systems*, © Combustion Engineering Inc.)

- Depends on burner design for maximum pressure to carry the load and minimum for stable fire.
- Consider burner an orifice — basic $\sqrt{\ }$ relation pressure (pressure drop to flow) but density changes considerably.
- Plot burner flow curve min to max

Example

ASSUME 1 psi minimum, 8 psi maximum — 100%

$$7 \text{ psi flow} = \sqrt{\frac{21.7}{22.7}} \times \sqrt{\frac{7}{8}} \times 1.00 = 0.91455$$

$$6 \text{ psi flow} = \sqrt{\frac{20.7}{22.7}} \times \sqrt{\frac{6}{8}} \times 1.00 = 0.827$$

continuing down to

$$1 \text{ psi flow} = \sqrt{\frac{15.7}{22.7}} \times \sqrt{\frac{1}{8}} \times 1.00 = 0.288$$

Turndown $= 1/0.288$ or $\boxed{3.47 \text{ to } 1}$

If minimum stable fire is reduced by 0.5 psi to 0.5 psi,

$$\text{then } 0.5 \text{ psig flow} = \sqrt{\frac{15.2}{22.7}} \times \sqrt{\frac{0.5}{8}} \times 1.00 = 0.2046$$

Turndown $= 1/0.2046$ or $\boxed{4.89 \text{ to } 1}$

(A) Subcritical
Figure 18-6 Turndown of Gas Burners

A low NOx burner from another manufacturer is shown in Figure 18-9. Note that this burner is also provided with dual air registers for use in mixing the fuel and air in stages.

Figure 18-5, which was shown previously in connection with gas burners, shows a tangential burner with nozzles arranged for coal and with provision for a warm-up oil gun. Secondary air, which is added to the primary air to complete the combustion process, issues from separate air nozzles in streams that are parallel to the coal/primary air streams.

In both of these types of burners the pulverized coal supply to the boiler is changed by adjusting the speed of the coal feeder that admits coal to the pulverizer and by changing the primary air flow rate. A change in primary air flow immediately affects the flow of the coal

- Basic — Follow subcritical curve to 0.47 of absolute inlet pressure, then straight line to maximum pressure.

Example

ASSUME Atmosphere = 14.7 psia,
pressure drop to atmosphere = 0.47 of absolute pressure at X psig.

$$0.47\,(X + 14.7) = X$$
$$0.47X + 6.909 = X$$
$$0.53X = 6.909$$
$$X = 13.03 \text{ psi for critical velocity}$$

ASSUME 25 psig maximum, 1 psig minimum.

$$25 \text{ psig} = 100\% \text{ or } 1.0$$
$$13 \text{ psig} = 27.7/39.7 = 0.698$$
$$1 \text{ psig} = \sqrt{\frac{15.7}{27.7}} \times \sqrt{\frac{1}{13}} \times 0.698 = 0.145$$

Turndown = 1/0.145 or $\boxed{6.90 \text{ to } 1}$

- If minimum can safely be reduced to 0.5 psig,

$$0.5 \text{ psig} = \sqrt{\frac{15.2}{27.7}} \times \sqrt{\frac{0.5}{13}} \times 0.698 = 0.1014$$

Turndown = 1/0.1014 or $\boxed{9.86 \text{ to } 1}$

(B) Critical Velocity
Figure 18-6 continued

that is already ground and stored in the pulverizer. Refer to Figure 5-7 for the coal feeding and furnace supply arrangement.

18-3 Fuel Oil Burners

Fuel oil burners for boilers usually use the same air register mechanisms as those used for gas burners. An oil gun is inserted into the center of the burner assembly. The oil gun atomizes and sprays the oil outward into the whirling combustion air stream. Though some gas burners

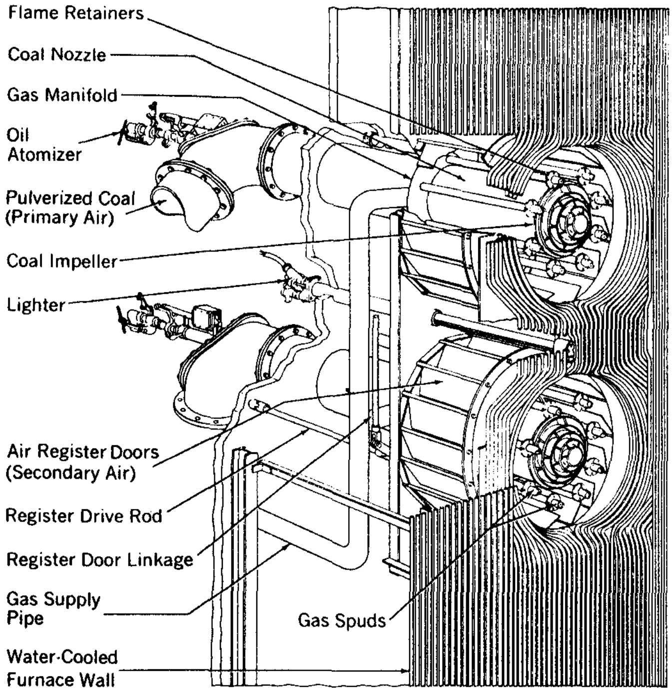


Figure 18-7 Pulverized Coal Combination Burner

(From *Steam, Its Generation and Use*, © Babcock and Wilcox)

do not require what are known as "diffusers" or "impellers," these devices are necessary for oil and pulverized coal burners and larger gas burners (see Figure 18-10).

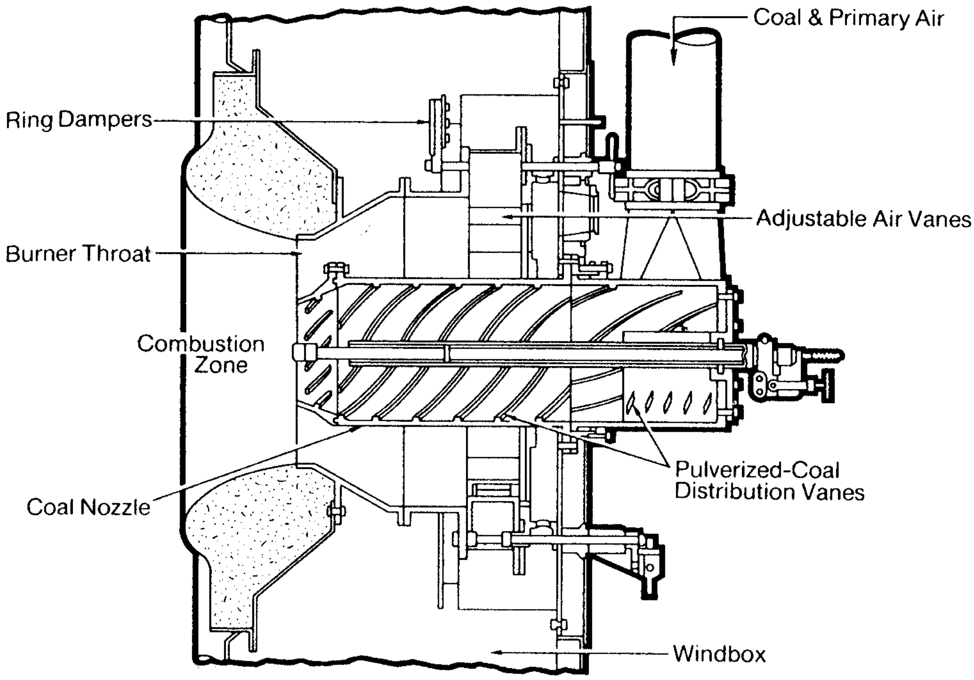
The purpose of the diffuser (impeller) is to produce more stable ignition by splitting the combustion air stream into primary and secondary portions. A portion of the air (primary air) passes through the diffuser slots and creates a fuel-rich zone as it (primary air) mixes with the fuel issuing from the burner gun. The remainder of the combustion air flows around the outer edge of the impeller and is added as secondary air to the basic fuel/primary air mixture that is ignited.

The oil guns for different burners are similar but may differ in the atomizing method. The classifications are:

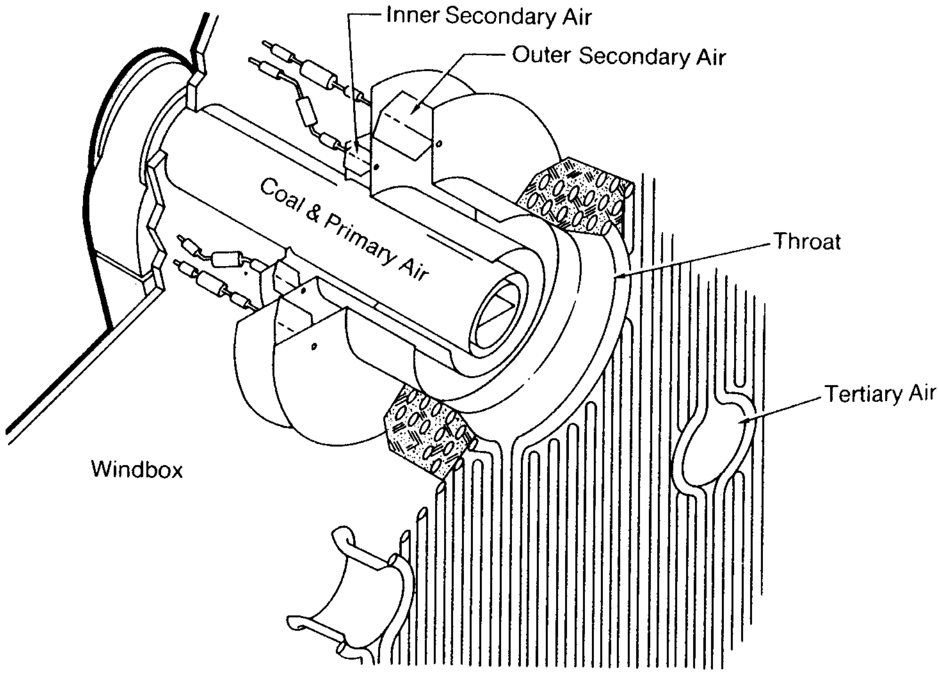
(1) steam or compressed air atomizing,
(2) mechanical pressure atomizing, and
(3) mechanical return flow pressure atomizing.

The steam or compressed air atomizing burner uses the mechanical energy of steam or compressed air to atomize the oil and is not dependent upon the thermal content of the atomizing fluid. In view of this, air and steam may be substituted for each other in the same burner. As shown in Figure 18-11, the steam and oil arrive at the burner tip through concentric tubes. The jets of oil and steam (or air) merge at the tip and the oil stream is atomized. The oil supply is regulated by a control valve in the supply line. Atomizing steam or air is regulated with a differential pressure-regulating valve that controls the steam or air at a given set pressure above the burner oil pressure.

A typical burner gun tip for steam or compressed air atomization of fuel oil is shown in Figure 18-12.



(A) Horizontal Firing Pulverized Coal Burner



(B) Staged Combustion Pulverized Coal Burner

Figure 18-8 Pulverized Coal Burners

(From *Fossil Power Systems*, © Combustion Engineering Co. Inc.)

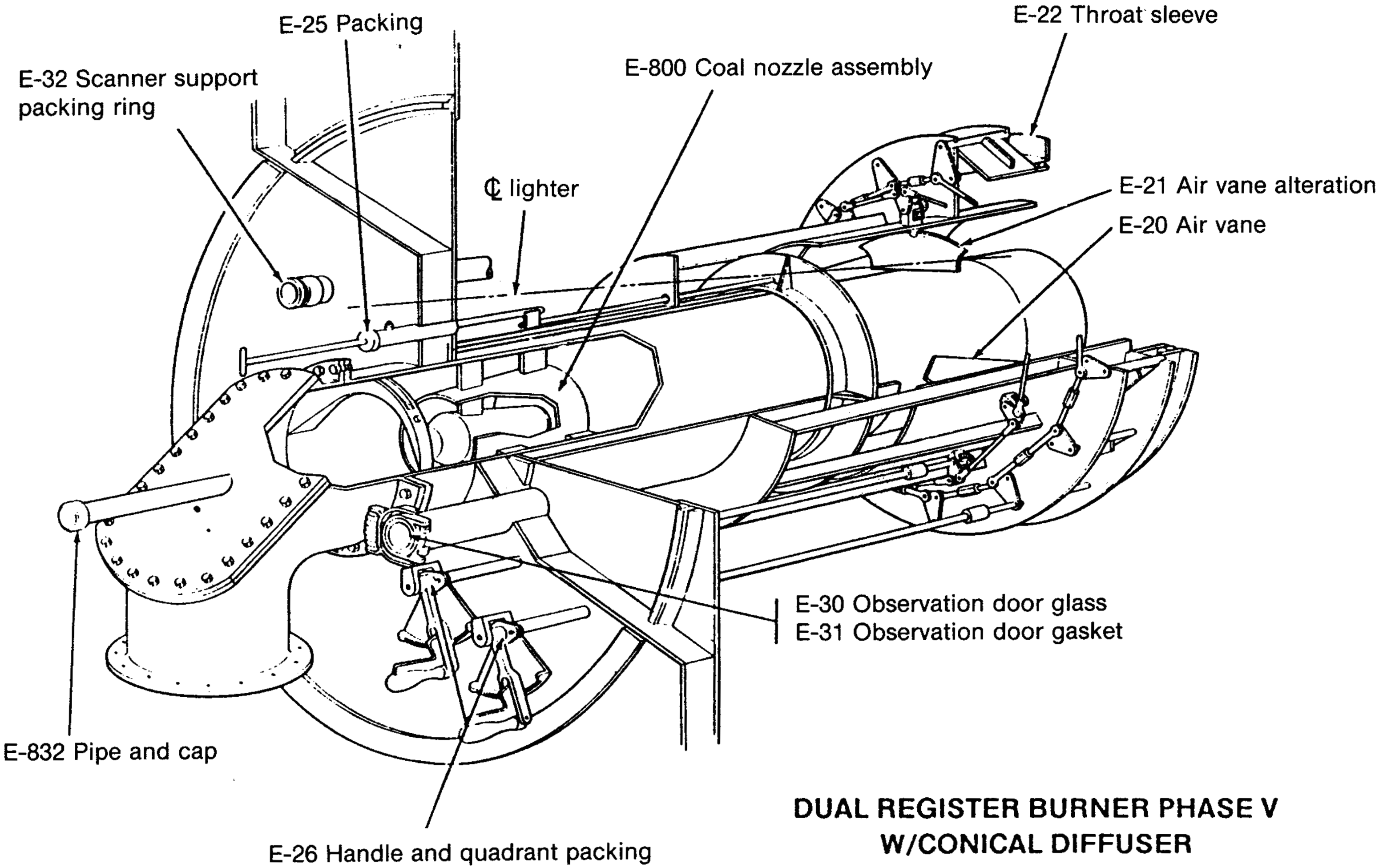


Figure 18-9 Low NOx Pulverized Coal Burner

(From Babcock and Wilcox Co.)

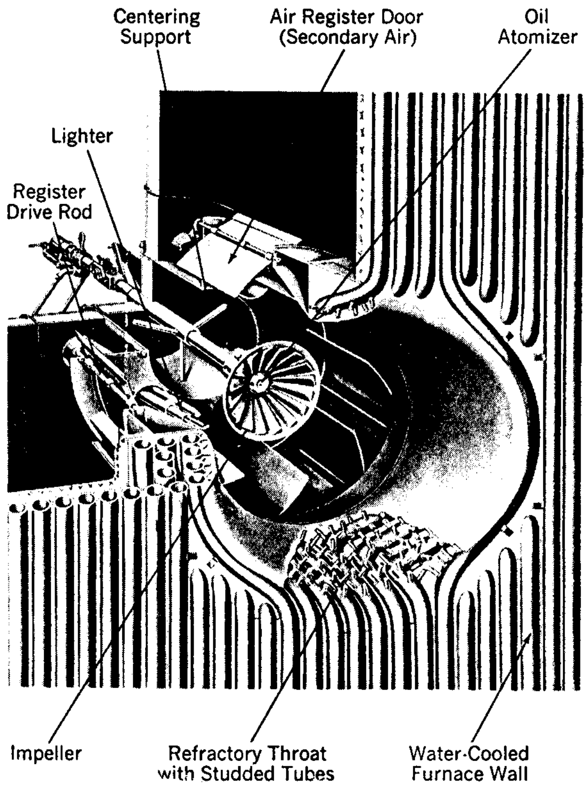


Figure 18-10 Impeller (Diffuser) for Gas, Oil, and Pulverized Coal Burners
(From *Steam, Its Generation and Use*, © Babcock & Wilcox Co.)

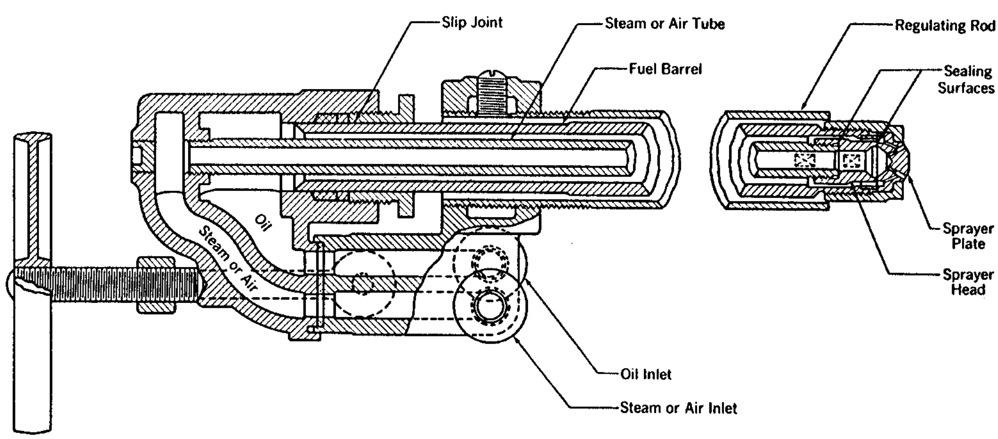


Figure 18-11 Steam (or Air) Oil Atomizer Assembly
(From *Steam, Its Generation and Use*, © Babcock and Wilcox Co.)

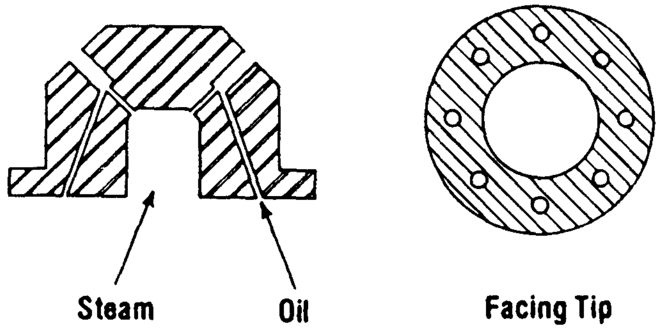


Figure 18-12 Steam (or Air) Atomizing Y Jet

(From *Improving Boiler Efficiency*, Kansas State University)

Caution: In some steam atomizing burners the steam and oil mix inside the gun. The use of differential pressure control for atomizing steam on such a burner may make the flame unstable. In those cases it may be necessary to control the steam in parallel with the fuel.

In the mechanical pressure atomizing burner, the gun may have a single oil tube to the tip or it may have a concentric return flow line. The tip orifice includes tangential slots that cause the oil to whirl as it sprays from the burner and to be atomized by the potential energy of the oil supply pressure. When the burner is in service, any return oil passage is closed, and oil supply is regulated by a control valve in the supply line. A tip for this type of burner is shown in Figure 18-13.

The return flow pressure atomizing burner can operate at the highest supply pressure of all oil burners. It utilizes concentric tubes in the gun, with the supply oil flow in the outer tube and the return flow in the center tube. If the return flow is unrestricted, the oil flows to a whirling chamber near the tip, whirls, and by centrifugal force enters the return flow passage instead of being sprayed into the furnace. Restricting the return flow with a control valve causes a greater or lesser flow to be sprayed from the tip into the furnace.

Since oil flows in both supply and return lines, the oil burned can be determined only by measuring both flows and subtracting the return from the supply flow. It is necessary to carefully match the flow calibrations of these meters in order to avoid large errors in "oil burned" at low furnace input rates. Figure 18-14 shows a cross section of a typical burner tip for a

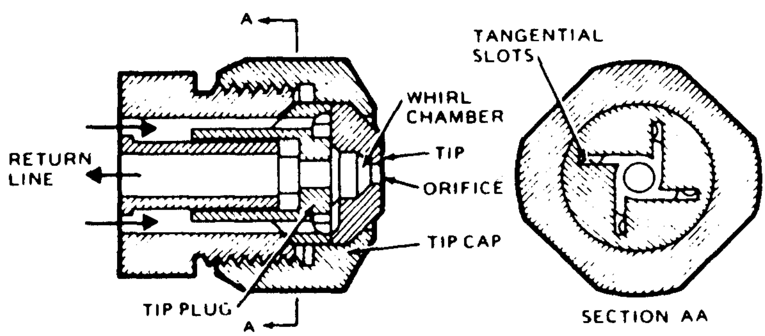
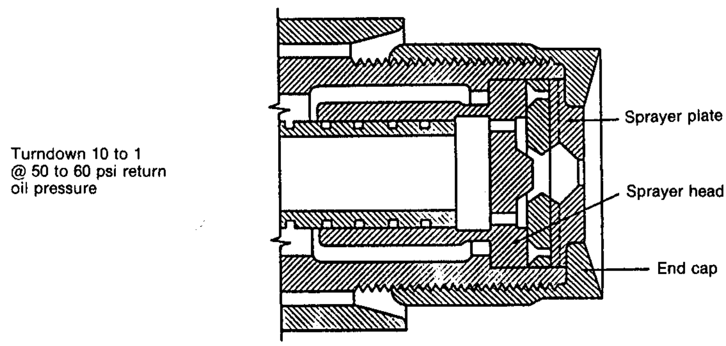
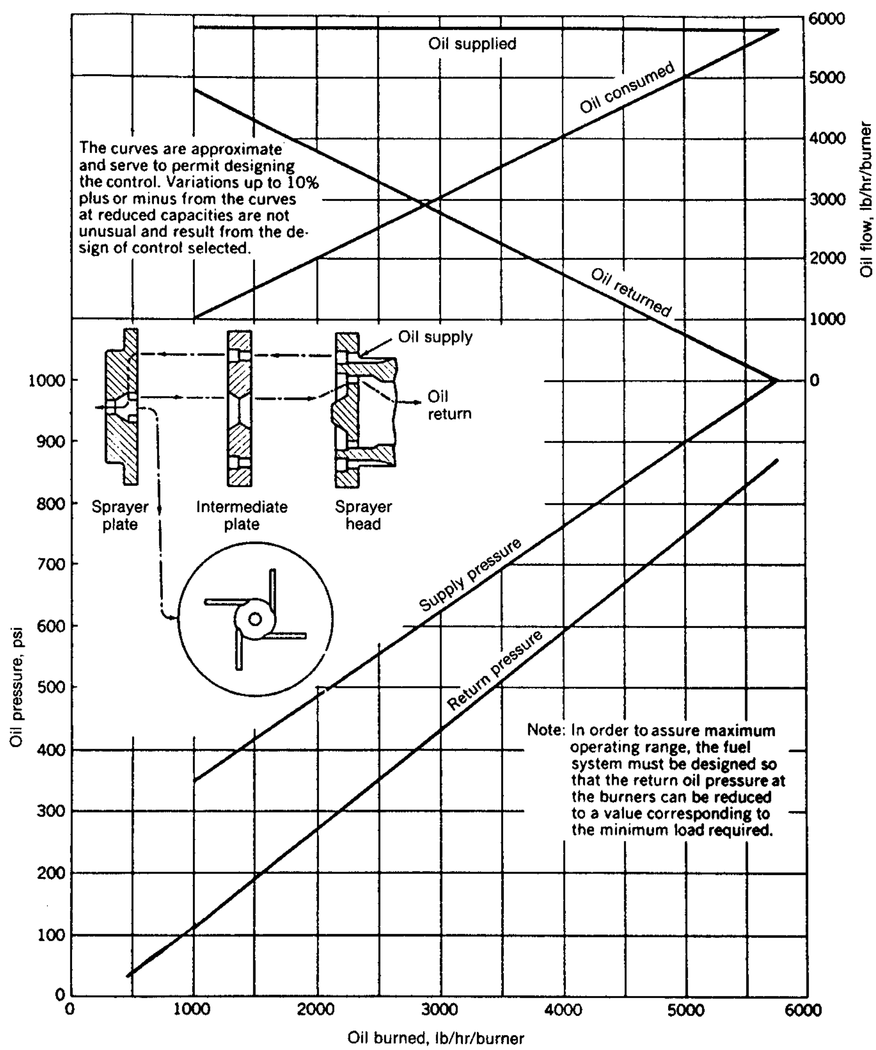


Figure 18-13 Pressure Atomizing Oil Burner Tip

(From *Improving Boiler Efficiency*, Kansas State University)



(A) Mechanical return flow oil atomizer detail at furnace end of atomizer assembly showing sprayer head, sprayer plate, and end cap.



(B)

Figure 18-14 Return Flow, Mechanical-Atomizing Oil Burner

(From *Steam, Its Generation and Use*, © Babcock and Wilcox)

Type of burner	Approximate size (input)		Atomizing pressure (PSIG)		Turndown ratio[b]	Usual applications
	Gal Per hr	BTUH[a]	Oil	Air or steam		
Vaporizing	0.15 to 2.5	20 thousand to 350 thousand	—	—	2 to 1	Small capacity residential stoves, furnaces, and water heaters.
Air atomizing, low pressure	0.5 to 530	70 thousand to 80 million	—	½ to 2	3 to 1 up to 8 to 1	Most versatile; warm air furnaces, boilers, and process furnaces.
Air atomizing, high pressure	10 to 500	1.4 million to 75 million	—	25 to 150	3 to 1 up to 8 to 1	Major-sized industrial plants utilizing compressed air in their process applications. Particularly adaptable for converting to combination gas-oil burners.
Steam atomizing	10 to 500	1.4 million to 75 million	—	25 to 150	3 to 1 up to 8 to 1	Major-sized industrial plants using steam generators, particularly water tube boilers. Particularly adaptable for converting to combination gas-oil burners.
Mechanical atomizing, nonrecirculating	0.5 to 80	70 thousand to 12 million	75 to 300	—	2 to 1 (on-off control only)	Domestic warm air furnaces, boilers, and small industrial furnaces.
Mechanical atomizing, recirculating (return flow)	25 to 1200	3.5 million to 180 million	100 to 1000	—	3 to 1 up to 10 to 1	Most economical atomizing burner. Wide range, from domestic oil burners to major-sized boiler plants, including marine boilers.
Horizontal rotary	5 to 300	750 thousand to 45 million	—	—	4 to 1	All types of installations — domestic, industrial, and commerical.
Vertical rotary	0.3 to 15	40 thousand to 2 million	—	—	4 to 1	Residential and small industrial burners.

[a]British thermal units per hour, based on a heat content of 140,000 Btu per gal for light (distillate) oils, and 150,000 Btu per gal for heavy (residual) oils.
[b]Ratio of the maximum firing rate to the minimum firing rate at which the burner will operate satisfactorily.

Figure 18-15 Oil Burner Application

(From *Steam, Its Generation and Use*, © Babcock and Wilcox)

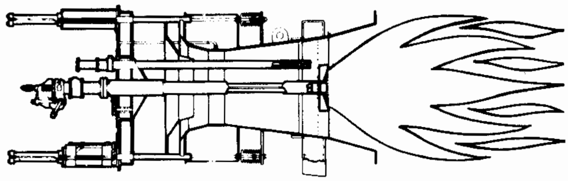


Figure 18-16 Low Excess Air Burner

(From *Improving Boiler Efficiency*, Kansas State University)

return flow type of burner. Also shown are the flow characteristic curves that relate supply and return flows to the pressures involved.

Figure 18-15 is a chart showing application guidelines and other information for these and other fuel oil burners as a group.

In many locations NOx requirements govern the sort of burners used. Some of the burners described are typical of existing boiler-burner applications. Should any of these applications be outlawed due to low NOx requirements, the regulations may require that the burners be replaced with burners designed specifically for low NOx operation. Since NOx requires an availability of oxygen, a key part of NOx reduction strategy is the reduction of the percentage of excess air. Low excess air also results in the economic benefit of fuel savings. Low excess air burners are usually longer and require a deeper windbox. A typical low excess air oil burner is shown in Figure 18-16. Generally the control requirements are the same as for the other burners, except that maximum precision of control must be obtained due to the very low 3 to 5 percent excess air margin.

Section 19
Solid Fuel Burning Systems

Solid coal, wood, and municipal or industrial solid waste are generally burned in boilers as a fuel bed on a grate. The mechanism is usually a mechanical stoker. Cyclone furnaces or pile burning are also used.

Coal may also be finely ground and blown into the boiler furnace as pulverized coal. The control characteristics of pulverized coal are similar to those of burning gaseous or liquid fuels and are quite different from those for other solid fuel burning. For that reason the burners are often combination coal-oil-gas burners. The fuel handling of pulverized coal is covered in the section on gaseous and liquid fuel burners.

For all other coal burning systems except the cyclone furnace, some form of stokers are used. One key difference among the several types of stokers is fuel residence time in the furnace. All types of stokers use a fuel bed or grate, but the control characteristics for different types of stokers are somewhat different.

19-1 Types and Classification of Stokers

A basic difference that is related to the type of stoker is the amount of residence time for the fuel in the furnace prior to its combustion. The period of fuel residence in the furnace causes the distillation of the volatile elements of the fuel into gases that easily mix with combustion air. Because of this, the response of such boilers is more immediately related to combustion air flow than to fuel flow. This response is relatively short lived and will dissipate unless additional fuel is distilled on the grate.

Stokers are of three general types:

(1) Spreader stoker
(2) Underfeed stoker
(3) Overfeed stoker

The Spreader Stoker

In the spreader stoker, coal is flipped by a distributor at the bottom of the stoker hopper onto the grate. A portion of the coal is burned in suspension, with the heavier pieces falling to the grate.

The coal feed is a volumetric feed that is regulated by a control lever that adjusts the amount of coal admitted to the spreader. Most of the combustion air is admitted from underneath, through the grate and the fuel bed, to the furnace. It is adjusted by a single control device.

In order to increase the turbulence and complete the combustion process, secondary combustion air is added as jets of overfire air above the grate. This turbulence mixes the secondary air and any remaining unburned distillation gases. Steam jets have also been used for this purpose, but this requires that the total combustion air flow be admitted from beneath the grate. The amount of and characteristics of the overfire air requirement are individual to an installation and are affected by the overfire air pressure and angle of the jets.

There are three basic subtypes of spreader stokers. The fuel feed is essentially the same for these; the difference is in the handling of the fuel bed and ash discharge, as follows:

(1) Dump grate — The grate sections are periodically turned at 90 degrees and the accumulated ash is dumped to the ashpit. A single longitudinal section of grate is dumped at any one time. These stokers are usually built of three or more of these longitudinal sections. During

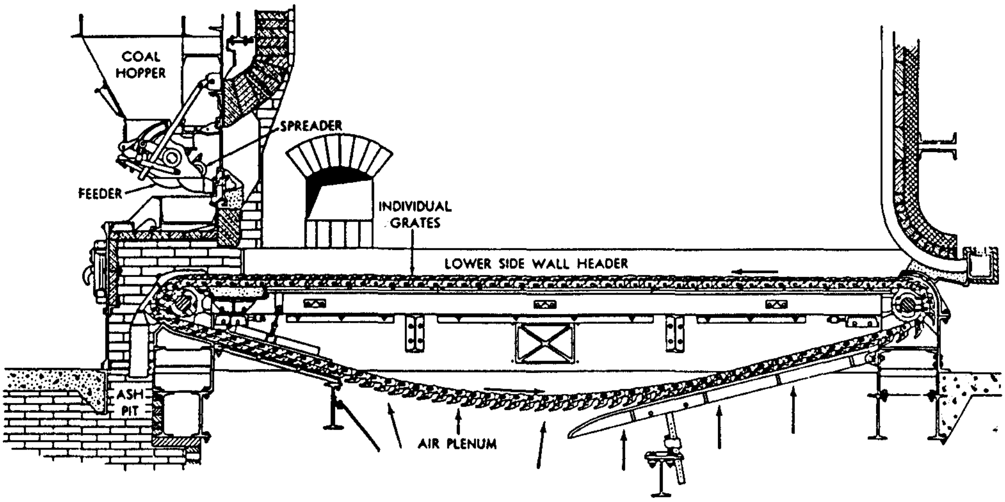


Figure 19-1 Traveling Grate Spreader Stoker

(From *Steam, Its Generation and Use*, © Babcock and Wilcox Co.)

the grate cleaning period, it is often desirable to increase the combustion air and lower the furnace draft set point.

(2) Traveling grate — In this type, shown in Figure 19-1, the grate moves slowly forward, discharging the ash to an ashpit at the lower front of the boiler. The grate speed is adjustable as is the coal feed rate.

(3) Vibrating grate (similar variations are the oscillating grate and reciprocating grate) — The grate is vibrated, causing the burning coal particles to move forward on the grate. The ash is discharged to an ashpit at the lower front of the boiler.

The heat flow response of the spreader-type stoker is quite rapid since a large percentage of the coal is burned in suspension. Some coal energy is stored on the grate and released gradually during the grate burning process. A wide variety of coals may be burned successfully with this type of stoker, but coals that melt and fuse in the combustion process may plug the combustion air entry to the fuel bed.

Because of the rapid response, a spreader stoker is useful for following fluctuating loads. Because of the suspension burning, the fly ash carryover is high, and low load smoke is a problem. This may require considerable attention to the overfire air and the reinjection of the fly ash and cinders to the grate. The spreader-type stoker is also the primary fuel-burning mechanism for burning wood waste and bark.

The Underfeed Stoker

The second basic type is the underfeed stoker. For a small boiler, an underfeed stoker may be a "single-retort" type, as shown in Figure 19-2. In this type a variable-speed ram pushes new coal into the furnace from underneath the center of the retort. The burning surface of the coal is constantly being broken up by the new coal coming in. For this reason an underfeed stoker is suggested for coals that might form clinkers by fusing the ash on the grate.

Primary combustion air is fed to the burning zone through air tuyeres on each side of the center. Secondary air, as overfire air jets above the burning zone, is also necessary. The ash is discharged at the sides.

Maintenance is relatively high on this type of stoker. It will burn a variety of coals since the burning surface is constantly broken up. Because there is very little suspension burning, fly ash and cinder carryover are low. Since all of the fuel is stored in the furnace for a period

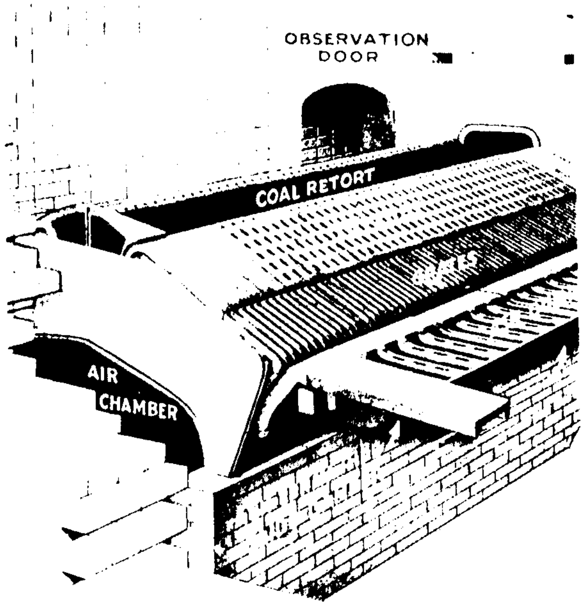


Figure 19-2 Single-Retort Underfeed Stoker

(From Detroit Stoker Co.)

before actual burning, most of the immediate response is due to a change in combustion air flow.

The multiple-retort underfeed stoker for larger boilers is shown in Figure 19-3. The basic difference from the single-retort is that the ash cannot be discharged at the sides because other

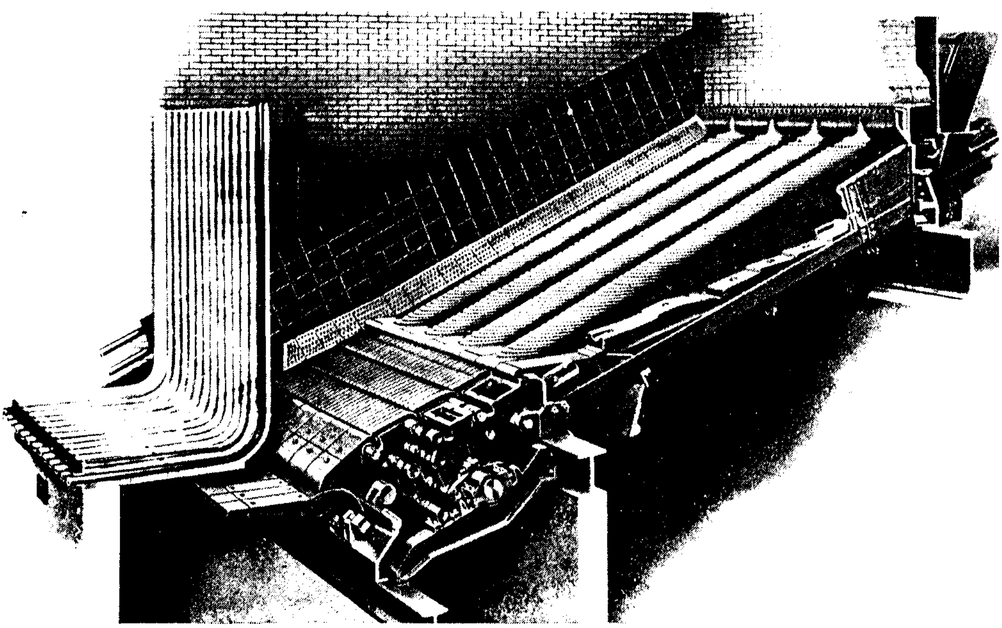


Figure 19-3 Multiple-Retort Underfeed Stoker

(From *Steam, Its Generation and Use*, © Babcock and Wilcox Co.)

retorts are on the sides. The grate is, therefore, inclined so that the ash will accumulate at the end of the retorts. In other respects, single-retort and multiple-retort underfeed stokers have similar characteristics.

The Overfeed Stoker

The third basic type of stoker is the overfeed stoker. There are two subtypes: the chain grate and the vibrating grate.

In the chain grate overfeed stoker, an endless chain type of grate moves slowly (usually over an approximate range of 1 to 6 feet per minute) under the bottom of the stoker coal hopper and drags coal into the furnace. In the virating grate the surface is sloped so that the coal fuel bed will be dragged into the furnace as the grate is vibrated. The thickness of the coal bed is adjusted by a coal gate or dam that has a fixed but adjustable height. In almost all cases the coal gate adjustment is a manual function that is performed by the boiler operator. The volume of coal fed to the furnace is a function of the coal gate height and the rate of horizontal movement or speed of the grate. The coal flow is regulated by adjustments of the grate speed.

The ash is discharged at the rear of the furnace as the grate sections turn downward. Primary combustion air is fed from underneath the grate. Overfire air jets above the burning zone feed a portion of the combustion air as secondary air. This arrangement is shown in Figure 19-4.

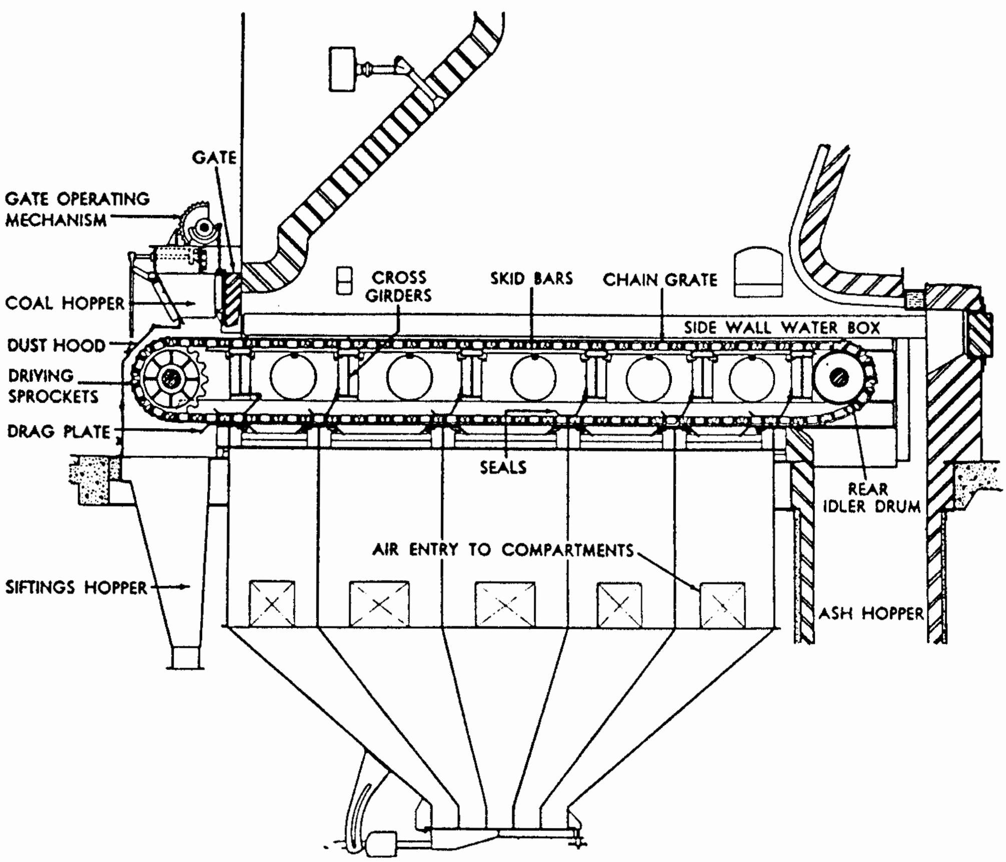


Figure 19-4 Chain Grate Stoker

(From *Steam, Its Generation and Use*, © Babcock and Wilcox Co.)

Immediately after the coal gate at the front of the furnace is a refractory ignition arch. As the coal enters the furnace, the heat from this arch starts the distillation process and ignites the coal on the moving grate. This must be a fuel-rich zone, as too much combustion air at this point will cool the arch and cause the ignition to move away from the boiler front.

The undergrate air section of overfeed stokers is divided into several compartments. Each has an adjustable air damper that is used by the operator for distributing the undergrate air. This is done to ensure a proper pre-combustion zone immediately following the coal gate and to establish the fire line, the end of visible combustion, and the desired burning profile.

The proper manual adjustment to the air compartment dampers is very important to the successful operation of overfeed stokers. Improper setting of the compartment dampers may result in the ignition zone moving away from the front refractory arch, high excess air, and excess coal being carried over into the ashpit.

Under equilibrium conditions of a given load and firing rate, a percentage of the total fuel is in the furnace in preparation for burning. A change in combustion air flow alone will increase the heat release. This is temporary, since ultimately there must be a higher fuel rate, and the higher fuel rate must be prepared to burn without significantly moving the ignition point or the fire line.

This suggests that the control system should use some derivative control to immediately get more fuel into the burning preparation zone and follow this with a reduced fuel rate that matches the new load requirement. Otherwise, at the slow speed of the grate, too long a period is required to obtain enough new fuel into the ignition zone. The result is that the increased grate speed may move the ignition zone away from the refractory arch.

With the operator in control of the air distribution, the stoker at a fixed speed and fixed gate height, the ignition point will be fixed at a point close to the arch, and there will be a fire line establishing the end of the burning zone. After this point there will be a significantly reduced combustion air flow controlled by the operator adjustment of the compartment dampers. If too much air is admitted here, the result is high excess air. If the operator does not provide enough air ahead of the fire line, there will probably be increased loss due to carbon in the ash. If the operator makes no adjustments when the load changes, the burning coal volume will increase and carry the burning past the fire line, with a resulting loss of carbon in the ash. Theoretically, the only way the fire line could be maintained at one point and the ignition remain at one point would be to vary the gate height as the load on the boiler changes and adjust the compartment dampers as the load changes. A compromise is often made of establishing about three different gate heights over the load range.

The other type of overfeed stoker is the vibrating grate stoker. This arrangement is shown in Figure 19-5. The grate is inclined in this stoker so that the material on the grate will gradually move toward the lower end as the grate is vibrated. This type of stoker also uses the manually adjustable air compartment dampers for distributing air flow to the different grate sections and movable gate height. Both of these two overfeed stoker types require the use of overfire air to increase turbulence in the burning zone.

Overfeed stokers disturb the burning mass of the coal to a relatively small extent. For proper burning on this type of stoker, fuels must be free-burning or be only weakly caking types. These stokers are unsuitable for those coals that tend to fuse together while burning. Such action with an overfeed stoker would tend to shut off the flow of combustion air.

Of all the stokers, overfeed stokers have the highest storage of coal in the furnace prior the actual burning. The heat flow response to changes to fuel flow is slow and must await the distillation of the volatile fuel elements.

Stoker mechanisms other than the stokers described are used for some applications involving industrial solid waste. For burning wood waste, a Dutch oven and conical pile combination as shown in Figure 19-6, is sometimes used. As fuel is added, it drops to the top of the conical pile and air is blown around the pile. As the fuel on the outside of the pile is burned, the ash moves toward the center of the cone. One method of regulating the combustion air flow is by

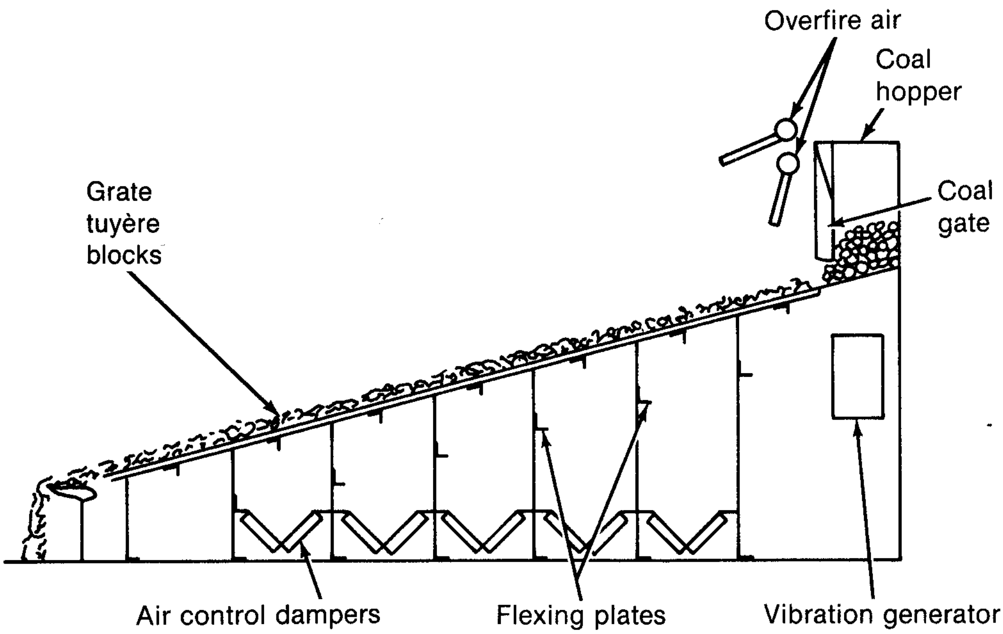


Figure 19-5 Vibrating Grate Stoker

(From Detroit Stoker Co.).

monitoring the brilliancy of the flame. Either too much or too little combustion air tends to reduce the flame brilliancy. The inside of the Dutch oven is all refractory lined and there is little if any flame cooling except for the moisture in the fuel and the water formed by the combustion of hydrogen.

In the burning of bagasse, a highly sloped stationary grate with ash discharge at the bottom

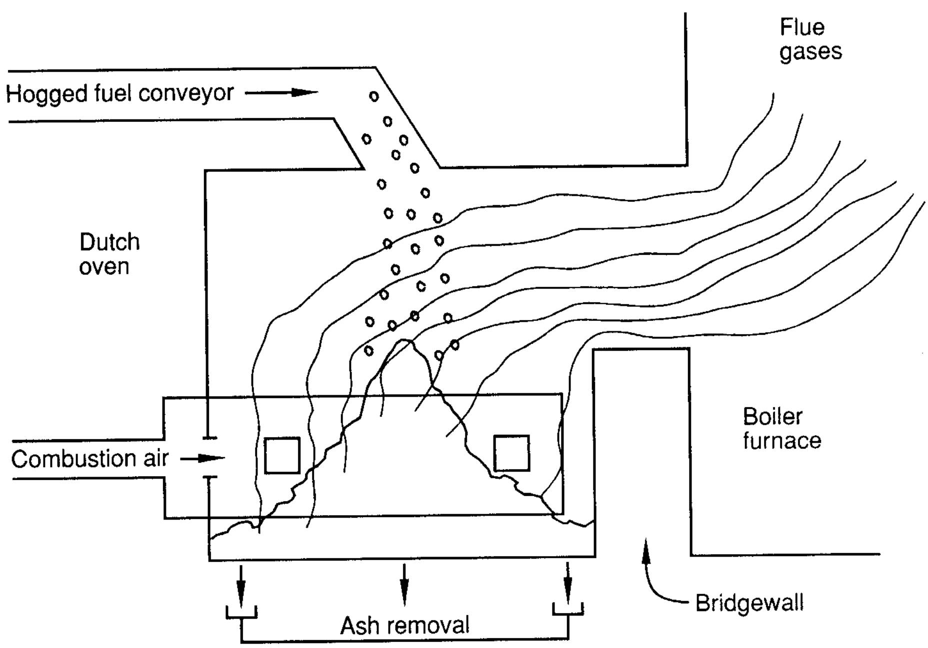


Figure 19-6 Pile Burning of Wood Waste

is often used. The bagasse is fed to the top of the grate by conveying machinery. Combustion air is admitted through the grate and secondary air is blown against the material on the grate to create turbulance. The chamber is refractory lined to ensure a high temperature for the drying and combustion of the gases. The chambers are relatively smaller than the boiler furnace, and the boiler may have several such "cells." The hot gases exit from these cells into the boiler furnace over a refractory bridgewall.

19-2 Special Stoker Control Problems

Some special problems in the control of stoker firing are unrelated to the control of combustion. The grate temperature must be kept low enough so that the grates will not be damaged. In some cases the stoker may be capabable of clean burning to a relatively low excess air value. Lower excess air increases the furnace temperature, which can cause grate damage. The grates must also be kept covered and protected from the radiant heat of the furnace. This is a case where the reduction in excess air in order to save fuel is limited by grate maintenance costs. Water-cooled grates are now available, but such grate systems are in the minority.

A hole in the fire on the grate causes combustion air to rush through the hole, with insufficient air going to other sections of the grate. This requires the operator to intervene by using a long bar through a furnace access door to scrape fuel over the hole in the fire.

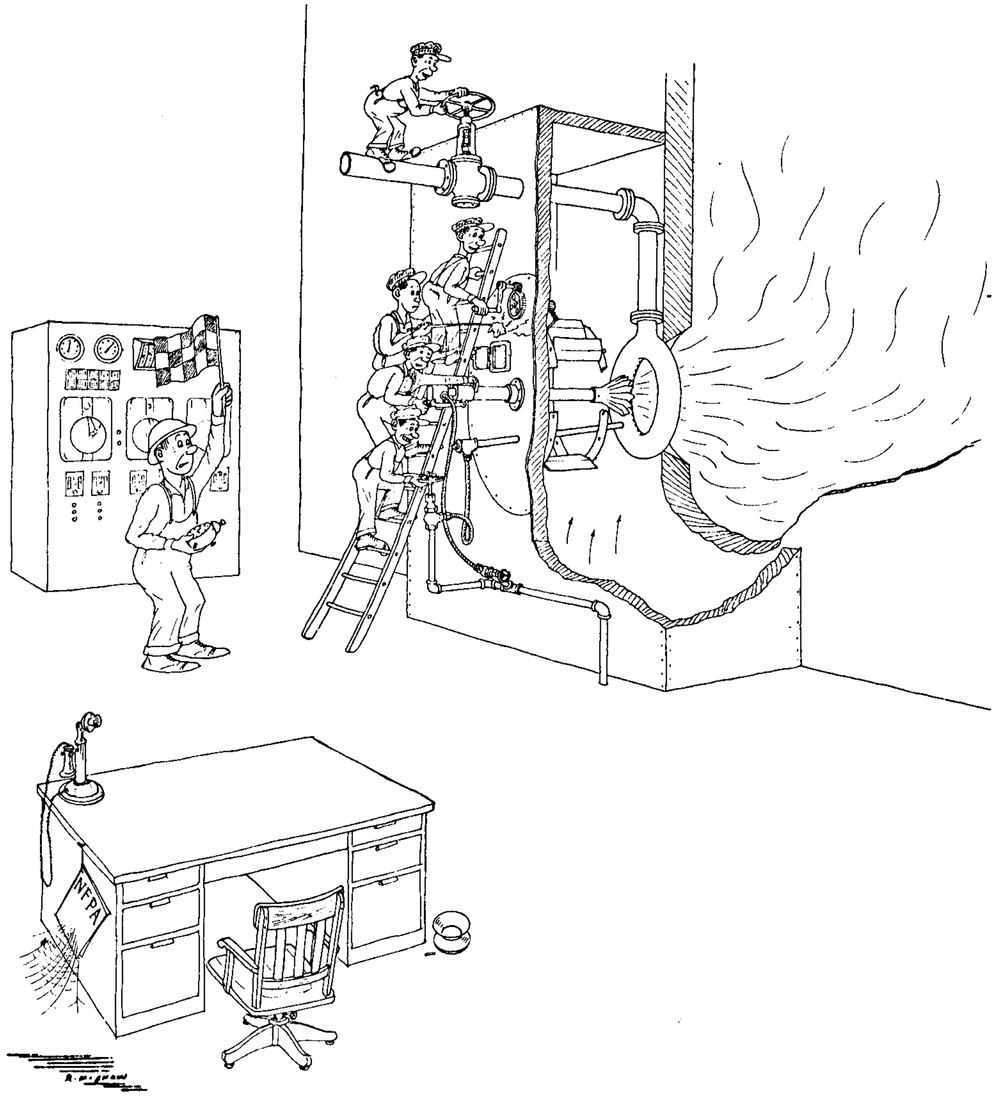
When burning fuels that may be fed intermittently, the grate may become bare and exposed to the full radiant heat of the furnace. In such cases a large amount of combustion air must be fed through the bare grate to keep it cool. Another problem special to stokers is the necessary manual adjustment of dampers for the proper distribution of the combustion air. The compromise between the characteristics of the load demand and the ability for stable response of the stoker to a heat flow demand is another consideration in the control of stoker-fired boilers.

Other coal burning problems that affect operator action and thus the ability of the control system to perform the entire control operation are listed below:

(1) Freezing of the coal in the bunkers or in storage
(2) Bunker fires
(3) Even distribution of coal on the grate
(4) Grate air distribution
(5) Overfeed stoker coal gate height
(6) Coal clinkering on the grate
(7) Incorrect type of coal for the stoker design
(8) Excessive smoke
(9) Excessive slagging of the ash
(10) Incorrect or large variation in coal sizing and percentage of fines

When burning waste fuels, other non-control problems may impact on the ability of the control system to perform its complete function:

(1) Fuel hang-ups in storage bins and screw feeders
(2) Size variation
(3) Intermittent admission of fuel to the furnace
(4) Large variations in fuel moisture content
(5) Large variations in fuel unit Btu values


NFPA

Section 20
Burner Management and Flame Safety Interlocks for Gas- and Fluid-Fired Boilers

Section 11 covered boiler and power plant interlocks in general and identified some of the more critical interlocking functions. Figure 11-6 demonstrated a macro-logic diagram for lighting a natural gas burner. This section expands on this very critical safety aspect of boiler operation.

Boiler explosions typically occur during the period of lighting off the boiler. To combat the boiler explosion hazard, a control science of interlock systems has been developed to reduce the hazard to a minimum. These systems are also designed to monitor the boiler operation, light off and shut down additional burners as necessary, and trip the fuel whenever the continued operation appears to be unsafe. The code authority that covers practices in this area is the National Fire Protection Association. The published documents are the 85 series, ANSI/NFPA 85 A-I. They can be obtained from the National Fire Protection Association, 1 Batterymarch Park, PO Box 9101, Quincy, MA 02269-9101. Any action to design or modify the design of boiler safety protection circuits should include adherence to these guides.

For the purpose of this discussion, a multiple burner gas-fired boiler is assumed. In general, the same end results are required when burning other fluid- or gas-fired boilers, but the nature of the fuel, the burners involved, and any fuel preparation required may alter parts of the logic. The arrangement of the hardware devices of such a multiple burner boiler fuel system is shown in Figure 20-1.

20-1 Basic Cause of Furnace Explosions

(Note—The paragraphs quoted below are the words of section 2.1 of NFPA 85B, Prevention of Furnace Explosions in Natural Gas-Fired Multiple Burner Boiler-Furnaces, 1989 Edition.)

"The basic cause of furnace explosions is the ignition of an accumulated combustible mixture within the confined space of the furnace or the associated boiler passes, ducts, and fans that convey the gases of combustion to the stack.

"A dangerous combustible mixture within the boiler-furnace enclosure consists of the accumulation of an excessive quantity of combustibles mixed with air in proportions that will result in rapid or uncontrolled combustion when an ignition source is supplied. A furnace explosion may result from ignition of this accumulation if the quantity of combustible mixture and the proportion of air to fuel are such that an explosive force is created within the boiler-furnace enclosure. The magnitude and intensity of the explosion will depend on both the relative quantity of combustibles that has accumulated and the proportion of air that is mixed therewith at the moment of ignition. Explosions, including "furnace puffs," are the result of improper procedures by operating personnel, improper design of equipment or control systems, or equipment or control system malfunction.

"Numerous situations can arise in connection with the operation of a boiler-furnace that will produce explosive conditions. The most common are:

(1) An interruption of fuel or air supply or ignition energy to the burners, sufficient to result in momentary loss of flames, followed by restoration and delayed reignition of an accumulation.

(2) Fuel leakage into an idle furnace and the ignition of the accumulation by a spark or other source of ignition.

LEGEND:
1. CONSTANT FUEL PRESSURE REGULATOR
2. MAIN SAFETY SHUT-OFF VALVE
3. CHARGING VALVE (OPTIONAL)
4. MAIN FUEL CONTROL VALVE
5. MAIN FUEL BYPASS CONTROL VALVE
6. INDIVIDUAL BURNER SAFETY SHUT-OFF
7. INDIVIDUAL BURNER ATMOSPHERIC VENT VALVE
8. BURNER HEADER ATMOSPHERIC VENT VALVES
9. LIGHTER SAFETY SHUT-OFF VALVE
10. LIGHTER FUEL CONTROL VALVE
11. INDIVIDUAL IGNITOR SAFETY SHUT-OFF VALVE
12. INDIVIDUAL IGNITOR ATMOSPHERIC VENT VALVES
13. HIGH FUEL PRESSURE SWITCH
14. LOW FUEL PRESSURE SWITCH
15. MAIN BURNER FLAME SCANNER
16. SPARK ELECTRODE
17. IGNITOR FLAME SCANNER
18. IGNITOR HEADER VENT

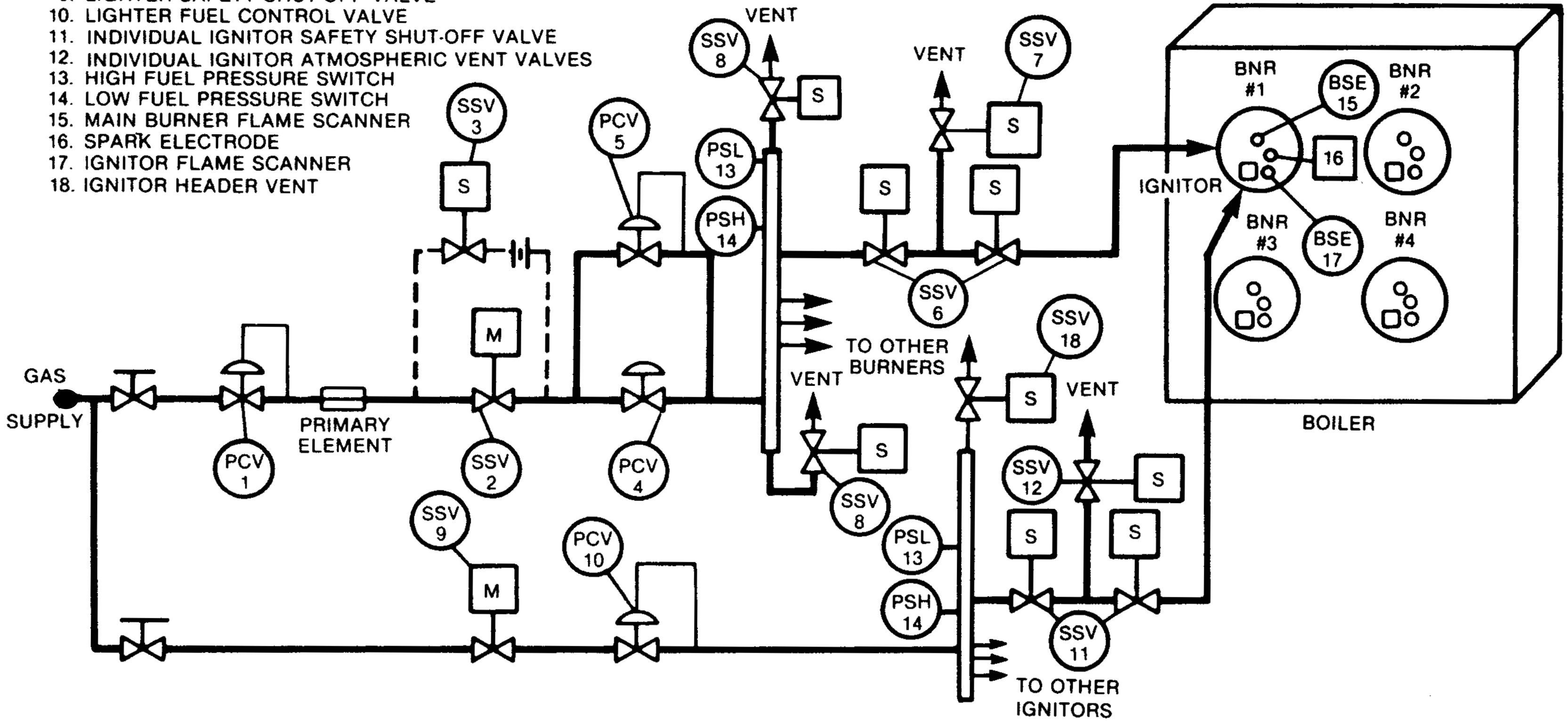


Figure 20-1 Typical Automatic Burner Management Hardware Arrangement

(From Bailey Controls Co. Application Guide)

(3) Repeated unsuccessful attempts to light-off without appropriate purging, resulting in the accumulation of an explosive mixture.

(4) The accumulation of an explosive mixture of fuel and air as a result of loss of flame or incomplete combustion at one or more burners in the presence of other burners operating normally or during lighting of additional burners.

(5) The accumulation of an explosive mixture of fuel and air as a result of a complete furnace flameout and the ignition of the accumulation by a spark or other ignition source, such as attempting to light burner(s).

"The conditions favorable to furnace explosions described in 2-1.3 are typical examples, and the examination of numerous reports of boiler-furnace explosions suggests that the occurrence of small explosions, furnace puffs, or near misses have been far more frequent than is usually recognized. It is believed that improved instrumentation, safety interlocks and protective devices, proper operating sequences, and a clearer understanding of the problem by designers and operators can greatly reduce the risks and actual incidence of furnace explosions.

"In a boiler-furnace, upset conditions or a control malfunction may lead to a air/fuel mixture that may result in a flameout followed by reignition after a combustible air/fuel ratio has been reestablished. There may exist, in certain parts of the boiler-furnace enclosure, or other parts of the unit, dead pockets susceptible to the accumulation of combustibles. These accumulations may ignite with explosive force in the presence of an ignition source."

The sequence of burner operation may be divided into the following interlock logic grouping. All of these play a very important part in safe burner operation:

(1) Boiler purge logic
(2) Ignitor header and main gas header valve management logic
(3) Gas burner management logic
(4) Main fuel trip (MFT) logic

20-2 Boiler Purge Logic

A particularly critical time period is that of lighting the first burner. For this reason purging the boiler-furnace of combustible gases is mandatory before any such action. The purging procedure is used only when a boiler is in the process of being started. This may be when the boiler has been unused for some time period or when the main fuel has tripped and the boiler is being restarted.

Purging is necessary to remove combustible gases. If the boiler has been unused, combustible gas, for whatever reason, could have leaked through the valve system into the boiler furnace. An important prerequisite to purging is to have all valves leading to the furnace closed, whether they are main fuel valves or ignitor fluid valves.

If the boiler is being restarted after a main fuel trip, combustible gas could have entered the furnace in the same way as they would in a previously unfired furnace. If several attempts to light off the first burner have failed, some combustible gas could have entered the furnace on each attempt.

In the following discussion it is assummed that the boiler and all its mechanical devices that control fuel and air are properly designed and in accordance with Section 4.1—Fuel Burning System—on pages 9 and 10 of NFPA 85B.

A summary of typical boiler purge logic is shown in Figure 20-2. Note that there are a number of progress steps with an indication or annunciation showing success of the operation up to that point. The sequence begins when all the permissives are satisfied. These permissives are as follows:

(1) The MFT (master fuel trip logic) of the burner management system on the unit must have been tripped. This is shown by an indicating light or annunciator. In this condition all devices that can admit fuel must be closed and other permissives satisfied for purging to begin.

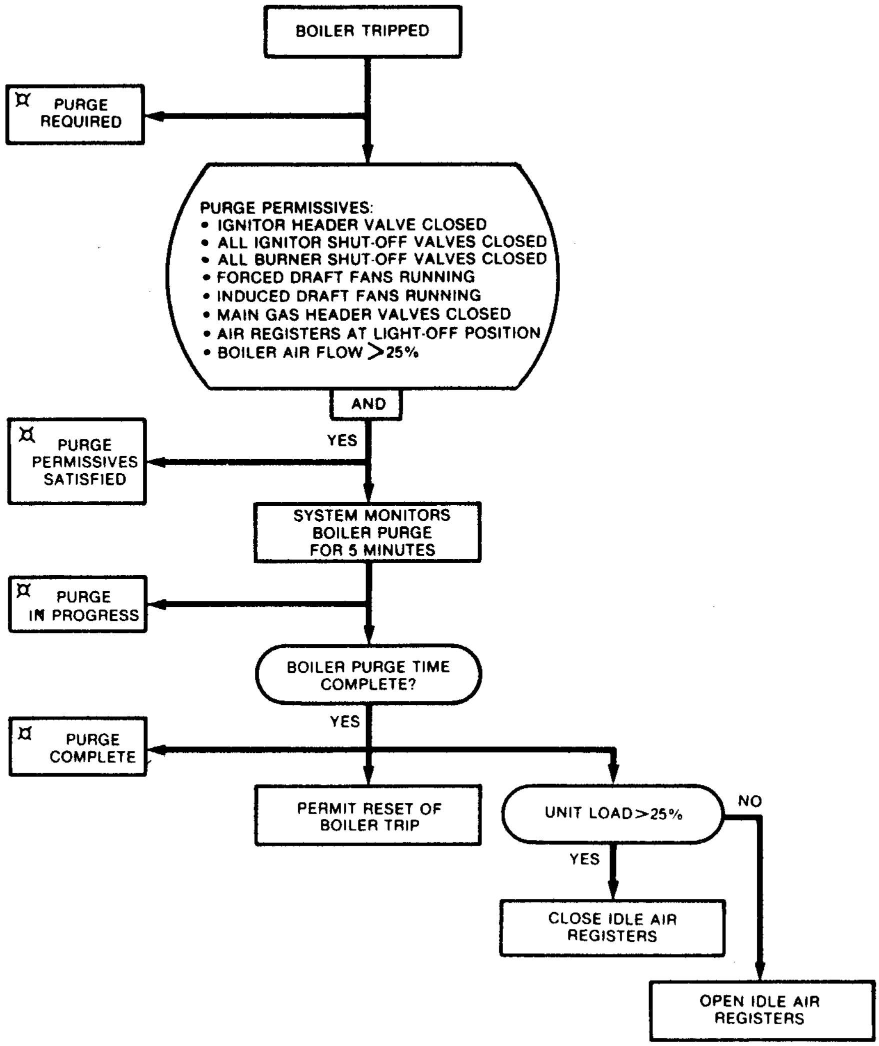


Figure 20-2 Boiler Purge Logic Summary

(From Bailey Controls Co. Application Guide)

(2) Proof that all devices that can admit fuel are closed must be signalled to the interlock logic system. These are the "in-line" valves of items 1 through 12 of Figure 20-1. Vent valves 8, 18, 7, and 12 open to assure no possible fuel pressure in the system.

(3) The forced and induced draft fans must be running to provide the possibility of combustion air flow. This may be signalled to the logic system by fan discharge pressures, contacts on the fan motor breakers, or centrifugal switches indicating at least a minimum rpm of the fan or fans.

(4) Valve 3 of Figure 20-1 is a small spring-loaded valve that bypasses the main fuel trip valve. Its purpose is to charge the main fuel header with gas so that the pressure will be within operating limits. The orifice in series with this valve limits the flow to a safe flow that could possibly pass into the furnace during the purge. The header is charged before purging to leak test the system. If the pressure is not maintained for the purging period, the "purge complete" step will not proceed and repurging is required. The leakage must be repaired before "purge complete" can be obtained.

(5) The air registers of all burners are opened to the light-off position.

(6) The control air dampers are opened to achieve an air flow through the unit of at least 25% of full load air flow.

An indicating light shows that the permissives are satisfied and a 5-minute purge timer monitors the purge time with an indicating light showing ''purge in progress.'' The timing is based on a certain number of volume of air exchanges within the boiler envelope. An assumption might be made that the time period can be shortened by purging at a higher air flow rate. This cannot be assumed. The induced draft fan is designed for hot gases, and the motor can easily be overloaded if the control dampers are opened too far on cold air. In addition, the light-off is to be made at the same air flow as the purge (5-1.5(c) of NFPA 85B), and light-off may be difficult at the higher air flow rate.

Until the purge has been completed, the loss of any of the permissives required for purge will cause the purge timer to stop. At this time any permissive that failed must be reestablished, and all other permissives must continue to be satisfied before the 5-minute purge can begin again.

When the indicator light shows ''purge complete,'' reset of the main boiler fuel trip logic (MFT) is permitted. If the first burner is to be lit, the unit load will be less than 25% and all idle burner air registers must remain open. After lighting-off additional burners and increasing the boiler load to more than 25% of its full load value, the registers on any idle burner are permitted to close and remain closed. They are permitted to remain closed until and unless the main fuel trip valve is closed. At this time the registers should open to the purge position, and the header vent valve should open automatically.

20-3 Ignitor Header Valve Management

The logic governing the action of the ignitor valve system and the main gas header valve system are similar. A macro-logic summary of the ignitor valve management logic is shown in Figure 20-3. There are many fine details to the logic that are not shown here.

To proceed with the process of lighting off the burner, there are three permissives. Two of these are the ''MFT reset'' and ''purge complete'' outputs of the purge logic. The other is ''ignitor header pressure within limits.'' This implies both high and low pressure limits and sensors to determine this. If this is not true, then the ignitor header shutoff valve will be closed and the ignitor header vent valves will open. These are the vent valves 18 and 12 and the shutoff valves 11 shown on Figure 20-1.

To charge this header with pressure and purge it of air, the lighter safety shutoff valve 9 is opened manually, valve 10 begins to control, holding the required pressure for light-off and completing the permissives. The ignitor header vent is open. It will be closed when the first burner is on or after a period of time if the first burner has not been lighted. If there is an MFT, ignitor header shutoff valve 9 will close and the header vent will open.

20-4 Main Gas Header Valve Management

The macro-logic for the main gas header is very similar to that of the ignitor gas header. This logic diagram is shown in Figure 20-4. Many fine details of the logic are not shown. The main gas header is tested for leakage during the purge cycle. All permissives are complete with purge complete, MFT reset, and ignitor header pressure and main gas header pressure within limits.

This condition will enable the opening of the main gas header shutoff valve. After this valve (valve 2 of Figure 20-1) is opened, fuel control valve 4 is positioned to the light-off position. At this time the minimum gas pressure regulator, valve 5, is holding the minimum operating pressure on the header with the gas flowing through header vent valve 8. This condition will continue until the first main gas burner has been successfully lit. The flow to this

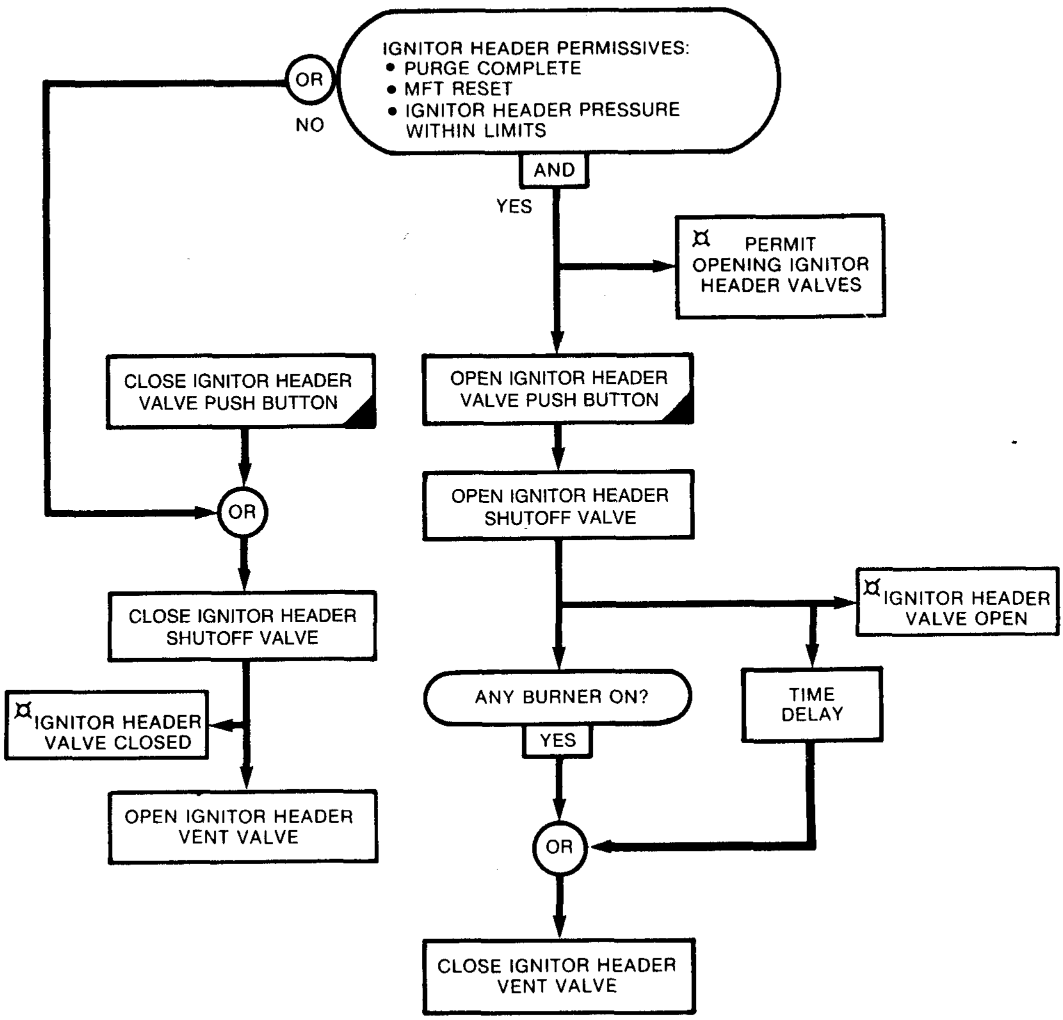


Figure 20-3 Ignitor Header Valve Management Logic

(From Bailey Controls Co. Application Guide)

first burner valve will be sufficient for valve 5 to control pressure on the header and vent valve 8 can be closed.

After an allowable period of time for lighting the first burner has passed and the first burner has not been lit, the vent valve will close anyway. If pressure in the header cannot be maintained without the vent flow, high gas pressure will cause an MFT, all fuel will be shut off, and header vent valve 8 will be opened. The MFT can be reset and the header can again be pressured through valve 3 to complete the permissives. This enables the reopening of main gas header shutoff valve 2 and a new period of time for lighting the first burner.

20-5 Gas Burner Management Logic

A macro-logic diagram for lighting-off burners is shown in Figure 20-5. With the purge, ignitor header, and the main gas header logic satisfied and the MFT reset, four of the permissives for lighting-off burners are satisfied.

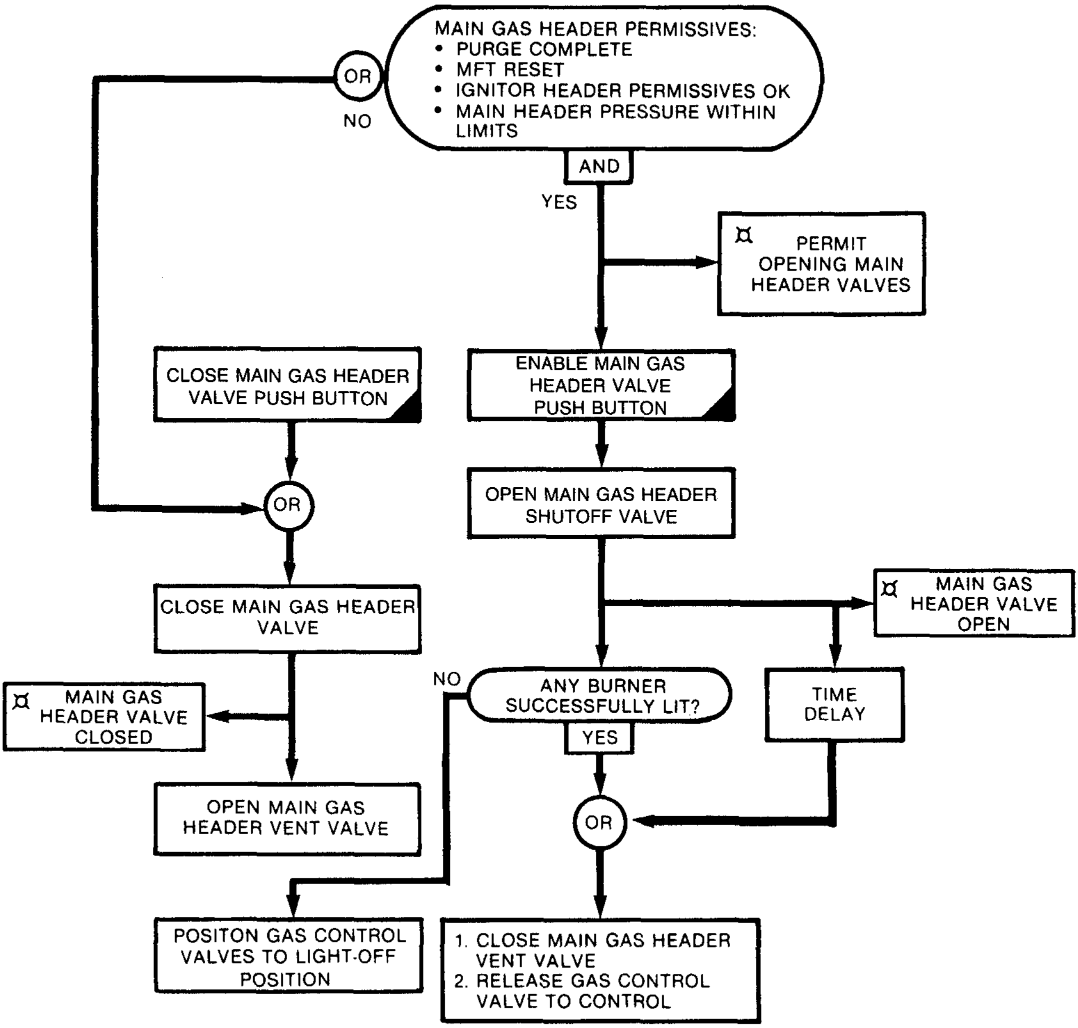


Figure 20-4 Main Gas Header Valve Management Logic

(From Bailey Controls Co. Application Guide)

Assuming that the first burner has not been lit, no flame will be detected. If flame is detected, this is an indicaton of flame detector malfunction that must be corrected before proceeding. Since no ignitor gas or main burner gas has yet been placed in service, all valves 6 and 11 must be proven closed. This can be obtained with a series circuit through "valve closed" limit switches. Alternately, an air pressure circuit can pass in series through taps in the valve bodies and be detected with a pressure switch. The taps through the valve body would provide a pressure or flow path only if the valve were fully closed. A single valve in the "not closed" position will block the air pressure passage. If the closed signal for all such valves cannot be obtained, this indicates either a sensor malfunction or a valve that is not closed. In either case the condition must be corrected before proceeding.

With all the permissives satisfied, a "burner ready" light signals for a "burner start" action. The air register of the burner to be lit must be adjusted to the light-off position. All registers of other burners remain in the purge positon. The light-off position of the register of the burner to be ignited may also be the purge position, but ease of lighting-off the burner

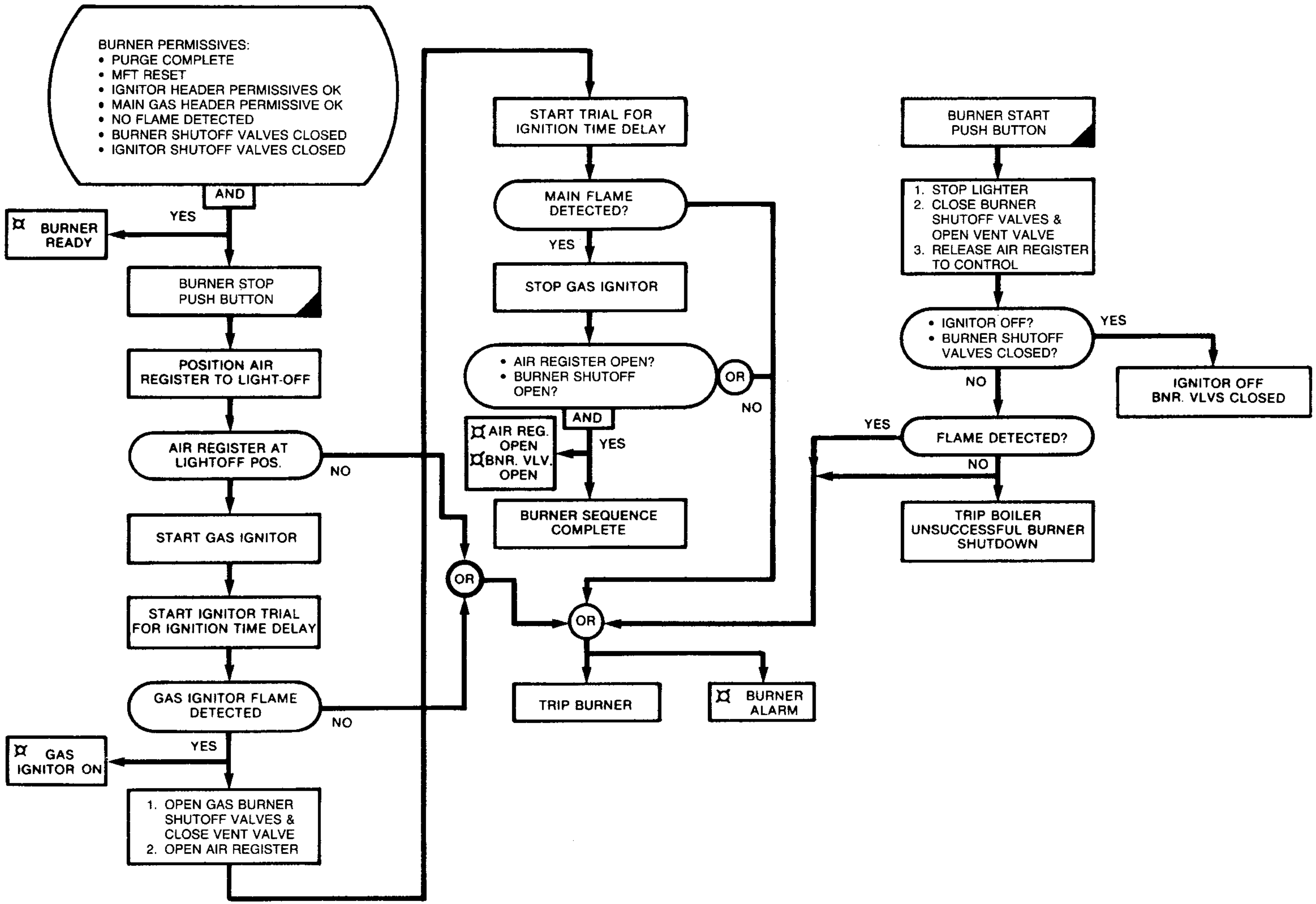


Figure 20-5 Gas Burner Management Logic

(From Bailey Controls Co. Application Guide)

may indicate a partial closing of the air register to obtain a more fuel-rich gas/air mixture. If the air register is not at the light-off position, the procedure will not continue and burner shutoff valve 6 will remain tripped until the register is adjusted to the light-off position.

Assuming the register position has been proven correct, the spark for the gas ignitor will be energized, ignitor shutoff valves 11 will open, atmospheric vent valves 12 and 18 will close, and "trial for ignition" of the ignitor will begin. If the ignitor flame detector does not see flame in 10 seconds, the ignitor shutoff valves 11 will close and the spark electrode will be deenergized. Before proceeding, the cause of the failure to ignite should be determined.

With air flow continuing at the purge rate, a repurge is not necessary, but there must be a waiting period of one minute before a retrial. Repeated retrials without investigating and correcting the cause of the malfunction are prohibited. If the purge air flow is not maintained at the purge air flow rate any time until the first burner flame is proven, a repurge and repeat of the whole procedure is required.

Assuming the ignitor flame has been proven, the main burner shutoff valves 6 are opened and atmospheric vent valve 7 is closed. If main burner flame is not proven with both the ignitor flame detector and the main flame detector seeing flame in 10 seconds after the individual burner shutoff valve leaves its closed position, a master fuel trip will occur. This will require a repurge. If this were not the first burner, only that burner would trip and an alarm would be indicated.

If main burner flame is established during the 10-second period, the air register is adjusted to its operating position, the ignitor gas is shut off, and the spark electrode is deenergized. At this time the burner must see flame on both the ignitor and main flame detectors or the burner will trip (an MFT, if this is the first burner). The burner sequence is now complete, the burner is operating, and indicator lights show that main burner shutoff valves 6 are open and that the air registers are in the proper operating position.

Subsequent burners are lit-off in the same manner. If any subsequent burner does not light properly, an alarm will sound and the problem must be investigated before proceeding. A delay period of one minute is required before a new "trial for ignition." As the boiler is transferred to automatic control, the air flow rate cannot be reduced below the 25% minimum, and the air flow registers on any idle burners will close. To remove operating burners from service, the burner stop push button will close the burner shutoff valves and close the burner air register on the burner taken out of service.

With one or more burners lit, the modulating control system can load the boiler as desired, with additional burners lit as required. Before lighting off additional burners, the burner pressure must be high enough so that all burner pressures will be above the minimum pressure after additional burners are lit off. Using signals from boiler load and burner pressure, the logic system can add or delete burners automatically as required.

20-6 Main Fuel Trip

The macro-logic summary for a main fuel trip is shown in Figure 20-6. This logic is an interlock that would trip the boiler automatically when any one of a number of potentially hazardous conditions arise. Typically, these are:

(1) forced draft fan not running,
(2) induced draft fan not running,
(3) boiler drum level low,
(4) air flow below minimum,
(5) total loss of flame,
(6) furnace pressure high or low,
(7) gas header pressure high or low,
(8) loss of flame detector cooling air, and
(9) operator initiated trip.

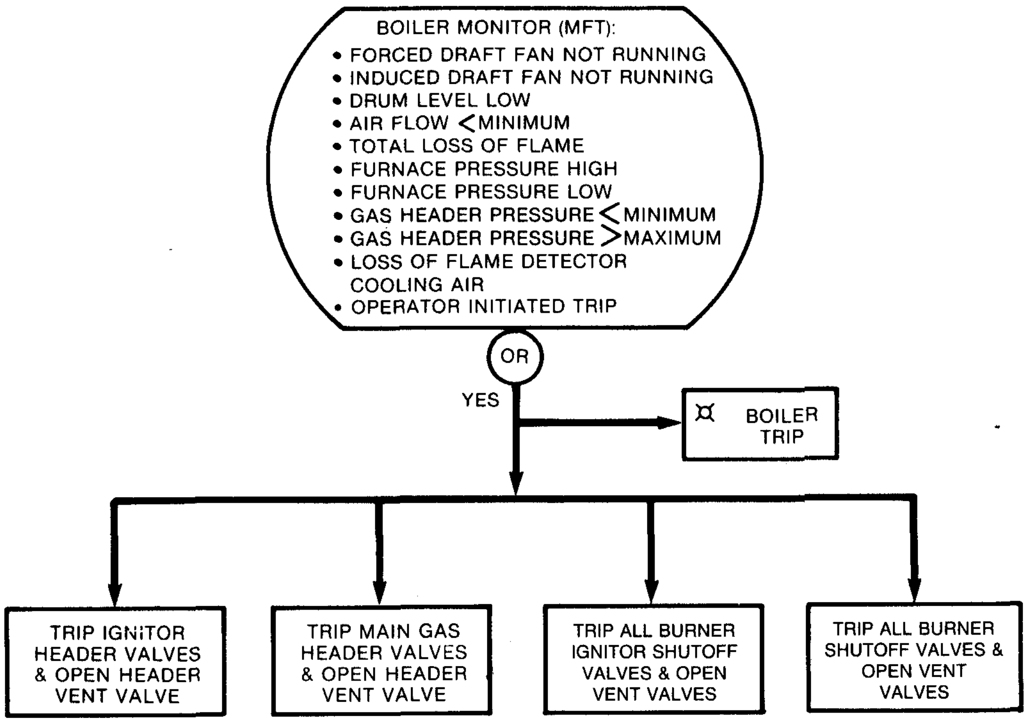


Figure 20-6 Main Fuel Trip (MFT) Logic

(From Bailey Controls Co. Application Guide)

Tripping the boiler consists of the following:

(1) Trip ignitor header valves and open header vent valve.
(2) Trip main gas header valves and open header vent valve.
(3) Trip all burner ignitor shutoff valves and open vent valves.
(4) Trip all burner shutoff valves and open vent valves.
(5) Open all burner air registers to purge position.

20-7 Degree of Burner Automation

The procedures discussed can be implemented with either a fully automatic system or a supervised manual system. An automatic system requires only an initiating push button with all purging, positioning, timing, and valve opening and closing taken care of automatically. In a supervised manual system, the operator performs the actions with a "permission to proceed" interlock at the break points along the way.

In either case the system should take action to trip a burner and close the air register of any burner that does not show flame on the flame detectors. If flame is lost on all burners or on a single operating burner, an MFT should take place. This puts a particular burden on the reliability of the flame detectors.

20-8 Reliability of Interlock Circuitry

As discussed in the section of this book covering interlocks, the circuitry design should include consideration of the hazard involved and the consequence of a boiler trip. In all cases the circuit should be designed to shut down the boiler if a hazard is indicated. This is sometimes called "fail safe."

An indicated hazard can be an external operating hazard or one involving faults in the circuitry of the interlock system. The consequence of a fault in the interlock system is that the system protection is destroyed even though the boiler could potentially continue operating without damage to equipment or personnel. If the boiler is a small heating boiler, there would probably be no real serious consequence if a boiler should trip and temporarily quit producing steam. The consequence becomes much greater depending on the criticality of continuous steam production. That is a question each operating organization has to answer.

To know that the flame detection is reliable, two independent flame detectors are required to see the flame. One of these can also be used to sense the ignitor flame during the process of lighting off the burner. After the burner is lit, the ignitor spark is deengergized and ignitor gas is turned off. If the flame detector sighted on that flame sees flame, it can only come from a main burner.

Flame detectors can fail in a condition in which their outputs would show flame to be present. This would mean that the protection has been lost. There should be some method for assuring that the flame detector output signals are good. One method is to use an automatic shutter that frequently drops into the viewing path of the detector for a momentary period during which the tripping input of that detector is momentarily disabled. If during this momentary period the detector sees flame, the detector has failed. Since there are two detectors monitoring flame, some systems are designed to alarm and operate temporarily with one detector seeing flame reliably. In such systems if a visual inspection shows that the burner is operating properly, the failed detector may be cut out of the trip circuit and the burner tripped only after the second flame detector sees no flame or is judged to be unreliable. Such systems require two detectors to prove main flame when lighting off a burner.

It is obvious that the boiler explosion hazard and the potential damage to personnel and equipment should require boiler tripping upon faults in the burner safety system that would eliminate its protection. The logic fault action could be accomplished with no fault tolerance using a single 1-o-o-1 circuit (one out of one required to hold) of electric relays, solid-state logic elements or PLCs (programmable logic controllers).

Where the consequence of tripping a boiler is critical to the rest of the plant, fault tolerance in the system without reducing its effectiveness is indicated. Some fault tolerance would exist with a 1-o-o-2 (one out of two required to hold) redundant circuit where a single fault could exist without tripping. An alarm could alert the operator to the fault. This would require 2 faults to trip. A circuit that is tolerant to a minimum of two faults and still provide protection can be implemented with 2-o-o-3 voting (two-out-of-three voting) fault-tolerant PLCs. This is discussed in Section 11 covering interlock circuits.

The NFPA 85 codes outline special testing and security action when programmable devices are used.

The foregoing discussion in this section is by no means complete. While the NFPA 85 codes must be followed, they cover minimum requirements and need amplification in many details. It is the intent in this section to be consistent with NFPA 85B, although many points included here are not included in NFPA 85B. Should there appear to be a conflict between what is written here and NFPA 85B, NFPA 85B should certainly be the guiding document.

Section 21
Combustion Control for Liquid and Gaseous Fuel Boilers

The basic difference in the approach to combustion control for liquid or gaseous fuel boilers from that for solid fuel boilers is that the the fuel can easily be measured. This basic difference applies, however, only for systems that incorporate fuel flow/air flow ratio or difference as part of the control strategy. Simple systems such as the single-point positioning (jackshaft) or parallel positioning systems can be applied to all types of boilers in a similar fashion.

As discussed in Section 8, the combustion control loops for all boilers respond to the Btu demand signals generated in the master control loop. The Btu demand signal is assumed to be linear with respect to Btu flow. A fully modulating control is used for almost all industrial boiler applications.

21-1 Single-Point Positioning Control

As shown in Figure 21-1, a single-point positioning or ''jackshaft'' system is a mechanical one. The position of the fuel control valve and the combustion air flow damper are connected together in a fixed relationship and move in unison to the demands of the master regulator.

A basic requirement of this type of system is the careful mechanical alignment of the fuel valve and the air damper positions. Fuel valves and air dampers tend to have different flow characteristics. Typical characteristics are shown in Figure 21-2. If the master regulator were to move each to the 50% position, then air flow for approximately 75% capacity would be provided while fuel for 25% capacity was being supplied. By making the flow characteristics linear, they can then be aligned.

In the case of the air damper, the alignment tool is the use of linkage angularity as discussed in Section 14. In the case of the fuel flow, the control valve is usually supplied with a cam arrangement for changing the perceived flow characteristic. The procedure is to linearize the air flow characteristic and then to match the fuel flow characteristic to that of the air flow. To perform this alignment procedure properly, it is necessary to perform combustion tests at several different boiler loads.

On the surface it appears that such a proper alignment would complete the requirements and that no further improvement is necessary. One weakness of this system, from a fuel/air ratio standpoint, is that the position of the fuel valve is not always a true measure of fuel Btu flow. Another weakness is that the fan damper or inlet vane position is not always a true measurement of the flow of oxygen for combustion.

A variation in either the flow of oxygen or fuel changes the combustion conditions and the ratio of excess air for combustion. With the parallel arrangement discussed and using gaseous fuel as an example, this can happen in several ways:

(1) Change in the fuel unit (scfh) Btu value. The fuel flow in scfh may be constant but total Btu flow changes.

(2) Change in fuel temperature. The fuel density changes and the fuel flow volume (scfh) changes. Total Btu flow changes even though unit Btu value remains constant.

(3) Increase or decrease in fuel specific gravity. The fuel density changes, and simultaneously the unit Btu value changes. These combine to change total Btu flow.

(4) Increase or decrease in fuel pressure. This causes density to change, thus changing

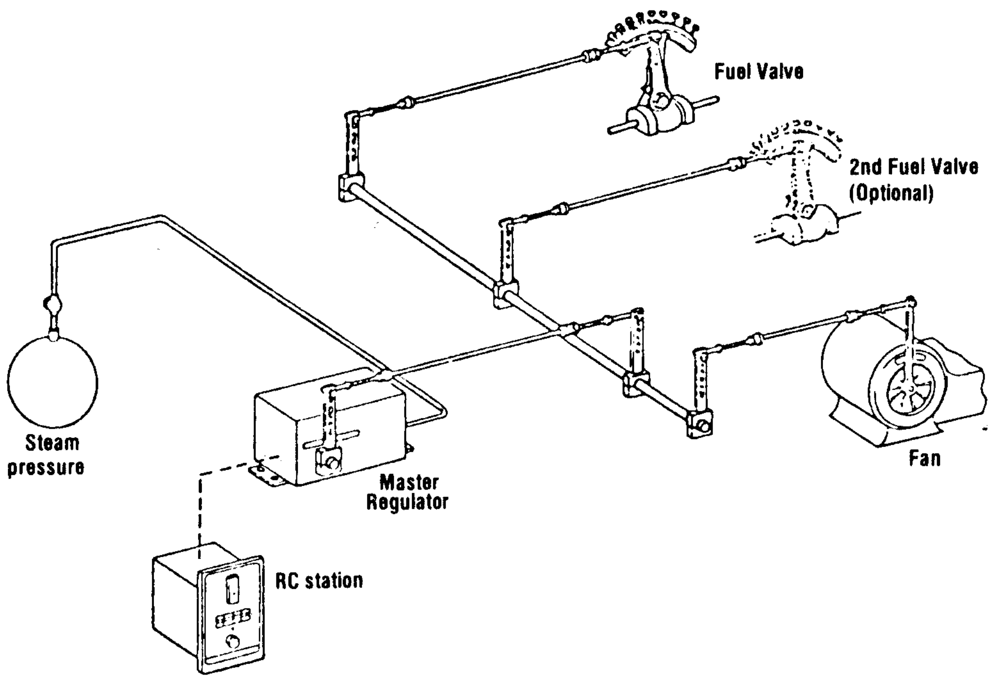


Figure 21-1 Single-Point Positioning Control System

fuel flow volume (scfh). It also changes scfh fuel volume through the changed pressure drop across the control valve. These combine to change total Btu flow.

(5) Increase or decrease in combustion air temperature. This changes the air density and delivers a changed amount of oxygen to the combustion process. Change in the density also affects fan delivery pressure and the total flow of combustion air.

(6) Change in the humidity of the combustion air. This changes the percentage of dry air in the total air, thus changing the flow of oxygen to the combustion process. The density of the air flow is also changed, affecting the fan delivery pressure and differential across the flow damper, thus affecting air flow.

(7) Changes in atmospheric pressure. This changes the fan total air flow delivery and pressure, thus affecting oxygen flow to the combustion process.

Since the fuel control valve and the combustion air damper are mechanically linked and the system does not include measurement of any of the above variables, the base system as shown cannot compensate for these variations. If we wish to compensate for these variations in fuel/air ratio, it is necessary to modify either the fuel pressure, the fuel control valve position, the combustion air control damper position, or the fan speed.

With the basic systems above, the fuel/air ratio may vary over a control error band of up to approximately 40% excess air. If the control system is adjusted for too low an excess air level, the control error band may at some time cause the boiler to operate with insufficient excess combustion air to burn all the fuel. Under such circumstances fuel may be wasted at 5 to 6 times that which would occur if the excess combustion air were too great by the same amount. With such systems it is, therefore, good practice to calibrate the system with sufficient excess air to accommodate the control error band.

Although the single-point positioning system is mechanically linked, a flue gas analysis trimming control loop can be applied to control the fuel/air ratio and reduce the control error

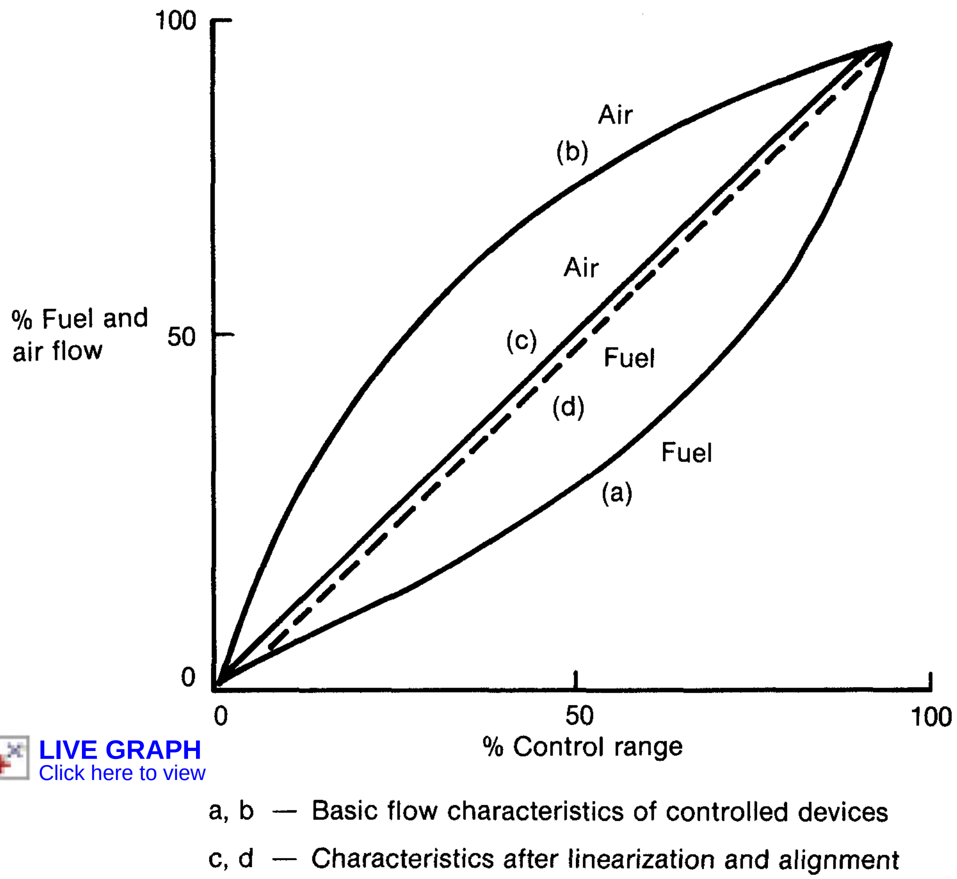


Figure 21-2 Flow Characteristics of Valves and Dampers

band. Figure 21-3 shows some typical arrangements for applying the trimming control. Other methods include changing the length of the link to the combustion air flow damper, changing the length of the drive arm connected to the link, or both.

In all of these arrangements the control signal from the trimming control originates in the control loops covered in Section 17. If there is a significant error in the basic system, the trimming control requires time to make the adjustment. If the basic system error is different at one load than it is at another load, time is required to readjust as the load is changed. This is usually not a problem unless the boiler load changes rapidly. In such cases the trimming control may have difficulty keeping up with the changes in excess air. This problem is demonstrated in Figure 21-4.

As shown in this plot, at 50% load, the trimming correction due to error in the basic system is 25% and is 12.5% at 25% boiler load. The controller output, therefore, must change by 12.5% as the boiler load is changed. This type of control is relatively slow due to the time delays described in Section 17. Some microprocessor-based "trim" control arrangements use memory of last time at this load vs. trim signal relationships to help move quickly to a new output signal as the load is changed. The plot also demonstrates that poor alignment of the basic system may create the need for an excessive amount of correction from the flue gas analysis trim control. In such cases trim control limits may prevent the amount of correction that is needed.

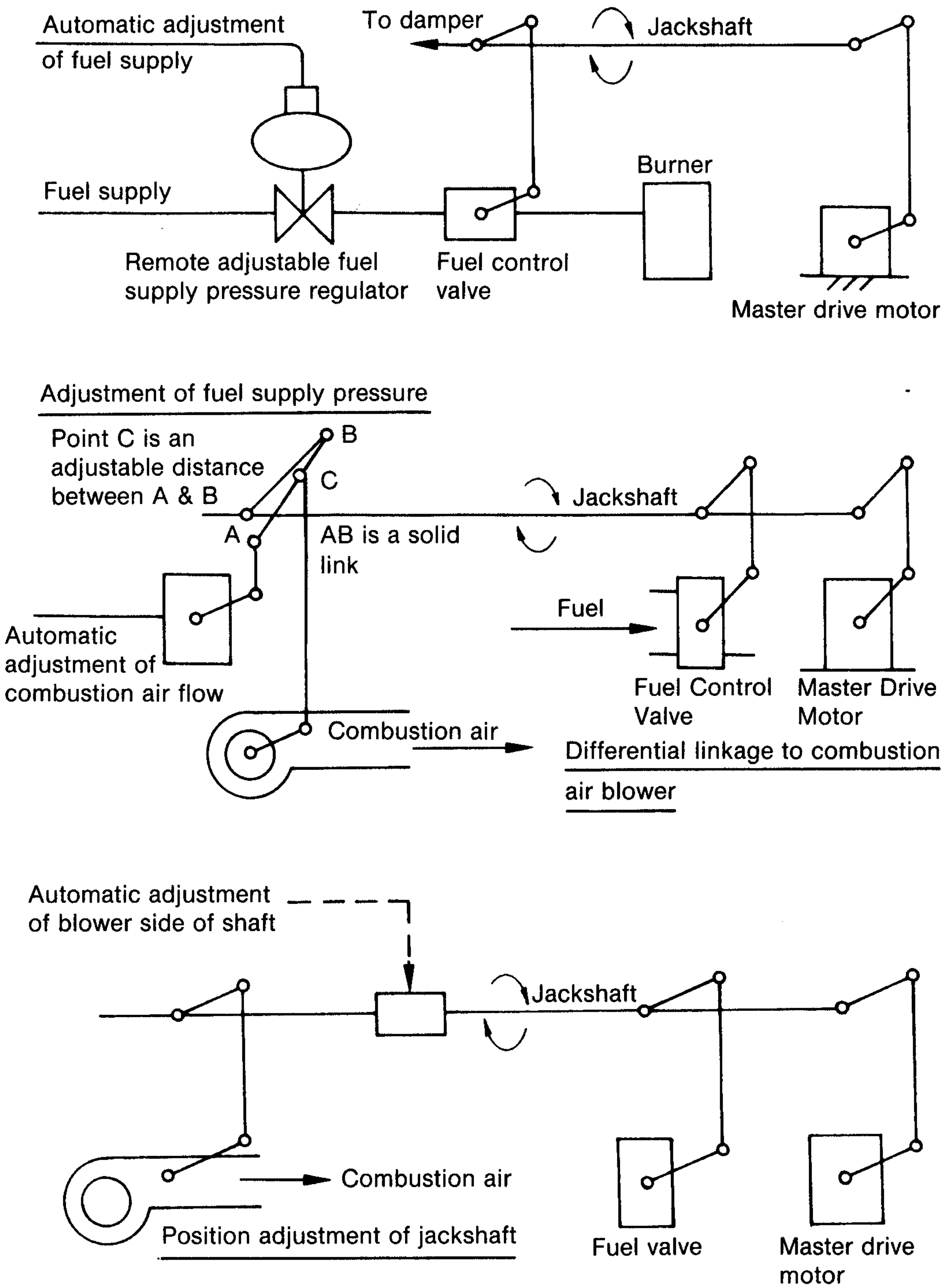


Figure 21-3 Automatic Control Methods for Changing Fuel-Air Ratio in Mechanical Control Systems

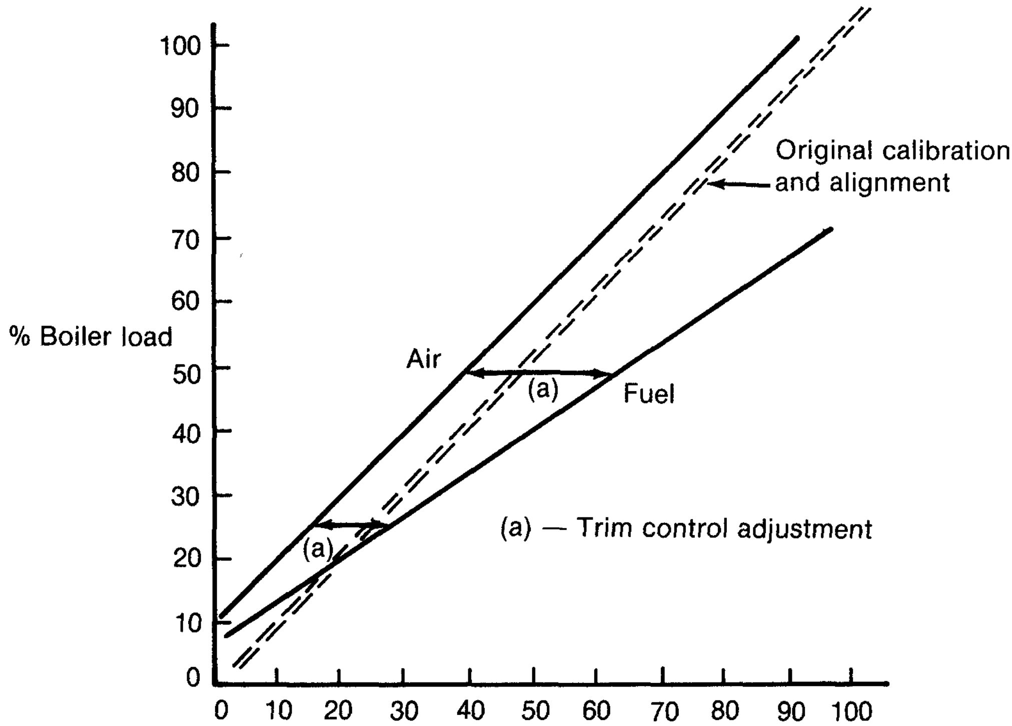


Figure 21-4 Alignment Problem: Air Is Cold, Fuel Pressure Is Low

21-2 Parallel Positioning Control

The functions shown in the mechanical single-point positioning arrangement are also performed using instrumentation control. Such systems, called parallel positioning control systems, link the functions pneumatically or electrically. A SAMA control logic diagram of a parallel positioning system comparable to the single-point positioning system of Figure 21-1 is shown in Figure 21-5.

Such control systems must be aligned in the same manner as the single-point system. In this case, a cam in the positioner of the fuel control valve is used to linearize fuel flow and align it with air flow. For air flow calibration, linkage angularity and/or the cam in the positioner of the damper operator may be used to linearize the air flow signal vs. flow.

A parallel positioning system has the same weaknesses and same control error band as the single-point system. Note that such a system may have a simple means of biasing the fuel/air ratio through use of the manual loading function, (a) in Figure 21-5. This adjustment means is useless without the use of a flue gas analyzer or some other form of combustion guide. In order that the system be aligned with the operator adjustment in the midpoint of its range, the firing rate demand input to summer (b) is set at a gain of 1.0. Assuming that the operator is provided a plus or minus 15 percent bias, the gain of the input from (a) is set at 0.3. With both inputs at 50%, and with no bias, the output of summer (b) would be 65%. The bias in summer (b) is then set at minus 15% so that the signal to the air damper control device will always match the firing rate demand signal.

The ease with which a boiler operator makes such a fuel/air ratio adjustment also makes it easier for the operator to cause system misalignment. One advantage of the parallel system

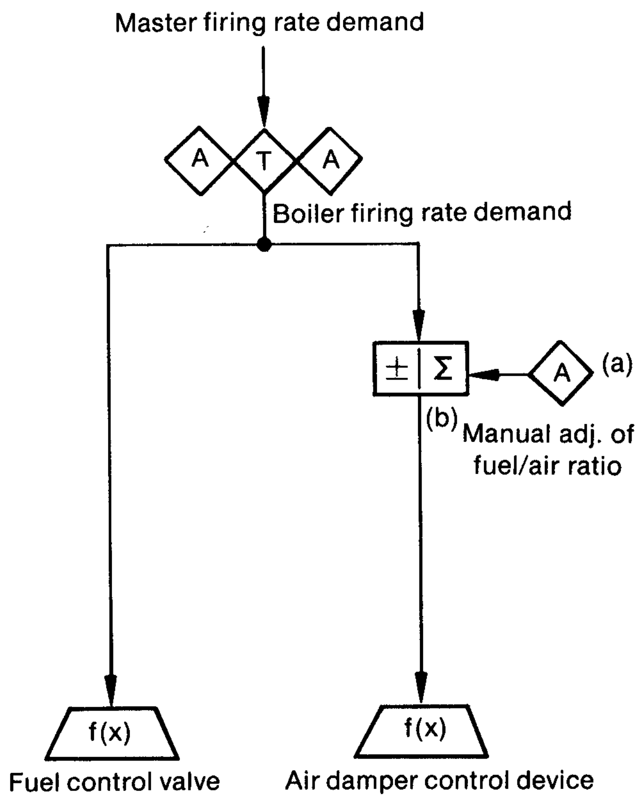


Figure 21-5 Parallel Positioning Control System

is that the timing of a fuel flow change or an air flow change can be modified by inserting a time constant into either of the two control signals to improve matching of actual fuel and air flow to the furnace. This makes possible improved dynamic operation.

Improvement of the control system in order to narrow the control error band is accomplished by the use of flue gas analysis trimming control. Since the basic system is an instrument control system, connection of the trimming control to the parallel positioning system is usually simpler than when connecting to a single-point positioning system. The arrangement of a parallel positioning system plus trim control is shown in Figure 21-6.

In the arrangement in Figure 21-6, the control signal (a) to fuel is used as the load signal in the flue gas analysis trimming control. The output of the trimming control (b) modifies the basic fuel control signal in the multiplier (c). The proportional plus positive bias (e) reduces the gain of the trimming control signal (b) and positions the output from (e).

Assuming that the trimming effect is to be an air flow multiplication of 0.85 to 1.15, the gain setting of the proportional plus bias (e) would be 0.3 and the bias would be 0.85. This would provide a multiplication of 1.0 at the midpoint of the flue gas analysis trim control output. The output of the multiplier (c) is a modified basic signal that acts as the control signal for air flow. If the furnace is a balanced draft furnace, the connection to the furnace draft control loop is shown at (d). This connection is not necessary if the furnace draft is controlled with a simple feedback control loop.

In some cases a summer is substituted for the multiplier (c). This is theoretically incorrect since the effect of the flue gas analysis trim control would then be greater at lower boiler loads that at higher boiler loads.

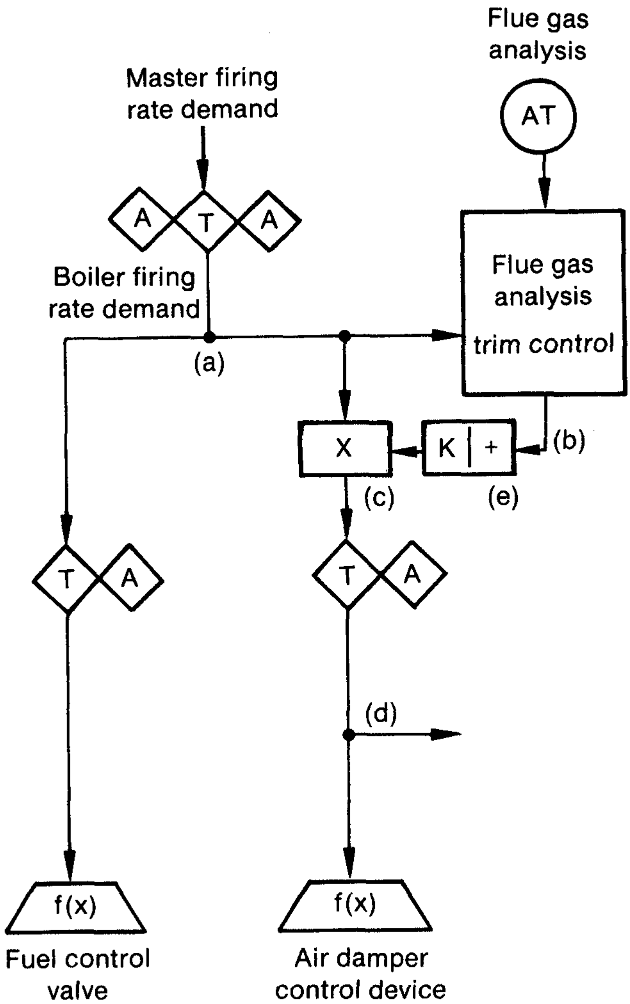


Figure 21-6 Parallel Positioning Control with Flue Gas Analysis Trim

21-3 Metering Control Systems

The weaknesses of the basic single-point positioning and parallel positioning systems can be overcome by including measurements of fuel and air flow in the control strategy. The evolution of several application methods for such metering systems has resulted in what is now generally recognized as a standard control arrangement. This control arrangement shown in the Figure 21-7 block diagram also includes active safety constraints. Such an arrangement is suitable for any liquid or gaseous fuel or fuel combination in which the unit Btu values do not vary by significant amounts (more than approximately 10 %). Several names, which all designate the same control logic, have been ascribed to this system. Such names are, "cross-limited," "lead-lag," "self-linearizing," and "flow-tieback."

In the type of system shown in the SAMA diagram of Figure 21-8, the firing rate demand signal (a) acts as a common set point for the fuel flow controller (b) and the air flow controller (c). Since the fuel flow measurement signal (d) and the relative air flow measurement signal (e) are linear, the base fuel/air ratio is established by the calibration of the air flow measuring device. As described earlier the relative air flow measurement is calibrated by combustion

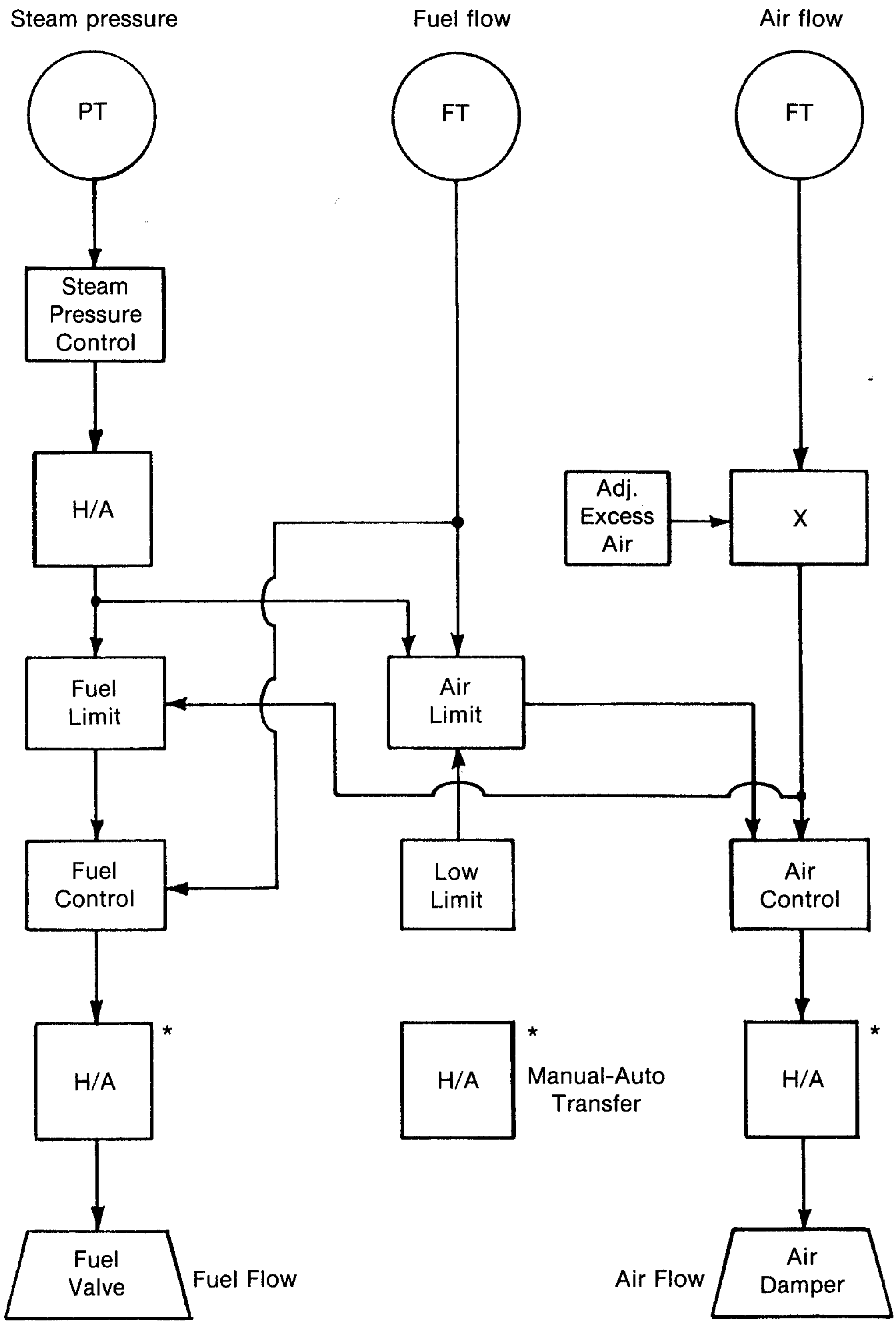


Figure 21-7 Metered Cross-Limited Boiler Control System

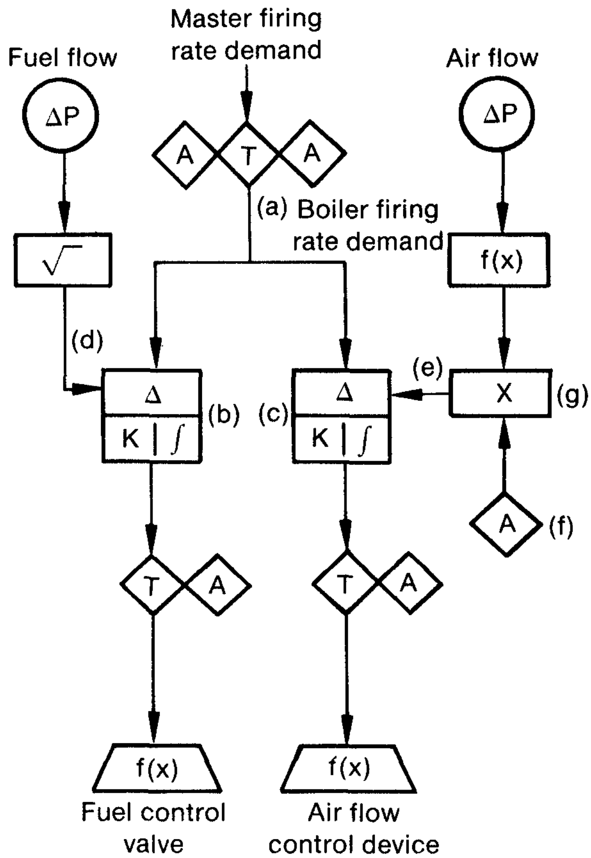


Figure 21-8 Basic Metering Boiler Control System

tests at various boiler loads. The result is that when the combustion conditions are correct, the percentage reading of both the fuel flow and the air flow measurements are equal. The manual signal (f) and the multiplier (g) provide a convenient means for the operator to alter the calibration of the air flow measurement and thus modify the fuel/air ratio.

To the basic parallel flow controller arrangement described above, high select, low select, bias, and gain functions are added as shown in Figure 21-9. These functions add active safety constraints to the system. The low select function (h) compares the firing rate demand signal (a) to the air flow measurement signal (e), and the lower of the two becomes the set point of the fuel controller (b). The result is that the fuel flow set point is limited to the level of the signal representing available combustion air flow. Similarly, the high select function (i) forces the air flow set point to the higher of the two signals that represent firing rate demand and fuel flow. The result is that actual fuel flow sets the minimum air flow demand.

The bias and gain functions (j) are added to provide a small 3 to 4% dead band between the application of the high and low select functions. This addition prevents the effect of process measurement "noise" from causing interaction between the fuel flow and the air flow control loops. The addition of the manual signal (k) connected to the high select (i) provides a minimum air flow control capability by preventing the air flow set point from being reduced below 25% of full range. The 25% minimum air flow setting is one part of the NFPA code.

This diagram also shows the application of a flue gas analysis trim control loop replacing the manual fuel/air ratio adjustment. By its control action, the air flow measurement signal (e)

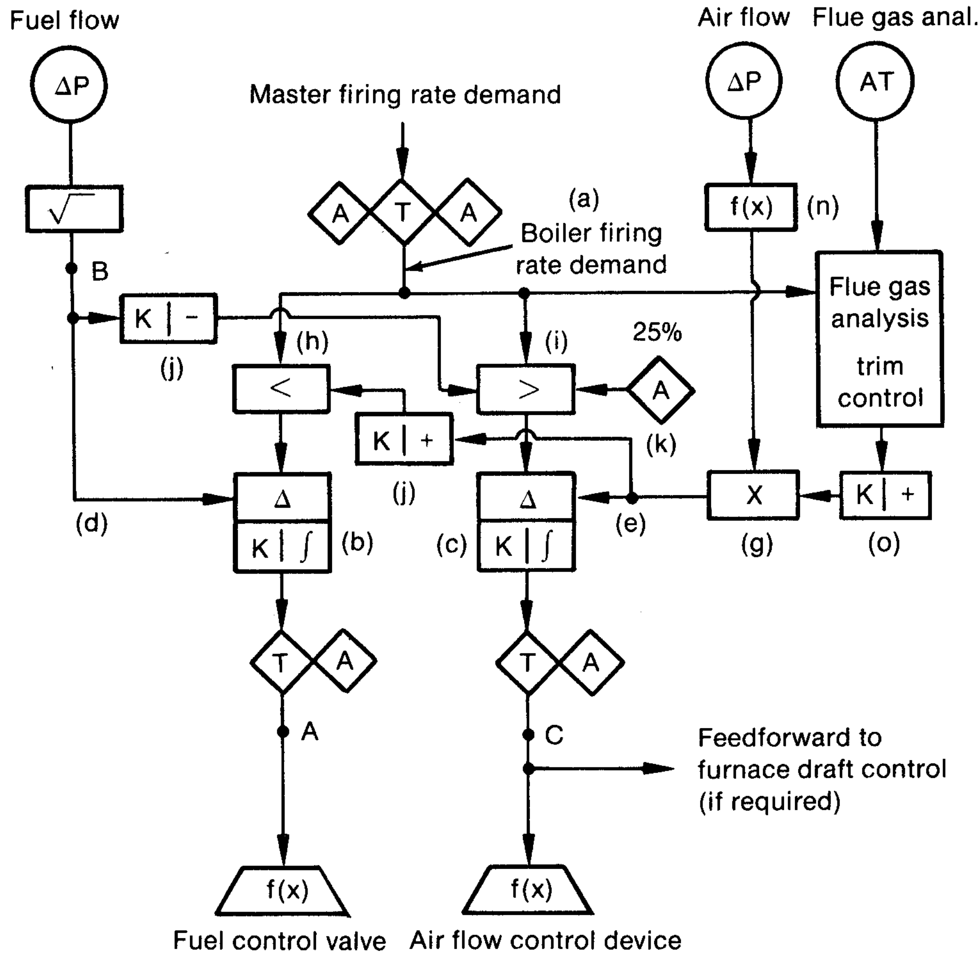


Figure 21-9 Metered Cross-Limited Combustion Control System

is continuously calibrated so that the programmed flue gas analysis will be obtained. The firing rate demand signal (a) is used as the "boiler load" signal in the arrangement shown. The fuel flow signal (d) or a steam flow measurement signal could also have been used. The points marked A, B, and C are the points at which this control system arrangement can easily be modified for application to the use of other fuels or fuel combinations or for other air flow control arrangements as covered in Section 16.

The control system alignment consists of:

(1) calibration of the air flow function generator (n);
(2) setting the bias values of approximately ± 25% and gains of 1.0 to items (j);
(3) setting the minimum air flow signal (k) at 25%; and
(4) setting the gain of proportional plus bias (o) to 0.3 and setting the bias to plus 85%.

After the alignment procedure, the next activity is to tune the flow controllers (b) and (c). This procedure is typical of the tuning of any flow control loop. The optimum gain setting will probably be in the 0.5 range and the optimum integral setting will probably be several rpm (repeats per minute). Dynamic load change testing may reveal a deviation in the desired excess air during the load change. In this case, either the fuel flow or air flow control loops should be detuned until their dynamic response is the same.

The 25% bias values of items (j) avoid any effect from the limiting control during this tuning period. After the flow controller tuning operation, these settings are adjusted to their operational settings. This is accomplished by gradually reducing the bias values until interaction with the flow controllers is indicated. This will normally show up best as the process noise band of flue gas analysis.

In some installations it may be necessary to alter the arrangement shown in Figure 21-9 because of excessive process flow measurement noise or the need for very rapid firing rate changes. By revising the control application to that shown in Figure 21-10, the primary control response is that of a feedforward system with a reduced-gain feedback trimming control from the flow control loops.

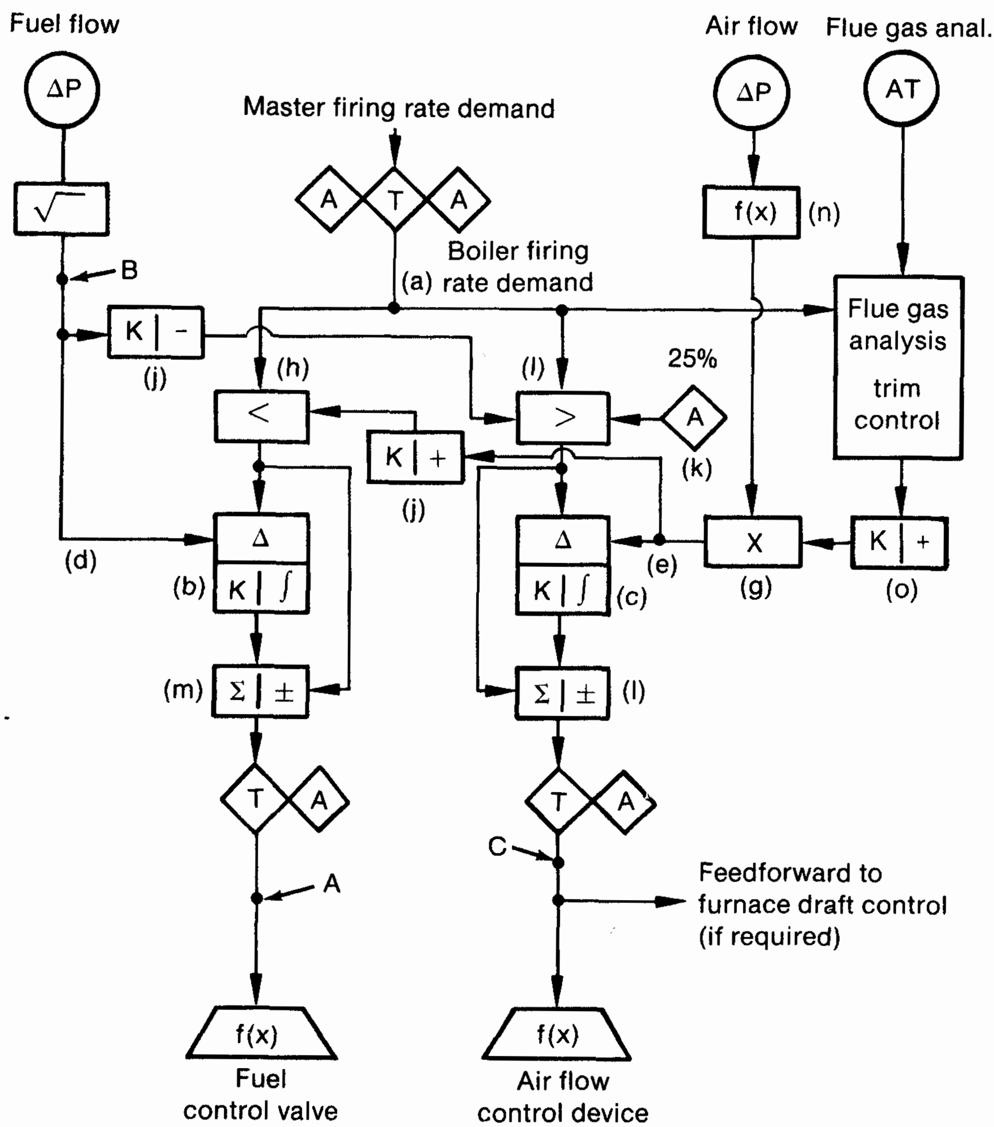


Figure 21-10 Metered Cross-Limited Combustion Control System

The additional items in the control logic are the summer and bias units (m) and (l). Both inputs to each of the summers are set at gains of 1.0. This allows the detuning of the flow control loops to accommodate the process flow measurement noise without affecting the response of the system. Since the feedback portion of the control must provide only a small portion of any control output change, the flow controllers may be considerably detuned while still obtaining stable and responsive control.

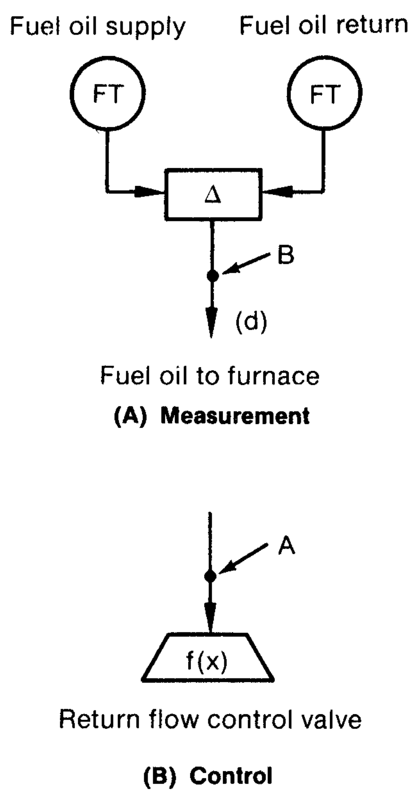
One additional requirement has been added. In the system shown in Figure 21-9 it was not necessary to parallel the control signals to the fuel control valve and the air flow control device. In the system shown in Figure 21-10 the initial feedforward signal attempts to position these devices to obtain the desired fuel/air ratio. It is, therefore, necessary to calibrate these devices for matching flow vs. control signal characteristics.

Tracking requirements are as follows:

(1) When the air flow is at the 25% minimum value, the percent oxygen controller should be in the tracking mode. A signal monitor on the air flow signal can initiate this.

(2) Whenever any one of the manual/auto stations shown is in the manual control mode, the input to the station should track the output. The implementation of this will vary depending on the control logic arrangement ahead of the station.

If a single fuel such as fuel oil is used instead of fuel gas, the modification consists of exchanging a linear fuel gas measurement (d) for a linear oil flow measurement and substituting a fuel oil control valve for the fuel gas control valve. Figure 21-11 shows the functions connected to points A and B when using return-type fuel oil burners. In this case, the "supply" and "return" flow measurement transmitter must be very carefully calibrated so that the mea-

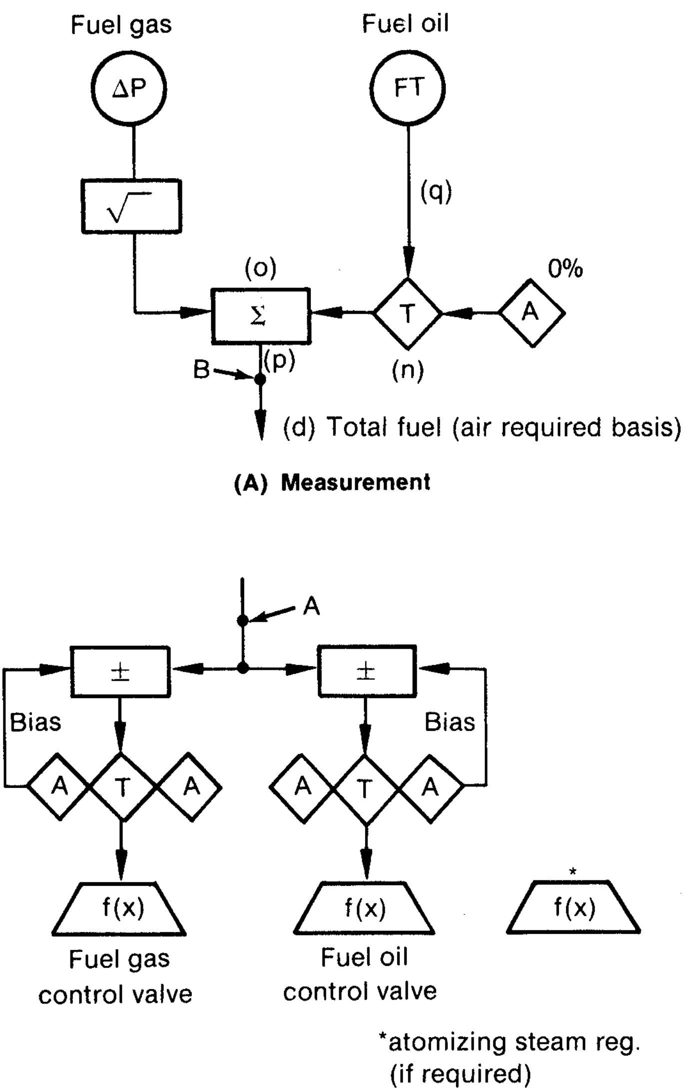


Figuer 21-11 Modification to Basic Combustion Control System for Fuel Oil Firing in Return-Type Burners

surement errors in the individual flows do not accumulate in the signal that represents oil flow to the furnace.

Other potential modifications at points A and B are those for combination fuel firing. An example of this is the combination of fuel oil and natural gas or the combination of natural gas and a relatively stable process-generated gas such as coke oven gas. The modification for burning either fuel oil or natural gas is shown in Figure 21-12.

In this modification the two fuels are totalized in the summer (o) on a basis that combines the effect of difference in Btu flow ranges, differences in air required for combustion, and differences in excess air requirement for the individual fuels. The gain value applied to the individual fuel flow signals into the summer (o) are calculated as follows. Assume that there are two fuels, natural gas @ 1,000 Btu/scfh and fuel oil @ 19,000 Btu/lb. The natural gas



(A) Measurement

(B) Control

Figure 21-12 Modification to Basic Combustion Control System for Fuel Gas or Fuel Oil Firing (Not in Combination on Auto)

flow range is 0 to 200,000 scfh and the fuel oil flow range is 0 to 10,000 lb/hr. The primary fuel is used as the base input with a gain of 1.0.

Assuming that the primary fuel is natural gas, the 0 to 100% range of the total fuel flow signal is therefore 0 to 200 MBtu/hr. The oil flow range is 0 to 190 MBtu/hr. On a theoretical basis, fuel oil requires approximately 7.3 lbs of combustion air per 10,000 Btu, while natural gas requires approximately 7.2 lbs of air per 10,000 Btu. Assume that the base of total combustion air for the natural gas is 110% (10% excess air) and 115% total air (15% excess air) for the fuel oil. The gain for the fuel oil flow input to the summer (o) can then be calculated by the following formula:

$$\text{Gain} = (190/200)*(7.3/7.2)*(115/110)$$

The result is 1.0065. If there were 3 or more fuels, the primary fuel would be the base and the other two or more would be individually matched to the primary fuel.

The totalized fuel flow signal (p) then enters the control system in the same manner as the flow measurement of a single fuel. The interlock shown (n) prevents the fuel oil flow signal (q) from being used when fuel oil is being circulated prior to burner light-off.

This control arrangement should properly be used only when there is a capability for two fuels, but they are burned one at a time. Switchover from one fuel to the other can be accomplished properly on automatic control if only one fuel is on automatic. To use this arrangement with simultaneous automatic firing of both fuels causes the fuel flow control loop gain to double due to doubling the fuel capacity available to the total fuel control signal.

Simultaneous automatic firing of two or more fuels without altering the fuel control loop gain can be accomplished with the point A and B modifications shown in Figure 21-13. In this arrangement the fuel control signal is split so that the capacity available to this signal does not change. The manual signal (r) sets a ratio for one of the fuel control signals relative to the total fuel control signal. The delta block (s) subtracts this signal from 100% of the signal value. The result is that the sum of the two fuel control signals is always equal to the fuel control signal from controller (b) of the system arrangement shown on Figure 21-9 or from summer (m) of the system arrangement shown on Figure 21-10.

An alternate control modification that will hold the fuel control loop gain constant while the boiler is simultaneously firing a combination of fuels is shown Figure 21-14. In this arrangement, the fuels are totalized on an ''air required'' basis, as in Figures 21-12 and 21-13. The control signals to the individual valves are added in the summer (t) and balanced against the basic fuel control signal in a controller (u), which produces the control signal to the control valves.

The fuel control valves will probably be of different capacities, the fuel stream of different fuels will have different Btu values, and the number of fuels being used at any one time must be accommodated. This is accomplished by adjusting the input gains of the individual valve control signals into the summer (t). These gains are the ratio of the individual valve capacities in Btu value to 100% of the desired total Btu range. The desired total Btu range is that required for achieving full boiler load plus necessary overfiring capability. For example, if the base fuel had 100% capability and two auxiliary fuels each had 50% capability, the gain for the base fuel would be 1.0 and for each of the others, 0.5.

When waste process-generated gases are available, it is usually desirable to burn these gases on a priority basis before using purchased fuels. Figure 21-15 is an example of one method of ''as available'' or ''priority'' fuel control, shown as a modification at points A and B of the basic control arrangement in Figure 21-9.

The pressure controller (v) is applied to the waste gas system. In operation with sufficient waste gas available, the pressure of the waste gas is at or above the set point and the output of the pressure controller is 0. At this time the signal to the waste gas control valve subtracts from the signals to the fuel gas and fuel oil control valves, placing these signals at a minimum

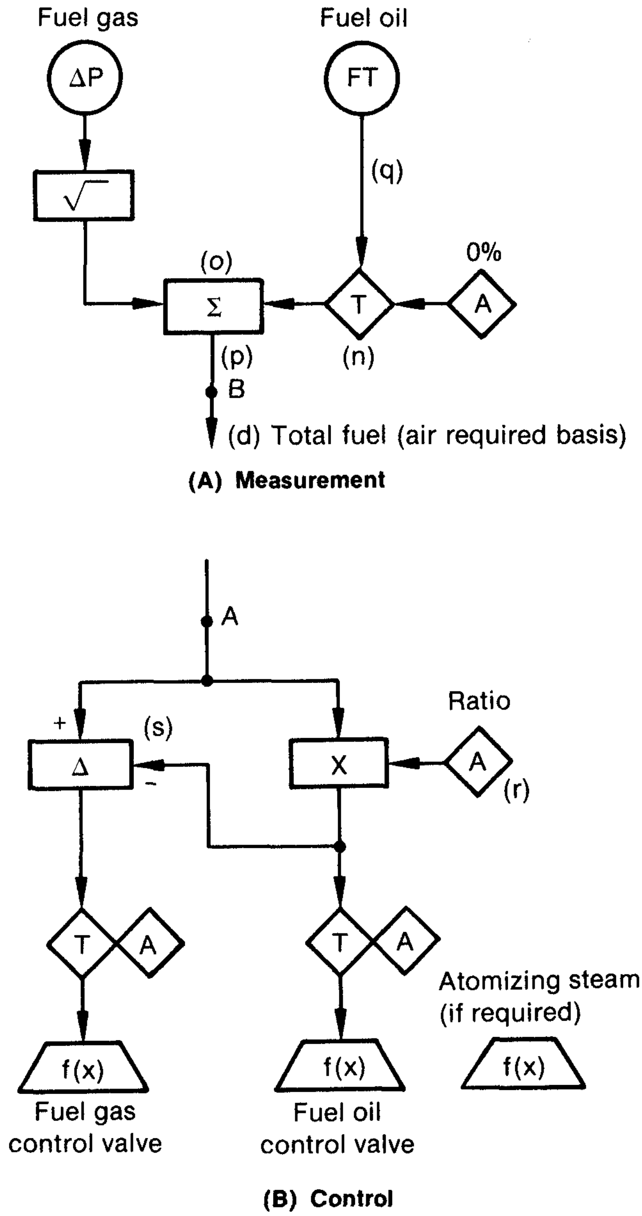
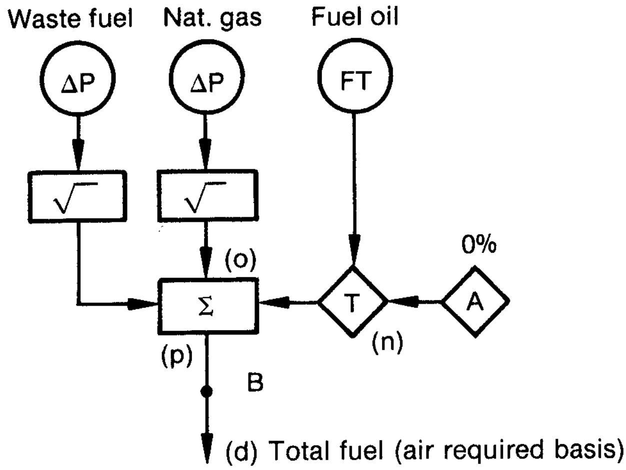


Figure 21-13 Modification to Basic Combustion Control System for Firing Fuels in Combination

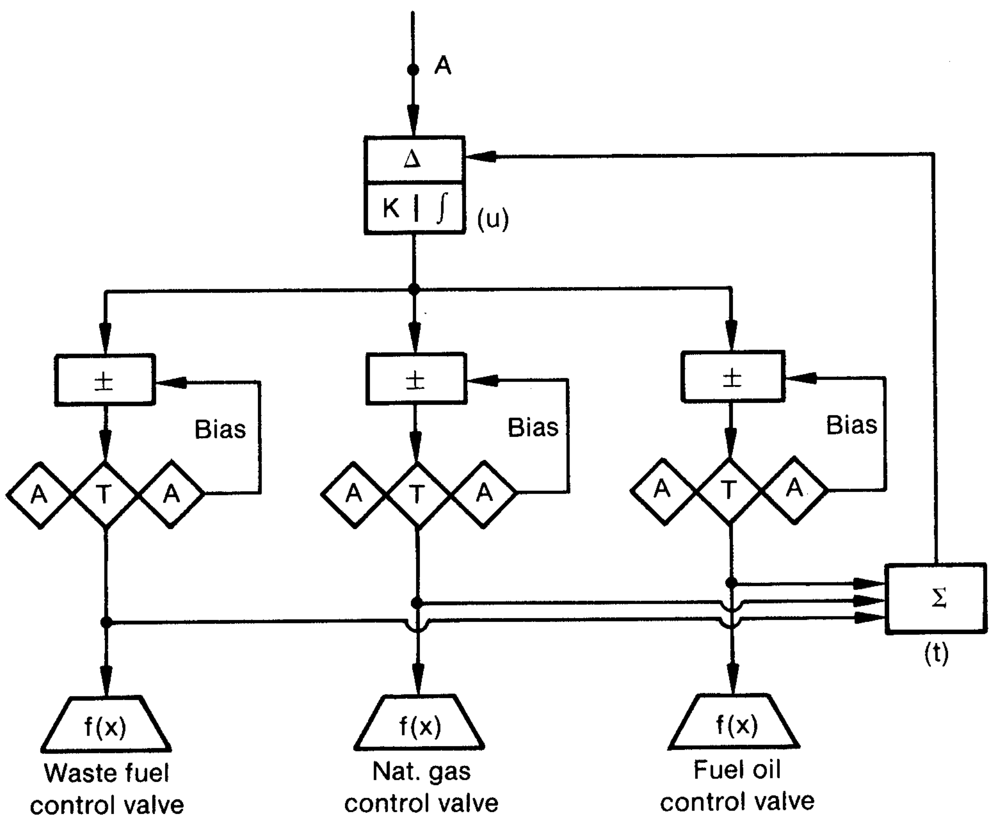
value with only a pilot flame sustaining combustion of these fuels. With less waste fuel capability than that required for the boiler load, the pressure of the waste gas will fall below the set point of controller (v), causing its output to increase. This subtracts, causing the output of delta (w) to decrease and reduce the waste gas flow until the pressure controller (v) is satisfied.

The difference between the input from A and the output of delta (w), which is equal to the output of the controller (v), is then added to the fuel gas and fuel oil control signals. This satisfies the total firing rate demand with the waste gas as a priority fuel.

In the previous arrangements involving combination fuels, it is assumed that it is not necessary to ratio the flows of the fuels involved. If the ratioing of flow is needed for ''least fuel



(A) Measurement



(B) Control

Figure 21-14 Modification to Basic Combustion Control System for Combination Fuel Firing

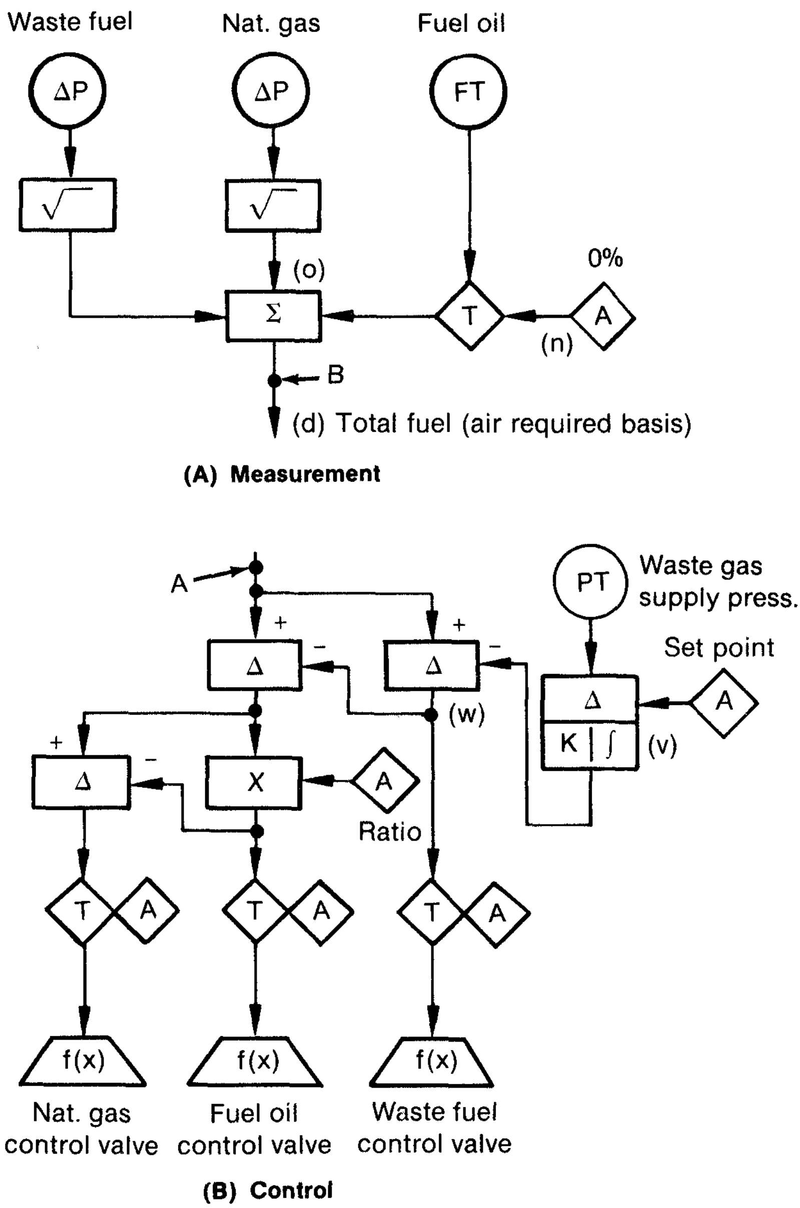


Figure 21-15 Modification to Basic Combustion Control System for "Priority" or "As Available" Waste Fuel

cost" control action or for other reasons, then the basic control of Figure 21-9 is altered in accordance with Figure 21-16. The flows are totalized as before on a "total air required" basis. The output of this totalization acts, however, only on the active constraint of "high select" function, which forces the air flow set point to the value of that required for the total fuel being burned. For the normal control functions, the fuel flow feedbacks respond to the individual set points of their controllers. The set points are generated through a set point ratio arrangement that is the same as that used for splitting the control signal, as shown in Figure 21-13. The ratio set may be generated manually, as shown, or through other control functions that are not shown here.

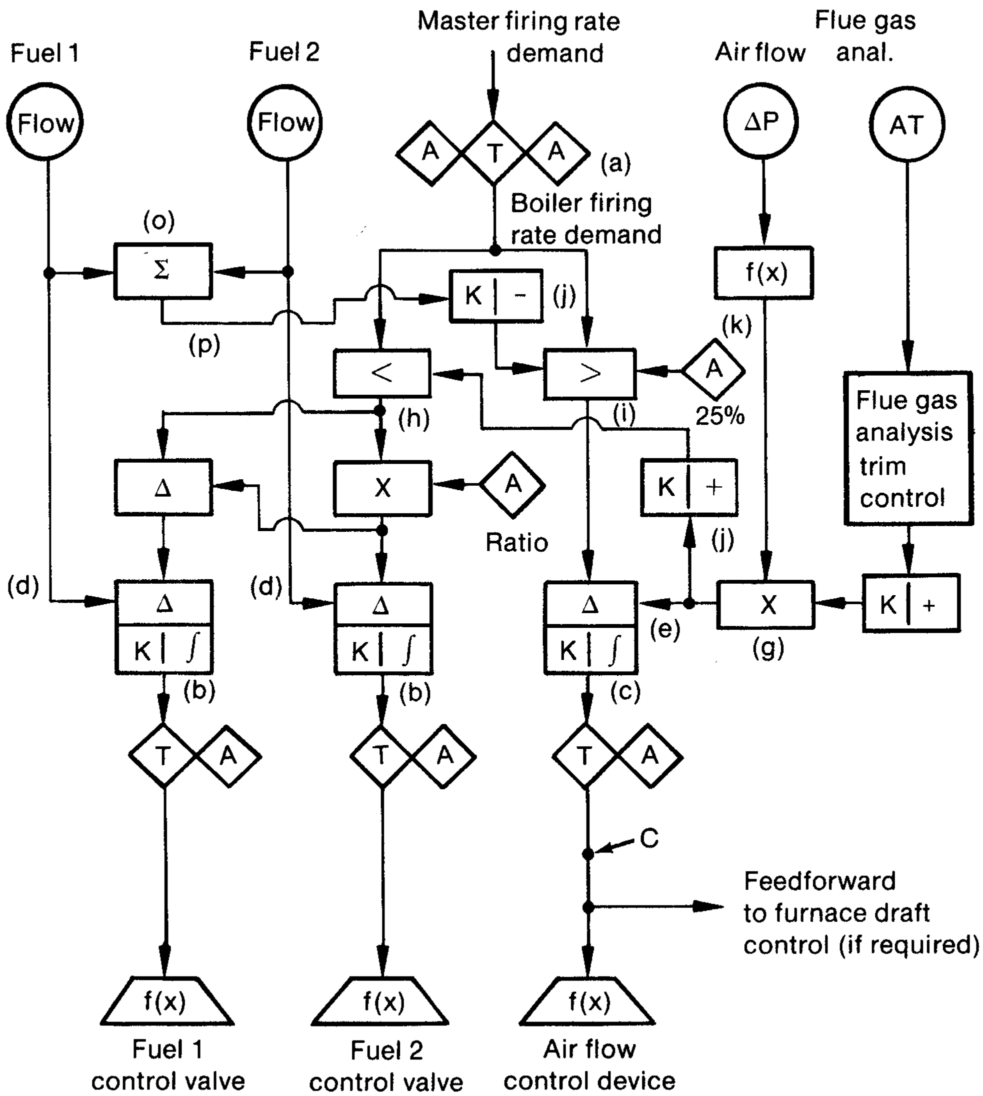


Figure 21-16 Metered Cross-Limited Combustion Control System with Adjustable Ratio Firing of Fuels in Combination

21-4 Effects of Fuel Btu Variation

The control system arrangements shown above are not the most desirable arrangements if the fuel Btu varies significantly (more than approximately 10%). Assuming a gaseous fuel, a fuel unit Btu variation of 10% with the same volume flow will change the total Btu delivery to the furnace by 10% while the air flow remains constant. This causes a shift of 10% in total air for combustion. Such a deviation would normally be unacceptable but is still within the range for a ± 15% flue gas analysis trim control.

In fact, for hydrocarbon fuel gases a fuel Btu variation is accompanied by a specific gravity change. For the same volumetric flow, this alters the differential pressure across an orifice and tends to counteract the fuel-air ratio control effect of the unit Btu value change. If a hydrocarbon fuel such as natural gas is the fuel used, the net effect on % excess air for a 10% change in fuel Btu value is approximately 3.5%, the approximate threshhold of acceptability without flue gas analysis trim control.

If the change in Btu value were 20% or 30% as is often the case with the use of refinery gas, the change in fuel Btu would significantly alter the total Btu input to the boiler, changing the steam pressure and causing the firing rate demand signal to change. In correcting the total Btu input to its previous value, the combustion air flow would be adjusted to an incorrect value. A flue gas analysis trim control could be used to recorrect the air flow, but this would be after the fact of the process disturbance and the effect on excess combustion air.

A better solution is to alter the fuel flow before the process is disturbed. A flue gas analysis trim control that modifies the fuel flow or the fuel flow demand signal will prevent most of the process disturbance described.

Since the air flow required is nearly constant for a given total Btu flow, then a change in fuel unit Btu value (and, thus, total Btu input) will cause an immediate change in the flue gas analysis. The change in the flue gas analysis can therefore be used to control a multiplication that recalibrates the fuel flow or fuel demand signal so that total Btu value will be the same

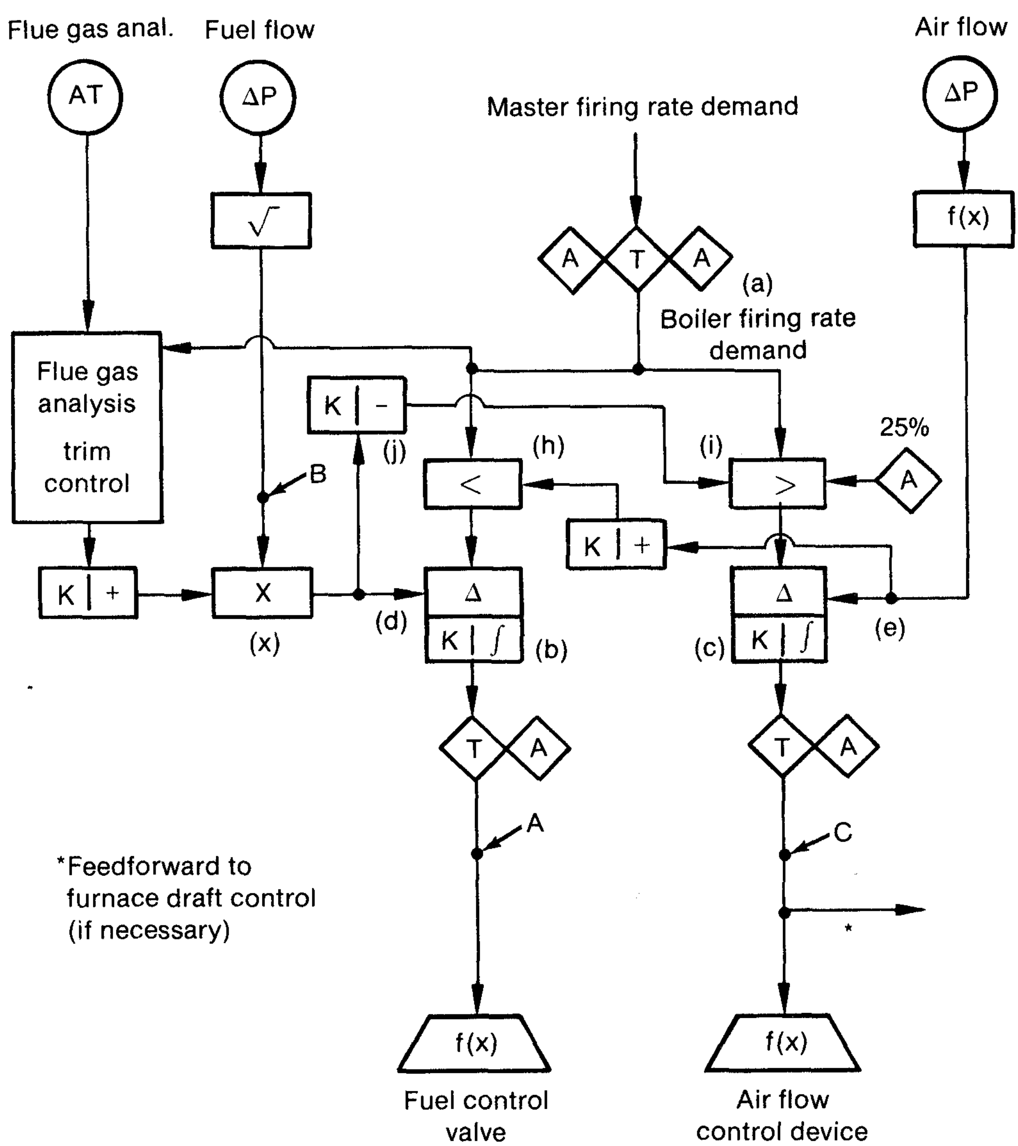


Figure 21-17 Metered Cross-Limited Combustion Control System — Variable Btu Fuel

as before. While there is some dead time in this action, the flue gas analysis response is many times faster than the total boiler response. In this way the disturbance to the process is minimized. Such an arrangement is shown in Figure 21-17.

The system shown is the same as that of Figure 21-9 except for the application of the flue gas analysis trim. As shown, the flue gas analysis trim control loop drives a multiplication (x) of the fuel flow measurement signal. The multiplication range is a function of both Btu value and the offsetting effect of change in specific gravity on the the flow measurement. This arrangement is satisfactory if the maximum multiplication expected is not more than approximately 1.15 or less than approximately 0.85.

If the maximum multiplication exceeds the approximate 0.85 to 1.15 range, the change in the feedback loop gain becomes more significant. Expected multiplication ranges of greater than the approximate 0.85 to 1.15 range require the multiplication on the demand or set point side of the controller in order that the feedback loop gain will be unaffected. This change is shown in Figure 21-18. This diagram also shows a divider (z), driven by the flue gas analysis

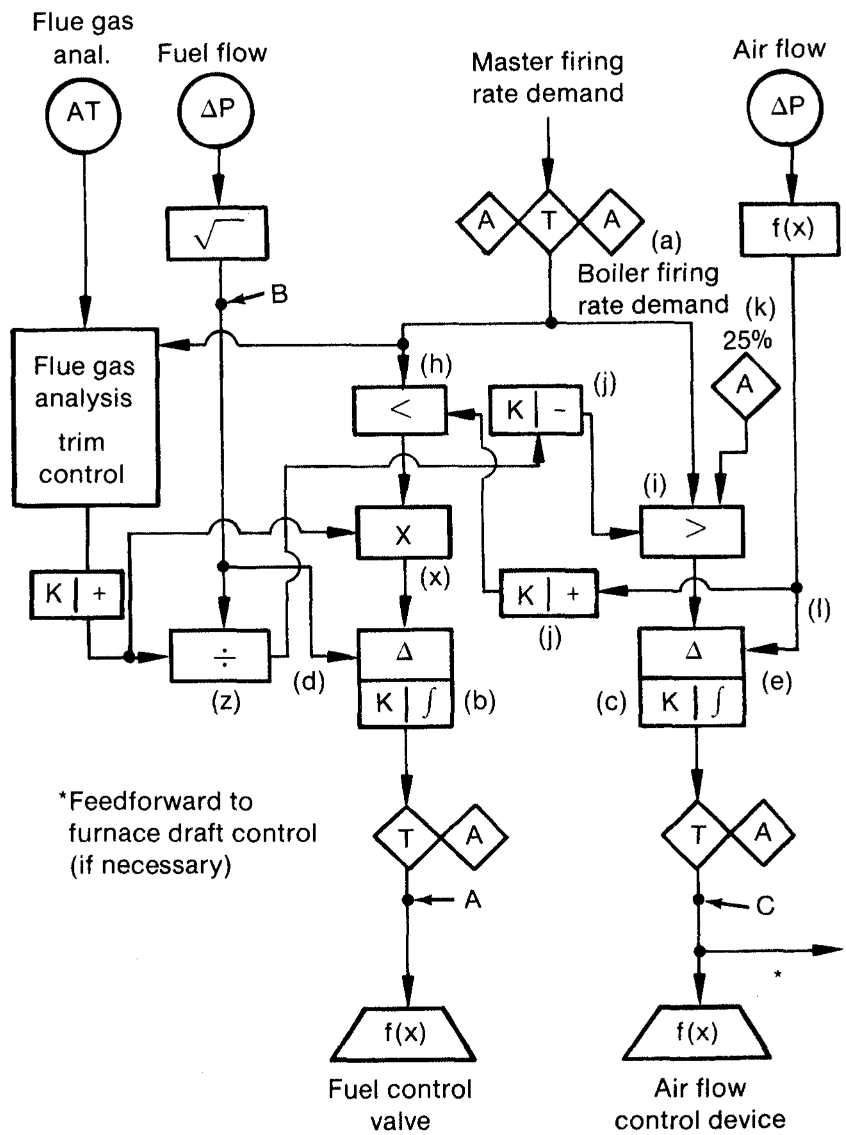


Figure 21-18 Metered Cross-Limited Combustion Control System — Variable Btu Fuel

trim control. This divider, by keeping the total "air flow required" range of the fuel measurement constant, assures that the air flow limiting function will remain correct. The technique described above is also suitable for installations that burn combination fuels with one or more of the fuels having a variable Btu content.

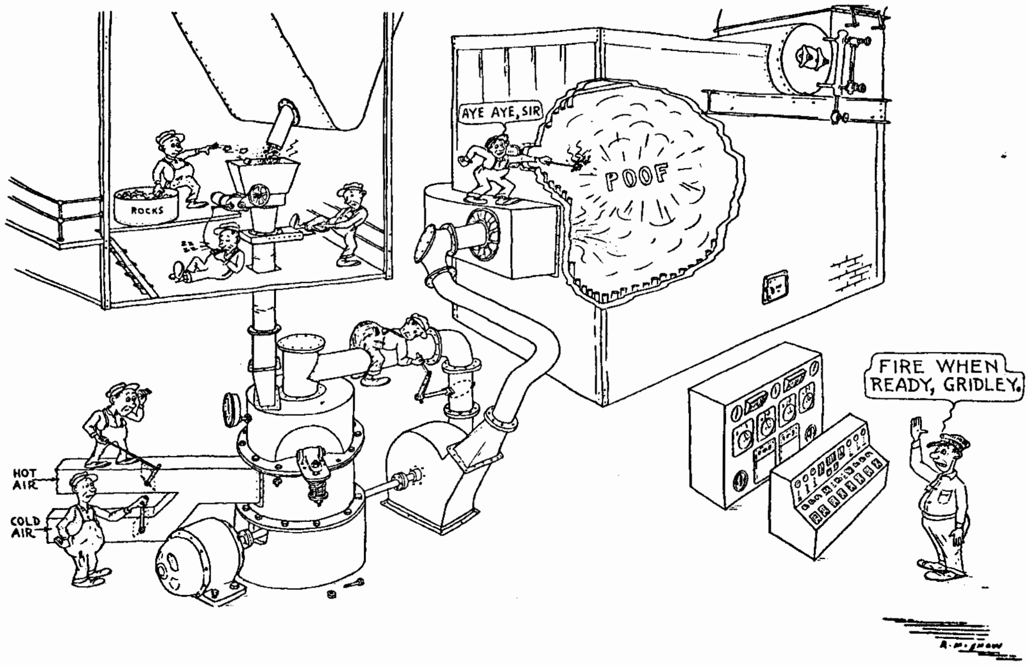
The desirability of any variable Btu fuel techniques is related to the amount of process interaction that is involved. The requirement is that the correction or compensation be handled within the control system without involving process temperature or pressure changes in order to obtain the necessary correction.

The technique described and shown in Figures 21-17 and 21-18 results in some process interaction since the process feels the Btu change and sends "after the fact" information on % oxygen back to the control system. The interaction period is based on the analysis and control time period, a fraction of the total process time constant.

Another approach that may be used is to measure specific gravity and use its value as a Btu value. In many cases this can be done because of the straight line relationship between specific gravity and Btu content of hydrocarbon gases. Whether this approach would be better depends on the time response of the specific gravity measuring instrument.

An emerging technique, which appears to be fast and thus not interactive with the process, uses two fuel measurements in series. One measurement is an orifice-type meter that is affected by specific gravity. The other meter is a volumetric meter such as a vortex shedding meter that is not affected by specific gravity. The result of a continuous calculation using a formula involving these two flows, the specific gravity-Btu value relationship, and the relationship to combustion air requirements can provide an immediate and proper total Btu feedback to the fuel flow controller.

Other techniques for controlling the combustion of variable Btu fuel are more complex than that shown in Figures 21-17 and 21-18. Among these are correction of fuel flow to mass flow (including specific gravity correction) and unit Btu measurement and compensation of fuel flow values. The measurements required have some time constants, and their compensations should be fast enough so that they may correct the fuel flow before any appreciable disturbance to the process. When these variables cannot be measured, a slower method of using the boiler as a calorimeter may be used. This method is suitable for longer term fuel Btu variations and will be discussed in the section covering control systems for coal-fired boilers.


ROCKS
AYE AYE, SIR
POOF
FIRE WHEN READY, GRIDLEY.
HOT AIR
COLD AIR

OK, LET'S TALK!

Section 22
Pulverized Coal and Cyclone Coal Burning Systems

In pulverized coal-fired boilers the coal is ground by a coal pulverizer to a fine powder and blown into the furnace. From a control standpoint the burning of pulverized coal is similar to that of a gaseous fuel with rapid heat response to changes in fuel flow and minor or slow response to changes in air flow. A typical arrangement of the unit pulverized coal system is shown in Figure 22-1. As shown in Section 18, a pulverized coal burner is similar to a gas or oil burner with coal transported on a stream of primary air and with secondary air, the main body of combustion air, added at the burner. In the unit system shown, which is the present-day system of general use, the coal is ground as it is used. Depending on the type of pulverizer used, there is more or less pulverized coal storage in the pulverizer prior to burning. Each pulverizer serves more than one burner. This may require that orifices be placed in the coal-primary air pipes to each burner so that the coal flows to the different burners are balanced.

A typical unit pulverizer system contains four basic elements:

(1) The coal feeder
(2) The pulverizer and classifier
(3) The primary air flow supply or exhauster fan
(4) The pulverized coal drying system

Cyclone furnaces are also used to burn coal in electric utility boilers. In these systems the coal is crushed instead of pulverized. The same type of coal feeder used with coal pulverizers is used with cyclone furnaces.

22-1 The Coal Feeder

The coal feeder feeds the raw coal from the overhead bunkers to the pulverizer. The coal should be fed to the boiler in approximate synchronization with the rate at which it is burned in the furnace. There are two basic types of feeders: the volumetric feeder and the gravimetric feeder.

A volumetric feeder feeds coal by volume. As the speed of the feeder increases, the volumetric rate of coal feed increases. Several different types of volumetric feeders are shown in Figure 22-2. The weakness of the use of the volumetric feeder is that the bulk density of the coal may vary, resulting in variation in the weight of coal fed and thus variation in total Btu to the furnace. This is an appreciable variation since the bulk density for most coals (different for lignite) ranges from approximately 38 lbs/ft^3 to 49 lbs/ft^3 for a moisture reduction range of 6 percent. This represents a $\pm$ 18.3 percent variation in total Btu feed for the same volume of coal.

The gravimetric feeder is both a weight flowmeter and a feeder that feeds coal to the pulverizer on the basis of coal weight. The weight of coal feed to the boiler is directly proportional to the control signal to the coal feeder. Prior to the 1980s, the typical gravimetric feeder arrangement was as shown in Figure 22-3. A belt weighing device inputs to a servomechanism that adjusts the leveling bar to maintain 100 lbs of coal on the belt at all times. The weight of the coal fed is thus directly proportional to the speed of the belt. The speed of the belt is thus a coal weight flow rate and the height of the leveling bar indicates the coal bulk density.

The newer type of gravimetric feeder uses a fixed-height leveling bar and electronic load cell for weighing the belt. The coal feed is the product of belt weight multiplied by belt speed.

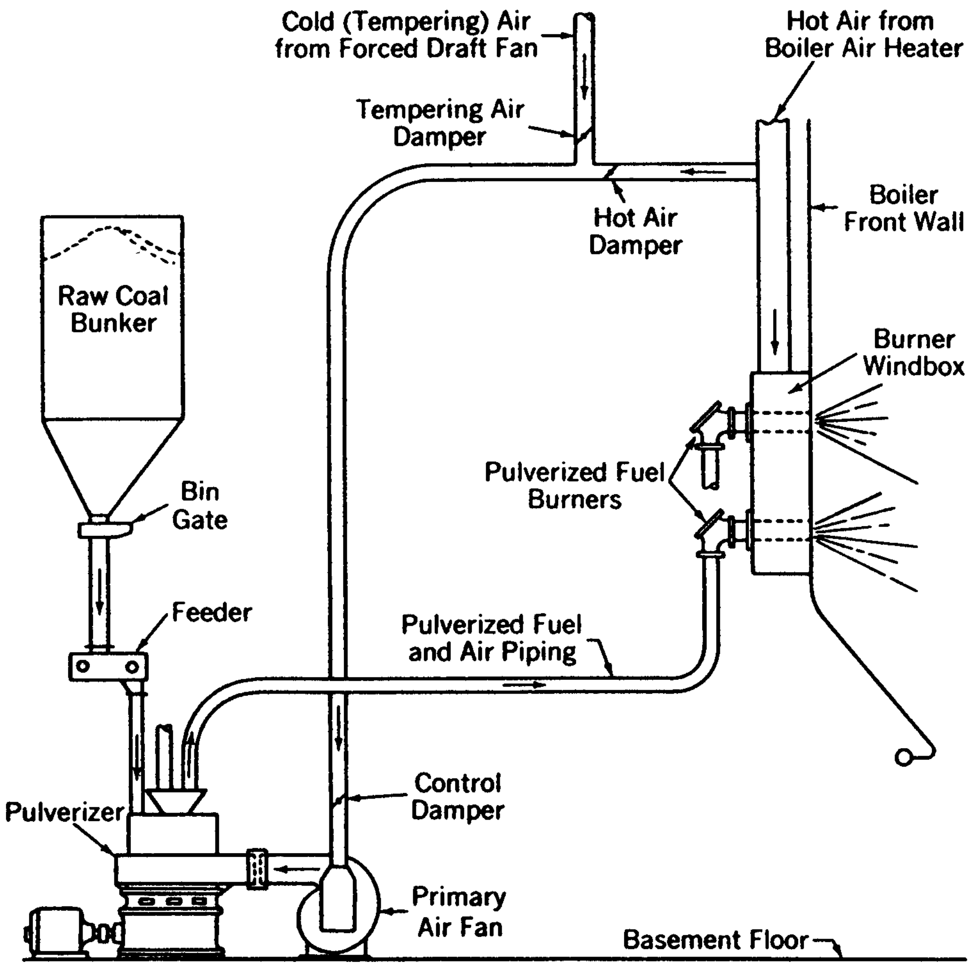
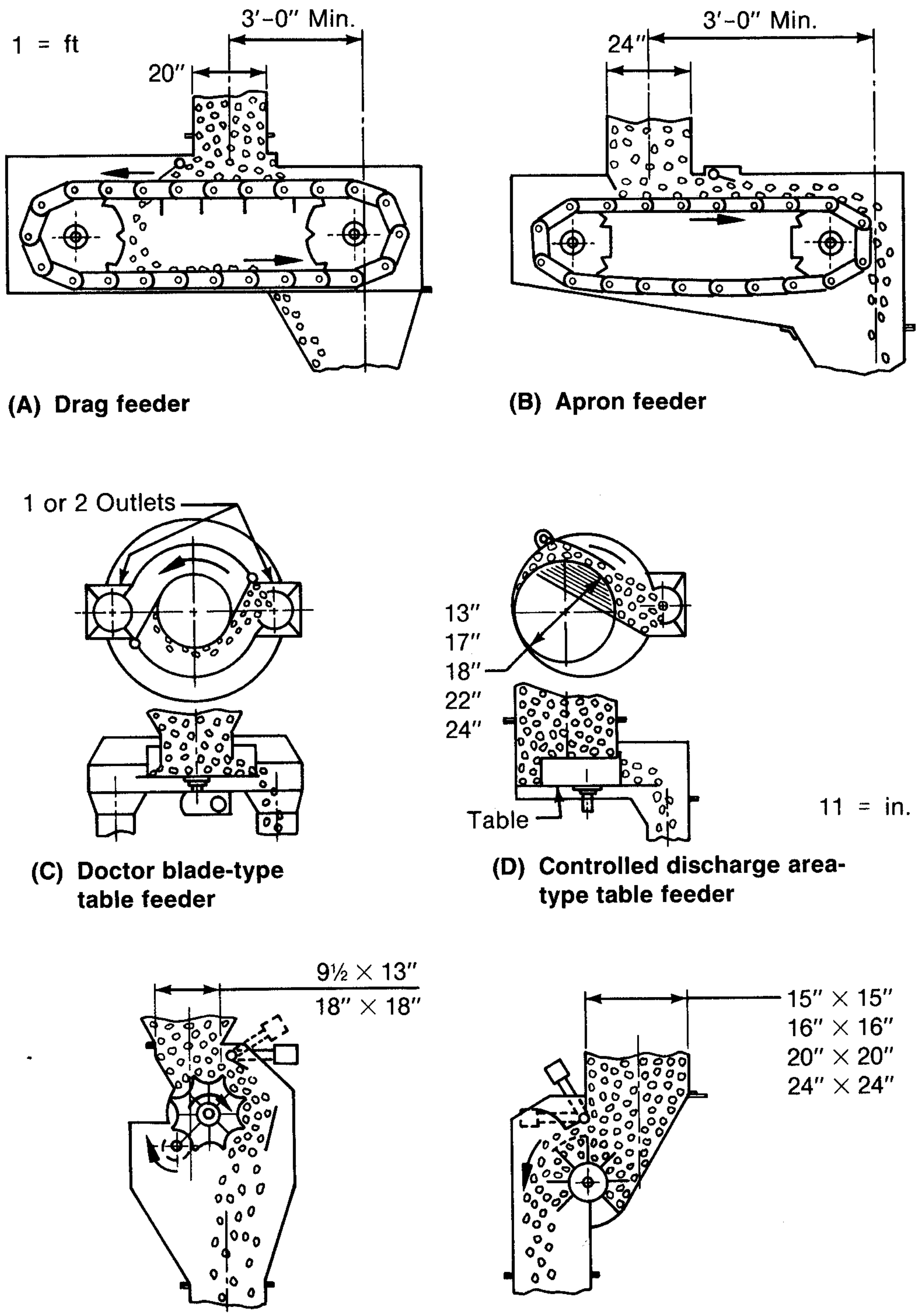


Figure 22-1 Direct-Firing System for Pulverized Coal
(From *Steam, Its Generation and Use*, © Babcock and Wilcox Co.)

With the moisture vs. bulk density relationship above and for a constant coal weight feed, the total Btu input variation is assumed to be $\pm$ 3%, the variation in the coal moisture content. On this assumed basis, a typical comparison between a gravimetric and volumetric feeder is shown in Figure 22-4. If the combustion air does not change, these variations in Btu feed to the furnace can create changes in total air for combustion and adversely affect both steam pressure and temperature. At high pressures and temperatures of steam, the Btu value changes very little due to pressure variations but is directly affected by temperature variations.

The basic benefit of a gravimetric feeder is that it can compensate for variations in bulk density and thus volumetric Btu content that arise from changes in coal moisture.

The gravimetric feeder does not compensate for Btu input variation that results from changes in ash content of the coal. While an increase in moisture content causes the bulk density to decrease, an increase in ash content usually causes it to increase. If the coal is of poor quality with wide variation in ash content, the effect of such variation on total Btu input to the furnace can be greater with a gravimetric feeder than with a volumetric feeder. The decision on whether



(A) Drag feeder

(B) Apron feeder

(C) Doctor blade-type table feeder

(D) Controlled discharge area-type table feeder

(E) Self-cleaning pocket feeder

(F) Stationary drum picket feeder

Figure 22-2 Different Types of Volumetric Coal Feeders

(From *Stock Equipment Co.*, technical paper)

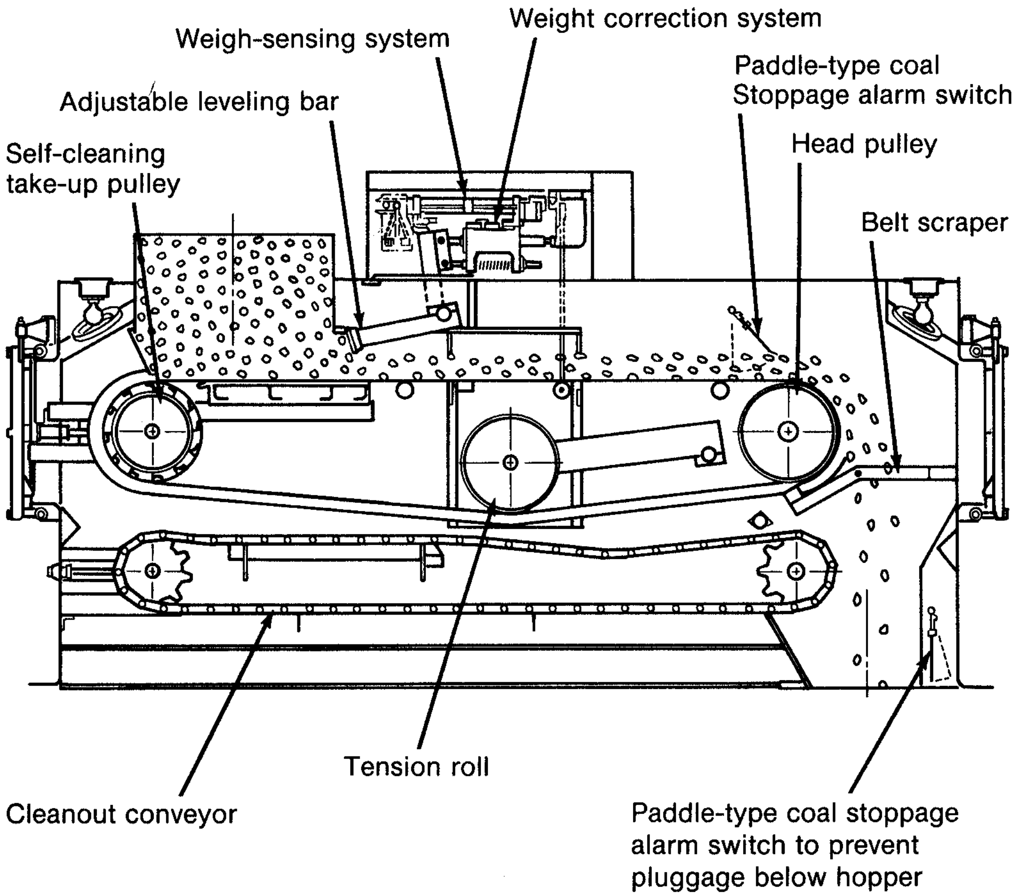


Figure 22-3 Gravimetric Coal Feeder, Belt Type

(From *Stock Equipment Co.*, technical paper)

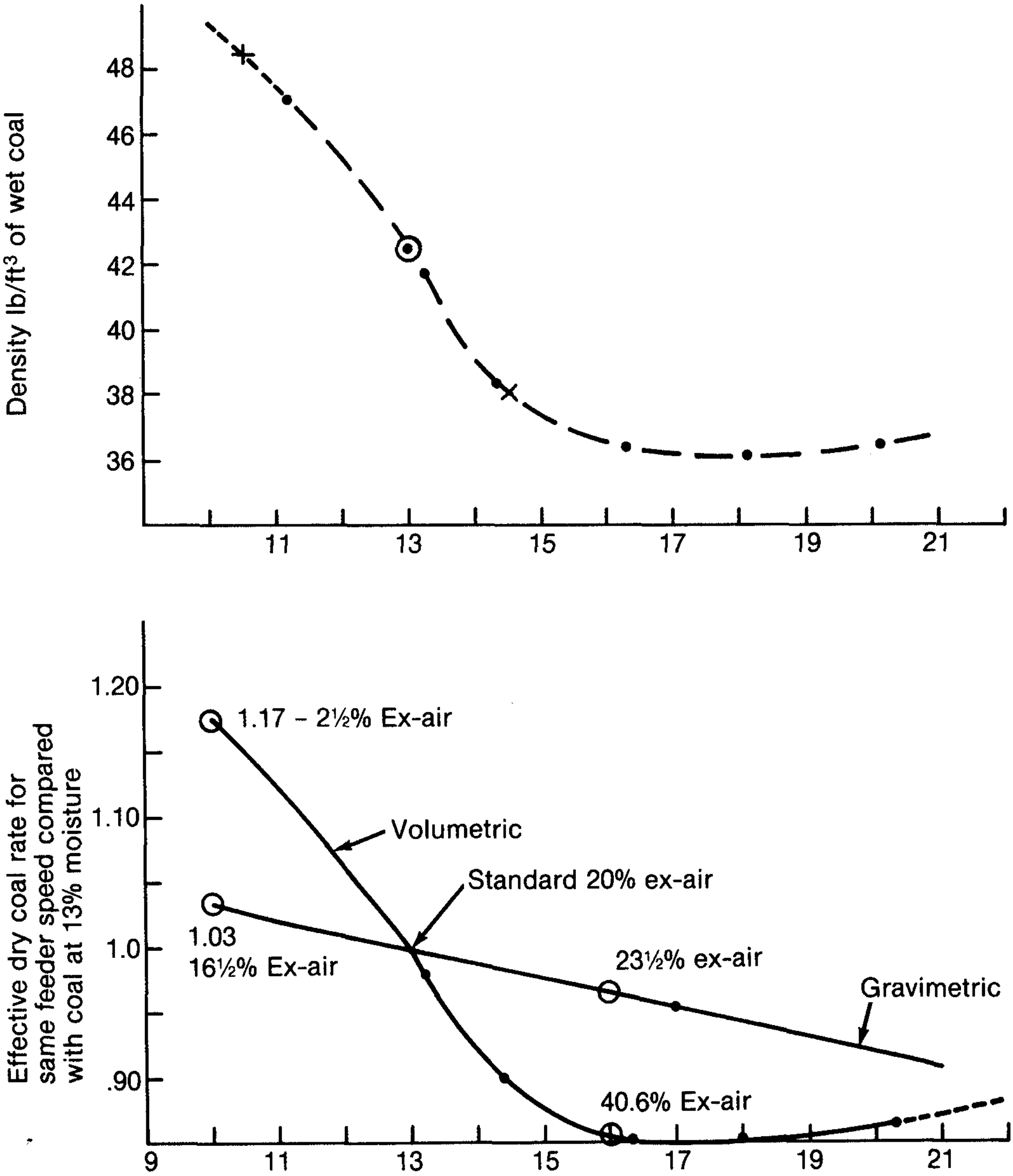
to use gravimetric or volumetric feeders in the control strategy should take into consideration the variation of both ash and moisture.

22-2 The Pulverizer and Classifier

The pulverizer is an electric motor-driven rotating mechanical grinding mechanism that crushes the coal between rotating balls and a race, as in Figure 22-5; between a roller and a bowl, as in Figure 22-6; between tumbling steel balls in a cylinder partially filled with the balls, as in Figure 22-7; and various types of hammer or impact grinding mills, as in Figure 22-8. Some coals require a particular type of grinding mill. A good example is that ball mills are more successful in the pulverization of very hard or abrasive coals such as anthracite or meta-anthracite coals.

In some pulverizers, the fineness of the grinding is determined by spring force on the grinding surface devices. In other pulverizers, centrifugal action applies greater contact force between the grinding surfaces as rotational speed is increased. The classifier rejects the larger particles back to the pulverizer. The ground coal size is thus determined by the size that is rejected by the classifier. In a typical installation the coal fineness is in the range of 70 to 80% through a 200-mesh screen.

Coal is continuously fed to the pulverizer and primary air is continually passed through



LIVE GRAPH
Click here to view

Figure 22-4 Comparison of Excess Air Effect for Gravimetric and Volumetric Coal Feeders

(From *Stock Equipment Co.*, technical paper)

the pulverizer to pick up the coal that has been pulverized. The amount of pulverized coal that is lifted from the pulverizer depends on the fineness of the pulverized coal, the level of coal in the pulverizer, and the square of the primary air flow. The turndown control range for a pulverizer is approximately 2:1. Typically, excessive coal moisture or hardness may reduce a pulverizer's capacity to as low as 50% of its rated capacity.

Because of these limitations on turndown and capacity, plus the fact that pulverizers are built in standard sizes, a pulverized coal boiler installation will usually require multiple pul-

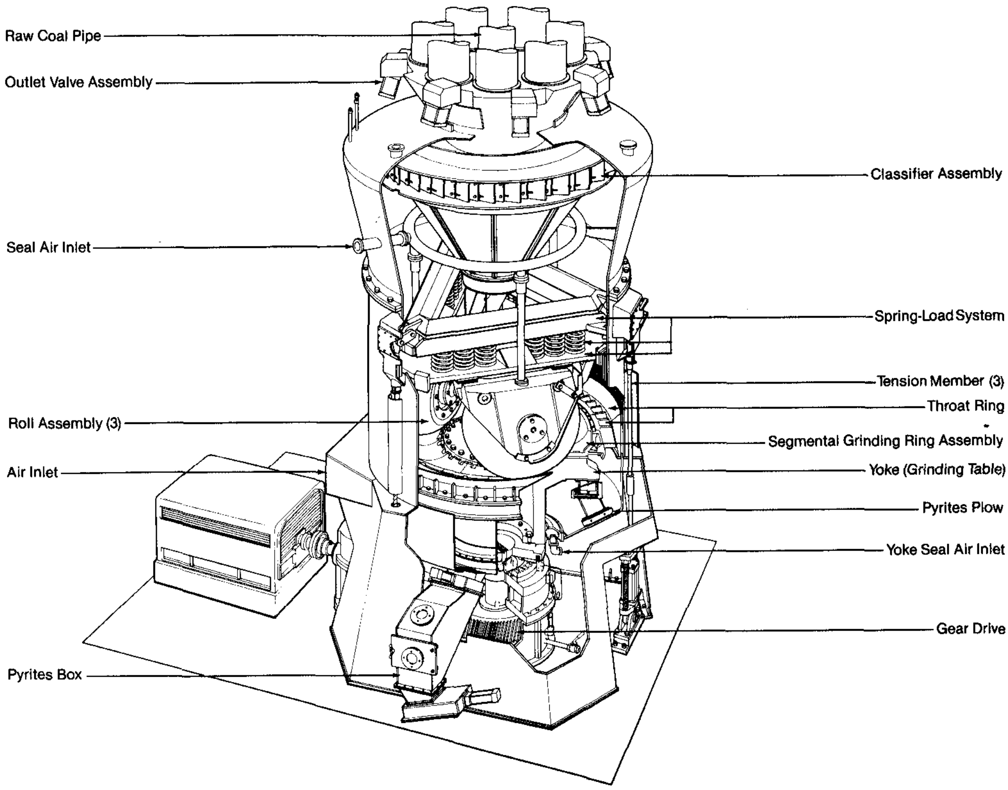


Figure 22-5 Medium-Speed Pulverizer, Roller Race Type

(From *Babcock & Wilcox, Steam, Its Generation and Use*, 40th edition)

verizers. Because the pulverizer is a heavy duty piece of grinding machinery that is subject to considerable wear and maintenance requirements, a typical installation will include an additional installed spare pulverizer.

As the coal is ground and lifted in the air stream, it first passes through the classifier. In the classifier, centrifugal action is imparted to the stream through static or rotating elements. The larger and heavier particles are thrown to the outside and returned to the pulverizer for further grinding.

From a control standpoint, there are different ways in which the same pulverizer can be controlled. There are, however, some generally accepted methods by which particular pulverizers are controlled. The differences are based on the manufacturer's control philosophy, whether the pulverizer is pressurized or under negative pressure, the type of pulverizer (which determines the amount of pulverized coal storage in the pulverizer), and whether the pulverizer is operated at a fixed or variable speed.

22-3 The Primary Air Fan or Exhauster Fan and the Coal Drying System

The source of the air flow that carries the pulverized coal stream from the pulverizer to the burners is a primary air fan or an exhauster fan. If the pulverizer is under air pressure, the fan pumps clean air to the pulverizer, through it and to the burner. If the pulverizer is a type that is operated under negative pressure, the exhauster fan is between the pulverizer and burners. Exhauster fan systems can also be used with pressurized pulverizers. An exhauster fan pumps the primary air-pulverized coal mixture from the pulverizer to the burners. A separate exhauster fan is necessary for each pulverizer. Because of the abrasive characteristic of the

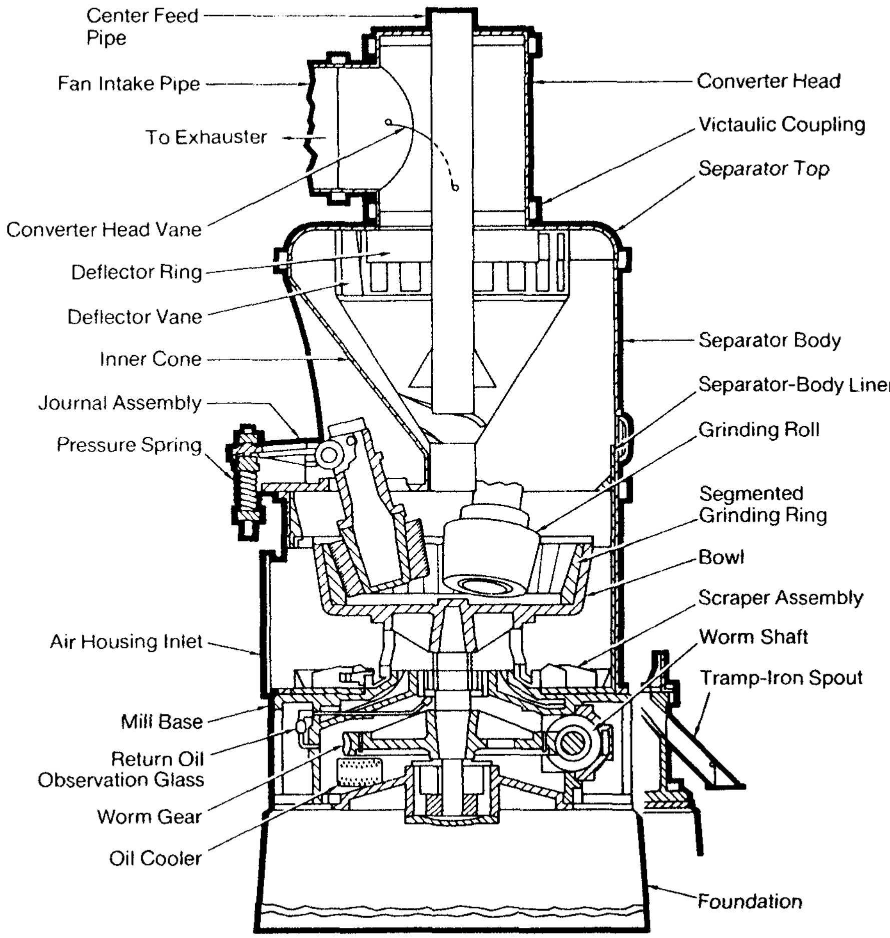


Figure 22-6 Medium-Speed Pulverizer, Roller-Bowl Type

(From *Fossil Power Systems*, © Combustion Engineering, Inc.)

pulverized coal, an exhauster fan must be of rugged construction and of simple and less efficient design to stand up to the wear on the fan parts.

For pressurized pulverizers the primary air flow for both types of fans is controlled on the clean air side of the pulverizer. The air supplied to the pulverizer is a mixture of hot, preheated air from an air preheater and relatively cold ambient temperature air, called tempering air. While the amount of coal lifted from a pulverizer at constant coal level is approximately proportional to the square of primary air flow, changing the coal level also has a very significant effect on the coal lifted in the air stream. The change in pulverizer coal level is a function of the difference in flow rate between raw coal input and pulverized coal output.

The coal is dried by the primary air stream mixture of cold and hot air. If the coal contains more moisture, then more hot air and less tempering air are used in the mixture. A temperature sensor that measures the temperature of the pulverized coal-air mixture feeds back to control the relative amounts of preheated air and tempering air. The controlled set point of the coal-air mixture temperature is usually in the range of 140-170°F. If the set point is too high, a hazardous condition may develop from pulverized coal that is too dry. If the set point is too low, the coal may build up on surfaces of the system, because it is too damp to flow cleanly and freely in the coal pipes.

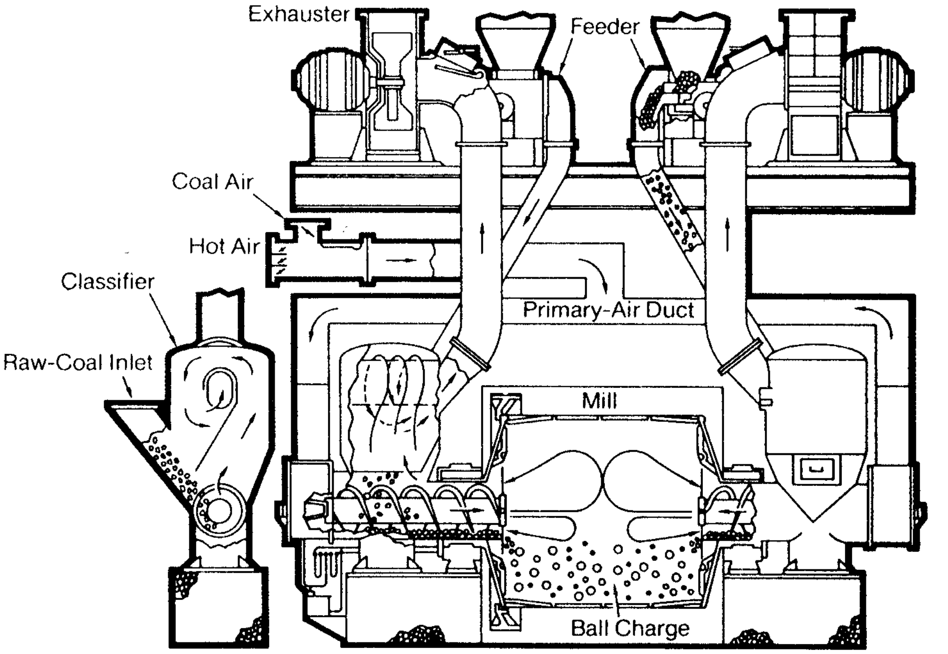


Figure 22-7 Slow-Speed Conical-End Pulverizer with Segregated Ball Size Indicated

(From *Fossil Power Systems*, © Combustion Engineering, Inc.)

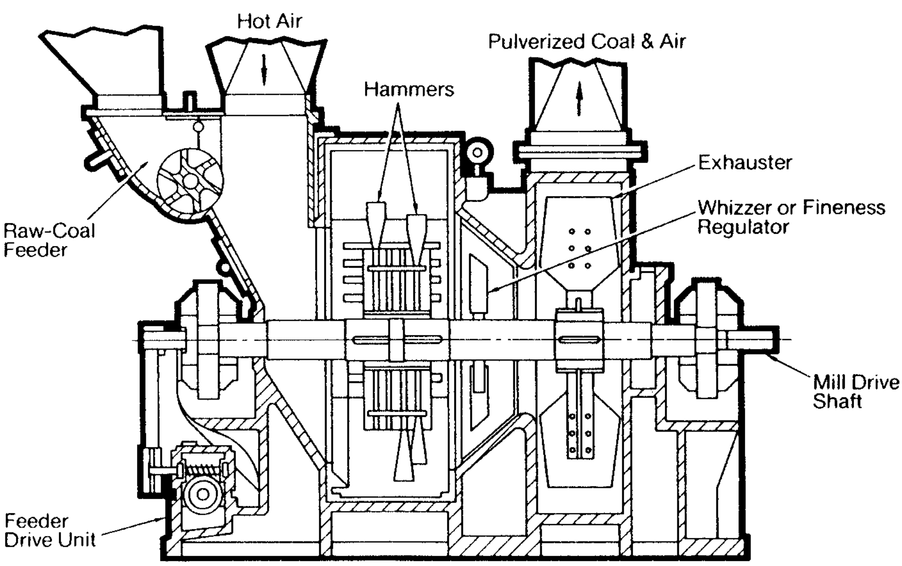


Figure 22-8 High-Speed Pulverizer, Impact Type

(From *Fossil Power Systems*, © Combustion Engineering, Inc.)

Just as in handling all volatile fuels, there is some hazard in pulverizing and drying the coal. Since the mixture is in the combustible range, a potential problem of fires in the pulverizers or coal pipes may exist. An emerging fire detection technique is to analyze the mixture of pulverized coal and primary air for the presence of carbon monoxide or other combustion produced gases. The presence of these gases indicates some burning taking place. Fires are extinguished by steam or inert gas blanketing.

22-4 Pulverizer Control Systems

A block of control logic is needed for each individual pulverizer. This control logic block contains all the modulating control functions for controlling the pulverizer. These functions are the coal-air mixture temperature control, the coal feeder control, and the primary air flow control. As indicated above, different control schemes are normally used for different types of pulverizers. Some of the blocks of control logic that are used for the well-known pulverizers are shown and discussed below. Digital logic that is necessary for start-up and shutdown sequencing and for safety interlocks is not shown but will be discussed later in this section.

Since the pulverizer manufacturers are normally required to guarantee such capabilities as capacity and fineness, the particular manufacturer should be consulted before finalizing the control concept.

A typical arrangement for the control of a modern Babcock & Wilcox ball and race or roller and race pulverizer and similar low storage pressurized pulverizers is shown in Figure 22-9. In this arrangement the demand for pulverized coal (a) originates as the boiler firing rate demand signal, which is further processed in the boiler combustion control system. Through the bias adjustment (b), the operator has the ability to bias the pulverizer load relative to that of other pulverizers.

The pulverizer coal demand (c) acts as the set point for the primary air flow controller (d). A primary air flow measurement (e) feeds back to satisfy the set point demand. This measurement also is processed through the proportional function (f) to limit the feeder speed to that proper for the available primary air flow. A minimum primary air flow is established through the high select (j) and manual signal (k). An actual feeder speed signal is sent from the pulverizer control logic block to the boiler control system, where it is summed with feeder speeds from other pulverizers.

The remainder of the control functions are implemented by the coal-air mixture temperature controller (h), which regulates the relative position of the hot air and tempering air dampers. A bias provision (i) is provided for adjusting the relative positions of the hot air and tempering air dampers. This control logic may be implemented by having the tempering and hot air dampers linked together in a particular fashion. All of the Babcock & Wilcox pulverizers use primary air fans with the pulverizer operating under pressure.

An earlier type of Babcock and Wilcox pulvizer used a simpler control strategy. The coal-air mixture temperature control uses the same general logic previously described, with implementation using a single control drive for both the hot and cold air damper. The fuel demand demand signal positioned the primary air damper (a). The damper flow characteristic was made as linear as possible with linkage angularity or a cam positioner on the positioner. The pulverizer feeder was a volumetric feeder of the table type, using an on-off two-speed motor to drive the feeder. The feeder was started and stopped by a contactor, which measured the balance between primary air flow differential pressure and a differential pressure derived from the level of pulverized coal in the pulverizer.

The special contactor is no longer available, and an improved method is now used for retrofitting the control of such pulverizers. The control stategy is shown in Figure 22-10. The two-speed motor of the feeder is replaced with a variable-speed motor (a). Instead of the contactor, a ratio control (b) between the two air pressure differentials (c) and (d) controls the

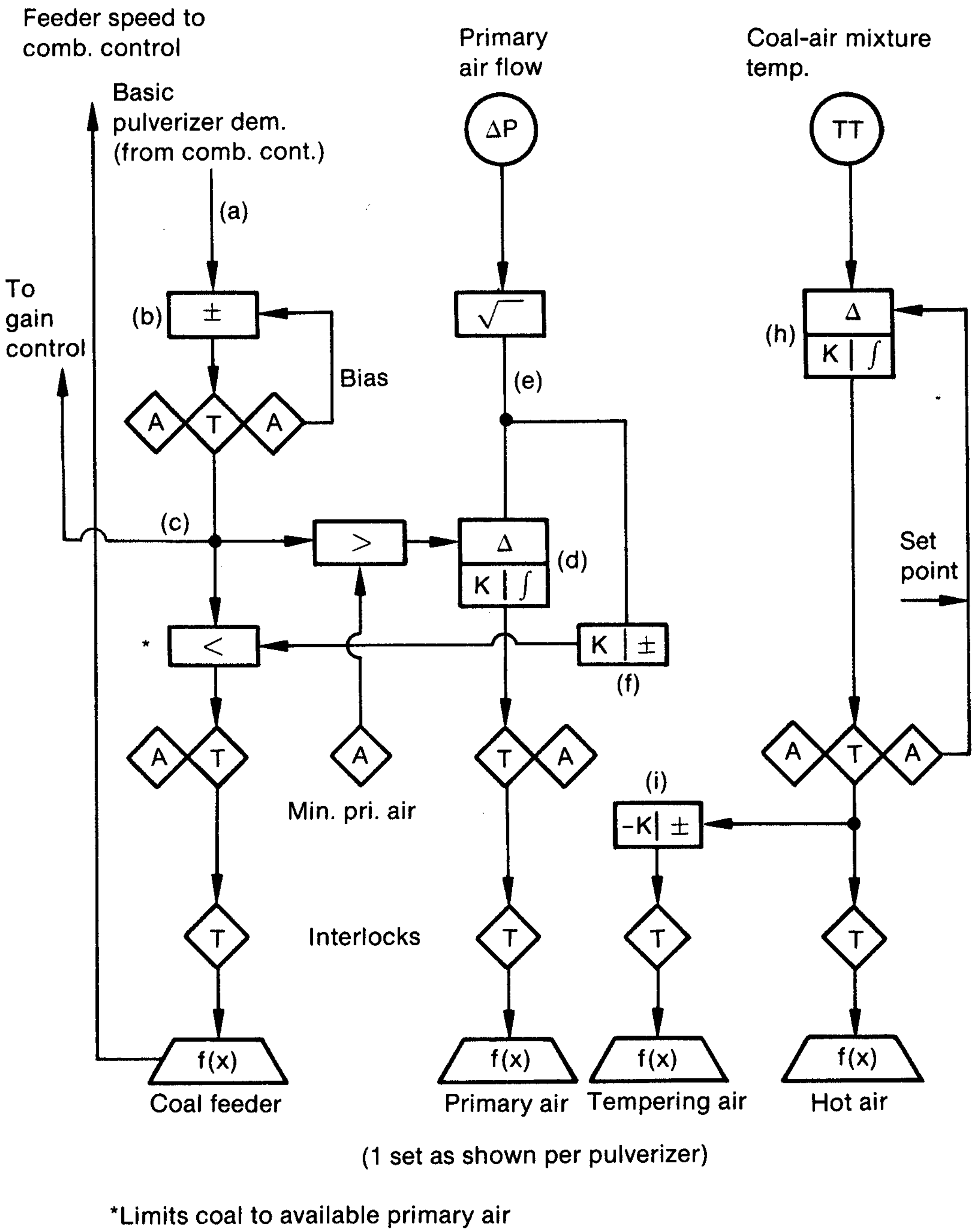


Figure 22-9 Control System for Babcock and Wilcox Low Storage Pulverizers

speed of the continuously operating variable-speed motor. When the pulverizer is furnished with new grinding balls, the ratio is adjusted and gradually readjusted as the balls wear.

Combustion Engineering pressurized pulverizers may be designed to operate under pressure or may be designed as draft systems. In Figure 22-11 the control logic for a pressurized system is shown. The basic pulverizer demand signal (a) can be biased by the operator using the bias function (b). This provides a means of balancing pulverizer loads and smoothly bring-

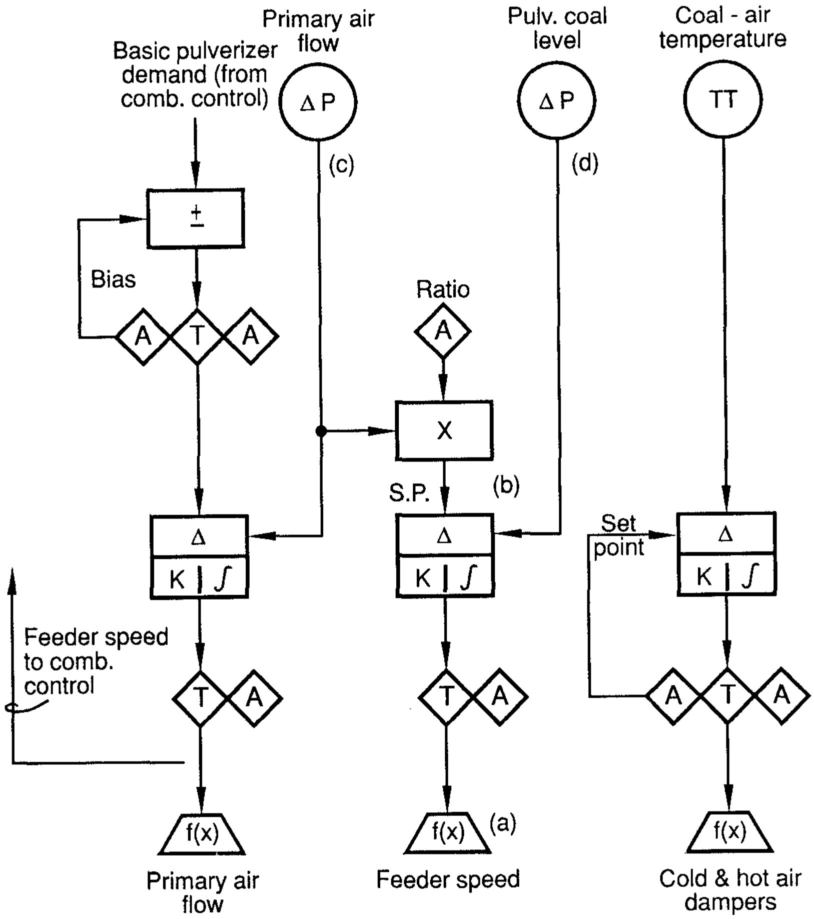


Figure 22-10 Control System for Babcock & Wilcox Pulverizer Using Table Feeder

ing the pulverizer on-line or taking it off-line. The resulting signal (c) positions the coal feeder directly and is further processed in proportional-plus-bias function (d) to a proper feedforward signal for the primary air flow. In this arrangement the primary air flow controller (f) is at a fixed set point, as adjusted by the operator, and is satisfied by the primary air flow feedback (g). The output of the controller is directed to the hot air damper and as a feedforward signal to summer (k) and then to the cold air damper.

The coal-air mixture temperature controller (i) regulates the temperature by changing the distribution of hot and cold air. As the temperature drops and more drying is needed, the hot air damper opens further and the cold air damper closes a like amount. In essence, both dampers control primary air flow and both dampers control coal-air mixture temperature. The tuning procedure for these control loops is that for typical temperature and flow control loops.

A signal of actual feeder speed is sent from the control logic block shown to the boiler control system. Note that since the primary air flow is constant, the pulverized coal flow is essentially a function of pulverized coal level in the pulverizer. Upon an increase, the feedforward from (d) temporarily draws down the pulverizer storage to obtain response. A new equilibrium mill coal level is established by grinding the additional coal fed to the pulverizer.

For a Combustion Engineering negative pressure pulverizer and other low volume pulverizers that are operated under suction rather than pressure, the control logic can be the simpler

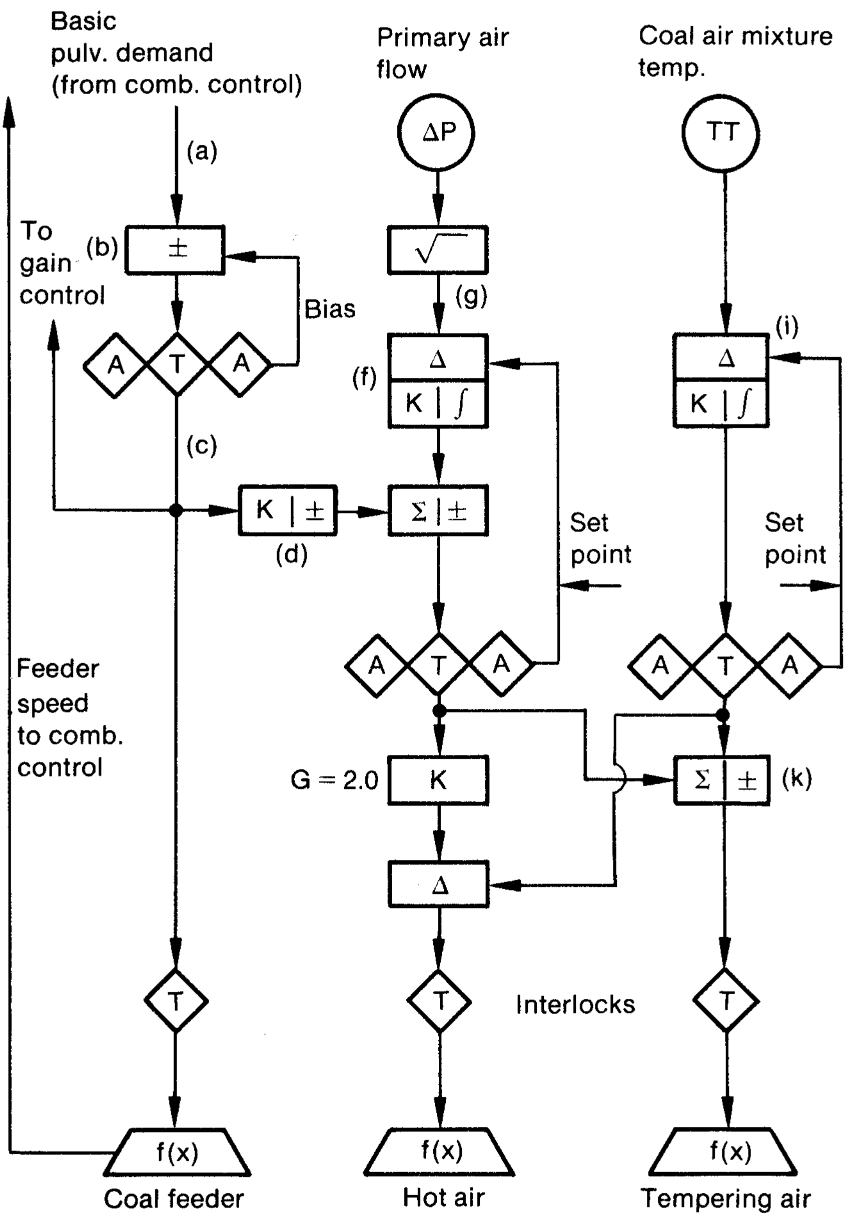


Figure 22-11 Control System for Combustion Engineering Pressurized Pulverizers

arrangement shown in Figure 22-12. In this case, the pulverizer demand signal (c) positions the feeder speed and the primary air (exhauster) damper in parallel. The tempering air is introduced through a mechanical balanced draft damper, which opens as a suction is created at the pulverizer air inlet. The coal-air mixture temperature controller (i) is a simple feedback controller that controls only the preheated air that mixes with the tempering air.

A typical Foster Wheeler pulverizer is of the ball mill type shown in Figure 22-7. Pulverizers of this type may have several minutes' storage of pulverized coal. At constant coal level they are relatively insensitive to immediate changes in the raw coal feed rate but have a significant amount of ground coal available for immediate response to load demand. Figure 22-13 shows how such a pulverizer may be controlled.

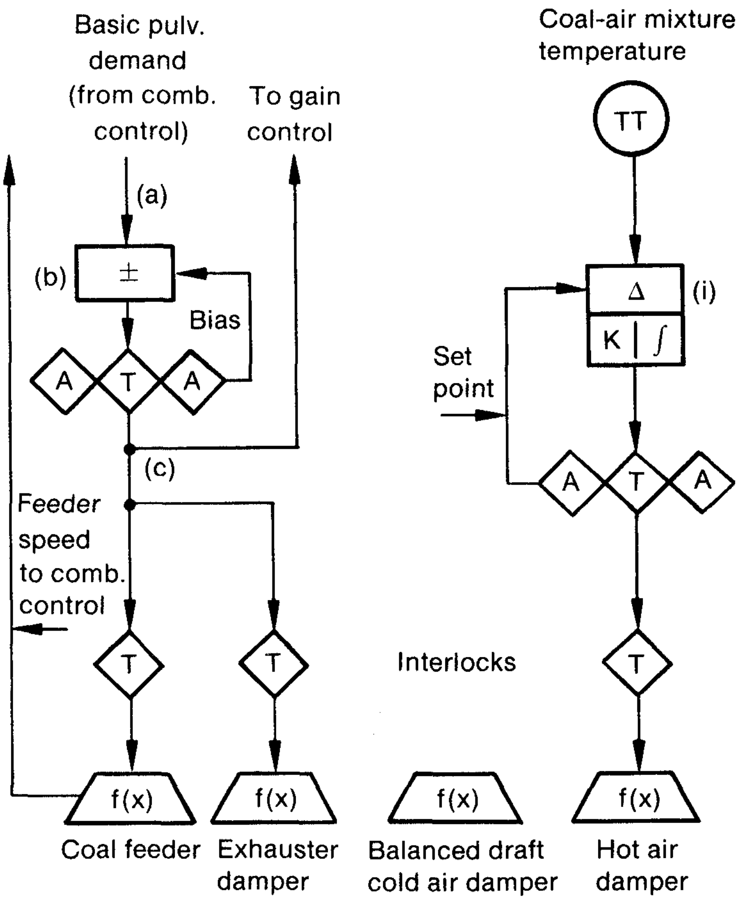


Figure 22-12 Control System for Combustion Engineering Negative Pressure Pulverizers

The coal level (a) in such pulverizers is controlled to a constant level and is sensed with an air bubbler or air leak-off probe. As the level changes, the coal level controller (b) signals a change in the coal feeder speed to satisfy the change in demand for coal. The demand signal for pulverized coal (c) is the set point for the primary air flow controller (d), which positions the damper on the exhauster fan. An increase in primary air flow lifts more pulverized coal as it passes through the pulverizer.

The classifiers are separate devices in the flow stream between the exhauster fan and the pulverizer. The classifier pressure differentials (e) are summed and used as a rough pulverized coal flow feedback to the combustion control system. It is assumed that these differentials are roughly proportional to the square of the primary air flow that lifts the coal from the pulverizer. Since pulverizer coal level and fineness are relatively constant, primary air flow alone is responsible for the flow of pulverized coal that leaves the pulverizer.

As the change in the exhauster damper changes the draft in the pulverizer, the draft controller (f) changes the tempering air flow. This action is similar to the action of the mechanical balanced draft damper of the Combustion Engineering pulverizer. Changes in the coal-air mixture temperature cause the temperature controller (g) to change the position of the hot air damper.

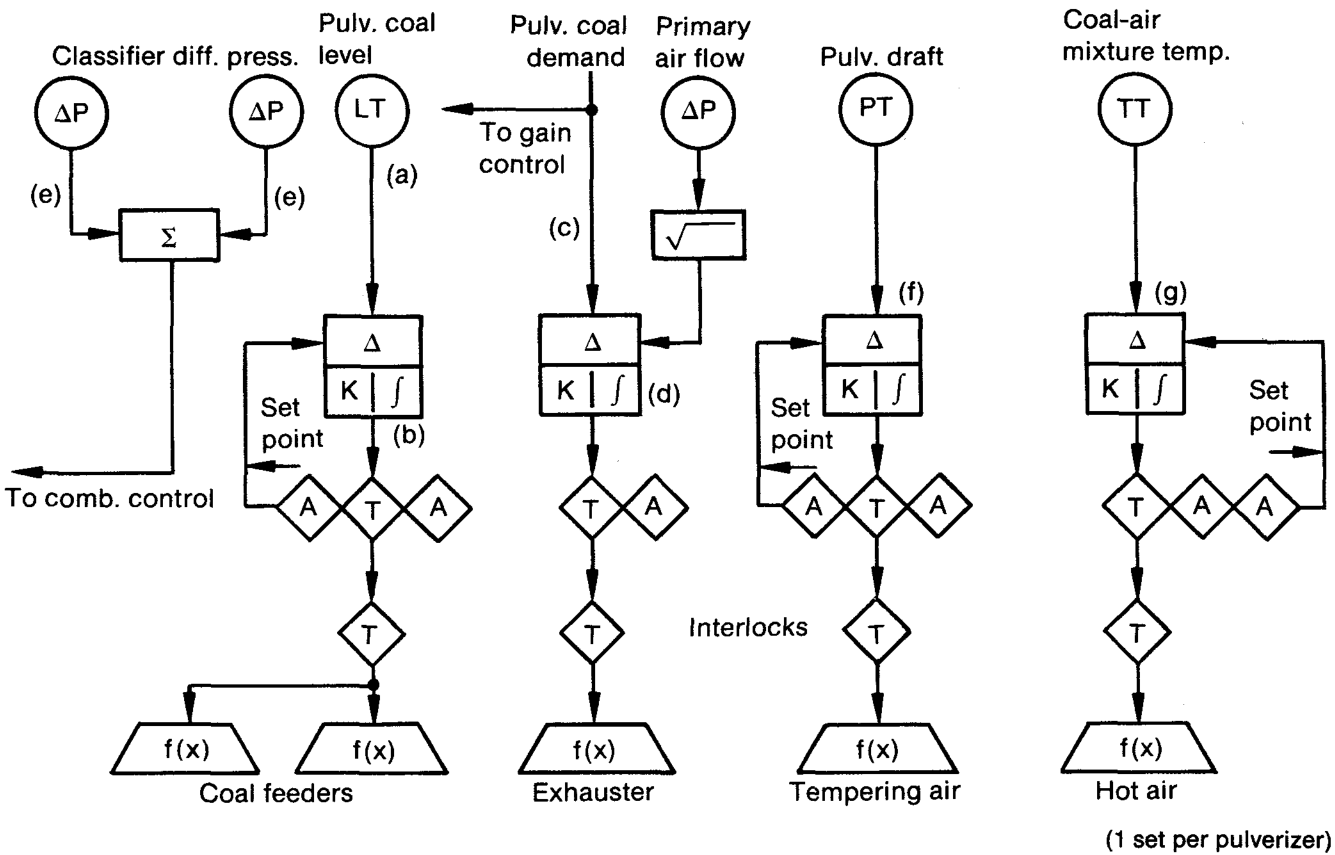


Figure 22-13 Control System for Foster Wheeler Ball Mill Pulverizers

Another variation for controlling this type of pulverizer eliminates the weakness of the classifier differential as a pulverized coal flow measurement. This arrangement is shown in Figure 22-14. The exhauster flow damper is positioned directly by the pulverizer demand signal (c). The coal level control of the pulverizer is a feedforward system. The signal to the exhauster is fed forward to the summer (h) as the preliminary control signal for the coal feeder, with the coal level controller (b) acting as a trim control. The input of the coal feeders is then in phase with the pulverized coal output from the exhausters.

If the feeder input causes a temporary rise in the coal level, the increase in the level causes the exhausters to deliver more pulverized coal. The feeder speed summation is then gained by the level error (i) in multiplier (g) before being used in the boiler control system. This shows that the exhauster coal flow is different from the raw coal feed because the level is in error. The calibrated actual feeder speed summation (j) is then the feedback to the combustion control system. The feedforward variation of the coal-air mixture and draft control shown here can be used with either of the basic control arrangements.

The Riley ball mill pulverizer is very similar to the Foster Wheeler system, but the control philosophy appears to be quite different, as shown in Figure 22-15. In the Riley system the coal feed can be controlled by pulverizer level, as in the Foster Wheeler system, by using the pneumatic probe type of pulverized coal level measurement. Riley also provides an alternative system (called Powersonic R) that uses inputs of sonic level, the sound level generated by the grinding process, and the kW rate of the pulverizer drive motor to control the raw coal feeders.

The flow rate of pulverized coal is determined by controlling the pulverizer differential (e) according to the demand for pulverized coal. This control adjusts the mill rating damper that changes the primary flow through the pulverizer. The summation average (e) of the pulverizer differential pressures can then be used as a rough pulverized coal flow feedback to the boiler combustion control system.

The classifier-to-furnace differential is controlled to a set point value by the action of controller (g). The set point is a function of the pulverizer fuel demand. The output of this controller regulates the mill bypass dampers to bypass a portion of the primary air flow around the pulverizer. Since this differential is affected primarily by the primary air flow, the primary air flow is approximately a constant value for all coal loads with a variable content of pulverized coal in the mixture. The flow through the pulverizer is a part of the flow through the classifiers and thus contributes to the classifier-to-furnace differential pressure. The control signal to the mill bypass dampers is therefore used as an input to summer (f) in order to act as a feedforward signal to the mill rating damper. As in the other systems, the coal-air mixture temperature, which is not shown, controls the distribution of hot and preheated air in the primary air stream.

The control arrangements shown are for the larger and more complex pulverizers that are typically used in electric utility plants. If simpler or smaller pulverizers are used, simpler control arrangements are generally adequate. Typically these mill controls resemble those for the Combustion Engineering Co. pulverizer in Figure 22-12.

All of the pulverizer control arrangements shown, or which may be applied, require one set per pulverizer. The boiler systems usually require three or more pulverizers for full load operation (two or more operating and one spare), although they may be operated with a fewer number at low loads. The boiler combustion control system should compensate for the number of pulverizers in use by automatically changing the gain of the pulverizer demand signal before it reaches the individual pulverizer controls.

All of these pulverizers or mills have some inherent time lag between the raw coal feed and the output of pulverized coal. This has no effect during steady-state operation but can create control problems due to delay of fuel response when boilers have rapid changes in load. The normal practice is to add derivative response in the combustion control system output signal ahead of the pulverizer control functions in order to improve the response.

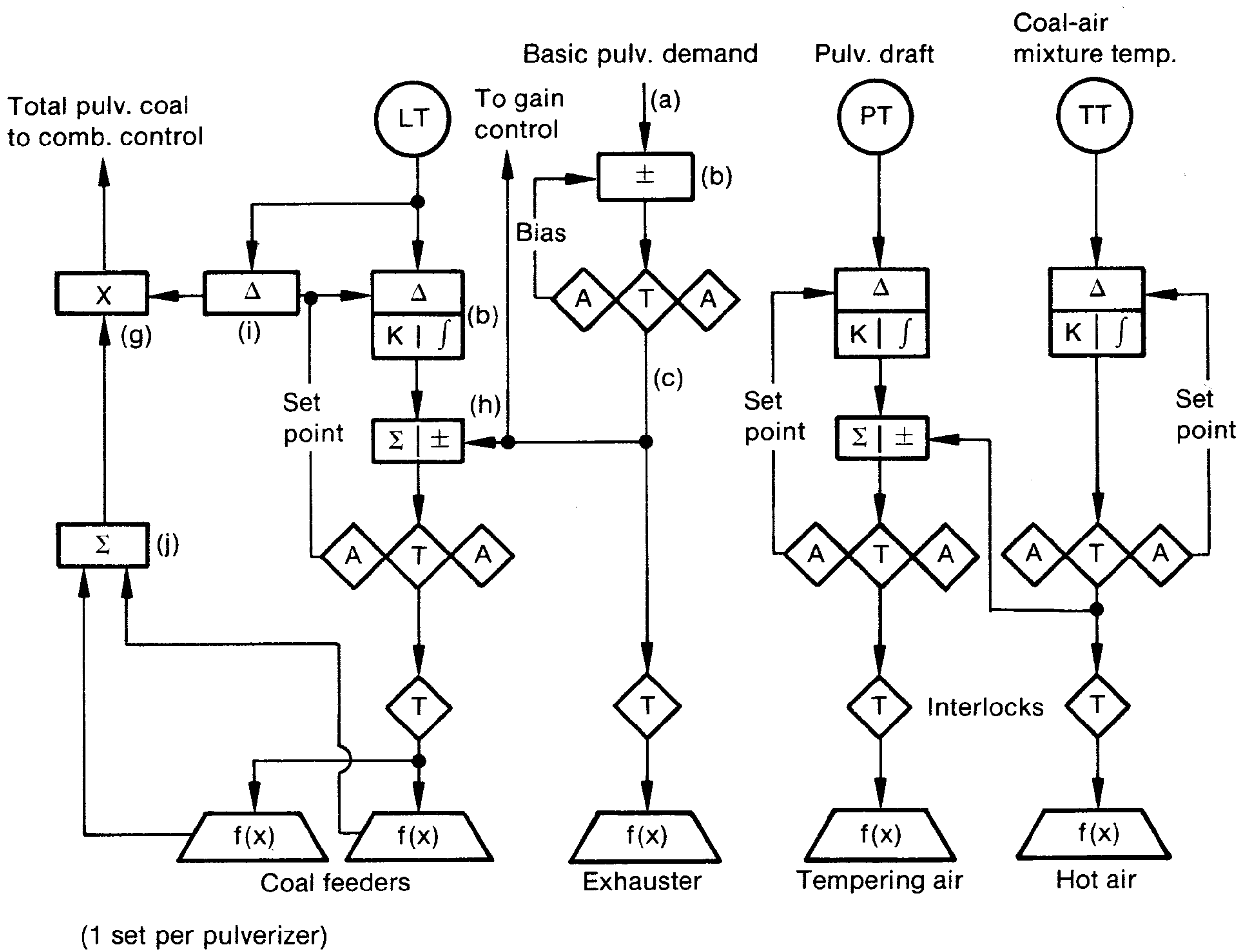


Figure 22-14 Control System for Ball Mill Pulverizers Using Gravimetric Coal Feeders

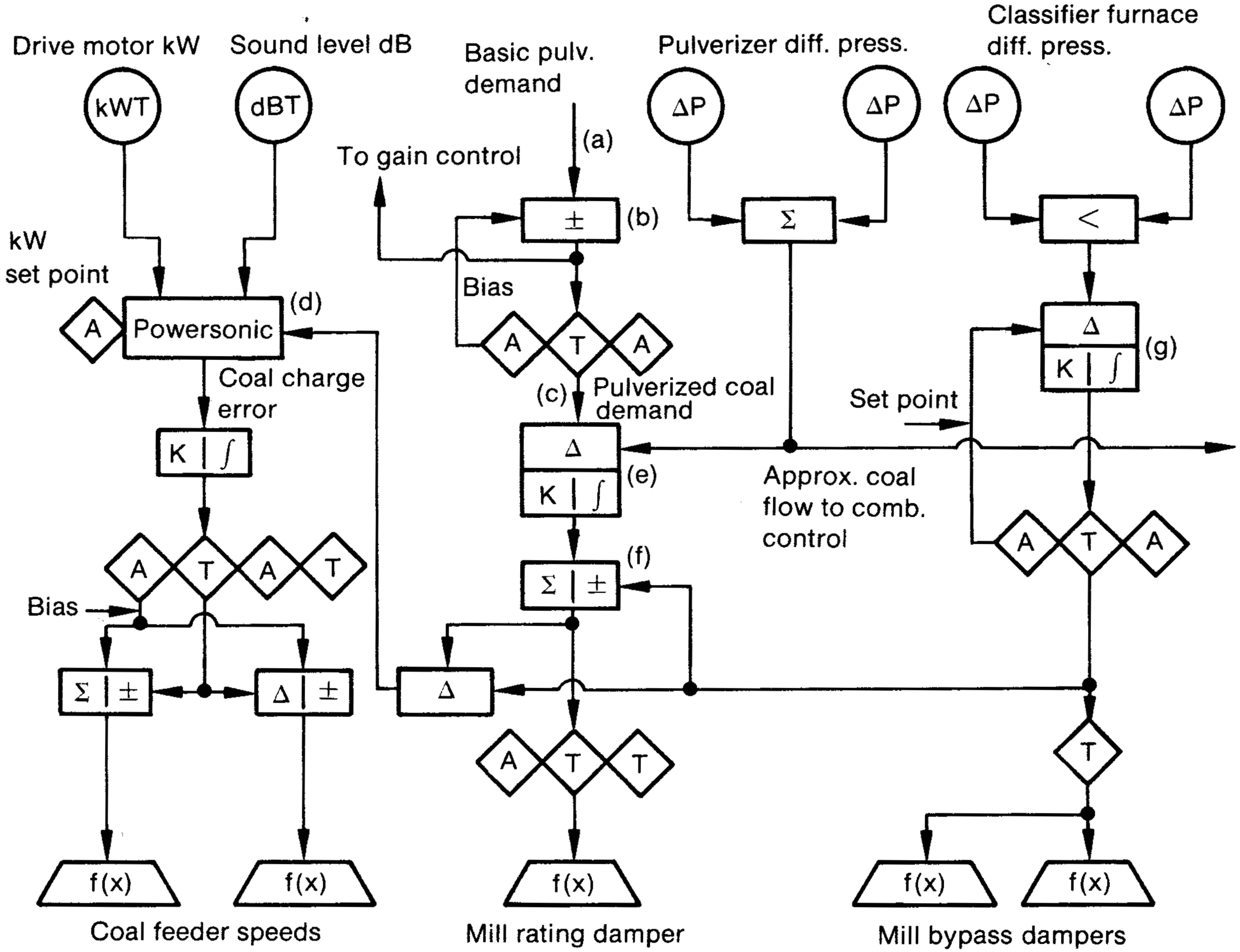


Figure 22-15 Control System for Ball Mill Pulverizer with Powersonic® (Riley)

22-5 Compartmented Windbox Pulverized Coal Boilers

Typical fluid fired boilers such as those burning gas, oil, or pulverized coal have a common windbox for all the burners. All of the combustion air (except where primary air is used) flows from the forced draft fan or the air preheater to this common windbox. By boiler design, burner location, and spacing, etc., the air flow is assumed to be uniformly split between burners. Generally this is reasonably correct but may not be precisely true at any or all loads. In addition, the flow of fuel to all burners is not necessarily uniformly split because of piping configuration, unbalanced pulverizer flows, and so on.

To allow for such lack of precision, excess air levels are often padded to avoid remnants of unburned gas leaving the furnace. Some final balancing of fuel and air ratio takes place in the furnace as streams of fuel-rich mixture are mixed with streams of air-rich mixture. A result is higher than desired levels of unburned gases and NOx.

In order to obtain better precision, in the 1970s Babcock & Wilcox Co. developed the compartmented windbox boiler. As shown in Figure 22-16, with this design the windbox is divided into the same number of compartments as there are pulverizers. Each pulverizer is connected to the burners of a particular windbox compartment. The air flow from the forced draft fan and air preheater feeds into a plenum chamber and is controlled to a set point pressure. From this plenum chamber air flow enters each windbox compartment through an air foil measurement and air flow control damper.

The overall arrangement is shown in Figure 22-17. The air flow to each compartment is controlled and shut off by the control dampers for each compartment. There are no air registers in the conventional sense. This requires some flow of cooling air through idle burners in order to protect burner parts, ignitors, and flame detectors from the high temperature radiation of the furnace. The air flow to each compartment is flow controlled in proportion to the coal flow feeding the pulverizer that is connected to the burners of that compartment. The air flow

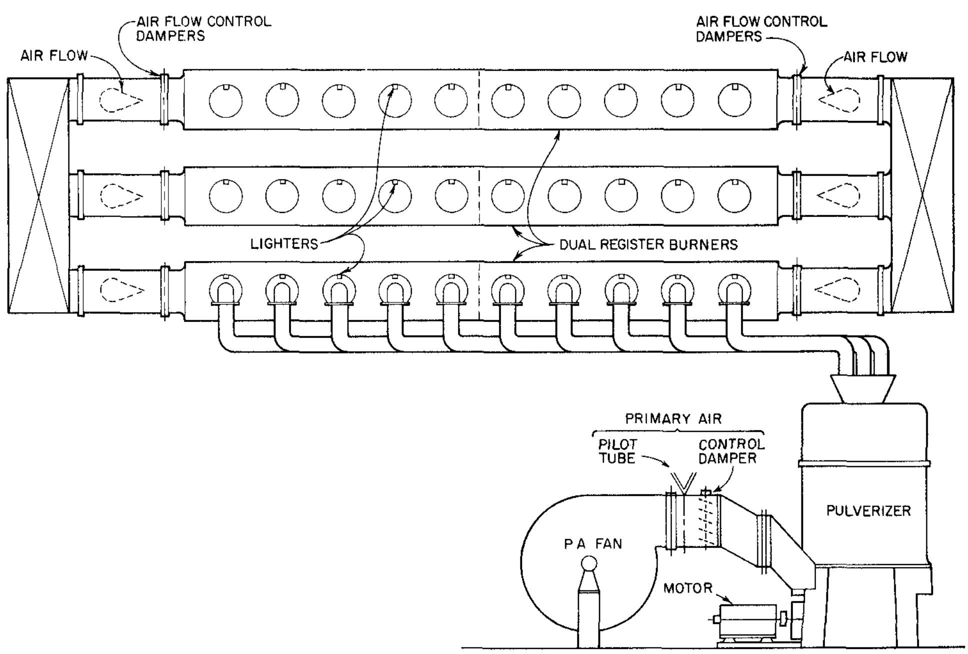


Figure 22-16 Arrangement of Pulverizers and Burners

(From Bailey Controls Co. Technical Paper)

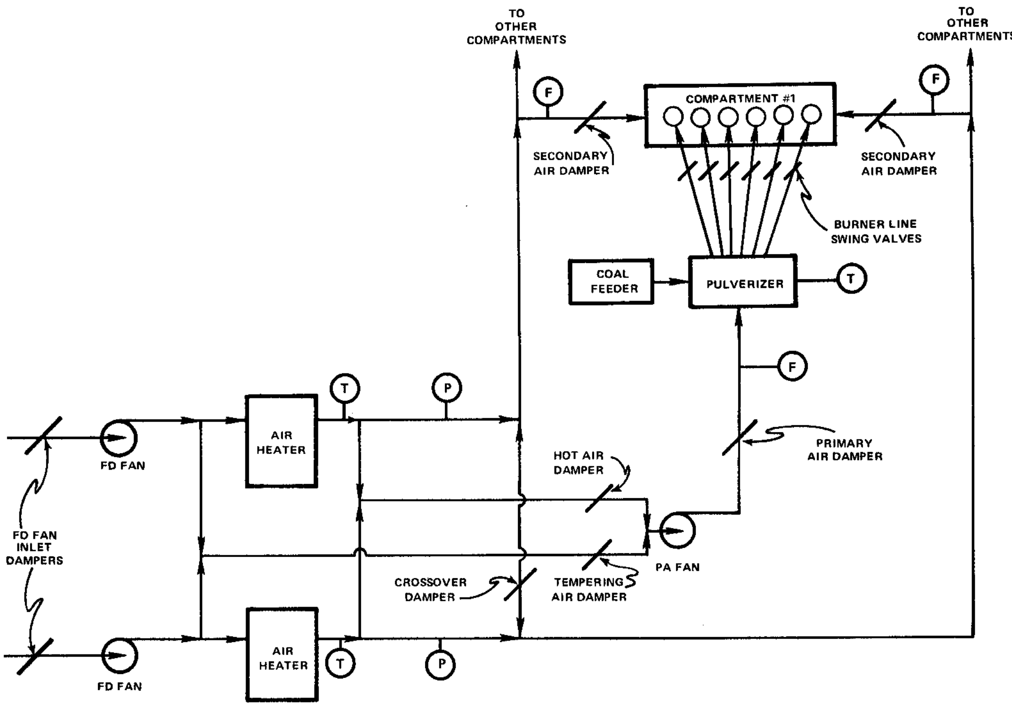


Figure 22-17 Compartmented Windbox Piping and Instrumentation

(From Bailey Controls Co. Technical Paper)

measuring air foils shown in Figure 22-16 are individually calibrated based on Pitot tube air foil velocity tests.

Essentially the control system tries to accurately control the fuel/air ratio of each individual pulverizer instead of the conventional approach of doing this on a total boiler basis. For precision of control, the primary air flow is accurately measured and totalled with the secondary air flow in the control loop controlling total air flow to the compartment.

In order to achieve lower NOx production, as shown in Figure 22-18, a specially designed burner is used with this type of arrangement. The accurate measurement of primary air flow allows the burner to be adjusted for a particular fuel-rich mixture for initial combustion and a

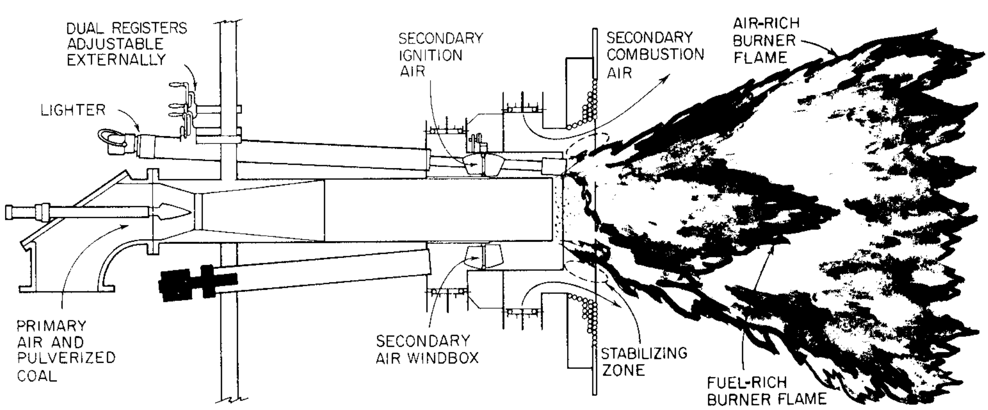


Figure 22-18 Dual Register Burner for NOx Control Pulverized Coal Firing

(From Bailey Controls Co. Technical Paper)

zone near the burner. Secondary air flow adjustments allow the shaping of the flame and the air-rich combustion zone for completing the combustion process. This is staged combustion, which is well known for reducing NOx in automobiles and other combustion processes.

The operation of the individual pulverizers and their controls is the same as if the windbox were not compartmented.

22-6 Start-Up and Management of Pulverizers and Their Burners

Pulverized coal-fired boilers are somewhat more difficult to start up than gas fired boilers. The valves of a gas-fired boiler have been replaced with pulverizers. The valves required only opening and closing, while an entire pulverizer start-up and operation logic is required for each pulverizer. There are very significant differences in the operation and start-up logic of pulverizers from different manufacturers and different models from the same manufacturer. This requires a wide variety of logic arrangements for starting up a pulverizer and placing it on-line.

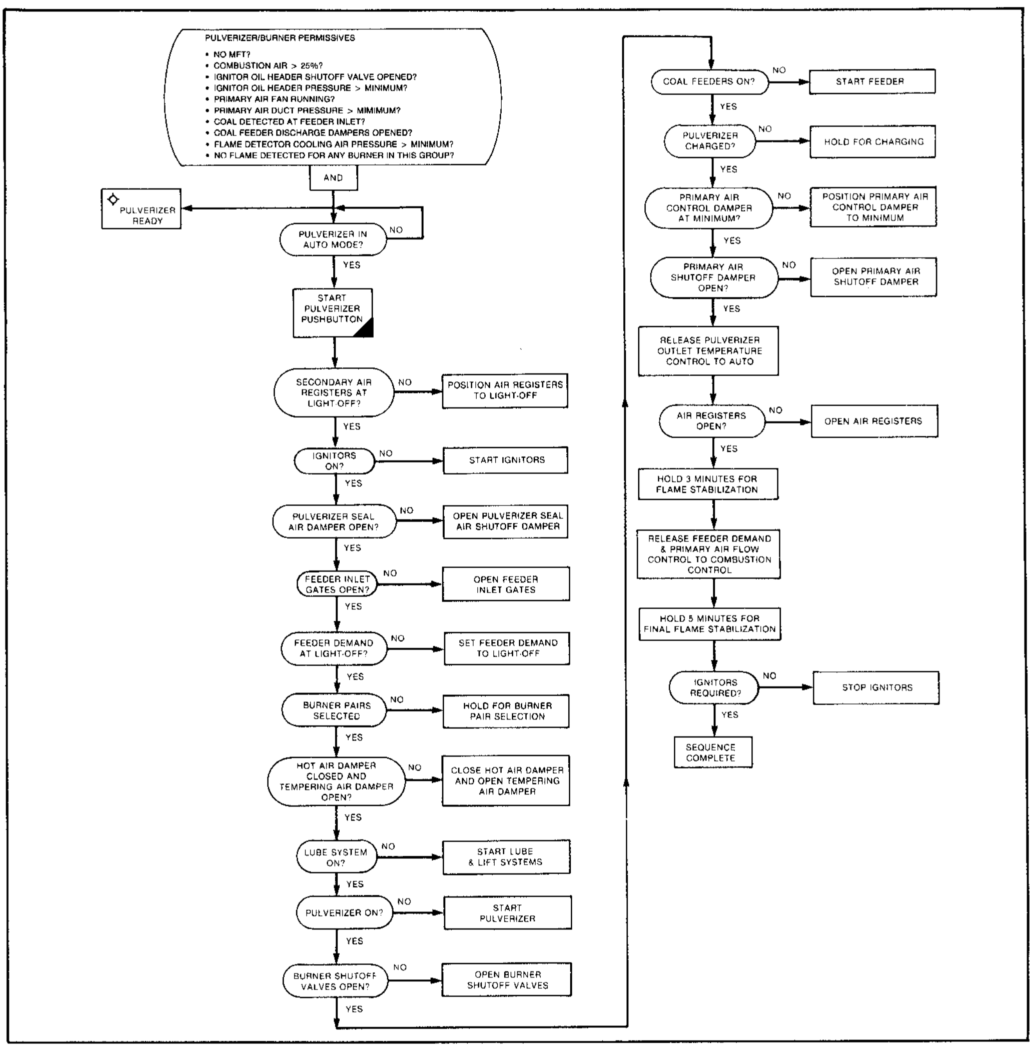


Figure 22-19 Typical Start Sequence for Foster Wheeler Ball Tube Pulverizers

(From Bailey Controls Co. Application Guide)

A typical pulverizer start-up control logic system includes a logic system for starting the ignitors, the primary air fan, the pulverizer motor, the coal feeder motor, and the burner line swing valve. In addition, other logic is required for purging, ignitor fuel header control, and other permissives. The permissives involved are measured; some are values from the modulating control system, and a number of the outputs include actions to the modulating control system equipment.

To demonstrate the wide variance of the logic, Figures 22-19 and 22-20 show the typical

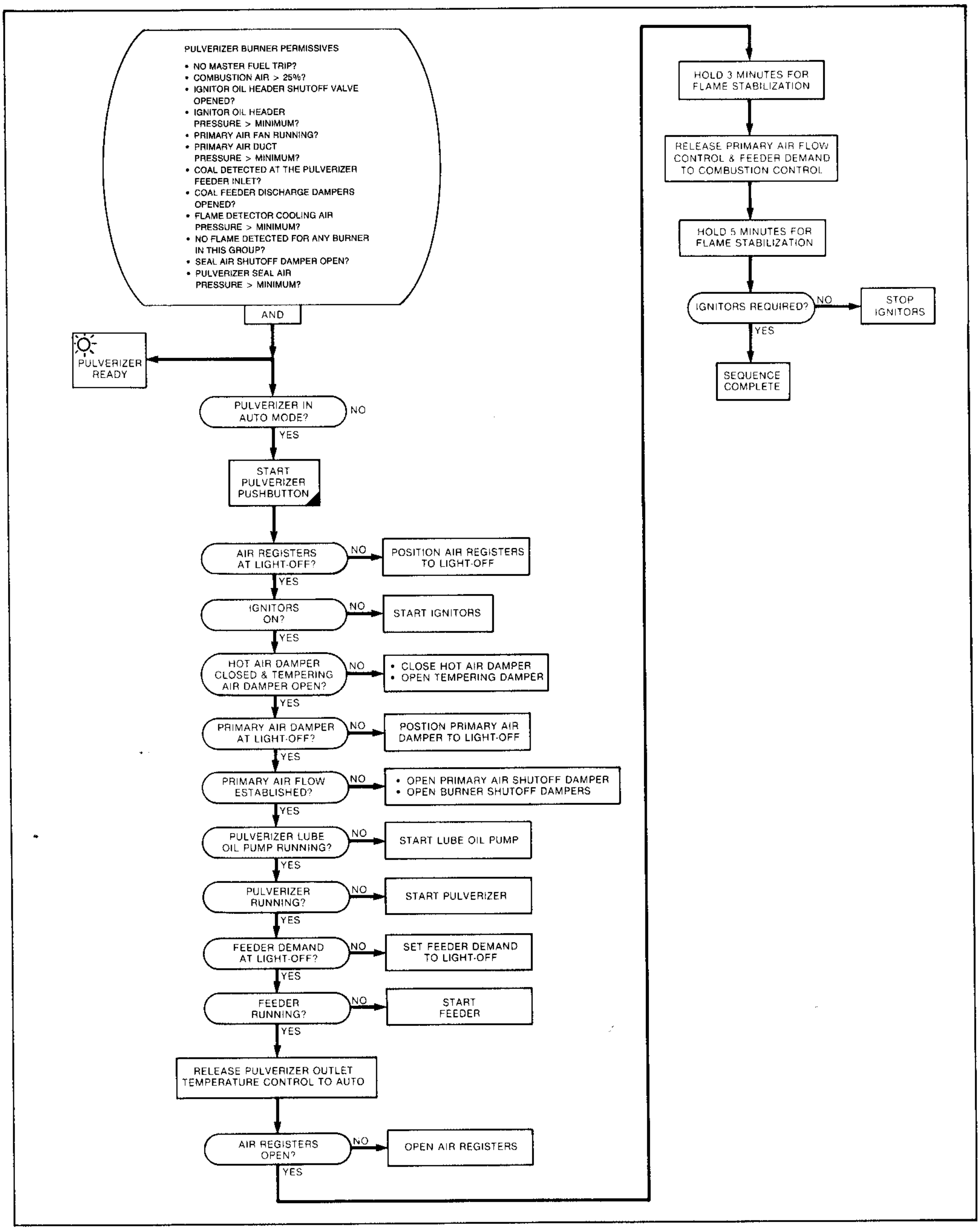


Figure 22-20 Typical Start Sequence for Foster Wheeler Type MB Pulverizers

(From Bailey Controls Co. Application guide)

start sequence for two different pulverizer models, both of which are manufactured by the Foster Wheeler Co. While the general structure of these two sets of logic is similar, a detailed study of the two show wide differences.

The pulverizer manufacturer is asked to take responsibility for the performance of his pulverizer. Just as the method of modulating control should be either dictated or cleared and approved by the pulverizer manufacturer, so should the procedures for starting up the pulverizers.

The overall procedure for purging, starting up and stopping pulverizers, lighting off and shutting down burners, and MFT (main fuel trip) are under NFPA jurisdiction. The procedures for pulverized coal-fired boilers are covered by the national standard ANSI/NFPA 85E-1985. The pulverizer manufacturers had a part in the development of the NFPA codes, and their pulverizer start-stop requirements will be consistent with the NFPA 85E code.

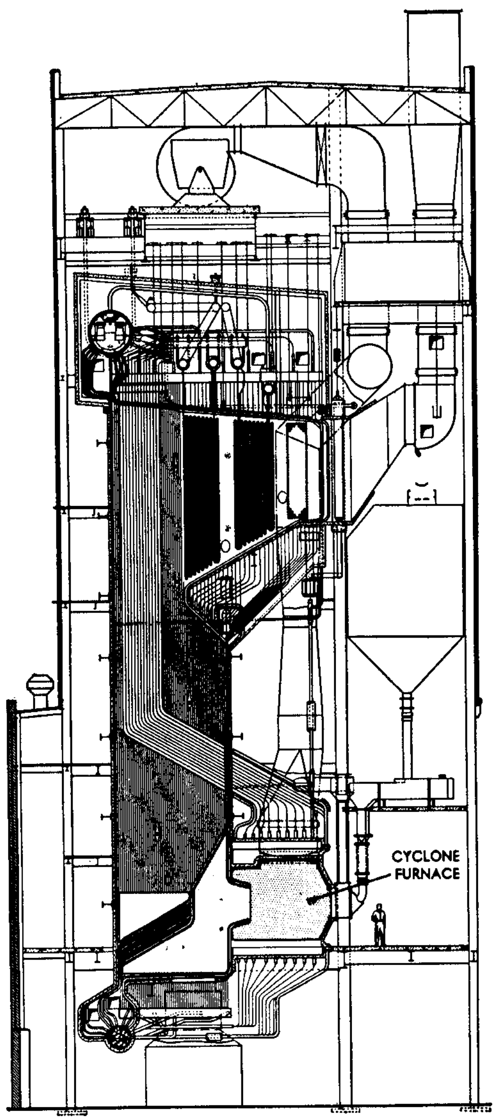


Figure 22-21 Radiant Boiler with Cyclone Furnace

(From *Steam, its Generation and Use*, © Babcock & Wilcox Co.)

22-7 The Cyclone Furnace

A cyclone furnace is a smaller ancillary furnace that typically attaches horizontally near the bottom of the the main boiler furnace. Figure 22-21 shows how cyclones are attached to a large boiler. Cyclones can also be installed on opposite sides of the furnace. Two or more of these cyclone furnaces are used, depending on total boiler capacity. A large electric utility boiler might have ten or more cyclones. They are intended for firing the lower grades and ranks of high-ash, low fusion-temperature coal with approximately 15 percent excess air. Heat is released at extremely high rates and combustion is complete within the combustion space of the cyclone.

Figure 22-22 shows the detailed arrangement of this device. The diameters of the main barrel range from five to ten feet. The fuel enters the small cylindrical section at the front of the cyclone in a vortex of primary air flow. The primary air flow is approximately 15 percent of the total air flow. Secondary air with a velocity of approximately 300 fps is admitted tangentially at the roof of the main barrel of the cyclone and imparts a further whirling and more centrifugal action to the coal particles. The necessary velocity is attained by the pressure ahead of the secondary air leaf damper plus the opening of the air passage. It is not simply a matter of sufficient air flow. The coal is crushed so that about 95% will pass through a 4-mesh screen and be fed to the cyclone with a volumetric or gravimetric coal feeder.

Gas temperatures exceeding 3000°F are developed. This melts the ash to a molten slag

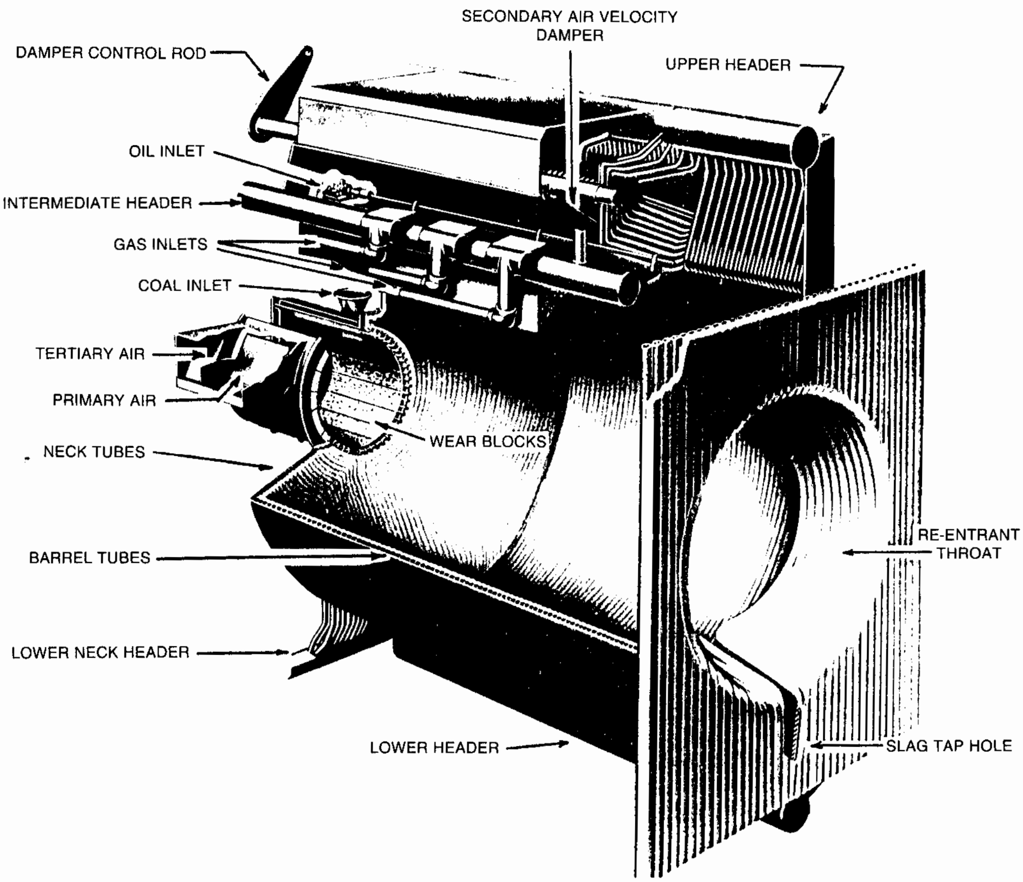


Figure 22-22 Cyclone Furnace Arrangement

(From © Babcock & Wilcox Co.)

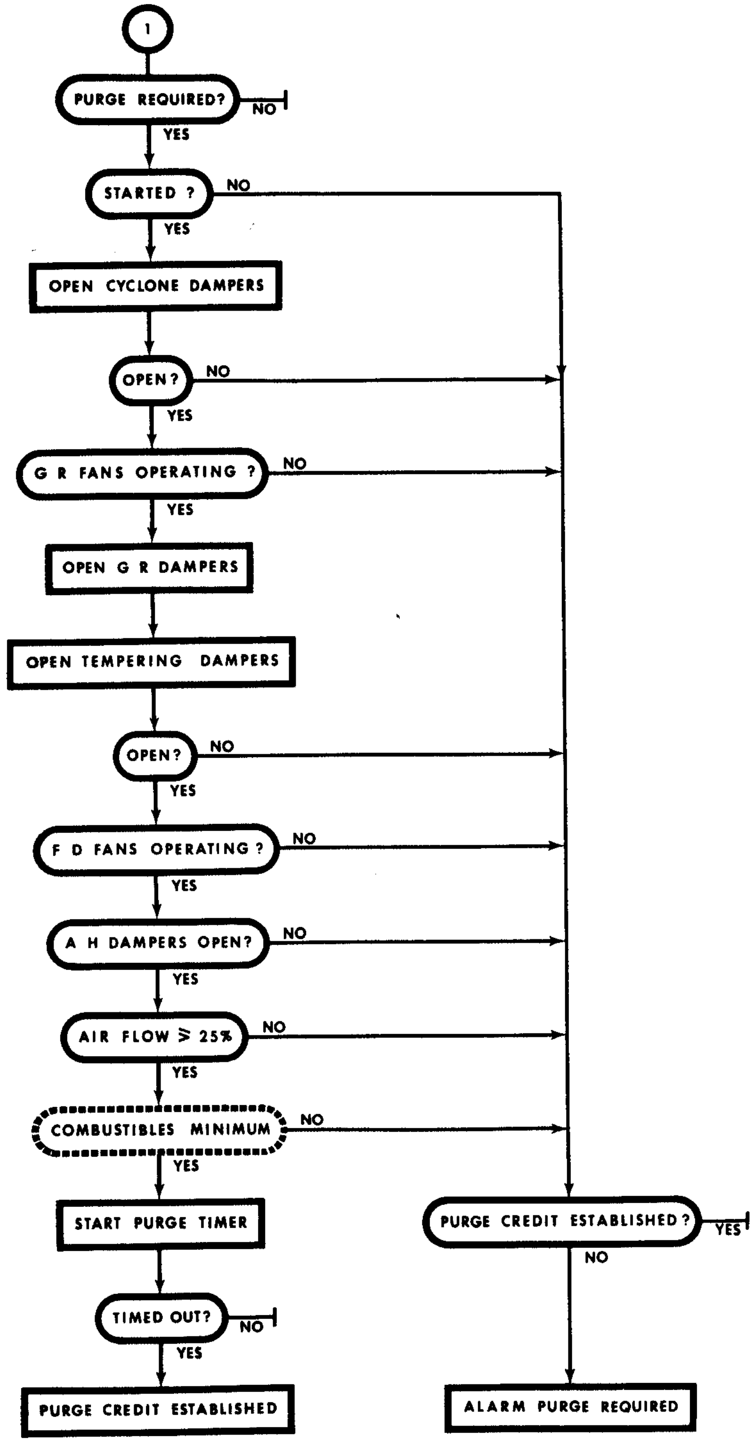


Figure 22-23 Cyclone Furnace Purge Logic

(From Bailey Controls Presentation)

that then forms a layer on the walls of the cyclone. The incoming coal particles are thrown to the outside by centrifugal force and are trapped in this layer of molten slag and completely burned by the scrubbing action of the high velocity tangential secondary air streem. As the molten slag builds up, it enters the main furnace through the slag tap hole and is tapped into a slag tank and disposed of in a convential manner.

The hot gases leave the cyclone and enter the main furnace through the water-cooled reentrant throat. An entrance for tertiary air in the center front of the cyclone allows adjustment for modifying the air flow pattern within the main barrel. The boiler furnace use is intended only for cooling the furnace gases.

The same cyclone can interchangeably burn oil, gas, hogged fuel, and coal. About 70% of the wide range of coals that are mined are considered suitable. When burning oil, an oil gun is centered at the front of the burner or sprays into the high velocity secondary air stream beside the secondary air velocity damper. When burning gas, the gas enters the secondary air stream just before the secondary air velocity damper.

One particular power plant uses their cyclone-fired boilers for the burning of RDF (refuse-derived fuel) in combination with coal. The refuse is shredded and blown into the secondary

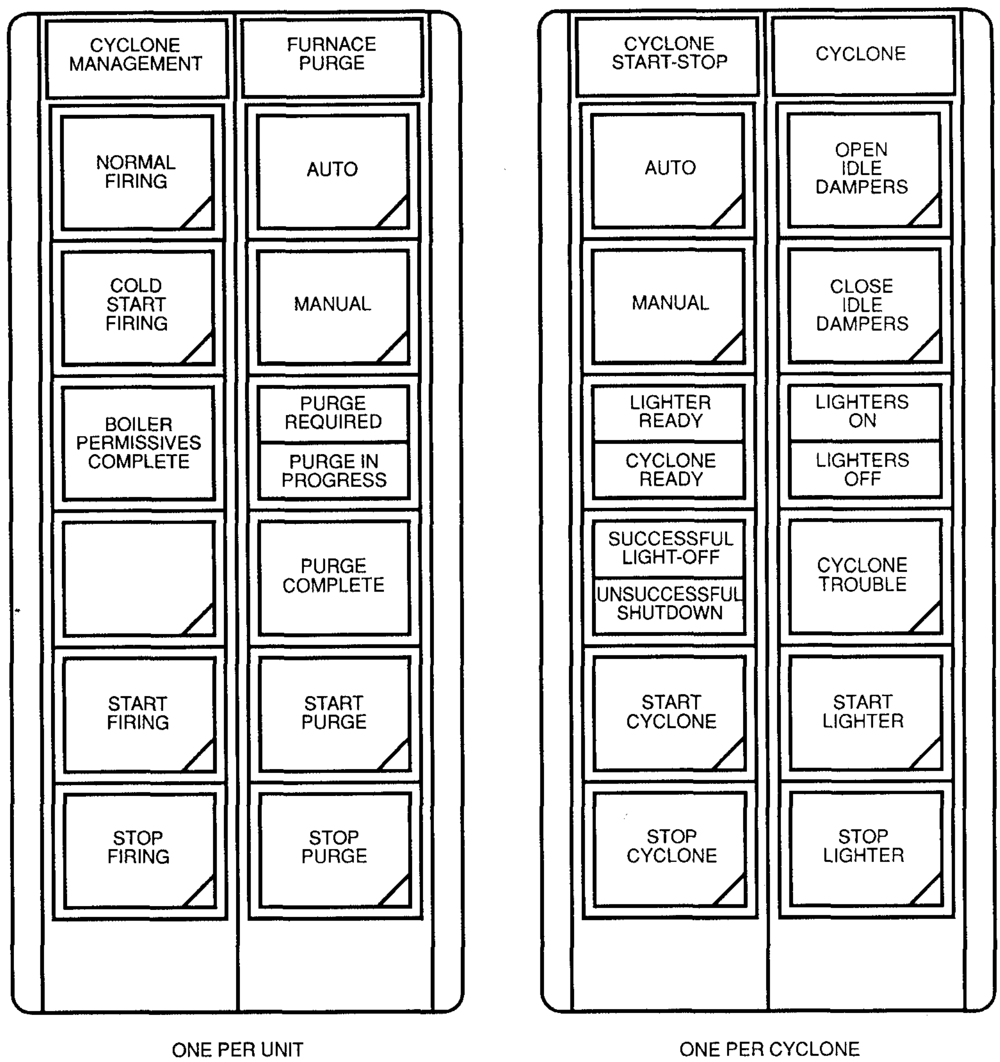


Figure 22-24 Cyclone Furnace Operator Interface

air stream. Up to 20 percent of the total cyclone fuel can be RDF when burned with coal in this manner.

The cyclone furnace system is very responsive to changes in fuel flow. A main drawback is that, due to the very high temperature within the cyclone, it is high in NOx production.

22-8 Start-Up and Management of Cyclone Furnaces

Cyclone furnaces have been described as the easiest way to burn coal but the hardest way to burn it right. The automation of a cyclone furnace in the starting up, shutting down, and normal operation is particularly important to assure consistent and correct operating procedure.

The suggested cyclone furnace purge logic is shown in Figure 22-23. Note that there is no mention of induced draft. Many of the cyclone-fired boilers were pressure fired and had no induced draft fans. If the boiler had been equipped with induced draft fans, their operation would have been part of the logic. Other than that there is little difference in this purge logic as compared to that of a gas-fired boiler.

After the purge, the start-up logic is quite simple and also compares closely to that of a gas-fired boiler. Typically, ignitors are lit, their flame proved, the primary and secondary air dampers manually set and the coal feeders turned on, the coal flame is proved, and an indication is given to the operator that the light-off has been successful. A typical operator interface station is shown in Figure 22-24. This station combines the necessary operator message annunciators, signal lights, and manual/auto transfer of the system operation.

The flame detectors are located so that the ignitor flame detector sees both the ignitor flame and the flame of the main burner. The location of these detectors is shown in Figure 22-25.

As on the other coal-burning systems, personnel with cyclone expertise should be consulted in designing a cyclone automation system. There is not a specific NFPA governing document for cyclone furnaces, but since cyclone furnaces can be used to burn gas, oil, or coal, the principles in the series ANSI/NFPA 85B-E and ANSI/NFPA 85 should be adhered to.

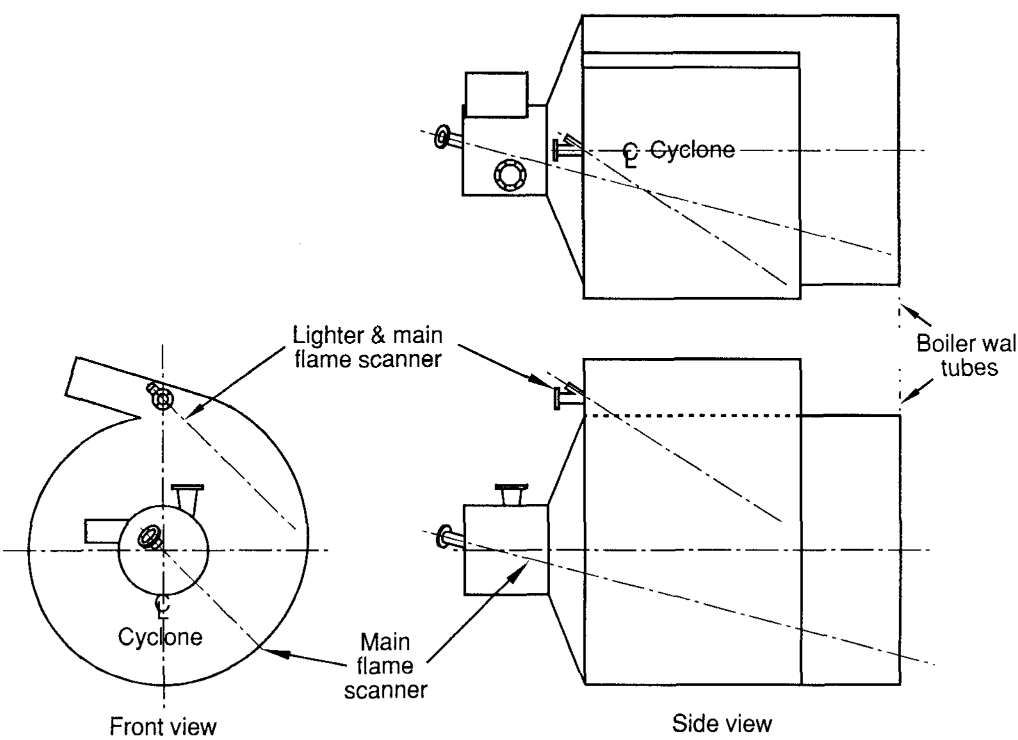


Figure 22-25 Location of Flame Detectors on Cyclone Furnace

Section 23
Combustion Control for Cyclone and Pulverized Coal-Fired Boilers

Combustion control systems for the firing of pulverized coal should be viewed as similar to those for gaseous or liquid fuel. The fuel flows in the current of primary air as a fluid, and the combustion and associated heat release after the fuel enters the furnace is very rapid. Differences that affect the control of combustion are the use of multiple pulverizers and the time delay for grinding the fuel. The variation in Btu content due to the moisture and the ash content variation create additional control problems. The fact that the fuel as it enters the furnace cannot be measured directly creates another challenge for the control system designer.

23-1 Coal Btu Compensation

The various pulverizer control loops in Section 20 contain some form of coal flow measurement. In some cases this measurement is the speed of a volumetric or gravimetric feeder. In other cases a rough measurement of pulverized coal flow is obtained with the classifier differential pressure or pulverizer differential pressure. A total boiler fuel measurement is obtained by summing these pulverizer coal flow measurements. The accuracy in coal Btu of this summation is affected by the accuracy of the basic pulverizer signals, their linearity, and the variation in moisture and ash content of the individual coal streams.

One of the methods of compensating for the potential inaccuracy of the signals and/or their summation is the use of the boiler as a calorimeter. If the heat release of the boiler is known and the coal input is known, then the heat release can be used to continuously calibrate the coal flow measurement. This enables one to obtain a more precise fuel Btu input for use in the control system. A rough measurement of the heat transferred—heat release multiplied by boiler efficiency—is the boiler steam flow. The steam flow measurement alone may be no more precise than the rough coal flow measurement.

A precise measurement of the heat transferred must account for changes in steam pressure, steam temperature, feedwater temperature, and the heat taken from or added to boiler energy storage. For noncontrolled extraction turbogenerators, the energy input is proportional to the first-stage shell pressure of the turbine. If all the boiler output goes to such a turbine, the turbine first-stage shell pressure measurement is a suitable measurement of the boiler energy output. Under steady-state conditions, with the boiler pressure and temperature constant and the feedwater temperature constant, this value is proportional to the heat transferred in the boiler. The above assumes a non-reheat boiler and constant boiler efficiency.

If the feedwater temperature changes, thus altering the heat content in each pound of feedwater, the heat transferred is a different value. If the pressure and temperature are changing, the heat added to or taken from storage is changing.

A measurement logic arrangement that accounts for the total heat transferred and thus the total heat release is shown in Figure 23-1. The derivative or rate of change of drum pressure recognizes that the boiler drum and piping system have a fixed volume and the quantity of steam in that volume is approximately linear with respect to pressure. This is not correct over wide changes in pressure but over the relatively narrow range of operating pressure can be assumed to be true. A rate of change of this pressure is therefore equivalent to a rate of steam generation that should be added or subtracted to the rate of steam usage in order to account for the total steam generated.

If there is no single boiler-single turbine relationship, a different technique (as shown in

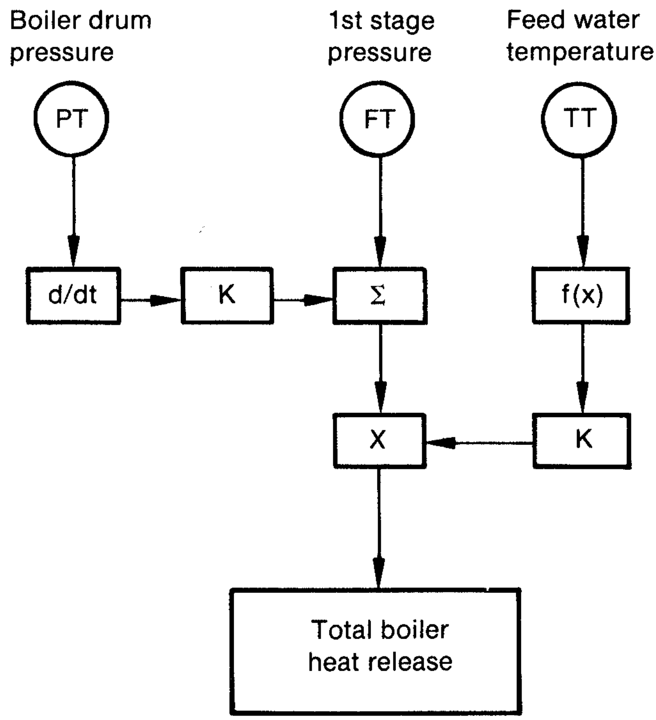


Figure 23-1 Computation Logic for Total Boiler Heat Release

Figure 23-2) can be used. In this measurement logic, the steam Btu flow is computed. The computation logic shown is identical to that of mass flow. The difference is that the functions of pressure and temperature shown combine the effects of Btu/lb and specific volume values.

The Btu flow is then multiplied by a function of feedwater temperature to obtain total energy transferred to the steam that is delivered to the steam system. This energy is then adjusted by the derivative of drum pressure to account for energy added to or taken from boiler energy storage. With constant boiler efficiency, the total is directly proportional to total heat release.

The total heat release values obtained by either of the above methods are, however, out of time phase with the coal flow measurement. The time delay between these measured values is the accumulated time for pulverizing the coal, transporting it to the burners, the combustion process, and the transfer of the heat generated. Any proper comparative use of these values must account for the time delay between coal flow measurement and the transfer of the heat.

The control logic by which the coal measurement is calibrated by the heat release value is shown in Figure 23-3. In this arrangement a summation (a) of the coal feeder speeds is used as the preliminary coal measurement. The calibration of this signal is effected in the multiplier (b). Essentially, the calibrated coal flow is compared to the total heat release in the delta (c). Any error between these two signals causes the integrator (d) to act until the signals are again in balance.

The time function (e) represents the delay discussed above and is adjusted so that the coal flow signal is in time phase with the heat transferred value. The signal from the time function is then an emulation of the actual heat release of the pulverized coal. It must be recognized that the integral (d) also has a time adjustment. This integral time must be slower than the time function (e) or the calibrating circuit shown will become unstable. The calibrated coal flow signal can now be used in the combustion control logic in the same manner as the measurement of fuel oil or gas.

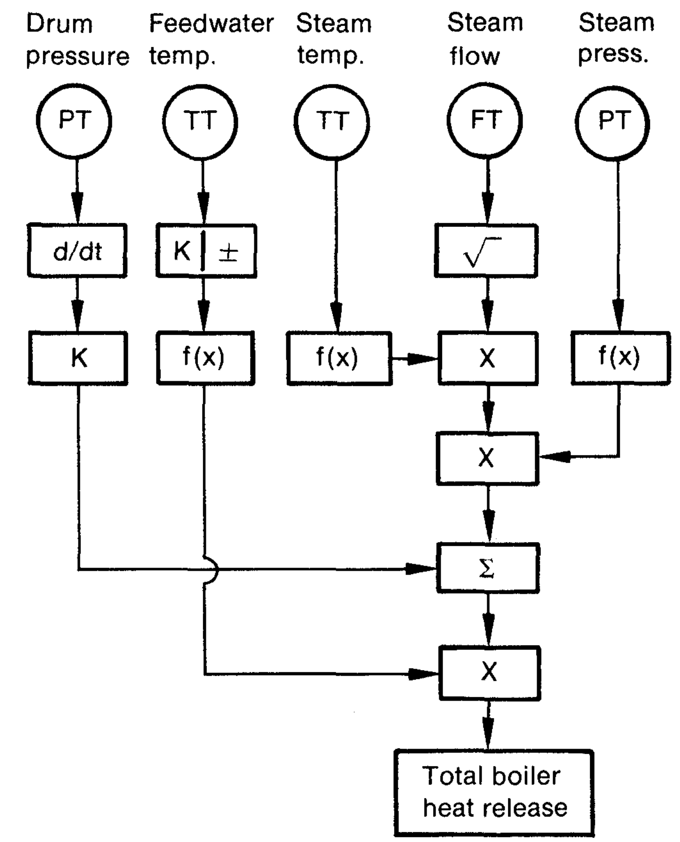


Figure 23-2 Computation Logic for Total Boiler Heat Release

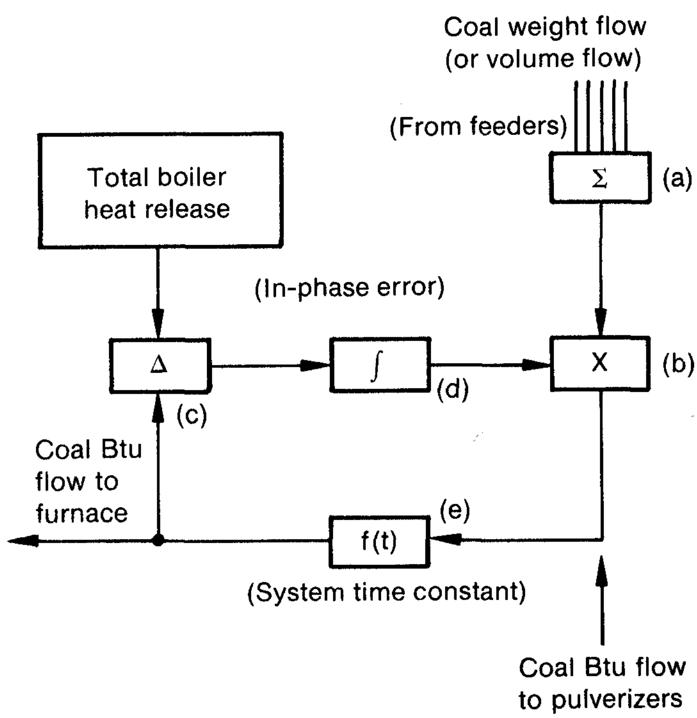


Figure 23-3 Calorimetric Calibration of Boiler Coal Flow

23-2 The Use of Multiple Pulverizers

The preceding paragraphs show how a total coal measurement is obtained when multiple pulverizers are used. Note that the summation of the coal flow from individual pulverizers is similar to the totalization of multiple liquid or gaseous fuels. The output of the combustion control system is handled in a fashion similar to that for multiple fuels. The particular control problem is that the gain of the control signal should change as the number of pulverizers in use changes. Two methods of accomplishing this are shown below.

In Figure 23-4, a multiplier is inserted into the control signal to the pulverizers. The desired multiplication is determined by the number of pulverizers in use and this implies some sort of "in use and on automatic control" pulverizer counting circuit. Such a circuit is shown in

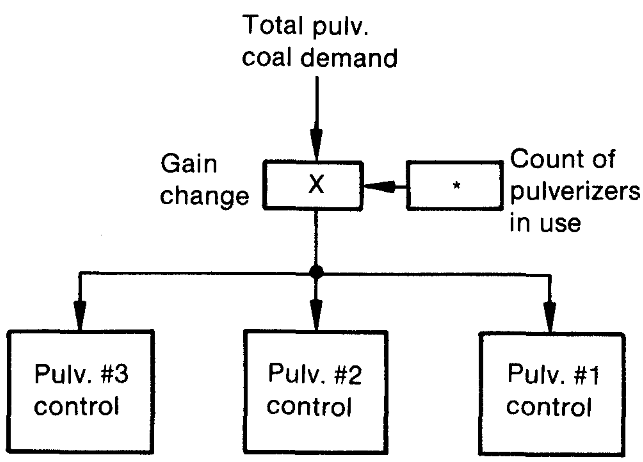


Figure 23-4 Gain Control for Number of Pulverizers in Use

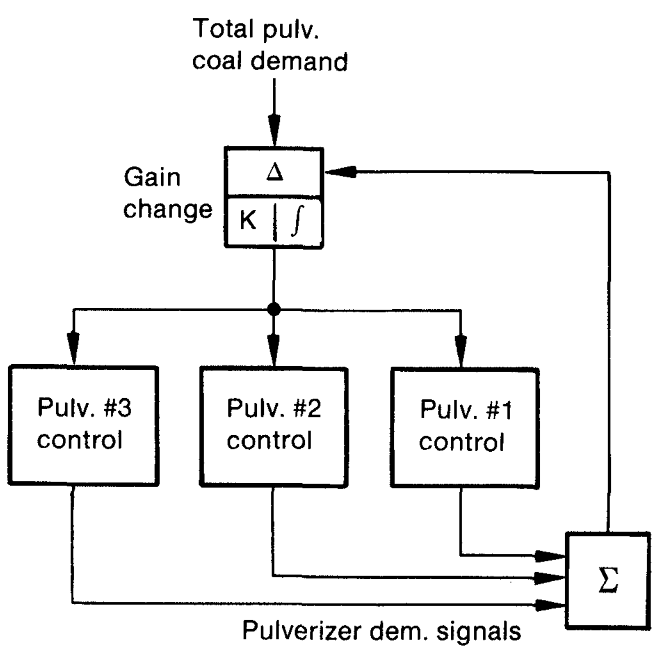


Figure 23-5 Gain Control for Number of Pulverizers in Use

Figure 8-13. In Figure 23-5, the pulverizer control signals are summed and compared to the total pulverizer demand control signal. Since the pulverizers are equal in capacity, the gain of their individual signals into the summation is equal. If two pulverizers are used for full boiler load, the gain should be 0.5; if 5 pulverizers are necessary, the gain should be 0.2.

It should be noted that this logic in Figure 23-4 is the "on-line" control logic and does not include the safety, start-up, and shutdown effects concerning when the pulverizers should be counted as "on-line." Similarly, Figure 23-5 does not show when the pulverizer control signal should be added or eliminated from the summation circuit. The details of the safety logic and the start-up and shutdown logic are not included in the scope of this discussion.

23-3 The Combustion Control System for Pulverized Coal as a Single Fuel

The combustion control system for a single-fuel pulverized coal-fired boiler can be an adaptation of the liquid or gaseous systems shown in Figures 21-9 or 21-10. In the same manner as the basic system is modified for various fuels or combination fuels, the tailoring points are those marked A, B, and C. Such a system for pulverized coal is shown in Figure 23-6.

The firing rate demand, the fuel controller, the fuel limit control, the air limit control, the air flow controller, and the flue gas analysis trim control are identical to those of Figure 21-9. The adaptation is that above point B the total coal measurement and its calibration circuit

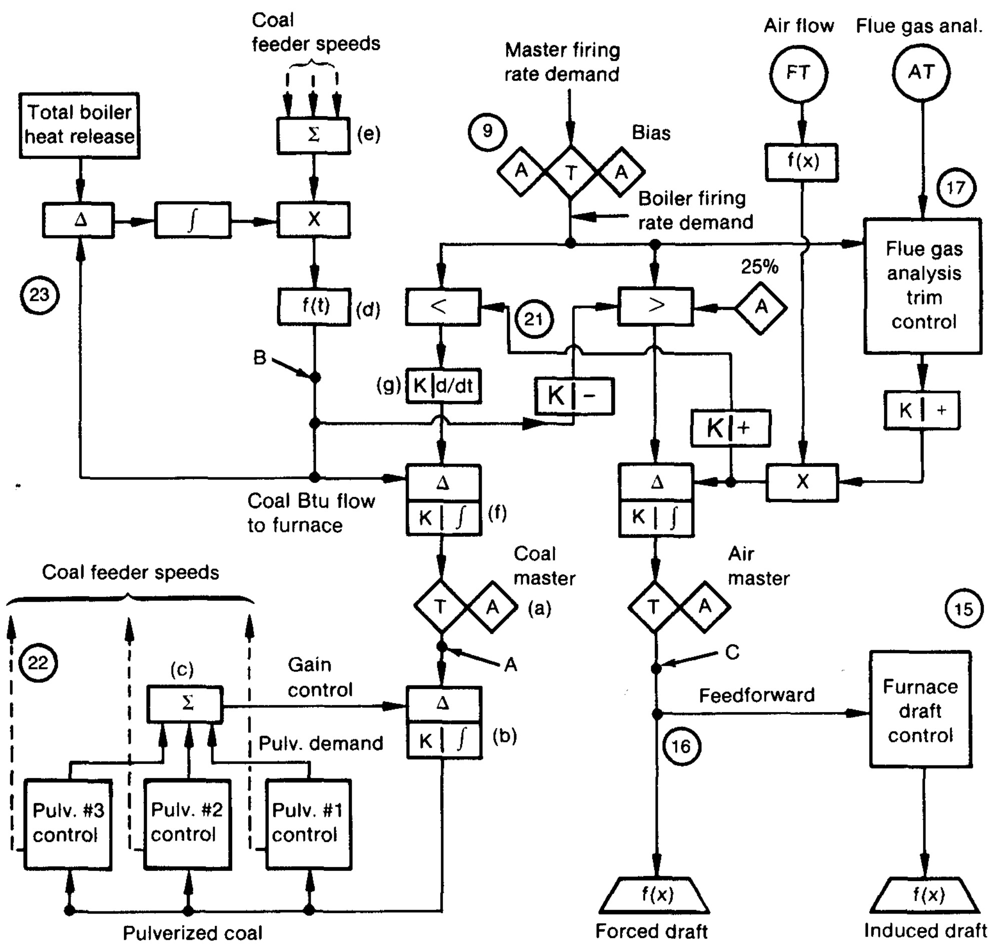


Figure 23-6 Control System for Single-Fuel Pulverized Coal-Fired Boiler

are connected In addition, below point A the pulverizer controls are connected. The control station (a) is considered the pulverized coal master manual/auto station.

The basic pulverizer demand signal is developed as an output of the gain adjusting controller (b). The pulverizer loadings can be kept in balance with each other by the use of the bias-type manual/auto station that is part of the pulverizer block of control logic. The output of the individual manual/auto stations feeds back to summer (c) as part of the gain control circuit. The arrangement shown is for three pulverizers. If there were four or more, another pulverizer block would be used and the input gains of the summer (c) would be changed.

The time function (d) between the feeder speed summation (e) and the fuel controller (f) delays the proportional feedback to the controller and tends to provide an output with some derivative action. Derivative action in the control of a coal feeder causes the pulverizer coal level to change in a direction that will improve the pulverized coal response. If the feedback time funtion does not provide enough derivative action, the insertion of the proportional-plus-derivative function (g) or adding derivative action to controller (f) will supplement that already available.

The circled numbers in Figure 23-6 refer to blocks of control logic that are developed elsewhere in this text. The numbers are the section numbers where the detailed discussion is found.

An alternate approach to controlling a single-fuel pulverized coal-fired boiler is the adapted use of the steam flow-air flow system used for spreader stoker-fired boilers. In Figure 23-7 the feeder speeds or other rough coal flow measurements are totalized in the summer (a) and the total fuel output is fed back to the fuel controller (b). A derivative function is added to this controller to temporarily increase the pulverizer coal level in order to improve the response of actual pulverized coal flow to the furnace.

A preliminary position of the forced draft and associated furnace draft control is signalled by the feedforward of the boiler master firing rate demand signal (c) input to the summer (d). The other input to this summer is the final readjustment signal (e) from the total heat release-air flow controller (f). Steam flow can be compared against air flow as in the spreader stoker system, but performance is improved if the more precise total boiler heat release is used in its place. The justification for using the more precise configuration is that pulverized coal-fired boilers are generally larger than stoker-fired boilers. The incremental cost of the more complex control system in order to achieve more precise control of combustion can usually be justified.

Since the total boiler heat release signal includes the ''put and take'' of energy from storage as the boiler load changes, it is not limited to steady-state comparison to combustion air flow. In view of this, the time function (g) may not always be required. The compensation for variation in coal Btu value or other factors affecting the coal flow measurement is accomplished by the integral function of controller (f). As in Figure 23-6 the circled numbers represent section numbers in this text where subsystem details are discussed.

23-4 Pulverized Coal in Combination with Liquid or Gaseous Fuels

When pulverized coal is burned in combination with liquid or gaseous fuels, the same burner air register assembly is used. The fuels have different nozzles but can be totalized on an air-required basis and fired together. The control application procedure is to add the coal flow to the flow of the other fuels to obtain a total fuel signal in terms of combustion air required. This is a modification above point B on Figure 23-6. In addition, below point A on Figure 23-6, the proper control actions to all fuels must be applied. Figures 23-8 and 23-9 show the modification to be made to the fuel control subsystem of Figure 23-6 when the boiler fires pulverized coal in combination with natural gas.

Since the total boiler heat release, as shown in Figure 23-8, represents the heat from all the fuel, the heat from the gas portion of the fuel must be eliminated before comparison with coal flow. This is done by assigning a gas-fired boiler efficiency value in proportional function

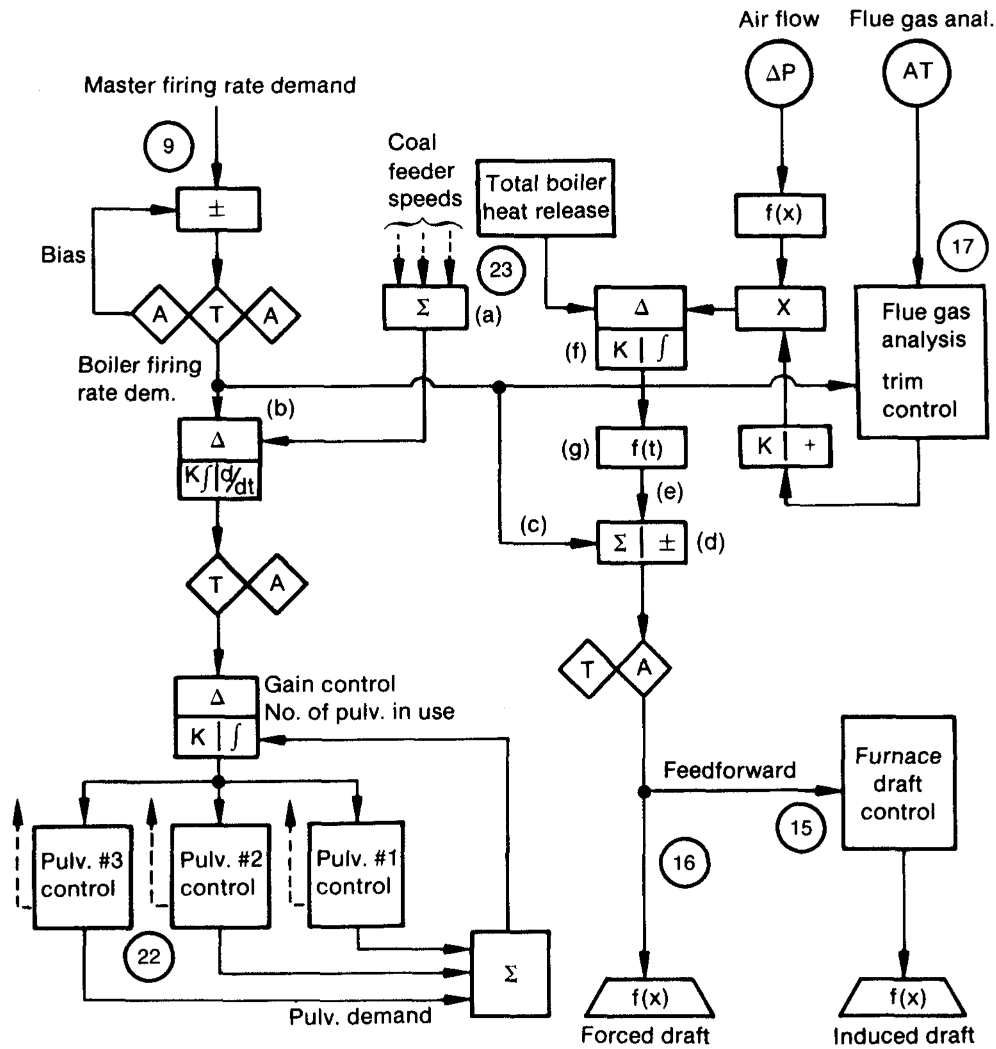


Figure 23-7 Parallel plus Total Boiler Heat Release-Air Flow Readjustment for Pulverized Coal-Fired Boiler

(a) and subtracting the result in the delta (b). The difference represents the heat release from the coal fuel.

The summation of rough coal flow in summer (c) can be compensated for Btu or other coal feed variations in the same manner described earlier. The total Btu values are then totalized in the summer (d) to provide a total fuel signal on an air required basis. The proportional functions (e) provide the small factors necessary for conversion from Btu value to combustion air required, difference in maximum Btu capacity, and difference in excess air requirement.

The output functions of the combination fuel system are shown in Figure 23-9. The coal/gas ratio logic (a) and (b) is the same as the fuel ratio logic in Section 21. The proportional plus derivative function (c) is intended to accelerate the pulverized coal response of the pulverizers by temporarily increasing their coal level. It is inserted here rather than in the total fuel controller to prevent the derivative action from affecting the fuel gas or other combined fuel. The gain change control (d) and (e) compensates for the number of pulverizers in use and on automatic.

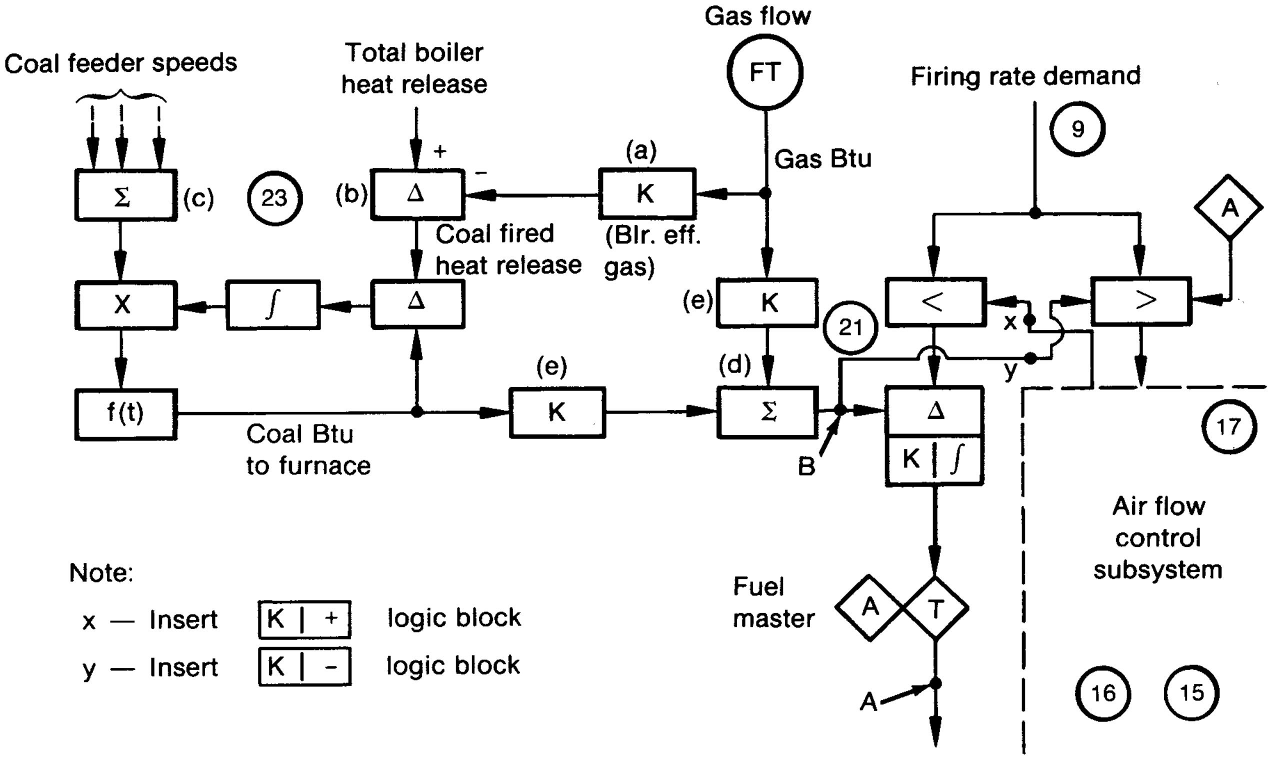


Figure 23-8 Measurement and Fuel Control for Combination Firing of Pulverized Coal and Gas

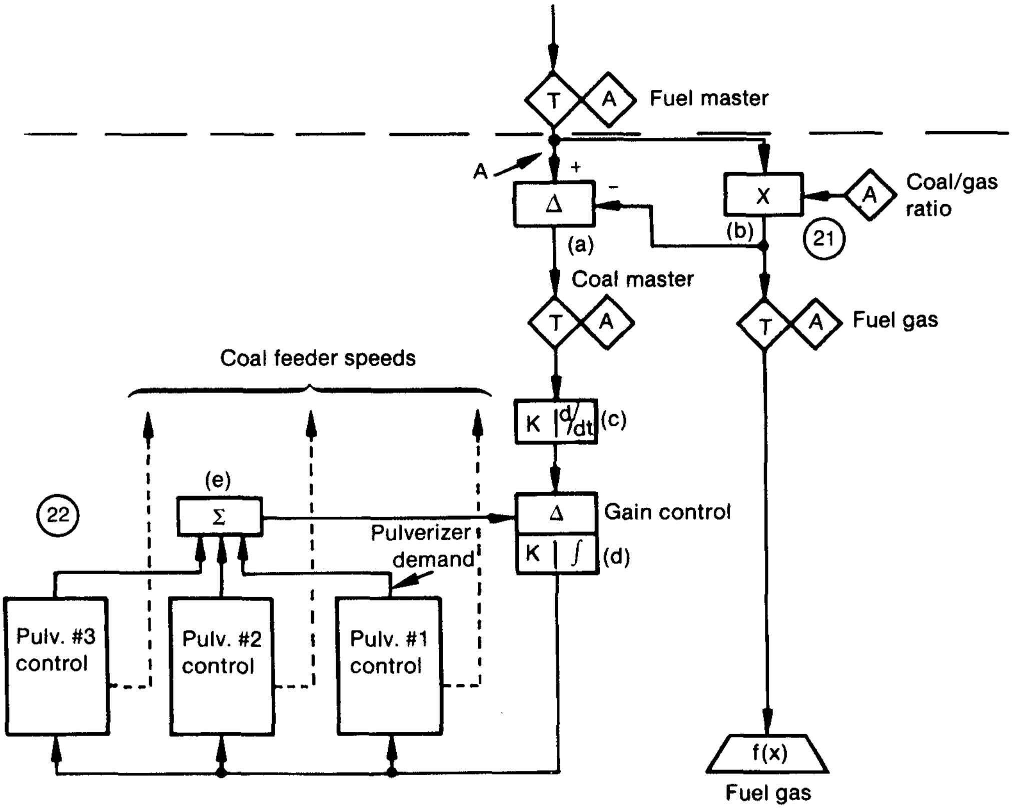


Figure 23-9 Measurement and Fuel Control for Combination Firing of Pulverized Coal and Gas

23-5 Compartmented Windbox Pulverized Coal Control Systems

The purpose of the compartmented windbox system is to obtain more precise combustion control on a burner and pulverizer basis. The systems discussed above are for application to boilers whose burners and pulverizers are controlled on the basis of the average for the entire installation.

For compartmented windbox installations, the strategy is for the coal to each compartment to control the air flows to that compartment. Figure 23-10 shows the arrangement for controlling a single compartment and how that compartment block of control functions ties into the overall scheme.

Assume that the pulverizer is a Babcock & Wilcox pulverizer with its block of control logic as shown in Figure 22-9. In Figure 23-10, all controls except the coal-air mixture temperature controls are shown. The coal-air temperature control loop for each pulverizer is required but was omitted from the system shown in Figure 23-10 for space reasons only. While the reference is to each compartment, this also refers to each pulverizer.

Items (a) and (b) provide the gain change adjustment needed for different numbers of pulverizers (compartments) in service. The primary air flow (j) is controlled based on the requirements of the pulverizer, but this flow must be carefully measured and added to the two secondary air flows for each compartment. This total compartment air flow is set in controller (f) to control the two secondary air flow sources to the compartment. The total air requirement is based on the compartment Btu demand (also the coal flow demand with constant Btu coal).

The coal-to-air ratio is adjusted based on a calorimetric action of comparing total boiler

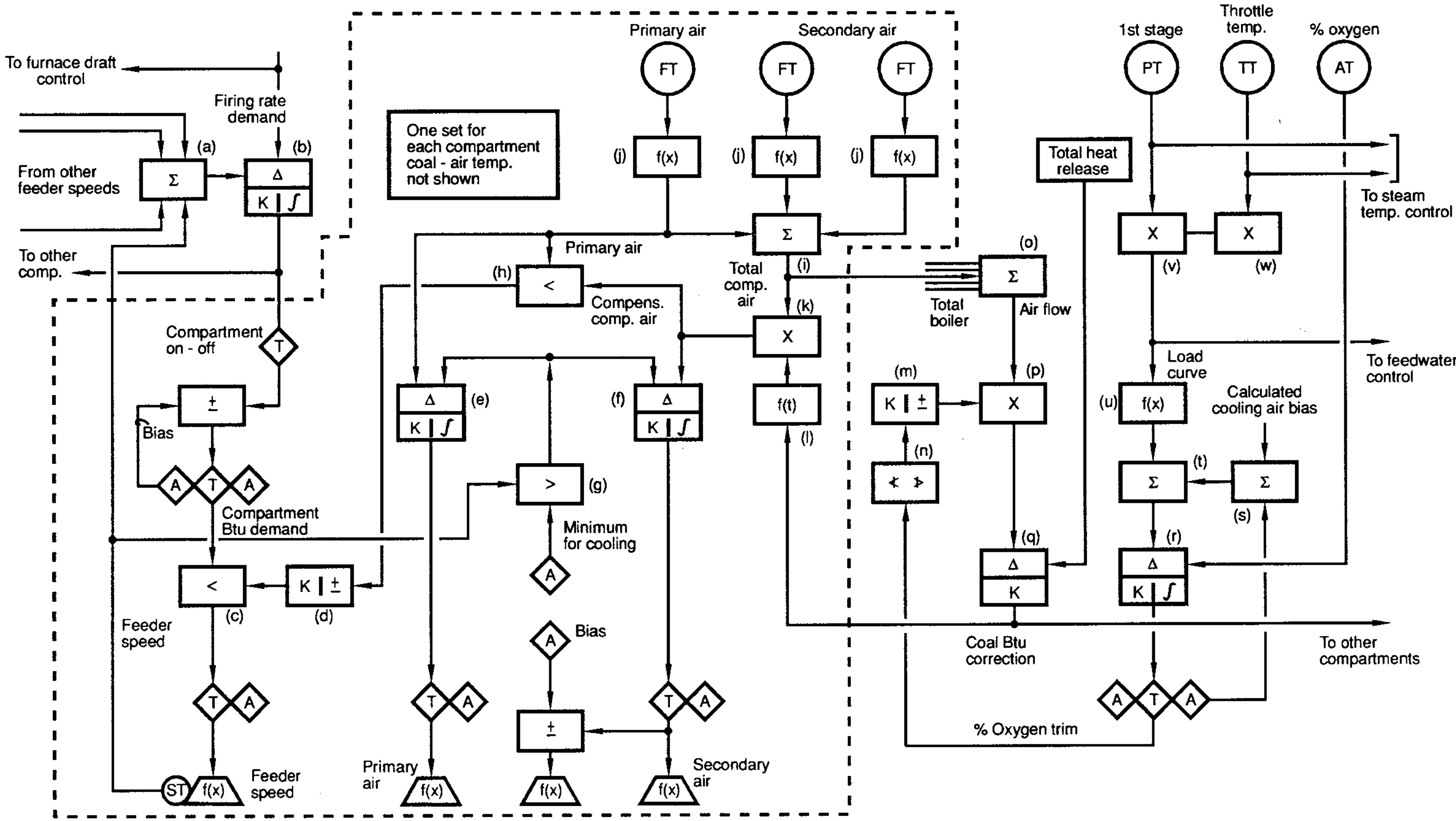


Figure 23-10 Combustion Control for Compartmented Windbox Boiler—1 Compartment Shown

heat release to % oxygen compensated total air flow in controller (q). The % oxygen trimming action takes into account that cooling air flow is necessary to protect the windbox when a compartment is idle. A calculation based on load, the number of idle and operating compartments, and the amount of cooling air to each are used to bias the % oxygen set point. In this way the air flow to operating compartments is not affected by the cooling air through idle compartments. The modification of the measured air flow effectively is a correction for variations in coal Btu value.

In addition to the control arrangement in Figure 23-10, a windbox supply pressure control loop is also required. This control provides a constant pressure ahead of the secondary air control dampers by controlling the forced draft speed or inlet louvres. This is shown in Figure 23-11. Also shown is additional detail of the secondary air flow measurement. The air temperature to the air foils will probably increase as the boiler load increases. More accuracy can be obtained if these air flow measurements are compensated for temperature. A single temperature measurement can be used for all compartments if the air from all air preheaters is mixed before arriving at the entrances to the compartments.

23-6 Control Systems for Cyclone Furnace Boilers

The control systems for cyclone furnace boilers are somewhat similar for those for compartmented windbox boilers. In both systems the coal flow to a compartment or cyclone controls the air flow to that device. The cyclone system is simpler since there is a single secondary air flow measurement and control, no pulverizer and no cooling air requirement.

As shown in Figure 23-12, actual coal flow sets the secondary air flow with controller (f) controlling the secondary air leaf damper. Primary and tertiary air are set manually. The duct pressure of the air supplying the secondary air to all cyclones must be at a pressure high enough to furnish the necessary seconday air rotative velocity. This pressure can be controlled to a

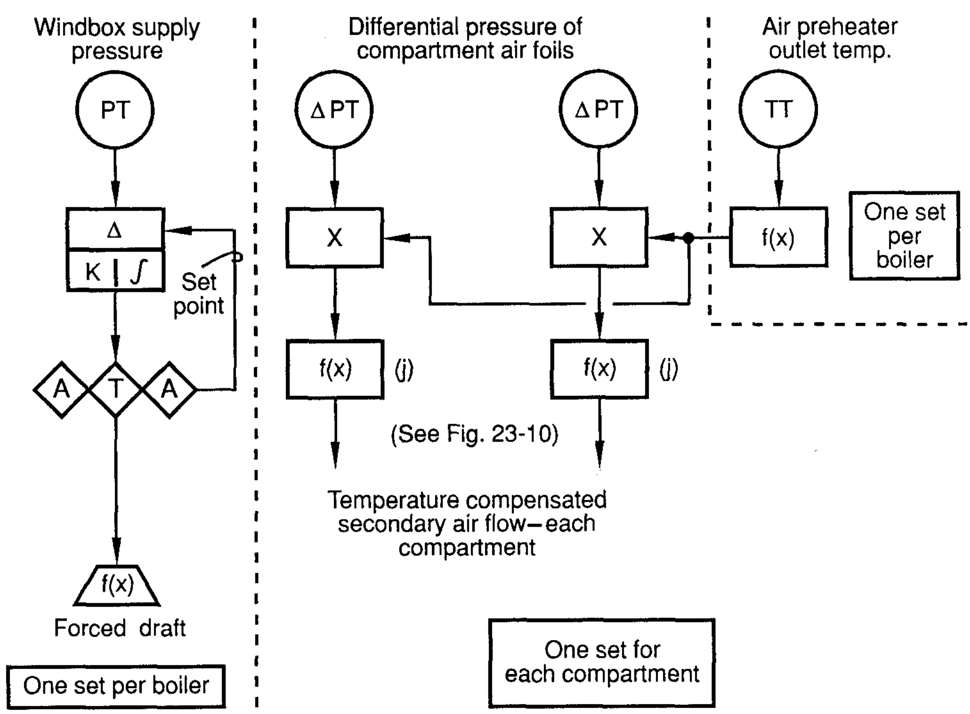


Figure 23-11 Temperature Compensated Air Flow—Windbox Supply Pressure

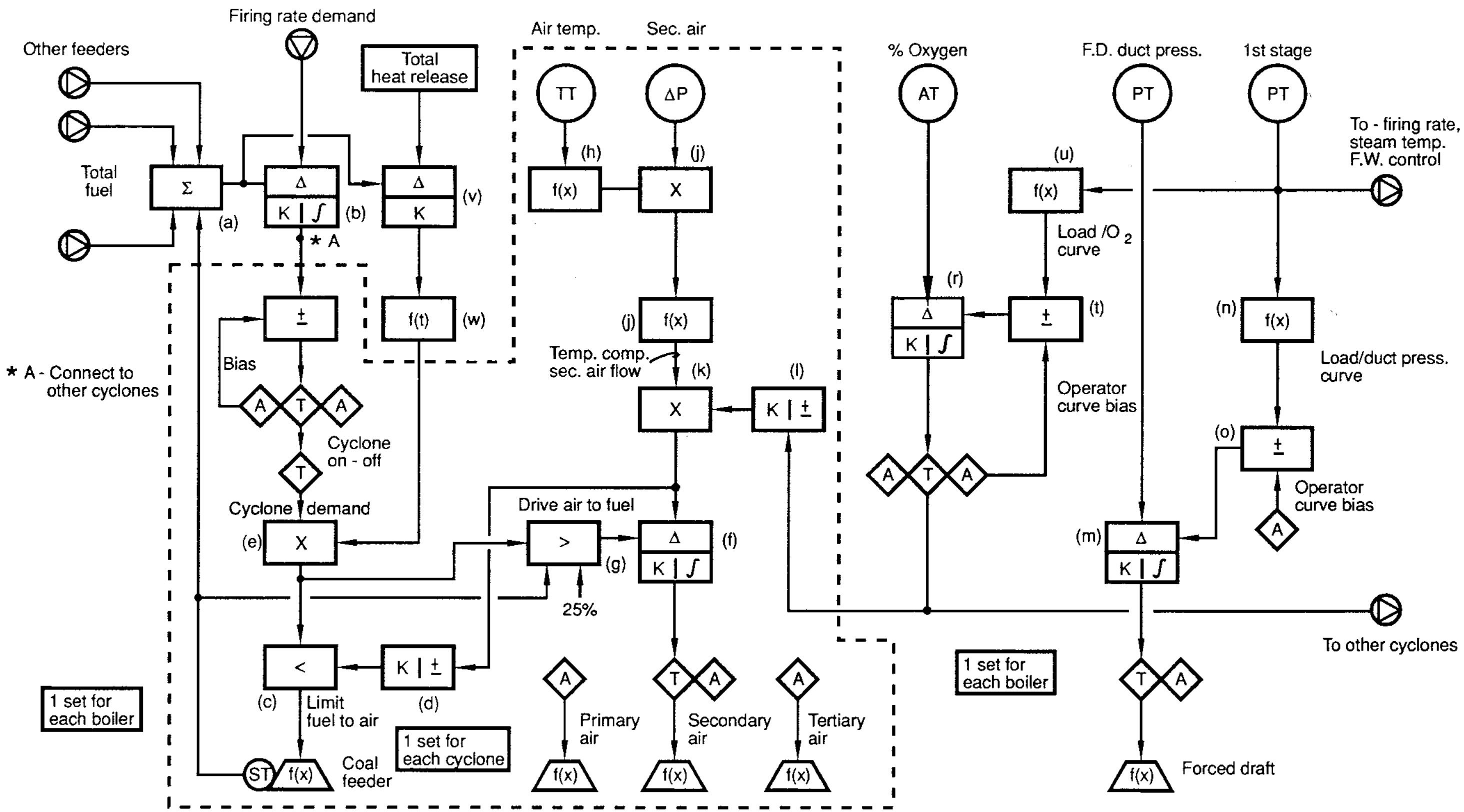


Figure 23-12 Combustion Control for Cyclone-Fired Boiler

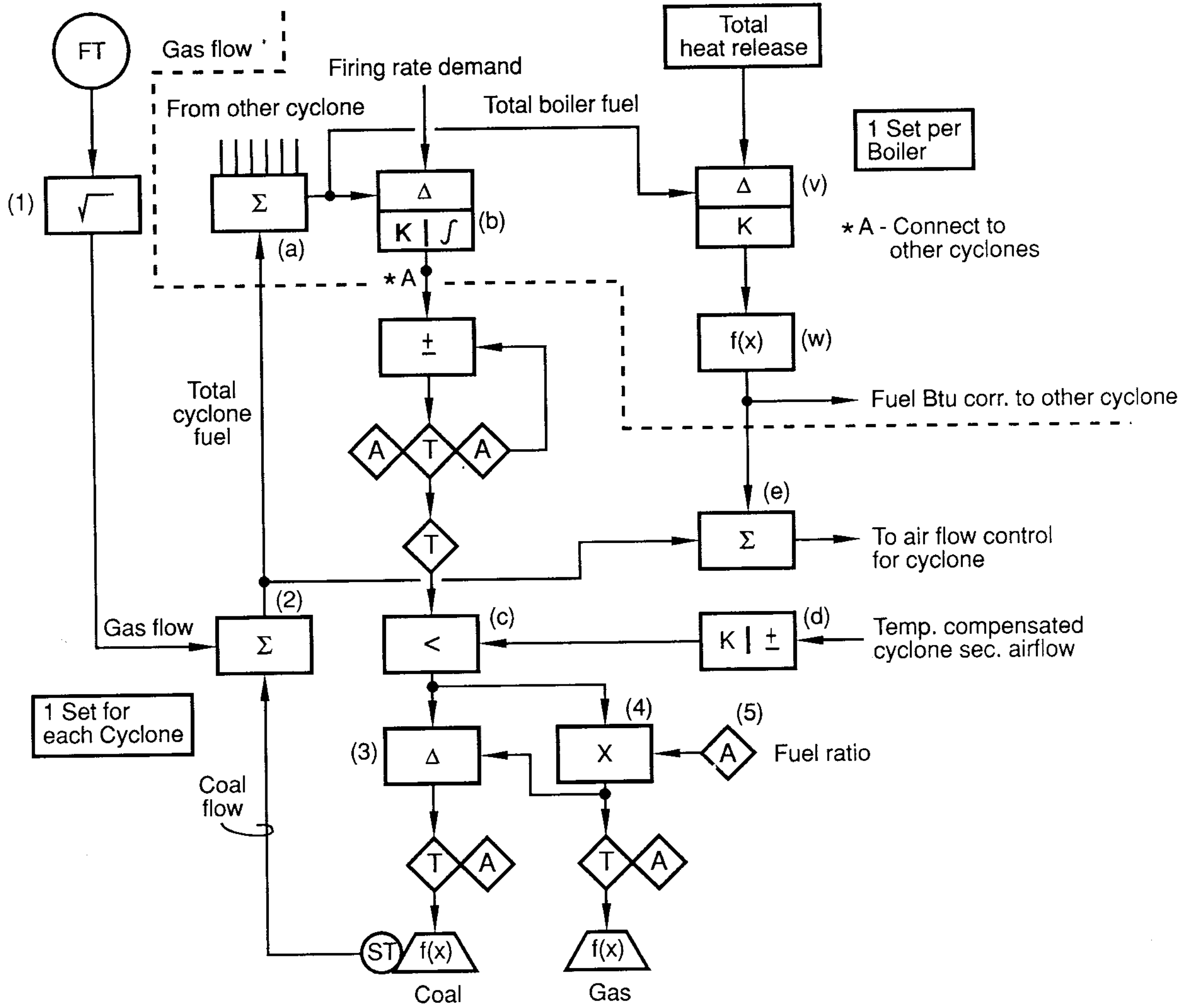


Figure 23-13 Coal and Gas Cyclone Firing—Individually or in Combination

fixed set point pressure or to a variable pressure as a function of load as showm in items (m), (n), and (o).

The coal Btu correction is made by comparing total boiler coal flow to total boiler heat release and having the result multiply the coal demand signal. The gain on this comparison is adjustable in controller (v), and the time constant of the process in converting coal flow to heat release is adjustable in time function (w). This method of fuel Btu correction could also have been used with the compartmented windbox system instead of the heat release vs. total compartment air flow that was shown in Figure 23-10. In the cyclone system of Figure 23-12 the % oxygen trim control, items (l), (r), (t), and (u) provides the final fuel-air ratio correction.

The control systems for cyclone furnace control can easily be modified for the burning of several fuels individually or in combination. Figure 23-13 demonstrates the modification to the block of cyclone control logic when gas and/or coal are burned individually or in combination. The gas flow (1) is added to the coal flow in the summer (2) on the basis of air required for combustion. The output of this summer is total fuel on the basis of air required for combustion. This, in turn, is added to the cyclone fuel totals of the other cyclones in summer (a) to obtain total boiler fuel. This total is used in the automatic gain adjustment controller (b) and the total boiler fuel vs. total boiler heat release (v) for developing the fuel Btu correction. This correction (e) could have been made in the fuel demand signal immediately after low select (c) as was done in Figure 23-12.

On the control output side, the fuel ratio network discussed in Section 21 is used for ratioing the control signal between the two fuels. If the ratio setter (5) is 1.0, the fuel is 100% gas; if the ratio setter is 0.5, the fuel is 50% gas and 50% coal. The fuel is 100% coal if the value of the ratio setter (5) is 0.

In the same manner that gas and coal are controlled in combination, other fuels can also be burned in combination. In one installation RDF (refuse-derived fuel) is burned in combination with coal in a quantity up to 20% of total cyclone fuel. The burning of the refuse in this manner has been quite successful, though the long term effects on the boiler are not yet known. In that installation a feeder injects the lightweight RDF into a 20-inch pipe enclosing a conveying air stream that blows the RDF to the cyclone. While not an accurate means, the speed of the RDF feeder that puts the RDF into the conveying stream is used as an RDF measurement.

Section 24
Combustion Control for Stoker-Fired Boilers

Since a continuous measurement of the fuel burned for use in stoker-fired boiler control systems cannot be obtained, inferential measurement of the fuel input is necessary for metering types of systems. Parallel positioning systems can be used but with the same inherent inaccuracies and weaknesses as when they are used with liquid and gaseous fueled boilers.

24-1 Parallel Positioning Control Systems for Stoker-Fired Boilers

Parallel positioning systems can be applied to spreader stoker-fired boilers in the manner shown in Figure 24-1. The fuel control is usually a lever that rotates a shaft linked to the feeding mechanism of each of the spreader units. The number of the spreader units is determined by the capacity of the boiler, with each spreader unit feeding coal to a longitudinal strip of the grate. The position of the link is an approximate measure of the volume of coal. If the density of the bulk coal were constant, it would also be an approximate measure of the weight of fuel. Since a large percentage of the coal in a spreader stoker-fired boiler burns in suspension, the response characteristic of the fuel and air should be approximately parallel. This is shown graphically in Figure 24-1.

The flow characteristic of the control signal vs. coal volume must be linearized and carefully matched to the flow characteristic of the required combustion air. This can be accomplished by using linkage angularity and cam shapes in the devices that position the combustion air control damper and the fuel control lever. The data for making these calibrations is derived from boiler combustion tests at three or more boiler loads spread over the operating range of the boiler.

Since all stoker-fired boilers are balanced draft boilers, the furnace draft must also be controlled, as discussed in Section 15. Overfire air requirements are somewhat unpredictable, although trends can be established over a period of time. The control for overfire air flow is normally a manual control when simple controls such as parallel positioning are used. The purpose of the overfire air is to provide turbulence above the grate for better mixing of the fuel and air. This turbulence is sometimes provided by jets of steam.

The rapid suspension burning of a large percentage of the coal makes spreader stoker-fired boilers more responsive to fuel input than to combustion air input. For this reason the direct control signal should be applied to the fuel device. Any correction through use of manual control or a more sophisticated type of control system is then applied to the air flow control device.

If an underfeed stoker is used, the combustion process occurs over a period of time. The coal enters the furnace and is heated; distillation and separation of the components then must occur before combustion. There is thus a fuel storage that involves a mass of coal in various stages of burning in the furnace at all times. The primary heat response is from changes in combustion air flow. The "preparation for burning" phase occurs rather rapidly since new coal enters directly into the burning coal mass.

This "preparation for burning" occurs more slowly in an overfeed stoker since the coal enters the furnace under an ignition arch of refractory material. The length and configuration of the ignition arch play a major part in the time for the distillation phase.

As shown in Figure 24-2, the direct control signal is then applied to the air flow control device, with the secondary signal directed toward the stoker speed device. Any correction is then applied to the less responsive fuel flow. It may be desirable in some cases to add a small time constant (j), as shown in Figure 24-2, to smooth the stoker speed signal and thus avoid

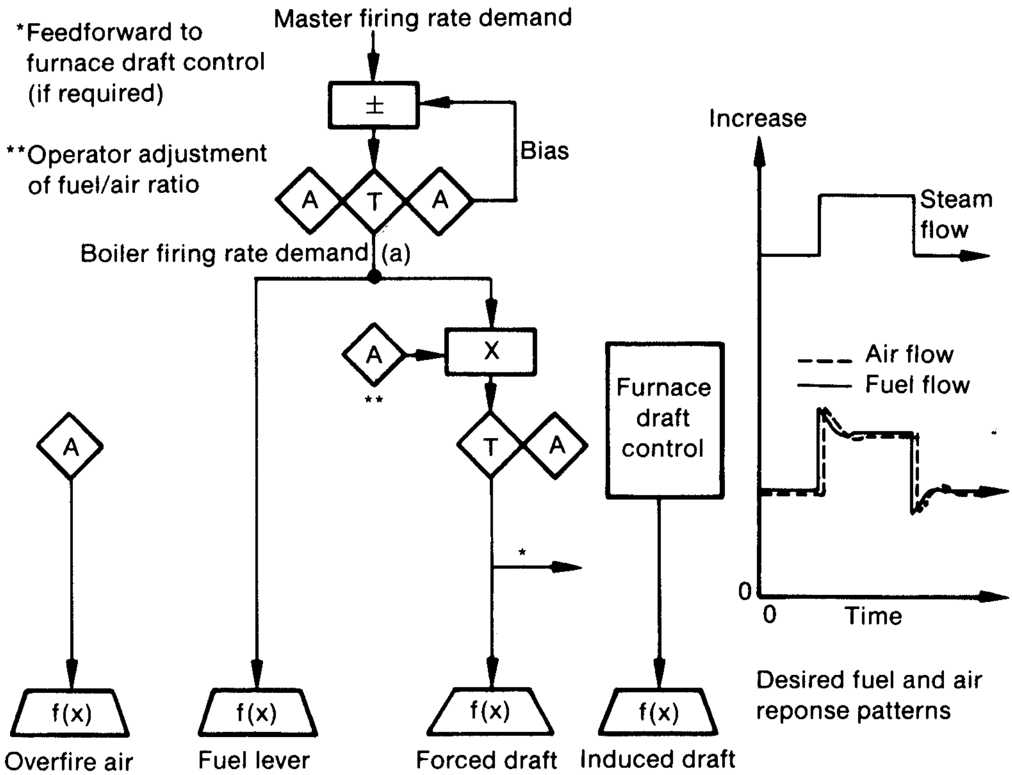


Figure 24-1 Parallel Positioning Control System for Spreader Stoker-Fired Boiler

rapid changes in the coal flow, which have very little effect on the overall unit response. This time constant may be a detriment to some overfeed stoker-fired boilers that have water-cooled furnaces and a short or small ignition arch. In these cases the distillation time may be long, slowing the distillation. The relative changes of fuel and combustion air are shown graphically in Figure 24-2. As with the spreader stoker control system, furnace draft control is also necessary for the balanced draft boiler. The overfire air is shown as a manual control.

When an overfeed stoker such as a chain grate stoker is used, the storage of fuel in the furnace prior to combustion makes the short-term heat response of such a boiler very insensitive to changes in fuel flow. Simple systems such as the parallel positioning are generally not adequate, and improved techniques are necessary. If a parallel positioning system can be used, the arrangement should be the same as that in Figure 24-2 for underfeed stokers. The primary signal should be directed to the air flow control device with a small time constant or, potentially, (based on load change requirements and boiler design), a small amount of proportional-plus-derivative control action inserted in the signal to the stoker speed control.

The approximate relationship between desired flow changes of fuel and air flow is also shown graphically in Figure 24-2. These general relationships, as shown graphically for the spreader stoker in Figure 24-1 and for the underfeed and overfeed stokers in Figure 24-2, should be used for any arrangements of control systems for these stokers.

Most stoker-fired boiler owners recognize the desirability of more complete control systems in order to operate the boilers at lower excess air values and thus save fuel. This requires some form of metering system and/or the addition of flue gas analysis trim control loops. The inability to measure fuel Btu input requires that a metering control system for a stoker-fired boiler use some form of inferential measurement of fuel Btu input.

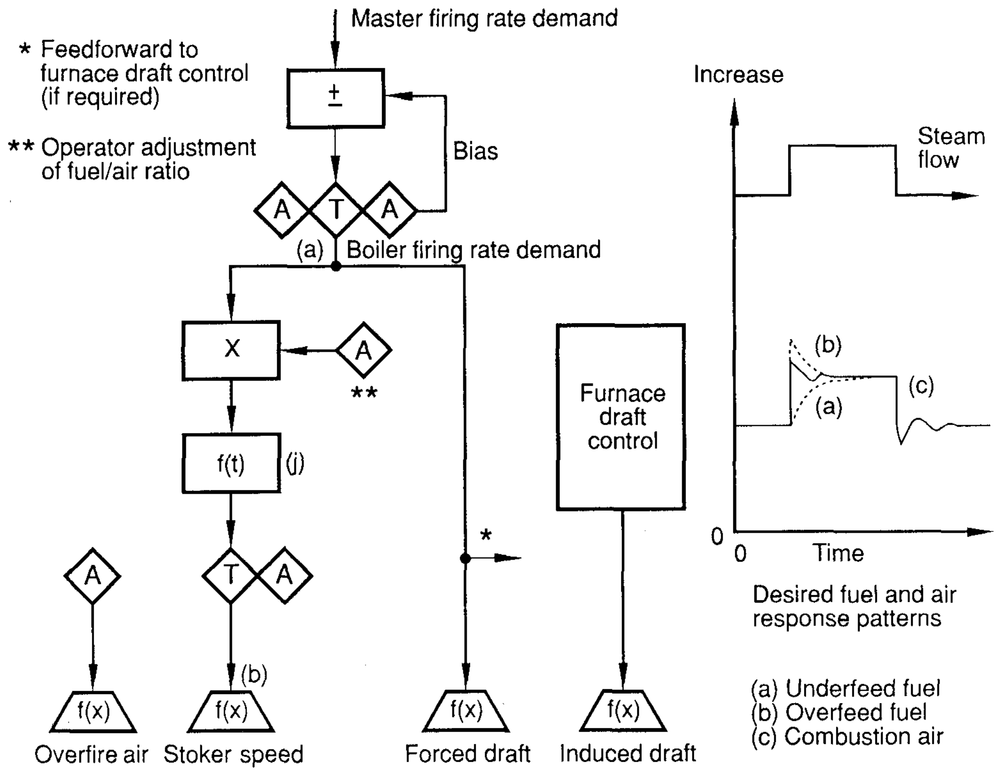


Figure 24-2 Parallel Positioning Control System for Underfeed or Overfeed Stoker-Fired Boilers

24-2 Inferential Measurement of Combustion Conditions in Boilers

The most common form of inferential relationship that defines combustion conditions is the steam flow/air flow relationship. This relationship was discovered in the early 1900s and patented as the first practical combustion guide for coal-fired boilers.

Steam flow is the usual measurement of heat output. If the combustion air is proportioned to the steam flow (heat output), certain combustion conditions will exist, thus producing a certain boiler efficiency. The steam flow heat output can be divided by the efficiency to produce heat input. Steam flow is thus an inferred measurement of heat input and, when correlated to combustion air flow, implies a relationship between fuel input and air flow. There are known specific limitations to the use of this technique:

(1) Steam flow is not proportional to heat input when load is changing and over- or underfiring is necessary to satisfy the heat storage requirements of the boiler.

(2) Changing the excess air changes the boiler efficiency and thus changes the inferred relationship between steam flow and heat input.

(3) The cleanliness of the boiler (soot on the fireside surfaces) changes the boiler efficiency and the relationship between steam flow and fuel heat input.

(4) Changes in feedwater temperature or boiler blowdown cause the relationship between boiler steam flow and heat output to change. The result is a change in the steam flow/heat input relationship.

(5) Operating at other than design steam pressure and steam temperature produces errors in the steam flow measurement. The result is a change to the measured steam flow/heat input relationship.

While the steam flow/air flow relationship is not a precise measurement of combustion conditions, the errors above are usually not large errors. The steam flow/air flow relationship has been used very successfully as a combustion guide and has been applied as an automatic control input for approximately 70 years.

The steam flow/air flow relationship is a field calibration based on the results of a series of combustion tests over the operating range of the boiler. The calibration of the air flow measuring device or devices must be adjustable so that its calibrated output matches the output of the steam flow measurement when the desired combustion conditions are obtained. The air flow measurement is, therefore, a relative one and does not have an output calibrated in terms of pounds or cubic feet. In this respect, the calibration method is the same as the relative air flow calibration used when burning liquid or gaseous fuel.

24-3 Parallel Positioning Control Systems with Steam Flow/Air Flow Readjustment

Because of the steam flow/air flow limitations listed above, the steam flow/air flow relationship is recommended for use as a corrective or trim control. During a rapid load increase, close adherence to the desired 1:1 relationship of steam flow to air flow will cause a reduction in excess air. The result may be smoke, higher furnace temperature, and potential damage to the grates. For this reason, there should be a deviation from the 1:1 relationship during load changes in order to maintain a more constant excess air level. Therefore, the steam flow/air flow influence should be delayed so that it is not felt or averaged into the control system during the under- or overfiring of load changes. An arrangement for the control of a spreader stoker-fired boiler that meets these conditions is shown in Figure 24-3.

The boiler firing rate demand (a) is the direct signal to the stoker lever driving device (b). This signal also acts as the input to function generator (c) with the output of the function generator positioning the overfire air damper (d). The function installed in this function generator is an average function based on observation of required values over a period of time. The operator has available a bias of the output of this function for more accurate positioning of the overfire air.

The boiler firing rate demand signal, through its input to the summer (e), serves as the initial and primary signal for controlling the forced draft. The initial calibration of the system is that of a parallel positioning system. Based on data collected from boiler combustion tests, the control signal vs. fuel flow and control signal vs. desired air flow relationships are linearized and aligned, using linkage angularity and positioner cam shapes.

During these tests, data is also collected for the calibration of the relative air flow measurement (f). This measurement is calibrated so that the value of the signal (f) and the value of the steam flow signal (g) are equal over the entire load range. This ensures that when steady-state combustion conditions are correct, the output of the delta (h) is equal to 0. Upon a change in boiler load that would cause signal (h) to change in a plus or minus direction, it is delayed in its input to summer (e) to allow a return to steady-state conditions before appreciable effect on the output of summer (e).

Note that there is no integral shown in this control loop. The proportional offset is adjustable through the gain adjustment on the steam flow/air flow relationship. The tuning is not critical due to the variable time constant adjustment in the time function (i). The overall tuning is a result of the gain in delta (h) plus the time constant adjustment (i).

Within limits, different combinations of the gain and the time constant will produce satisfactory results. A higher gain in order to produce a lower proportional offset requires a longer time constant. If a proportional offset exists during steady-state operation, the operator can use the fact of the offset to recognize the need for manual stoker adjustment or other action resulting from changing coal quality or boiler fireside fouling, etc.

Assuming that the system logic can accept signals with both positive and negative values,

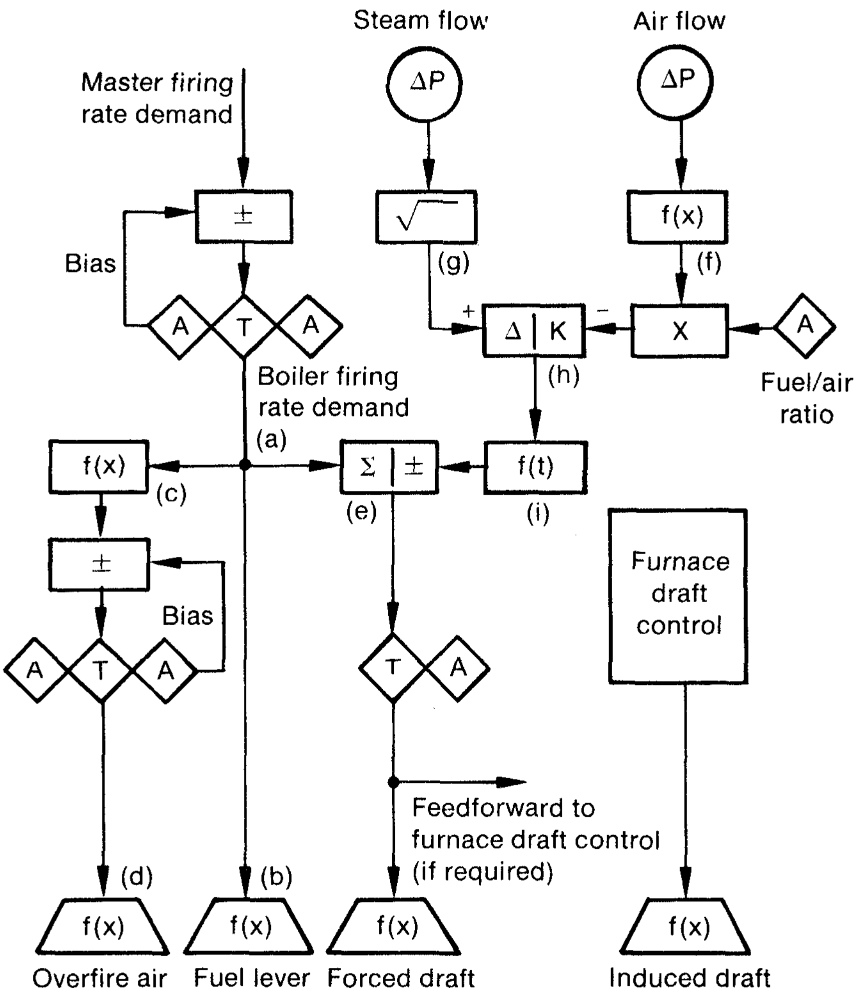


Figure 24-3 Parallel Positioning Control System with Steam Flow/Air Flow Readjustment for Spreader Stoker-Fired Boiler

the output of the delta (h) and the time function (i) are 0 or nearly so under steady-state operation. The signal (a) should have a gain of 1.0 at its input to summer (e). With a 0 signal from time function (i), the bias adjustment of summer (e) would be set at 0. If the system can operate only on positive value signals, a plus 50% bias would be added at the output of delta (h) and a minus 50% bias adjusted in the summer (e).

A parallel plus steam flow/air flow readjustment control system for underfeed and overfeed stoker-fired boilers is shown in Figure 24-4. There are two essential differences between this control arrangement and that for a spreader stoker. The signal from summer (e) is shown connected to fuel with the direct signal (a) connected to the forced draft. This is a result of the different basic response characteristics to fuel and air flow changes. The other change is the addition of the time function (j). This allows the fuel response to be adjusted to a desired fuel response curve as shown in Figure 24-2. As indicated previously, when overfeed boilers are used, depending on the load change requirements and the boiler design, the time function might be eliminated or reversed into a proportional-plus-derivative function. All other functions of the control system and their calibration are the same as that for the spreader stoker system of Figure 24-3.

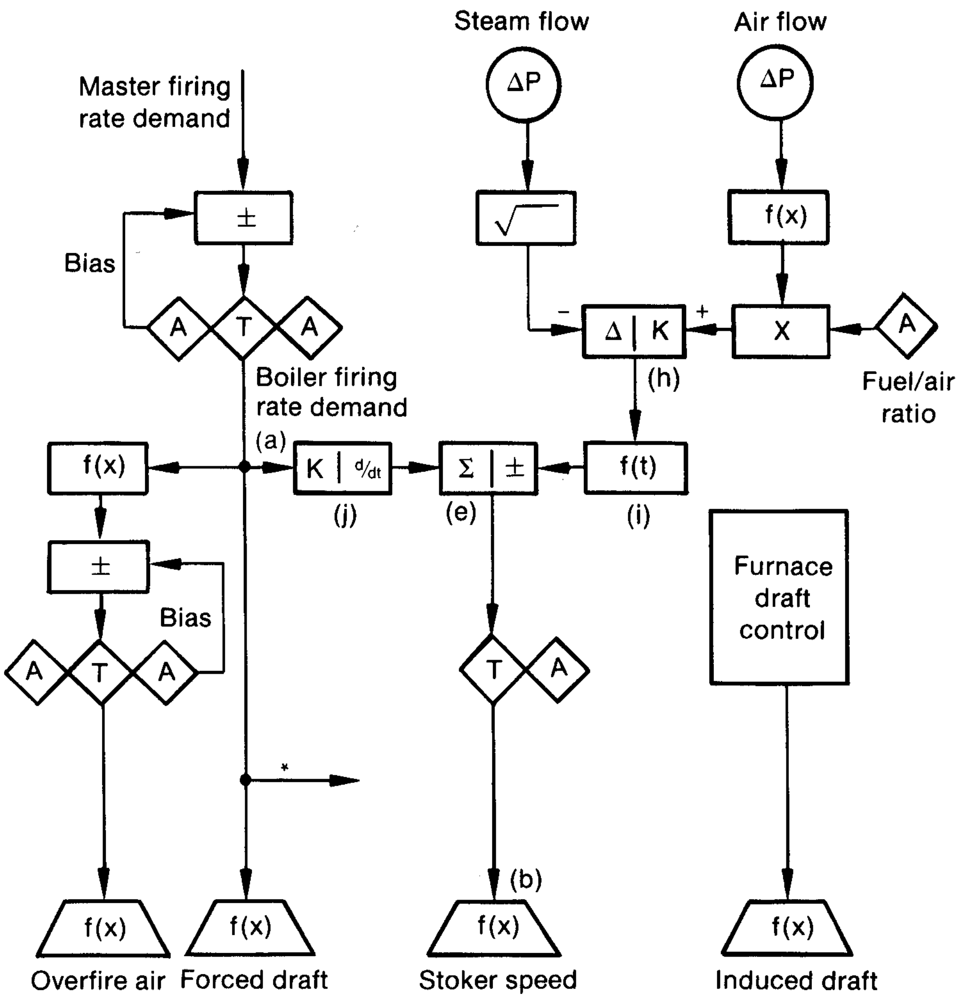


Figure 24-4 Parallel Positioning Control System with Steam Flow/Air Flow Readjustment for Underfeed and Overfeed Stoker-Fired Boilers

In addition to the operation of the control system, when using overfeed boilers, operator action is required. The operator must set the fuel bed thickness with the coal gate height. From an automatic control standpoint, only the stoker speed is controlled, and it is assumed the coal gate height will be constant. If a boiler were to be operated this way on a continually increasing load, the end of the fire line would gradually move toward the end of the stoker, and, ultimately, at the maximum load much of the coal would be lost over the end of the stoker. In recognition of this the operator should use indices of load as signals to reset the gate height. Gate height has been automated in a very few cases.

The operator must set air flow distribution with the compartment dampers. In the control arrangement shown, overfire air is controlled as a function of boiler firing rate, but the operator must bias the setting if necessary. A furnace draft control loop is also necessary with the balanced draft boiler. When blowing soot, the operator may need to increase the furnace draft set point to a more negative value. He may also need to increase the combustion air-to-fuel ratio during the soot blowing period. Therefore, the total control system is the result of many actions based on visual observation by a skilled operator and an imperfect automatic control arrangement.

24-4 Series Ratio Control Systems for Stoker-Fired Boilers

Another form of control system that is applied to stoker-fired boilers is known as the series ratio system. This type of system makes use of the steam flow/air flow relationship without the direct linkage of a parallel positioning system. The linkage of the fuel and air control is provided through the process. The net result is control action that is similar in its overall pattern to that of the parallel positioning plus steam flow/air flow readjustment. Figure 24-5 shows such a system for a spreader stoker-fired boiler.

The boiler firing rate demand (a) directs the fuel feed through adjustment of the fuel lever. In the arrangement shown, overfire air is controlled in the same manner as in the parallel-plus-readjustment type of system. The boiler firing rate demand is a measure of the requirement for steady-state steam flow plus or minus the need for adjustment to boiler energy storage. This signal need not be connected to the combustion air flow control if other signals that are analogous to the firing rate demand signal are available.

In the system shown in Figure 24-5, the firing rate demand signal (a) controls the fuel lever directly as in the parallel and parallel-plus-readjustment type of system. The signal from the proportional-plus-derivative function (b), (k), and (h) is analogous to the firing rate demand signal (a). The steam flow portion is measured steam flow with the proportional-plus-derivative function representing the rate at which the boiler energy storage should be adjusted.

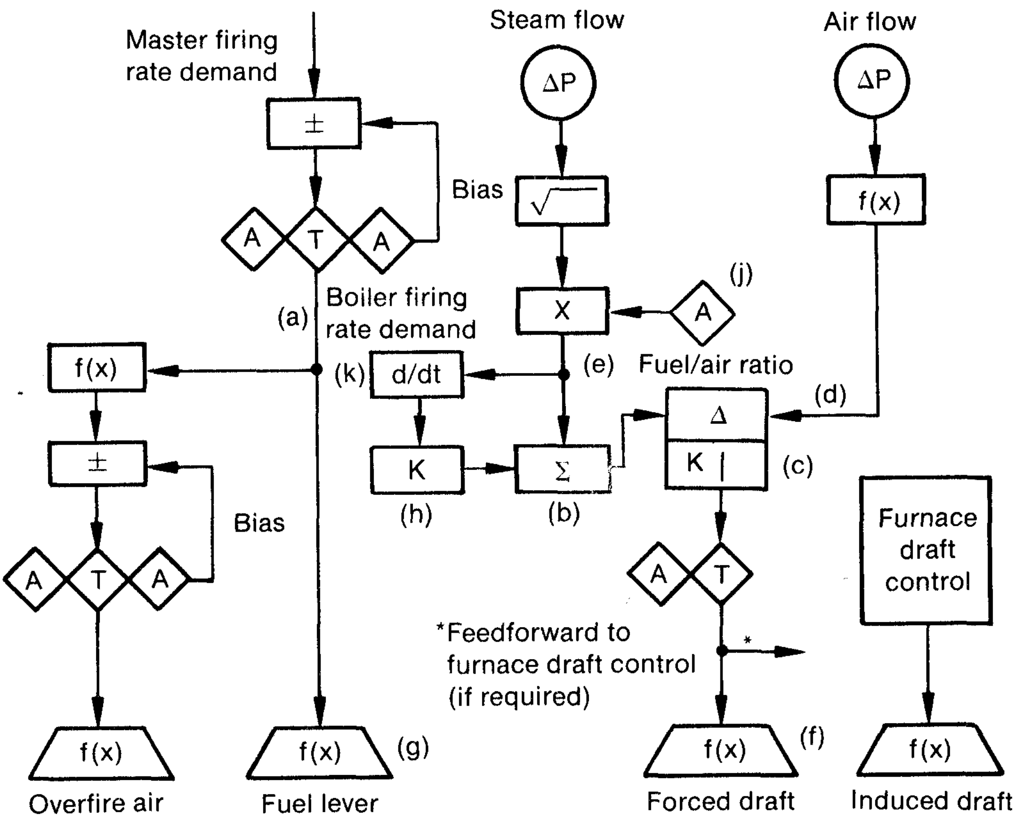


Figure 24-5 Series Ratio Control System for Spreader Stoker-Fired Boilers

In tuning the combined function (b), the proportional gain (h) is adjusted to provide the magnitude of the over- or underfiring necessary, with the derivative time (k) representing the time period over which the over- or underfiring takes place. The result is a proper set point for the combustion air flow proportional-plus-integral controller (c). The direct input from steam flow into the combining summer (b) should be set at a gain of 1.0 in order that steam flow and air flow will be together at all loads. Since the derivative action takes place only during a load change, the steam flow/air flow relationship is in 100% control of the air flow during the steady-state condition of boiler load.

Because the firing rate demand signal (a) is not used in the control of combustion air, the fuel (g) and forced draft (f) devices do not require the careful alignment calibration of the parallel-plus-readjustment type of system. They should be linearized so that control response will be equal at all loads. Since full reliance is on steam flow/air flow during steady-state operation, the relative air flow measurement (d) must be carefully calibrated to match the steam flow measurement (e). As described earlier, this calibration is based on a fuel/air ratio setting (j) of 1:1 and combustion tests at three or more boiler loads over the boiler load range.

A series ratio control system for an underfeed or overfeed stoker-fired boiler is shown in Figure 24-6. In this arrangement, since the boiler is much more responsive to air flow than fuel flow, the firing rate demand signal (a) is the set point of the air flow controller (h) with feedback from the relative air flow measurement (d). Assuming steady-state operation and a fuel/air ratio setting (j) of 1:1, the steam flow measurement (e) directly positions the stoker

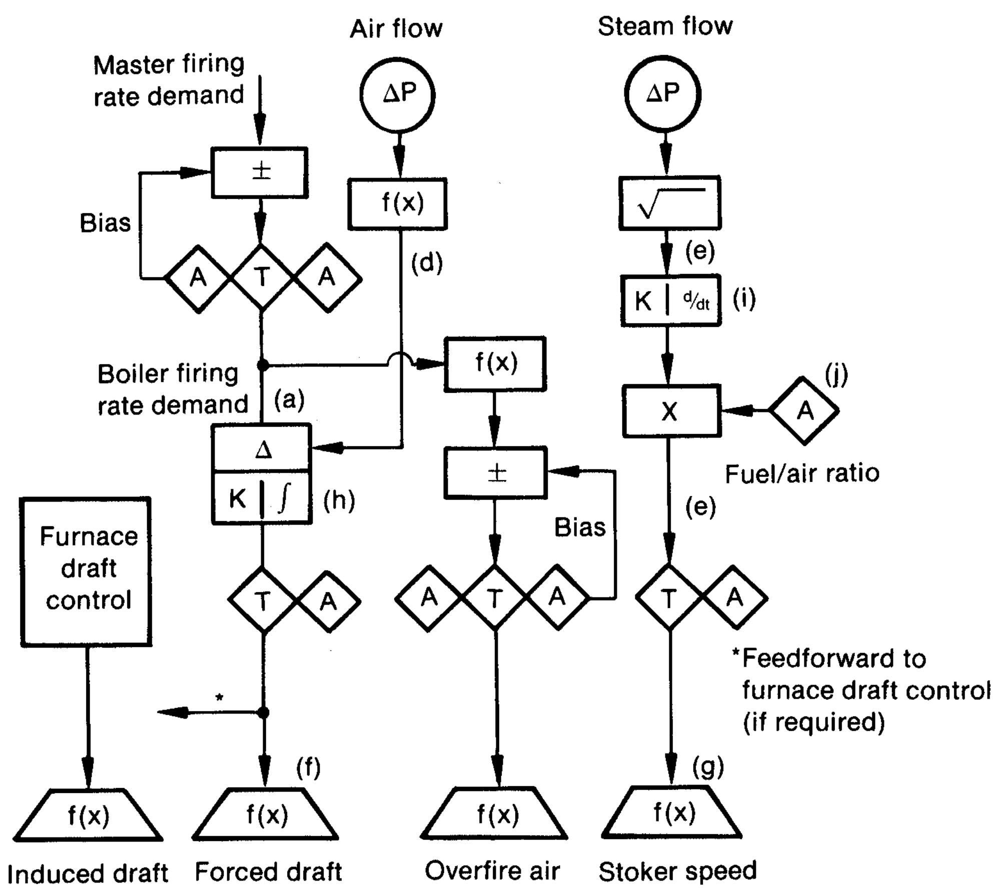


Figure 24-6 Series Ratio Control System for Underfeed and Overfeed Stoker-Fired Boilers

speed (g). During a load change, the time function, or potentially proportional-plus-derivative, (i) smooths and/or magnifies the change in fuel flow to the stoker as shown on the desired fuel response curve in Figure 24-2.

The characteristic of this system is that variations in coal quality or other operational factors affecting the system operation will create a steady-state offset in the steam flow/air flow relationship. As in the case of using proportional-only readjustment in the parallel-plus-readjustment system, the operator can use the offset indication to provide additional intelligence to his part of the overall control system operation. This type of system will produce an offset based on a steam flow/air flow gain of 1.0 instead of the adjustable gain of the other type of system.

The system is aligned and calibrated by matching the stoker speed control (g) characteristic to the air flow measurement signal (d). Since the stoker speed vs. coal flow characteristic is essentially linear, the calibrated air flow signal will also be nearly linear. As before, this calibration is based on boiler combustion tests at three or more loads and includes the variable excess air vs. boiler load relationship.

24-5 Applying Flue Gas Analysis Trim Control to Stoker-Fired Boilers

Flue gas analysis trim control for stoker-fired boilers is applied by substituting a block of trim control functions for the manual fuel/air ratio control shown in Figures 24-1 through 24-6. It should not be applied indiscriminately where factors other than flue gas analysis limit the reduction of excess air. These limiting factors to the reduction in excess air are discussed in Section 17. Factors that particulary apply to stoker-fired boilers are:

(1) carbon in the refuse,
(2) slagging, difficulty in ash handling,
(3) smoke and particulate carryover, and
(4) burning of grates from high furnace temperatures, other furnace and stoker maintenance.

If smoke is the limiting factor, the trim control in Figure 17-10 can be used. If any of the above except smoke is the limiting factor, then the % oxygen trim control of Figure 17-7 may be used. A restriction is that the effect of the limiting factor should be included in the development of the curve for load vs. desired % oxygen. A trim control arrangement of % oxygen and ppm CO in accordance with Figure 17-10 would be permissible if the lower edge of the % oxygen band would not cause the breaching of one of the limiting factors. If there are no other limiting factors and the only limit to excess air reduction is the optimization of boiler efficiency, % oxygen and ppm CO can be used without reservation as the final trim control index.

Note that the use of % carbon dioxide plus ppm CO has not been discussed as a trim control method. While % oxygen can be used alone, % carbon dioxide cannot. For any particular fuel, a change in % oxygen is always accompanied by a change in % carbon dioxide. When excess air is reduced and carbon monoxide appears, however, the relationship is distorted and % carbon dioxide becomes an unreliable index. If, however, ppm CO is used along with % CO_2, the ppm CO furnishes the intelligence for determining whether or not the % CO_2 is a reliable index. Applied in such a manner, % CO_2 plus ppm CO can be used interchangeably with % O_2 plus ppm CO.

A change in the hydrogen/carbon ratio of the fuel changes the excess air/% CO_2 relationship, while the same change has a very minor effect on the excess air/% O_2 relationship. In addition, % O_2 analysis is more precise and less costly than a % CO_2 analysis; therefore, the natural choice is % O_2.

The application of any trim control arrangement should be carefully examined to determine its potential for breaching any of the limiting factors to reducing excess air. For example, one particular combination of flue gas analyses for use in stoker trim control loops is % CO_2 plus ppm CO. The particular application describes the application of ppm CO to the control of overfire air, with the undergrate air used to optimize the % CO_2. It can be shown analytically that unless some other limiting factor is used to stop the % CO_2 optimizing action, the result will be an excessive amount of carbon in the refuse.

When applying the flue gas analysis trim control to a particular control arrangement, the tuning of the trim control will be affected by the control elements downstream from the point at which the trim control enters the loop. Figure 24-7 shows the application of trim control to the parallel-plus-SF/AF system of Figure 24-3. Assume that the trim control is a % oxygen system as shown in Figure 17-7. The integral time setting of the trim control must always be significantly slower than the time function setting of item (i). The overall system will show instability due to interaction between these functions as these time settings are moved closer together.

The interaction is similar to that of a cascade loop, where the tuning time constant of the downstream controller integral action should be an approximate order of magnitude faster than that of the upstream controller. The practice should be to adjust time function (i) for proper action with steam flow/air flow control. The % oxygen controller integral setting should then be adjusted for overall system stability. The limitation above also applies to the addition of flue gas analysis trim control to underfeed and overfeed stoker-fired boilers. This arrangement is shown in Figure 24-8.

When preheated air is used with overfeed or underfeed boilers, it may be desirable to use

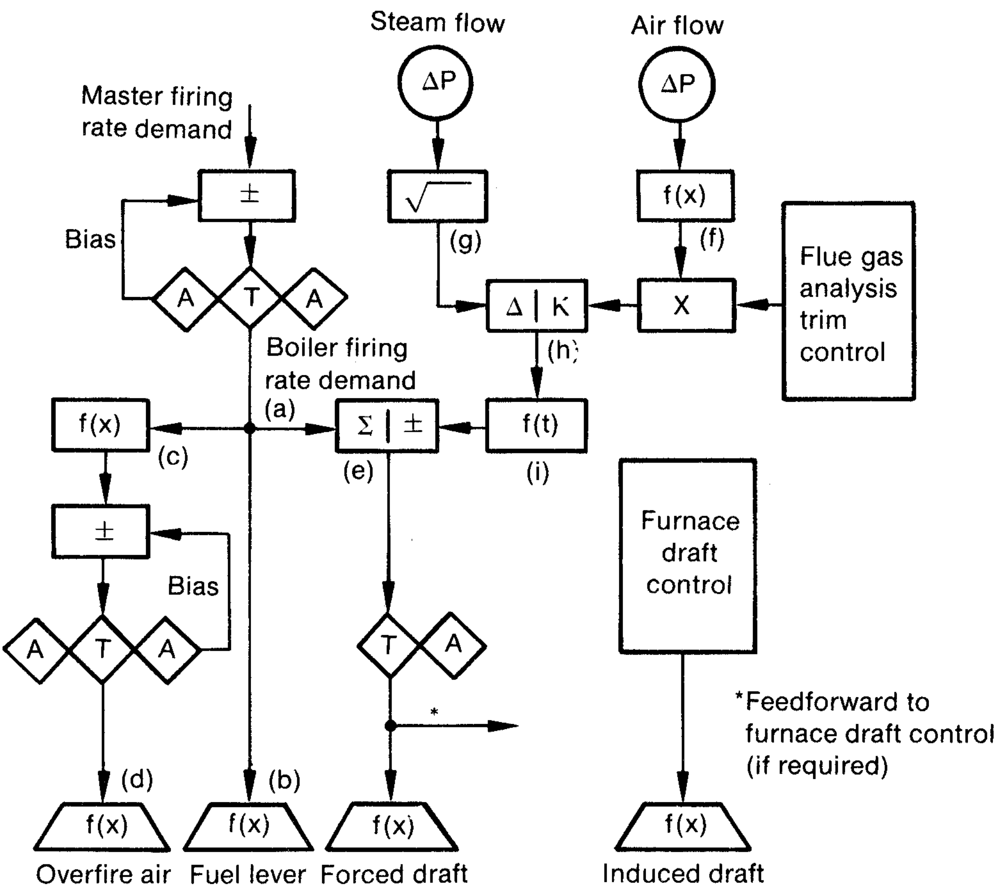


Figure 24-7 Parallel Positioning Control System with Steam Flow/Air Flow Readjustment and Flue Gas Analysis Trim Control for Spreader Stoker-Fired Boilers

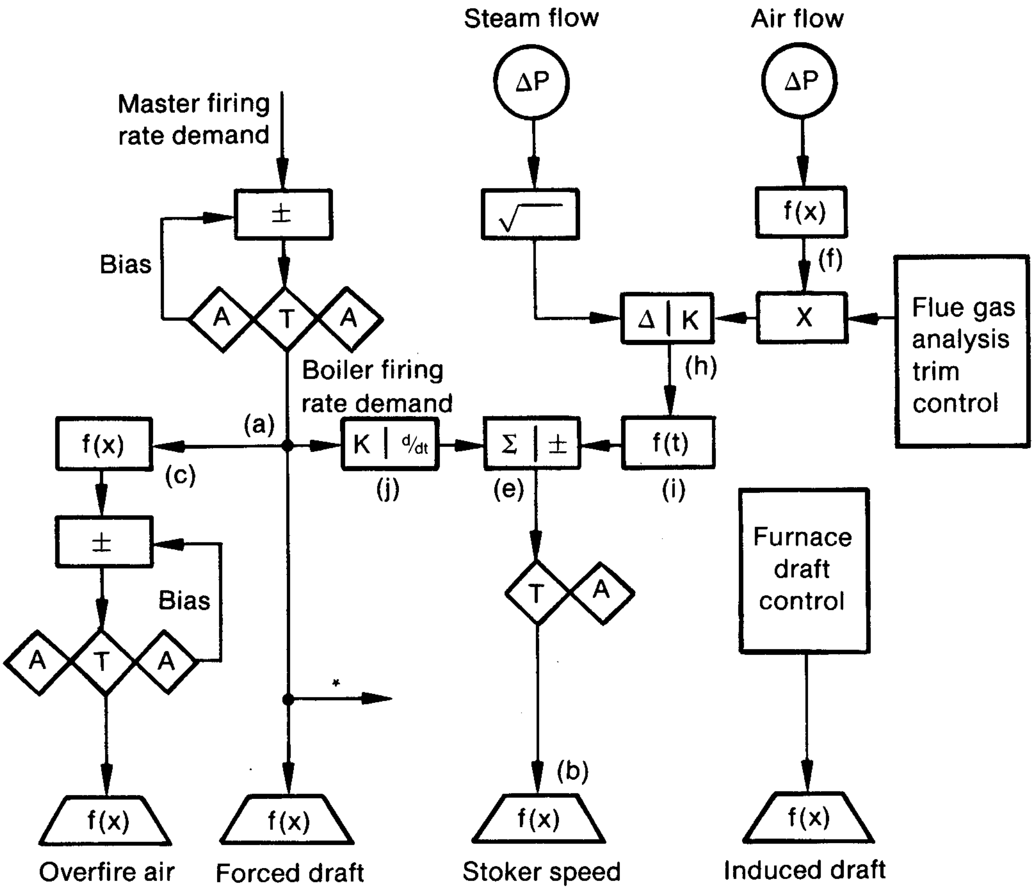


Figure 24-8 Parallel Positioning Control System with Steam Flow/Air Flow Readjustment and Flue Gas Trim Control for Underfeed and Overfeed Stoker-Fired Boilers

some method to limit the preheated air temperature. One method of producing a lower preheated air temperature is to install the air preheater as a parallel flow heater with the cold air and the hot flue gas entering the air preheater at the same end. To limit the air temperature by reducing the air preheater heating surface might cause a significantly higher draft loss across the gas side of the air preheater.

24-6 Combustion Control for Combination of Stoker and Liquid or Gaseous Fuel Firing

Some boilers are fired by using a combination of stoker firing of solid fuel and a separate burning system involving fluid or gaseous fuel burners. Typically, the solid fuel is a waste product such as wood bark or other wood waste, or solid refuse such as bagasse (the refuse from sugar cane) or other solid waste. The liquid or gaseous fuel may be burned as an auxiliary fuel to achieve a desired boiler steam flow, to temporarily replace the waste fuel, or to maintain ignition of the waste fuel.

The usual boiler air flow configuration is a single set of combustion air fans but with individual air flow control to the two sections of the boiler. In many installations the supply of waste fuel is intermittent, requiring the firing of auxiliary fuel. The intermittent supply of waste fuel may create periods when the stoker grate is bare and exposed to the radiant heat of the furnace.

All aspects of the boiler equipment arrangement and the operation needs must be considered when applying control equipment. Installations of this type are very individual, and there is no one good control application solution to all the potential operation problems. Essentially, two separate control systems are involved, and their manner of linkage determines the system results.

While the application solution may or may not be transferrable, the thought process required can be demonstrated by considering the needs of a particular installation. Assume a paper mill boiler that burns bark and wood waste with natural gas as an auxiliary fuel. The boiler is fired with a spreader stoker, with the gas burners mounted in the furnace wall above the stoker. The stoker and the gas burners have separate measurable and controllable combustion air systems. The combustion air is supplied by a single forced-draft fan with inlet vane control. There is also a single induced draft fan.

The boiler may be fired to full boiler rating on either fuel. The solid fuel is stored in a bin above the stoker and admitted to the stoker by variable-speed screws on the bottom of the bin. Additional solid fuel is admitted to the bin with a constant-speed belt conveyor that is not under the control of the boiler operator.

There may be times when the wood waste to the stoker diminishes or disappears through lack of supply to the storage bin or because screw feeders bridge over and do not supply fuel to the stoker. If there is no wood waste, the material on the grate will gradually be consumed and the bare grate will be exposed to full radiant heat of the furnace. If this happens, approximately 15% of full load air flow must be passed through the grates to avoid grate damage.

There are two desired operation modes: (1) the boiler steam loading follows the demand of the steam system for steam in parallel with other boilers connected to the plant header; (2) the boiler can be set at a given steam flow set point and can maintain that steam flow even though the plant demand for steam may change. Figure 24-9 is a control system application that might be designed for such requirements.

The master (plant) firing rate demand is adjusted to the desired boiler firing rate demand in the bias-type manual-auto station (a). When on automatic, the output of this station will follow the changes in demand of the plant steam system by changing the set point of the steam flow controller (b). If the station is placed in the manual mode, the steam flow will be set by the operator changing the manual output of the station. The output of controller (b) is the gas-firing demand, which actuates a typical gas firing system as discussed in Section 21.

The boiler firing rate demand from station (a) is also the firing rate demand for the stoker. The high select (e) limits the minimum value of the signal to 15%. The low select (c) limits the maximum value to approximately 25% higher than the amount of steam generated from the waste fuel. The stoker firing demand signal is applied directly to the speed of the stoker feeder and is also applied as a feedforward to the summer (k), which positions the combustion air flow to the bark or wood waste stoker. This part of the system can be recognized as the basic spreader stoker system shown in Figure 24-3.

The total measured steam flow cannot be properly compared against only the air flow to bark combustion, so a means must be found to determine the steam flow that is generated by the wood waste material. This is accomplished in the functions (f) and (g). The proportional function (f) provides a scaling function that involves the capacity of the gas flow measurement, gas Btu/scfh, Btu added per pound of steam flow, and boiler efficiency in burning the gas. The output of the function (f) is the steam flow rate for the gas that is being burned. The gas-burning steam flow rate is then subtracted from the total steam flow rate to obtain the rate of steam flow generation from burning the wood waste.

To protect the grates from overheating, the high select (h) prevents the air flow through the stoker from being reduced below approximately 15% of maximum stoker air flow. The control signals to the forced draft dampers for controlling the combustion air to the stoker, and the gas burners are totalized in the summers (i) and (j). The resulting total is used to position the inlet vanes of the forced-draft fan (m) and to act as a feedforward signal to the

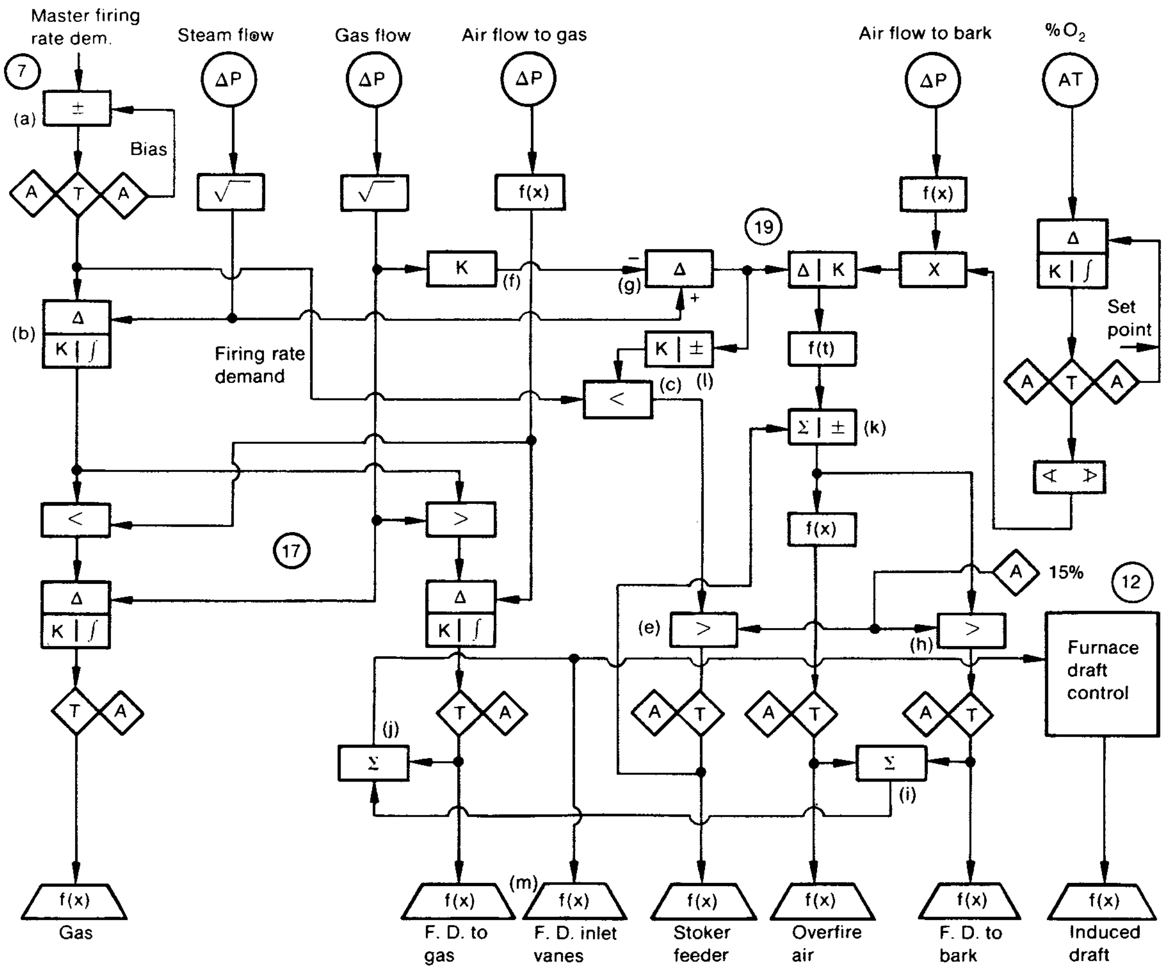


Figure 24-9 Control System for Combination of Stoker Firing and Firing Liquid or Gaseous Fuel

furnace draft control. Another approach to the forced-draft fan control is to use a forced-draft duct pressure control loop.

The flue gas analysis control, % oxygen in this case, is set at a constant % oxygen set point. When the boiler is on gas firing and the stoker grate is bare, the air flow cannot be reduced because of the 15% flow through the stoker grate. For this reason the flue gas analysis control is applied only to stoker firing.

Assume that the grate is bare and the boiler is operating at 50% capacity on gas fuel. The signal from (g) is 0% since there is no steam generation from the wood waste. The firing rate demand signal to the stoker is at 25% because of the bias set in item (l). Stoker steam flow at 0 plus 25% equals 25% from item (l). This 25% is compared in the low select (c) to the total firing rate demand of 50%. At this time, bark flow is admitted to the storage bin. This feeds bark at a 25% rate; the grate is gradually covered, and the minimum 15% air flow causes combustion and steam generation by the stoker. The increase in steam flow causes the controller (b) to reduce its output. This increases the stoker demand signal by increasing the maximum limit in (c). This will continue up to the capability of the stoker or until the value of the boiler firing rate demand signal is reached. The fuel gas will reduce to its minimum firing rate or intermediate firing rate as called for by controller (b).

The stoker firing controls will predominate and follow the boiler firing rate demand as long as there is waste wood or bark fuel available. If there is insufficient stoker fuel, the gas fuel will increase to satisfy the boiler firing rate demand. Should the stoker fuel be reduced to 0 for any reason, the stoker will burn the remaining fuel on the grate, and the boiler will again be maintaining the boiler firing rate demand with gas alone as the fuel.

There are a number of potential solutions to this control problem. One solution uses closed-loop control on the stoker feed by totalizing the speed of the stoker feeders and using calorimetric continuous calibration of the resulting total fuel signal. Generally, more elaborate and precise control applications are used as boiler capacities increase.

24-7 NFPA Purging and Interlock Requirements for Stoker-Fired Boilers

The code authority for purging and interlock requirements of stoker-fired boilers is ANSI/NFPA 85I 1989. This can be obtained from the National Fire Protection Association, Batterymarch Park, Quincy, MA 02269.

The following cold-start procedures are contained in NFPA 85I:

(1) Prior to starting ID fans, verify an open flow path from the inlet of the FD fan to the stack. Unless there is sufficient natural draft for initial firing, the induced draft should be started and normal furnace draft maintained.

(2) Verify that the grate is clear of ash and debris.

(3) Fill the feeder hopper with fuel, start the feed mechanism, and establish a bed of fuel on the grate.

(4) Place kindling on the fuel bed. Spray the kindling from outside the furnace with a light coat of distillate oil.

Caution: Gasoline, alchohol, or other highly volatile material must not be used for light-off.

(5) Open the furnace access door, light a torch, and ignite the wood by passing the torch through the door.

(6) When the wood on the bed of fuel is burning, start the ID fan (if not in operation), and place in automatic mode of operation.

(7) The overfire air should be started immediately to prevent damage from gases passing through the ductwork.

Caution: Undergrate air pressure should always be greater than furnace pressure to prevent reverse flow and potential unit damage.

(8) When the fuel is actively burning, start the FD fan with dampers at minimum position.

(9) Start the fuel feed. Observe operation and adjust the fuel rate and air as required until boiler pressure is at normal operating pressure.

(10) Place the fuel and air in automatic mode of operation.

The above indicates that the operation is completely manual. Purging is not mentioned, but interlocking the fans so that the forced draft fan cannot be operated without the induced draft fan is expressly implied. Note that this procedure is for lighting off the solid fuel. If other fuels such as gas, oil, or any volatile fuel is an auxiliary fuel and the stoker is not in operation, the entire NFPA 85 procedure for that particular fuel, including purging and flame proving, should apply. If the stoker is in operation, and the volatile fuel burner is to be lit, the procedure would be the same for lighting a second burner of a boiler with the first burner already lit.

For a hot restart after grate burning has stopped, follow the cold start procedure. If grate burning continues, follow the procedure omitting steps (2), (4), and (5).

NFPA 85I also describes certain types of action and operation of the combustion control system. There appears to be no conflict with the previous discussion in this book. Nothing in discussion of stokers and their controls in this book can be assumed to take precedence over ANSI/NFPA 85I 1989.

Section 25
Atmospheric Fluidized-Bed Boilers

The term fluidized-bed describes a process in which a bed of material is fluffed into a fluid mass by high velocity air or other gas that is applied to the underneath side of the bed. The effect is not unlike boiling water with the steam bubbles rising through the water and expanding the volume. For fluidized-bed combustion, the fluidizing air is the primary combustion air, with secondary air added as required to assure complete combustion.

The fluidized-bed process is not new and has been used for many years in the refining industry for fluid catalytic cracking and fluid bed hydrogenation and in other industries for such applications as coal drying and chemical vapor heating. A fluidized bed that operates essentially at atmospheric pressure is called an atmospheric fluidized bed.

Its use for boiler fuel burning is a somewhat newer process. Though a patent for the process was filed in 1944, fluidized-bed firing for boilers did not gain general recognition until the early 1970s. At that time the basic idea was for combustion to take place in the fluid bed, and direct heat transfer was obtained by locating boiler tubes in the fluid bed.

A basic concept is that combustion temperature is reduced by mixing a large amount of noncombustible bed material with the fuel. The effect of a reduced combustion temperature is a significant reduction in NOx formation. If the bed temperature is approximately 1600°F and the noncombustible material is a sulfur absorbing material such as limestone or dolomite, then sulfur in the fuel can be absorbed during combustion.

The material containing the absorbed sulfur is calcium sulphate or calcium magnesium sulphate, which becomes mixed with the chemically spent remains of the limestone or dolomite. This mixture of material is continuously withdrawn with the ash and replaced by fresh limestone or dolomite. Elimination of the sulfur in this way eliminates the need for costly flue gas scrubbers that would otherwise be required for the elimination of SO_2 and SO_3 from the flue gases.

If sulfur capture is not required, the fluidizing material can be sand or similar material. For such systems the material is not involved in the combustion process and does not require continuous removal from the bed. A small amount of makeup material to replace that mechanically removed with the ash or drained from the bed would be required.

A number of pilot installations were installed during the 1970s, with the fuel-burning capacity of successive installations growing larger or using various combustible materials. One of these combustible materials was the long-term accumulation of coal mining waste, called "culm," that contained a significant percentage of carbon. Fluidized-bed combustion was successful in recovering this wasted energy, and a number of installation have been made. Other installations were for industrial or municipal waste. During the 1980s an upscaling of capacity and new applications were being tested in sizes for utility plants up to 160 MW. The fluidized-bed boiler process is now being applied commercially where such installations can be justified on an economic basis.

Atmospheric fluidized-bed boilers are generally of two basic designs: "bubbling bed" and "circulating bed." Design variations may use elements of both of these. Circulating beds are lighter, with less material per unit volume, and are called "low density" beds. Low density is acquired by building up the fluidizing air pressure to achieve greater air velocity and expanded bed volume. Such a bed can also have a "high density" mode in which the fluidizing air is at a lower pressure and the material is fluidized to a lesser volume.

While a significant number of both types of fluidized-bed boilers have been installed, the process is still evolving. The control application practices are not firm and may be significantly influenced by the particular manufacturer's process design. The discussion in this section should, therefore, be viewed as a general guideline or starting point to be rationalized or modified in accordance with actual requirements.

25-1 Bubbling Bed Fluidized-Bed Boilers

A diagram of a bubbling bed fluidized-bed boiler is shown in Figure 25-1. The bed is located at the bottom of a stoker-type waterwall furnace with a significant portion of the steam generating tubes buried in the bed. Fuel is added by a screw conveyor, a gravity chute, pneumatic injection by an air stream of sufficient velocity, or a spreader stoker. Noncombustible bed material is added in the same manner as the fuel.

The bed is fluffed to the bubbling condition of a high density fluidized-bed by fluidizing air that is admitted below the bed. Since the bed is only bubbling, the flyash or other carryover from the bed is relatively low. In this respect the closest approximation is that of a spreader stoker. This carryover is cleaned from the flue gas at the end of the process, and, if significant amounts of carbon remain, it is returned to the bed.

Since the bed is in a bubbling and high density condition, changing the firing rate of an individual bed is limited to a maximum turndown of approximately 2:1. If greater turndown of the boiler firing rate is required, the boiler must use multiple bubbling beds. In this case, if a load of less than 50% is required, one or more of the beds would be slumped or allowed to settle by reducing the fluidizing air. Operation at the low loads may also require controls for alternately firing the individual beds, as they may tend to solidify if not in use for a period of time at the bottom of a hot furnace.

As in the change of fuel supply to a spreader stoker, the burning carryover from the bed and the fine fuel particles burn in suspension and provide some immediate load following response. Because of the large amount of unburned fuel inventory in the bed at all times, the major portion of load following response results from changing combustion air flow. Changing the total combustion air relative to the fuel input affects the bed temperature and can be used to control bed temperature. Draining or adding material to the bed also affects bed temperature.

Several general statements concerning the basic operating control of such a boiler can be made.

(1) The firing rate demand control and steam temperature control are no different from those of any other boiler and can be implemented in accordance with the discussion in Section 8. Typically, the firing rate demand change calls for the addition or reduction of fuel and fluidizing air in parallel.

(2) Fluidized-bed boilers are balanced draft and include both forced and induced draft fans. The furnace draft control and air flow measurement are the same as those of other boilers and can be implemented as discussed in Sections 14, 15, and 16.

(3) The feedwater control system requirements are identical to those of other boilers and can be implemented as discussed in Section 13.

(4) The key issue that differs from other boilers is the control of the fuel and the bed temperature. Bed temperature control may also involve the control of the ratio of fuel flow to combustion air flow.

If the fluidized-bed purpose is to capture sulfur from the fuel, it is particularly necessary to control bed temperature. The most efficient performance in sulfur capture occurs with the bed temperature between 1500°F and 1600°F. This temperature can be controlled by changing the mass of material in the bed, trimming the air flow to change the amount of fluidizing air with respect to fuel and other material that is being added to the bed, or changing the ratio of different air flow streams.

One manufacturer of smaller industrial bubbling bed boilers suggests trimming the control of air flow from bed temperature as shown in Figure 25-2. If this is done, the result may be a bed temperature change but also a change in fuel/air ratio. Such a change in fuel/air ratio would be indicated by a change in the analysis of the flue gases.

Another manufacturer suggests using bed temperature to trim the air flow control and, in addition, to use the flue gas analysis to trim secondary air flow. The net result would be no change in total air flow but with a lowered amount of fluidizing or primary air.

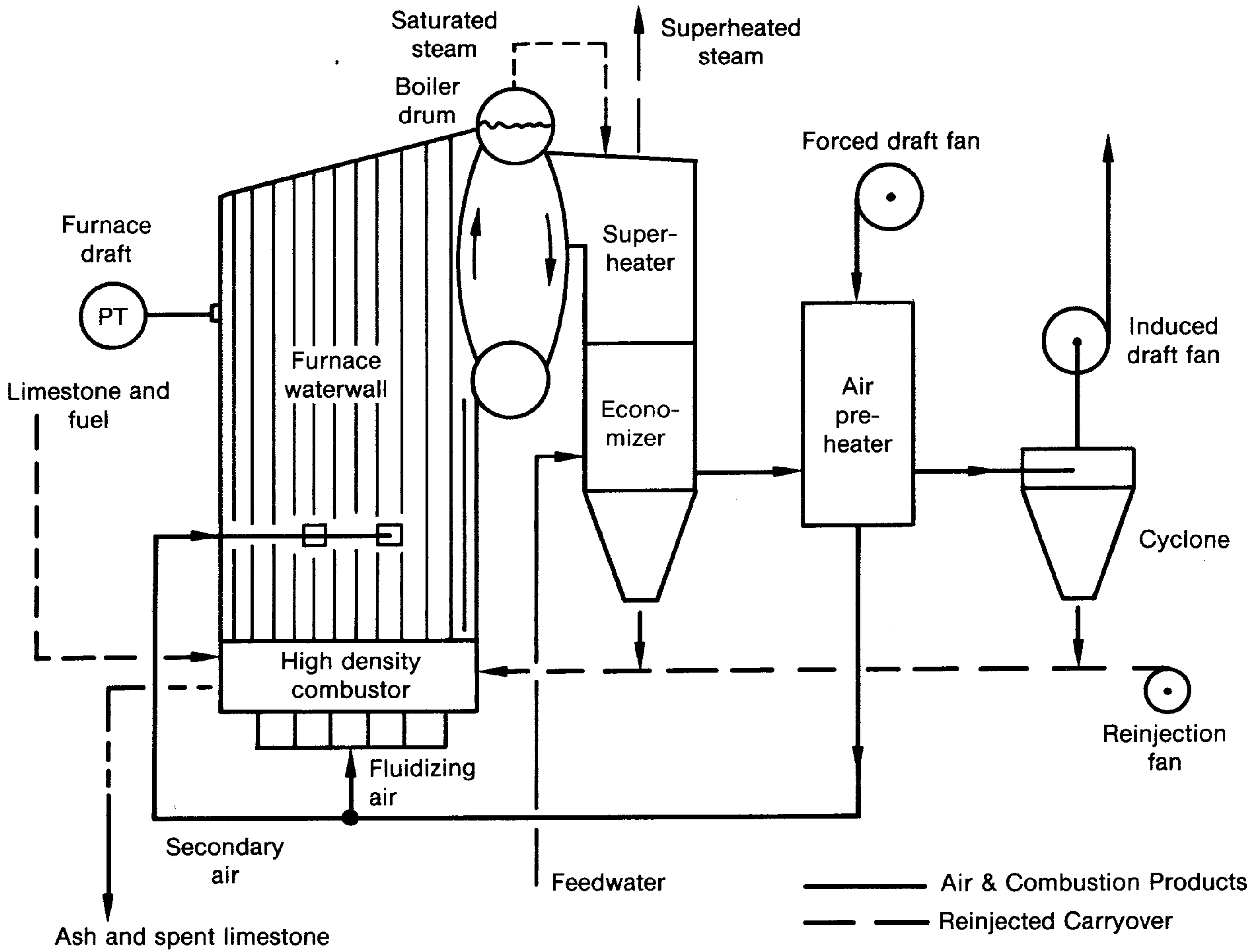


Figure 25-1 Bubbling Bed Fluidized-Bed Boiler

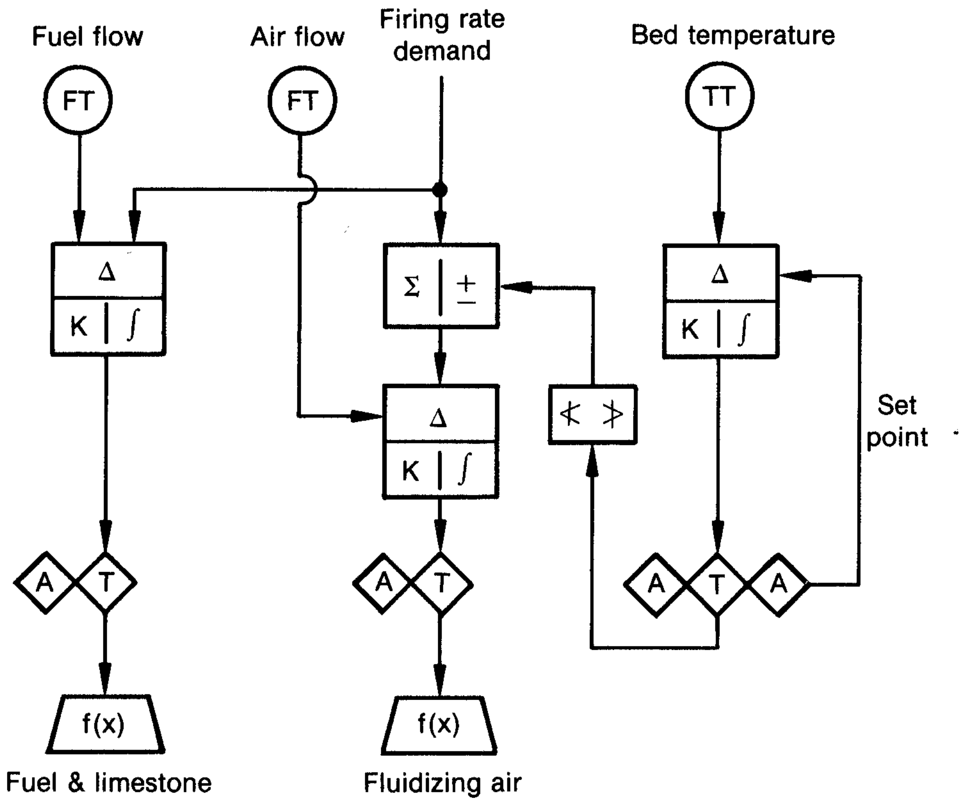


Figure 25-2 Bed Temperature Trim of Air Flow Control

An indication of bed temperature change may indicate an incorrect total amount of bed material or the amount of fuel in the bed relative to other material. If the problem is an incorrect total amount of bed material, the solution is to drain material from or add material to the bed. If the bed temperature is high, adding more material to the bed and increasing its height may cool the bed due to the cooling effect of more boiler tubes that are immersed in the bed. If the problem is due to an incorrect amount of fuel relative to the bed material, an alternative that appears to involve less process interaction would be to trim the fuel input rate from bed temperature. Flue gas analysis trim control in accordance with the discussion in Section 17 would then be used to trim the control of air flow.

For sulphur capture systems, fuel and limestone or dolomite are added in a set ratio. On a longer term basis the ratio is adjusted by measurement of the residual SO_2 in the flue gas. Due to the effect of bed temperature on sulfur capture efficiency, such a system requires a proper control of bed temperature.

25-2 Circulating Bed Fluidized-Bed Boilers

By increasing the velocity of the fluidizing air above that of the bubbling condition, the bed volume expands, and the result is a low density fluidized bed. With the expanded volume, higher velocity, and reduced density, a significant percentage of the bed fuel and other material leave the bed and are carried over to be collected and reinserted into the bed.

A diagram of a circulating fluidized-bed boiler is shown in Figure 25-3. At the bottom of this diagram is the fluidized-bed combustor containing the fuel and a relatively large amount of fluidizing material. Boilers with single combustors up to a capacity of approximately 900,000

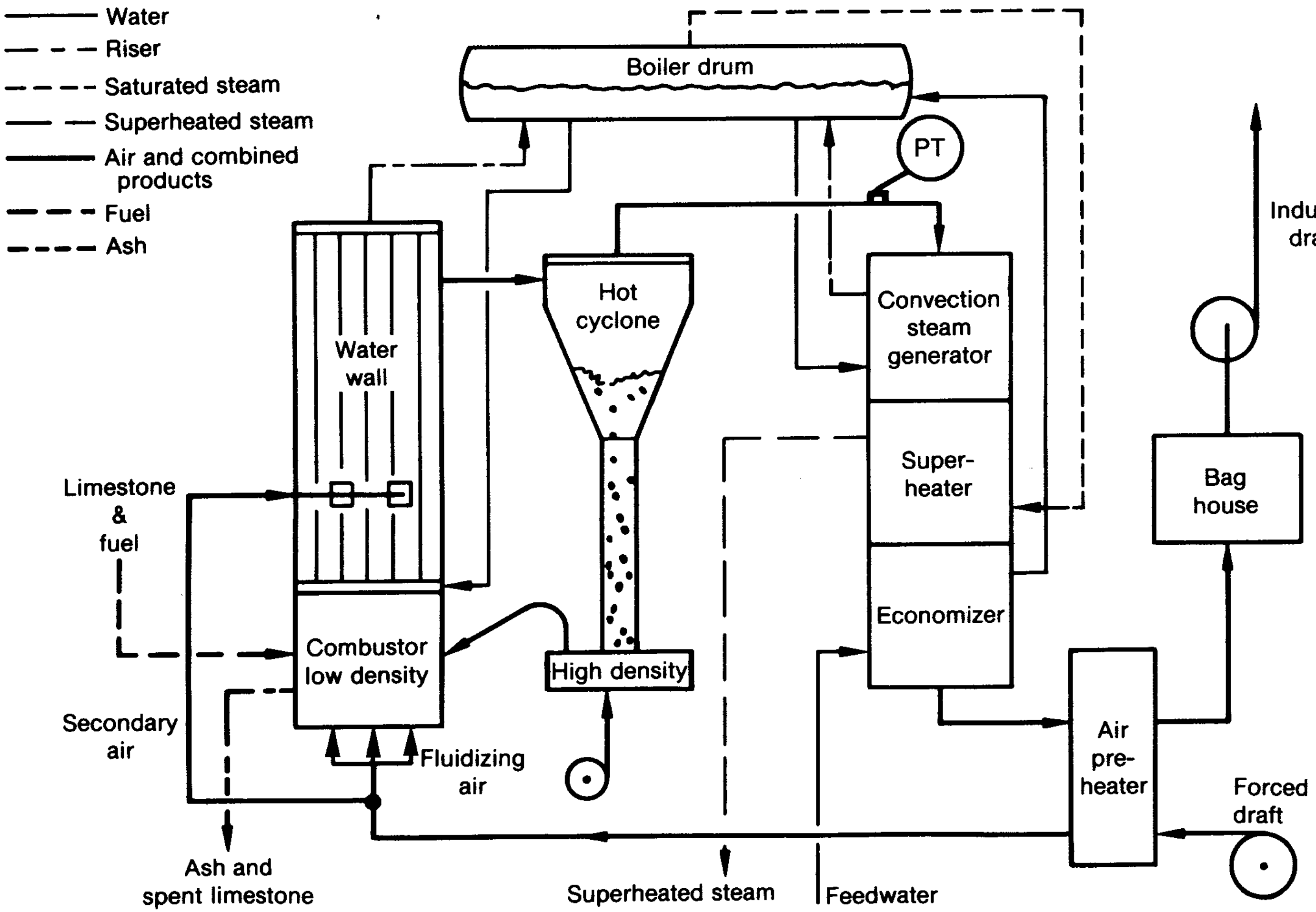


Figure 25-3 Circulating Bed Fluidized-Bed Boiler

lbs of steam per hour are now operating. The input of the fuel and the other material is added to this combustor.

A waterwall heat transfer furnace section is shown above the fluidized-bed combustor. As in the bubbling bed, the fluidizing air enters at the bottom of the combustor and acts as primary combustion air. As the combustion gases and carryover material leave the combustor, the combustion is incomplete.

The hot gases and burning particles leave the combustor, bounce off the waterwall heat transfer surface, and transfer heat by conduction, convection, and radiation before entering a hot cyclone or other type of particle separator. This simplified schematic drawing does not show that the primary and secondary air circuits are actually two complete air flow systems with individual forced draft fans and air preheaters. The secondary air that is used for fluidizing air is the higher pressure of the two. This secondary air may enter the bed from the underneath or be inserted at higher levels in the bed. A system of median selection of a number of bed thermocouple temperature measurements is used for proportioning the secondary air.

As shown in Figure 25-3, the hot gases exit from the hot cyclone particle separator and then travel into and through the convection steam generating section. This flue gas then transfers additional heat in a superheater and/or economizer before passing through a combustion air preheater, induced draft fan, and particulate collecting equipment such as a precipitator or bag house.

In the arrangement shown, centrifugal force throws the heavier solid material to the outside of the hot cyclone collector, and gravity causes these solid particles to fall. The material that is collected is then reinjected into the fluidized bed by gravity flow through a loop seal. To assist this gravity flow, a small bed at the bottom of the collector standpipe is lightly fluffed by high pressure air into a higher density fluid bed. The static head of the material in the standpipe below the particle separator overcomes the difference in pressure due to draft loss and acts on this small higher density bed, causing the material to flow into the low density bed of the combustor. Any fluidizing air of the small high density bed becomes secondary combustion air, with other secondary air admitted to the furnace above the fluidized bed. This circulation is continuous and can be increased or decreased by changing fluidizing air velocity and volume, thus changing the bed density and volume.

Controls for the circulating bed type of boiler are essentially the same as for the bubbling bed type. Some manufacturers may require controls for the circulation part of the installation, but others do not. As more material leaves the combustor, more is collected in the hot cyclone and returned to the bed.

As with the bubbling bed, the process is evolving, and the manufacturer's control strategy should be the first approach to control application design. Steam temperature control and feedwater control are the same as for other boilers. Controls for firing rate demand differ to some extent. The circulating fluidized-bed boiler can be turned down over a greater range and a number of multiple beds is not required to obtain greater turndown than the 2:1 available with a bubbling bed. This greater turndown availability eliminates the requirement for any controls to alternately fire the multiple beds under a low load condition.

The furnace draft control balance point is shown on Figure 25-3 immediately above the convection section, with the suggested initial set point value of 0.5 inches of H_2O. Furnace draft control application as discussed in Section 15 is appropriate for the circulating fluidized-bed boiler.

Most of the factors concerning bed temperature control with bubbling bed boilers apply equally to circulating bed boilers. An exception is that the circulating bed does not have the capability of bubbling beds to be cooled by adding bed material. In addition, the fireside waterwall outlet temperature, which includes the solid material temperature, can be affected by the distribution between the primary fluidizing air flow and the secondary air flow that is added above the bed.

Because of the much larger amount of material that is returned to the bed with the circulating bed boiler, the temperature of the solid material that is returned also affects bed temperature. One manufacturer uses this fact in the process design and the bed temperature control strategy.

With the number of factors that can affect the bed temperature and the differences in the way these factors interact with different boiler designs, some period of testing and evaluation of different process and control strategies will continue to be necessary for some time in order to solidify design practice.

25-3 NFPA Requirements for Atmospheric Fluidized-Bed Combustion System Boilers

The applicable code document for fluidized-bed boilers is ANSI/NFPA 85H 1989, ''Prevention of Combustion Hazards in Atmospheric Fluidized Combustion System Boilers.''

Because these boilers have auxiliary or start-up burners utilizing gas or oil, purging and other practices included in the NFPA 85 requirements related to that fuel must be adhered to.

There are a number of differences when compared to conventional boilers. The starting and shutdown sequences for fluidized-bed boilers are designed to preserve the temperature of the bed material and refractory while providing safe operating conditions. As a result, the warm-up cycle for cold start-up and hot restart as well as the shutdown sequence are different from a conventional coal-, oil-, or gas-fired boiler. For example:

(1) On a cold start-up, after a normal purge period, air flow (depending on the design) may be reduced below the purge value to provide for a proper warm-up rate.

(2) Another deviation involves a hot restart. If the bed material is above a predetermined minimum ignition temperature, fuel may be admitted to the boiler or warm-up burners may be started to preserve bed temperature without the normal purge cycle.

(3) Tripping the fans or diverting air flow from previously active bed sections is allowed shortly after a master fuel trip (MFT) without a normal post purge.

Other requirements in NFPA 85H 1989 describe many other operational details and requirements. A copy of this document should be studied for the additional insight it gives on the operation of fluidized-bed boilers. It is intended that nothing written herein violates the NFPA 85H 1989 requirements. Should anything in this section be so interpreted, the requirements of ANSI/NFPA 85H 1989 should be followed.

Section 26
Control System Complexity and Future Directions for Boiler Control

This text deals with those aspects that most relate to the control of boilers and to the control application that results. The control application is essentially independent of the particular set of hardware and software with which the control application is implemented, whether pneumatic, electric analog, or digital. At this time the hardware/software revolution is virtually complete, and over 90 percent of future boiler control systems will likely be implemented with digital control.

The use of digital control will facilitate the integration of the on-line control with the digital logic for start-up, shutdown, and safety monitoring. The use of digital control also makes much easier the implementation by software of more complex control algorithms that were previously not used because of the expense of considerable additional hardware when implemented with analog control. This advantage is of considerable benefit in the size range of industrial and small electric utility boilers, but it is of particular benefit to the control of the larger and more complex electric utility boilers.

26-1 Complex Control Systems for Electric Utility Boilers Using Embedded Process Models

The control application design of such units has until today been based almost entirely on the skill and intuition of experienced utility boiler control application engineers. The use of what is known as ''model-based control'' methods to solve the complex and interactive control problems of these large multivariable systems has not been used to any significant extent. The successful use of such methods relies on the development of and inclusion in the control system of an accurate boiler model that runs significantly faster than its real-time counterpart.

In one form of such systems, the boiler model is connected into the control system as an observer with the same set of process inputs as the basic system. A change in the inputs results in a rapid computation of model outputs that predict what the process measurements of the real system will be, based on the control actions being applied. Any deviations between the control system set point values and the predicted future process measurement values are then used to correct the control actions before the predicted deviations appear as real deviations. If the process model is accurate and the overall system is properly designed and tuned, more stable and precise control performance is obtained.

With such a system, success relies on expensive, accurate modeling. In the past, the errors of a poor or inaccurate model could require the basic system to carry the additional load of undoing incorrect control actions that result from incorrect predictions of the model. Similarly, errors in process measurements to both the model and the basic system may significantly reduce the benefit that could be obtained from an accurate model.

The model should, therefore, be able to learn and constantly improve itself and improve the accuracy of imprecise measurements. Work is now proceeding in this direction: an incomplete model that can ''learn'' and improve itself as a model in order to achieve improved control results. This appears to be the most promising direction toward greater use of ''model-based control.'' A simplified diagram of such a model incorporation is shown in Figure 26-1.

Such utility boiler control system designs are being tried in an increasing but still relatively small number of cases, and encouraging results of improved precision and control performance are being obtained. The expense of the model development and the lack of system understanding by the typical control systems engineer, who must take care of the regular day-to-day

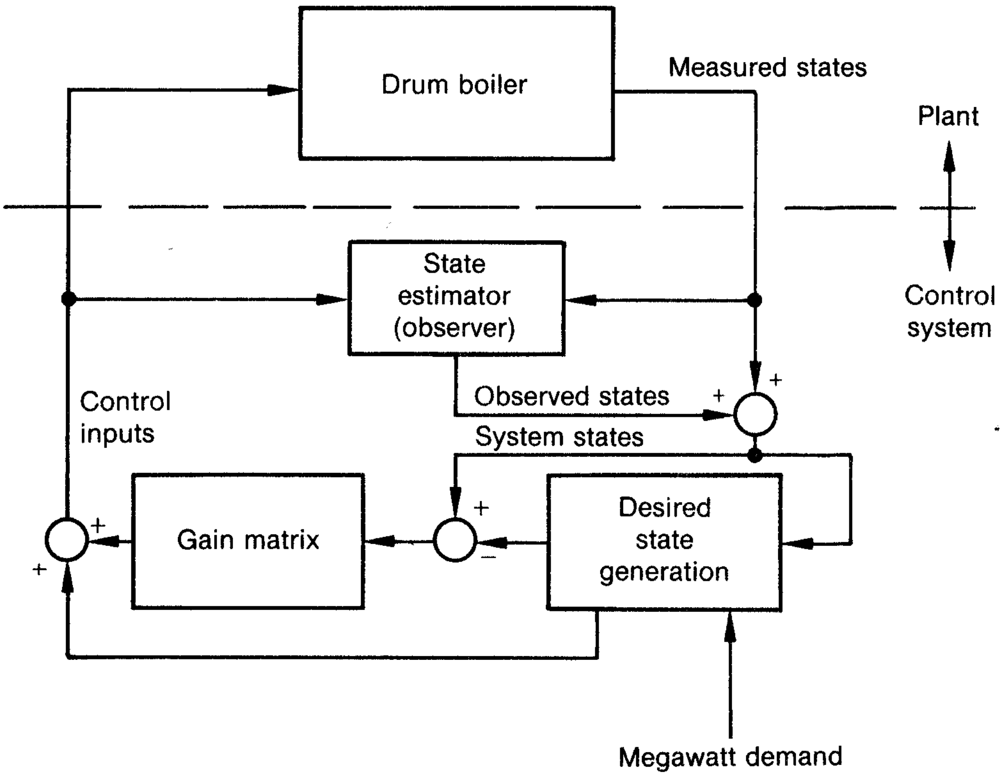


Figure 26-1 Functional Diagram of Fossil Plant Modern Control System Including an Observer Model

operation of the system, are two key drawbacks to greater use of this technique. The acceptance and general use of this technique is also slowed by the fact that few new power plants are being built today.

Such methods will also not be used unless it can be shown that present control performance standards result in losses that would pay for the cost of improved control performance. It is not enough that steam pressure, steam temperature, or fuel/air ratio be performed with greater precision. There must be an economic benefit to justify any increased cost. It is important to note that such systems would have been practically impossible without the use of digital control. More use of model-based systems can be expected as they become better understood and their implementation can be made at lower cost.

26-2 Improving Control Precision and Stability without Process Modeling

While the use of model-based control is a developing technique for more precise boiler control, other avenues are available for improving control performance. Some of these have received little or no attention by boiler control application engineers, while others have been used sparingly. These arrangements do not require costly boiler system modeling and are more easily understood by the typical user's control engineers. Digital control offers the opportunity to take full advantage of these techniques, though some of them can easily be implemented with analog control.

Self-Tuning or Adaptive Tuning

Self-tuning controller algorithms are now available for insertion into control systems. Such controllers automatically compensate the controller tuning as process or boiler conditions change. Adaptive tuning can also be implemented from load or some other variable of the process. The technique is shown in the diagram of Figure 26-2. Its use is demonstrated in a control system in Figure 13-19 of this text.

Another candidate for the use of this type of control improvement is in boiler steam temperature control. It is known that the optimum tuning of the gain and integral modes of the steam temperature controller changes as load on the boiler changes. Some software controllers in microprocessor-based systems allow the direct input of an external signal for adaptive controller tuning. If not, the objective can be accomplished by the combination of standard control algorithms shown in Figure 26-2.

The Calibrating Integral Circuit

An example of this is shown in Figure 8-10 of this text. This technique configures the proportional and integral functions in separate controllers. A long-term proportional error causes the integral controller to act in a manner that will correct misalignment in the gain of a feedforward or measurement signal. Another example in this text is shown in the fuel Btu compensation circuit shown in Figure 23-3.

Computation of Unmeasurable Variables

A typical example of the use of this technique is the computation of boiler heat release. Boiler heat release cannot be measured directly, but it can be computed by using the boiler as

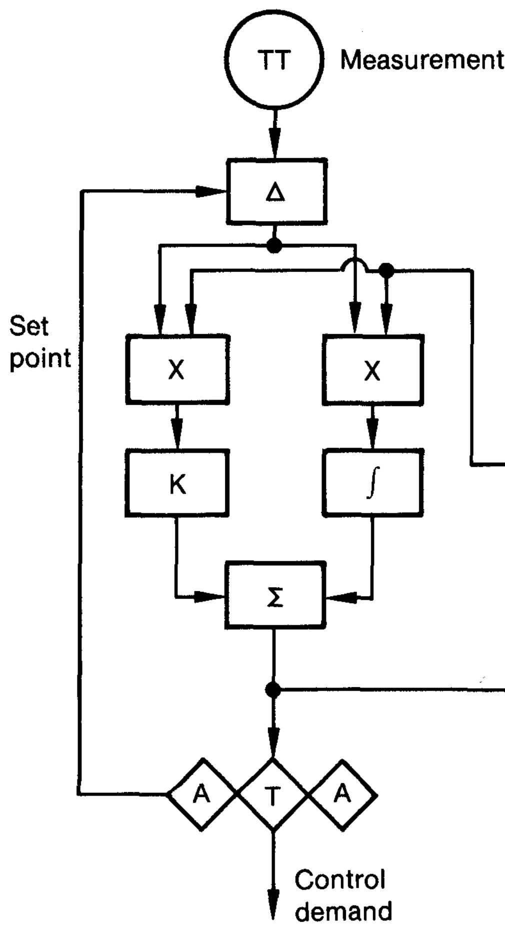


Figure 26-2 Adaptive Controller Tuning

a calorimeter. One form of such a computation is shown in Figures 23-1 and 23-2 and is discussed in Section 23. An addition to this computation, which would further enhance its precision in the control circuit, is another computation that represents variations in boiler efficiency.

The text also describes how the computation result can be combined with a calibrating integral circuit to improve a relatively poor measurement of boiler fuel input. This is shown in Figure 23-3 of this text. For control system performance improvement, the control engineer should investigate whether available measurements can be combined in a computing circuit and the calculated result used in place of an unavailable direct measurement.

Cross Coupling

The performance of many control systems is compromised by the interaction between the various sections of the overall system. In boiler control the interaction is typically among the feedwater control system, the combustion control system, and the steam temperature control system.

Assume a utility boiler operating at 2400 psig and 1000°F. Assume also that steam temperature is controlled with a water spray and firing rate is controlled from steam pressure. At a given point in time the steam pressure may be 2425 psig and the steam temperature 985°F. The conventional action would be to reduce firing rate in order to reduce steam pressure to the 2400 psig set point and to reduce spray water flow in order to increase the steam temperature to the set point of 1000°F.

Since a reduction in firing rate would further reduce steam temperature, the result would be a further decrease in spray water flow. Spray water flow is added to the boiler steam generation and the decrease in spray water flow would tend to further decrease steam pressure, thus causing an increased firing rate and subsequent pressure increase. The result is that the two control subsystems would interact and fight each other. Precision of control would be reduced.

An analysis of the thermodynamic steam and water properties shows that the heat content of the steam is due almost entirely to the steam temperature. Steam temperature is also the major factor in determining the specific volume of the steam. At 2400 psig the pressure values used in the example would have very little specific volume or heating value effect. For a given steam flow, the specific volume of the superheated steam, along with the volume flow, determines the steam pressure in the confined volume. In essence, pressure can change because of load changes, temperature changes, and spray water changes. Most systems consider only load changes.

Both pressure and temperature should, therefore, be used in such systems as inputs for controlling firing rate, and both pressure and temperature should be used for controlling the spray water flow. By cross coupling the steam temperature with appropriate gain into the firing rate control and steam pressure with appropriate gain into the spray water control, the overall system performance is improved.

Using the previous example, the 2425-psig pressure controller would reduce firing rate a smaller amount than before due to the increased firing rate request from steam temperature, with the remainder of the pressure reduction needed coming from reduced spray water flow. The steam temperature controller would call for a reduction in spray water flow. While in the right direction to also reduce the steam pressure, the amount of the change in spray water flow would be tempered by the steam pressure input.

Assume that the pressure is 2425 psig and the temperature is at the set point of 1000°F. Both the firing rate and spray water flow would be decreased by the steam pressure controller, with no action from the steam temperature controller. If the system is properly tuned, the firing rate decrease would have the effect of reducing steam temperature, and the spray water flow decrease would have the effect of increased temperature. The result is little, if any, effect on the steam temperature.

Nonlinear Control

There are several varieties of nonlinear control. One common method is the "error squared" algorithm in which the proportional error from set point is squared before the controller gain is applied. A nonstandard nonlinear gain can be applied to a measured linear deviation from set point and can be inserted into a control loop in the manner shown in Figure 26-3. In this case, the desired gain is computed as a function of controller error and can include a different error-gain function, depending on whether the set point error is positive or negative. The desired gain can be a function of other relationships as well.

A particular application of nonlinear controller gain is to make possible the use of the plant steam system as a steam accumulator. In some plant systems, sudden large but temporary energy demands may overtax the boiler system in meeting these large load swings. If a non-

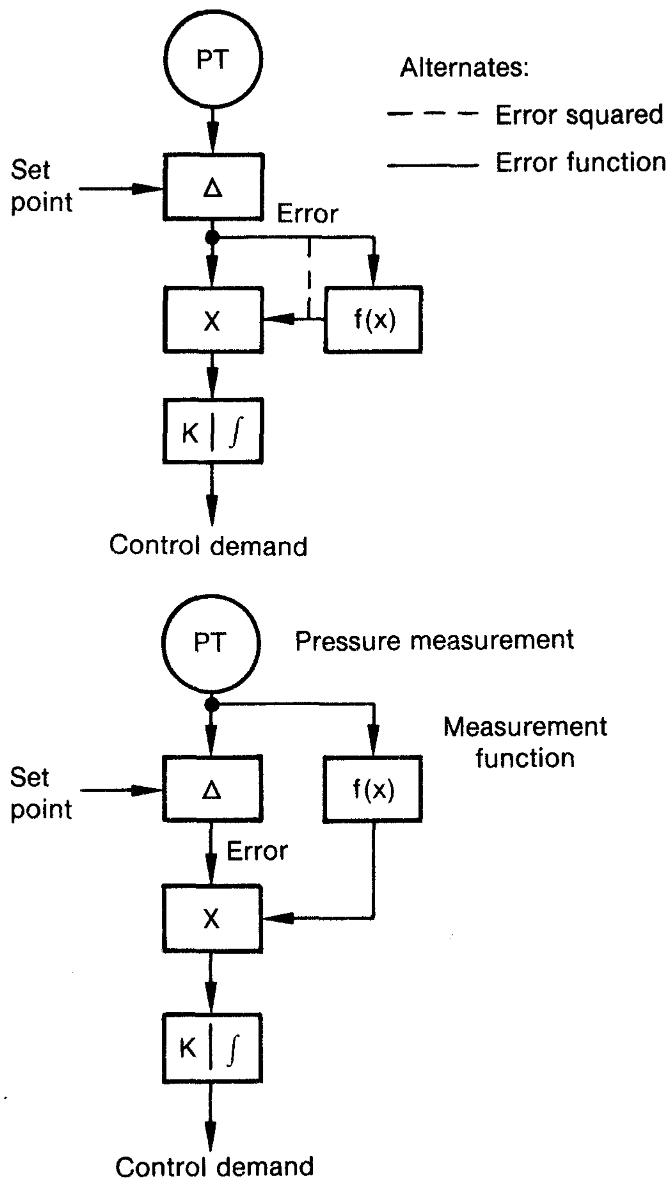


Figure 26-3 Nonlinear Control with Controller Gain a Function of a System Variable

linear steam pressure controller is used, as shown in Figure 26-3 with controller gain as a function of steam pressure, then the steam system energy storage can be used to assist the boiler in meeting these sudden large steam demands while changing the boiler firing rate characteristic to a more desirable pattern. Such a system would allow the boiler steam pressure to decay or build up over a wide pressure range at programmed rates. If this were done, it would be necessary to pressure compensate the measurements of boiler steam flow and boiler drum level.

26-3 Artificial Intelligence and Expert Systems

Self-tuning controllers are one aspect of this technique that inserts the methods of the expert individual into the control system. The expert control engineer observes the action and performance of the system before he makes tuning and alignment adjustments or reconfigures the system. In making his decisions he may use certain thought processes that he cannot fully identify, though many of these can be reduced to mathematical equations.

The expert takes control readings, draws curves, calculates calibration gains and biases and makes adjustments to realign or recalibrate the system. With the resources available in a digital system, there is no valid reason that the system itself could not perform this function. Self-tuning techniques can also be applied to the various controllers, but this can be a little more complex than simple adaptive tuning. There are occasions when it is desirable to loosen controller tuning to avoid oscillations in the combustion process. This information would also have to be imparted to the system. In some cases, such as fuel flow and air flow, the control loop timing must be tightly synchronized so that the precise amounts of fuel and air arrive at the burner precisely together. In this way, stable % oxygen in the flue gas can be obtained during quick load changes. This may require detuning the fuel or air so that the two match. This technique can be imparted to the system.

If these thought processes and techniques are properly inserted into the system as software, the system will be able to diagnose its ills and administer the proper remedies. All of the ideas discussed are available for use now with present-day digital systems.

Control engineers are working on more sophisticated techniques that use probability logic. If not already available, in the very near future these more sophisticated means will be available for solving very difficult control problems but would not be required for the great majority of boiler control problems.

26-4 A General Observation Relative to Boiler Modeling

Earlier in this section, the use of process models in advanced control application was discussed. The implication was that process models were expensive and time consuming to develop and that a model capable of learning to improve itself was desirable. This approach allows a relatively simple model as a starting point.

There are all kinds of models. A multivariable control system is a form of reverse model that closes the control loops. Loop-closing models of a boiler system can be developed that rely only on the intuitive skill and process knowledge of a boiler control application engineer. Such a model would probably be indicative of a wide range of boilers and rarely of a particular boiler. Such a model can be a starter model in a learning system.

Figure 26-4 is an example of the development of a simple model. The emphasis at this point is for a model that will act like the boiler whether or not it is based on any data of the particular boiler.

A SAMA diagram, the tool of the boiler control application engineer, can be used to diagram both the control system and the boiler model. If the SAMA block diagramming is used, the software system built around such control funtions can be used for the model as well as for the control system. This is an aid to the understanding of the control application and operation engineers.

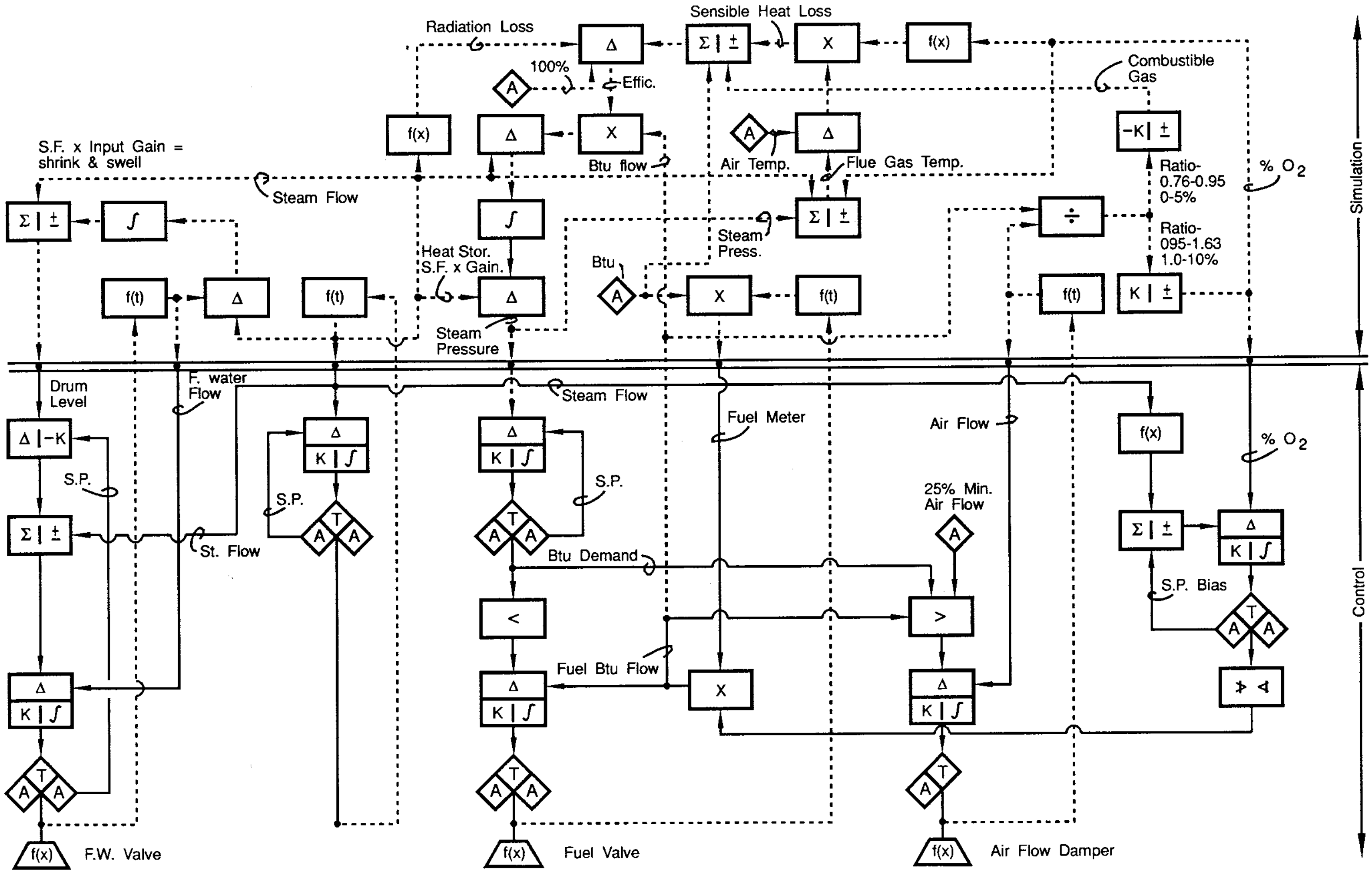


Figure 26-4 Simulation of Boiler Combustion, Steam Generation, Efficiency and Boiler Water System

Figure 26-4 is intended only to present the concepts involved and not to be used directly. These can be described as follows:

(1) The steam pressure is a result of the balance between heat input and output, heat storage, and the capacitance of the system. These can be calibrated and/or tuned to develop the steam flow and steam pressure.

(2) The boiler drum level is a result of the difference between the steam flow and the water flow, the capacity and size of the drum and the shrink and swell characteristics of the boiler. In the model, the functions can be tuned for any desired shrink and swell and capacity functions by changing control function gain and integral values. The feedwater flow is the result of the control output.

(3) Combustion produces flue gas. The analysis of the flue gas, % oxygen, % combustibles, and resulting excess air can be determined by dividing the air by the fuel. The burner develops combustible gases at some % oxygen, and this is tunable.

(4) The boiler efficiency is primarily a function of % oxygen and combustible gas, flue gas and air temperatures, fuel, and boiler load and capacity. Flue gas temperature is a function of steam pressure, boiler load, and boiler design. This can be calibrated into the model.

A boiler is a large integrator that takes all of various inputs and spews out values, many measurable and many that are not. Observation of these outputs are used to tune and calibrate a complex boiler control system. In like manner, observation of these outputs can used to tune and calibrate a model of a boiler.

In using such a model, all known values of the boiler design can be entered into the calibration of the various functions. The boiler can then be operated between steady-state load points and the model tuned and calibrated so that the model outputs seem to track the outputs of the actual boiler. This can be a starting point in the development of an improved model.

For an electric utility boiler, the model would be much more complex than that of Figure 26-4. The model simulation would include furnace draft control, reheat and superheat temperatures, economizer inlet temperature and other factors. Each of these can be developed using techniques similar to the above.

Another modeling approach uses test data of a boiler or a similar boiler. The results are individually curve fitted relative to load. Changing the load input to the model produces measured variable outputs. The curve-fitted data can also be individually curve fitted to each other.

26-5 General Observations Relative to Boiler Control Application

Boiler control application should be undertaken only by those individuals who have a thorough knowledge of the combustion and steam generation processes and a thorough understanding of process control. The process of solving a boiler control application problem is similar to that of solving any other problem. The following are three steps that some authorities have given weights of 50 percent, 40 percent, and 10 percent, respectively:

(1) Fully understanding and stating the problem—50 percent. To properly state the control problem, all of the objectives of the control system, including the necessary control performance, must be clear. An objective of $\pm$ 5°F for steam temperature control should take into account the economic benefit of $\pm$ 5°F as compared to $\pm$ 10°F or $\pm$ 15°F. If control systems and how different control elements interact (or the manner in which the different elements of the process act and interact) is not fully understood, the control problem cannot be stated correctly.

(2) Determining various alternative solutions to the problem—40 percent. After the problem has been properly stated, there may be several potential solutions with each having ad-

vantages and disadvantages. Proper potential solutions cannot be developed without the process and control system knowledge.

(3) Selecting the best alternative solution and implementing that solution—10 percent. This step also requires a thorough knowledge of both process and control systems.

The control solutions should be based on the functions needed rather than on the control means. The economic justification may require a simpler, lower cost solution, but this should not get in the way of the proper analysis of the problem. Though digital control is now the predominant control means, analog control could be the proper solution to a problem involving the matching of existing equipment, spare parts, and the retraining of operation or maintenance personnel.

With the advent of digital control, changes in other control application practices may also begin to emerge. In the past the boiler control functional separation has been based on a breakdown that was based on the modulating functions of combustion control, steam temperature control, feedwater control, the digital functions of burner management, the digital functions of burner light-off and flame safety control, and the digital interlock functions. The result has been that each installation is a custom application.

Since microprocessor-based systems can handle modulating and digital logic functions equally as well, it appears that simpler coordination and greater standardization of application can be made if we depart from the functional control system breakdown and orgainize the system around unit controllers for the type of equipment being controlled.

An example of such a controller would be a pulverizer controller. All of the modulating functions, the digital interlocking functions, the start-up and shutdown of the burners, and all of the safety function would be a part of that pulverizer controller. Other pulverizer controllers of that installation would be identical. A single demand for more or less pulverized coal would be sent from a master system, and the pulverizer controller would perform all necessary functions to supply the coal. This would include starting up or shutting down the pulverizer and all its auxiliary devices, if necessary. Other possible unit controllers would be the induced draft controller, the forced draft controller, feedwater pump controllers, and so on.

Boiler control has made tremendous strides in the past 50 years as we have progressed from direct-connected firing aisle control panels and direct-connected regulators to the distributed microprocessor-based systems of today. With all the power and flexibility of modern distributed digital control at our disposal, the boiler control application engineer now has all the necessary tools and the opportunity to use them.

In the future, the many new and more sophisticated boilers, turbines, and energy systems will require the design and implementation of control systems. Add to that the job of replacing 80 to 90 percent of all the boiler control systems that have been installed in the last 30 to 40 years.

What a wonderful time to be a boiler control application engineer and have a part in the next genererations of progress.

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