FACTORS LIMITING OPERATING CAPACITIES OF UNIVERSITY OF TENNESSEE CENTRAL POWER PLANT BOILERS

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INTRODUCTION AND PURPOSE OF INVESTIGATION

This work is intended to be an impersonal and unbiased study of the central power plant of the University of Tennessee with the purpose of ultimately being able to offer suggestions as to the operating difficulties encountered in the operation of the boiler and stoker units. The writer is attempting to present this material in such a way that it will be of interest and benefit to power plant operators in this plant and other plants which are probably operating under the same difficulties. Obviously, this work will differ very materially from the mass of writing on the subject due to the fact that each power plant has its own problems which differ from those of any other plant. As far as is known a similar study has never been made in connection with this plant.

The Walsh-Weidner Boiler in unit number one has a heating area of 3500 square feet and a combustion chamber of 597 cubic feet volume. The Taylor Multiple Retort Stoker which fires this unit was bought under a guarantee of being capable of efficiently burning, under a boiler of this size, sufficient coal, with a heat value of 12,500 B.T.U. per pound, to maintain a boiler horsepower rating of 250%. In actual practice it has been found that at around 100% rating the steam-coal ratio of this unit in pounds is between 7.50 and 8.50. Obviously, this could not be classed as efficient combustion. It has also been found that as higher ratings are obtained it becomes more difficult to get efficient combustion. Upon analysis, the flue gases show a low percentage by volume of carbon dioxide, a high percentage of free oxygen, with a negligable amount of carbon monoxide and a relatively low temperature. This proves conclusively that there is an abnormally high percentage of excess air figuring in the

combustion process. The boiler setting was thoroughly examined for leaks through which air might filter into the furnace. It was found that air was entering through three small leaks in the setting and around the fire door, but the amount of air entering here was small. The question to be answered now is why is all this excess air required and from where does it come? This condition will be discussed later under "Conclusions".

CHAPTER I

GENERAL SURVEY OF THE FIELD

The Stoker

For efficient and smokeless combustion of fuel in furnaces continuous feeding of the fuel and uniform distillation of volatile matter in the presence of oxygen are essential requisites. For this reason the mechanical stoker is more effective in producing high combustion efficiency and in preventing smoke than any apparatus accompanied by intermittent firing.

With a hand-fired furnace there is a rush of air into the furnace each time the fire door is opened. The stoker has overcome this condition.

The mechanical stoker is capable of responding promptly to high and sudden overloads and of being brought to full steaming capacity from a banked fire or cold grate in a very short time. It permits the use of cheap fuel with good economy.

In the year of 1785 probably the first mechanical stoker for steam boilers was designed by James Watt. This first stoker was a simple device to push the coked coal from the front of the grate back toward the bridge wall. European engineers were quick to take up Watt's idea and numerous stokers were patented. These stokers were not adopted at once in this country, partly due to their supposed lack of adaptability to American boiler practice and partly because of inadequate knowledge of the art of stoker operation. These early stokers were equipped with very poor driving mechanism and were very unmechanical in having parts that were not readily renewable and exposed to very high temperatures.

At the beginning of the twentieth century there were four types of stokers used in this country, representing the front inclined, side inclined,

chain grate and underfeed makes. At this time very few plants operated at above 125% rating. By 1910 this figure had increased to 150%, the increasing demand for steam always calling for higher boiler ratings until by 1915 performances of 200% were reached in some plants. Until about 1910 the majority of plants in America operated on natural draft.

At about this time the multiple retort underfeed stoker entered the field and completely revolutionized power plant practice. By means of this type of stoker ratings of 250% and 300% were soon reached. This was followed by larger boilers for which the underfeed type of stoker is admirably adapted. This stoker is well adapted to large boilers, there being no physical limit to the number of retorts that may be placed side by side to serve wide units. In the underfeed stoker the green coal is fed into the lower layer of the fuel bed and is gradually pushed upward, cokeing as it moves up. By the time it reaches the top layer it has given up its volatile constituents and become incandescent. The fuel bed is carried on an inclined structure made up of a series of narrow passages at right angles to the front of the furnace. These narrow passages which slope downward toward the back of the furnace are known as retorts alternating with tuyeres or stacks of air supply blocks. Coal from the hopper falls into the ram boxes from which it is pushed into the retort by the main feeding rams. After this ram has covered about two thirds of its stroke, it moves in unison with the distributing rams under the retort. The sloping faces of the retort raise and push the fuel forward so that when they are withdrawm there is room for the next charge of green fuel. The tendency of caking coals, as is used in most of the large eastern central stations, augments the action of stoker in producing a fuel bed of unusual thickness, and the pushing action of the feeding mechanism keeps the bed

broken up and porous.

Therefore, under the entire fuel bed there is a layer of green coal which is pushed up, step by step, into the coking zone, where it is gradually coked by the heat from the fire above, releasing its volatile gases, then into the combustion zone where it forms the bed of incandescent coke and is consumed to ashes, being carried to the back of the furnace where it is removed by the hinged ash dumper.

Low pressure air is admitted to the chamber under the retorts, entering the furnace through the tuyere openings which direct high-velocity streams into every part of the incandescent coke. Combustion is regulated by adjusting permanently the speed of each ram to the character of the fuel, keeping the air pressure as nearly constant in all parts of the retort as possible.

Furnace

The design, proportions, and details of the furnace or combustion chamber is of vital importance to the success of the entire steam generating unit. There are perhaps no two plants having exactly the same furnace design, for each fuel and set of operating conditions involve such a number of variables that each installation must be considered by itself. "Whatever may be the nature of the fuel or the conditions of operation, for complete and efficient combustion, the furnace must be constructed and operated in such a manner that the combustible gases will be brought into intimate contact with the proper amount of air, and maintained at a temperature above the ignition point until oxidation within the combustion zone is complete".[1]

In practice there is often a serious deficiency in furnace volumes. In the past, boilers have been set within two or three feet of the grates,

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1. Gebhardt, Steam Power Plant Engineering, page 209.

allowing inadequate space for combustion with the result that combustion was prolonged for a distance through the boiler passes, resulting in incompleteness of chemical union, hydrocarbon and CO losses, and low efficiencies. Increasing fuel prices have brought about a demand for increased efficiency. Low efficiencies may also accrue in a combustion space which has the proper theoretical volume, but which has a very small grate area. In a case of this kind an unusually high percentage of excess air is required for the production of the required amount of heat.

It has been found that if only the theoretical amount of air required for combustion is supplied, the products of combustion will consist chiefly of N_2 and CO_2, with small amounts of water vapor and SO_2. In case excess air is supplied or leaks into the setting, free oxygen will be present in the flue gases. Therefore if, upon analysis, the flue gas is found to consist chiefly of free oxygen, CO_2, N_2 and SO_2, it is an indication that the combustion has been complete but with an excess of air.

In practice, a certain amount of excess air is required, depending upon the nature of the fuel, design of furnace equipment, and the rate of combustion. If CO and H_2 or other combustibles are present in the flue gases, it is evident that combustion is incomplete, and if there is no free O_2, the air supply is deficient. "However the flue gases may contain CO, H_2 and other combustible gases, and considerable free oxygen. This is due to the fact that: (1) the combustible mixture of air and combustible gases in the zone of combustion was incomplete; (2) the furnace temperature has been too low for proper ignition; or (3) there has not been time for combustion to be complete. The actual volume of free O_2 and the percentage by volume increases with the amount of excess air. Therefore, either the CO_2 or the free O_2 is a true index to the air excess for complete combustion

of a given fuel." [2]

The proper basis of furnace design is B.T.U. liberated per cubic foot of effective furnace space per hour. In 1924, the efficiencies, capacities, and maintenance obtainable from such furnaces justified the designs on the basis of modern costs and fuel prices of furnaces with effective volumes of 1 cubic foot for each 18,000 to 24,000 B.T.U. liberated per hour. In actual practice many furnaces are found with low efficiencies, limited capacities, and smoke, due to the fact that the furnaces are much smaller than the size required. General results may usually be improved by enlargement of the furnace.

Draft

The efficiency of the whole combustion system is dependent on a flow of the gases from the region of high density to the region of low density. This difference in pressure, when spoken of with reference to combustion, is known as draft. All underfeed stokers use forced draft. "For the average fuel, including crude oil, lignite, hard and soft coal, it was shown that 7.65 lb. air per 10,000 B.T.U. per pound of fuel is necessary. The amount of excess air required with coal is approximately 80% to 100% for natural draft and 40% to 60% with stokers and mechanical draft. The corresponding weights of air required per pound of fuel consumed are, therefore, 13.8 to 15.3 lb. for natural draft and 10.7 to 12.2 lb. for mechanical draft per 10,000 B.T.U. calorific value of the fuel."[3]

Problems involving the flow of air through the combustion chamber arise frequently, both in design and operation. Draft of itself does not burn the coal. Air is required for the combustion and this air is supplied

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2. Gebhardt, Steam Power Plant Engineering, page 85.
3. Harding, Steam Power Plant Engineering, page 109.

by the draft. Therefore, the rates of combustion with various coals are related to the amounts of furnace draft. This draft, in the case of an underfeed stoker installation, is derived from two sources. The first is called natural draft and is created by use of a chimney or stack. In the chimney the hot gases leaving the furnace tend to rise by virture of their lighter weight due to increased temperature. The other or forced type of draft is usually by the direct method in which the air is supplied directly to the ash pit or through an intervening duct system and only that portion under the fuel bed is under pressure. With this system a pressure is maintained sufficient only to force the air through the necessary duct system, stoker setting and fuel bed. Pressures greater than this will not only create an outrush of smoke and flame into the boiler room when the fire door is opened, but will also cause an objectionable leakage of gases into the boiler room. It may be seem that the use of forced draft will require a higher chimney than an induced draft system, beacuse its effect practically ceases at the surface of the fire.

With the induced draft system the fan or blower is situated on the chimney side of the boiler and handles the smoke and hot gases leaving the furnace. In order that there will be no outward leakage of gases through the boiler walls and at the same time a minimum infiltration of air into the furnace, it is desirable to so adjust or balance the drafts on either side of the fuel bed so that the pressure just above the fire is slightly below that of the atmosphere or about 0.10 inch of water. Modern power stations have found it almost necessary to install both induced and forced draft fans because of the necessity of maintaining a well balanced draft under practically all conditions of firing. "Between the breeching or smoke flue and the furnace comes the heading surface of the boiler itself.

Draft losses occur here. They must occur, as there must be resistance to flow of the gases if proper heat absorption is to be obtained. The areas of the gas passages are fixed, but at different rates of combustion the volume of gases passing through these areas varies. Therefore, there is increased velocity of gases at the higher ratings, accompanied by increased resistance, that is, increased draft loss." [4]

Boiler

The boiler of today is, with the exception of a few special designs in recent years, practically the same as those of ten years ago. The boiler is the heat absorber of the unit and has nothing to do with the coal-burning process.

Each type of boiler has its particular advantages and disadvantages. All combustion should be completed in the furnace before the products of combustion reach the boiler. Heat is transferred from the incandescent fuel-bed and flame to the heating surface which is exposed to the fire almost entirely by direct radiation. The heating surface screened from the fire is heated by convection from the hot gases sweeping over the furnace.

A general classification of steam boilers is unsatisfactory because of the overlapping of the various groups. Probably the most common item made use of in classifying boilers is the relative positions of the heated gases and water. Upon this basis boilers are designated as water tube or fire tube, depending upon whether the water or products of combustion pass through the boiler tubes.

In so far as efficiency is concerned, exhaustive tests have proven

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4. Marsh, Combustion in the Power Plant, Page 170.

that there is no choice between the two types. The fire tube boiler is most commonly used in small or medium sized installations. This boiler consists of a steel plate shell, the heads of which form the tube sheets and into which the tubes are expanded. The shell is supported either by lugs, riveted to the shell, resting on wall plates or suspended by hangers from double channels supported by I-beam columns.

The gases pass under the shell over the bridge wall to the rear of the combustion chamber, then through the tubes to the front, out the smoke connection to the breeching, and thence, to the chimney.

The demand for boilers of larger size than is possible with the return tubular type, greater overload capacity, and ability to respond quickly to sudden demands has led to the selection of the water-tube type of boiler for practically all modern large steam installations. Among the advantages of the water-tube boiler possibly the predominating one is safety. When the shell of the fire-tube boiler starts to "bag" due to excessive strain together with high temperature or when a seam begins to weaken the chances for a terrific explosion are very great. This is due to the fact that when the opening does occur the entire contents are free to escape with very little obstruction and at a high velocity.

In a water-tube boiler a leaking tube does not cause an explosion, because, due to the obstructions encountered by the water and steam in reaching the opening, the rate of escape or velocity of discharge is too low to do serious damage.

Due to more efficient circulation, the water-tube boilers of modern design also offer advantages in conservation of boiler room floor area, accessibility for quick steaming, and capacity. Experience has proven that the rate of heat transfer through a metal surface from hot gases to water

is dependent upon the velocity of the water over the surface, and when rapid circulation is secured an increase in capacity is the natural result. It is not uncommon to operate water-tube boilers continuously at 300% of their rated capacity with 500% peaks.

CHAPTER II

GENERAL THEORY

Because of the fact that air and the gases distilled from the coal in a furnace do not mix perfectly, it is necessary to supply a certain amount of excess air to any combustion process. After this amount has been supplied, any further excess causes a loss in heat up the stack due to the heat required to raise the temperature of the unused air. This unused air only causes the combustion chamber to be cooler than it should be.

In using a furnace which is too small in grate area for the work required of it, an abnormally large amount of excess air is reqwired to burn the coal at the required rate. With this condition reducing the temperature of the furnace chamber, the result is that the evaporation of steam per pound of coal is decreased. As a result, when high ratings are to be obtained the combustion of the required amount of fuel will release enough heat to damage the furnace refractories. Also, under similar conditions, it might be found that the temperature of the fuel bed is in excess of the temperature of the ash fusion point. When this is true the ash will melt and tend to seal off the openings in and between the grate bars. With slag forming in the grate area, the air from the blower will naturally pass through that part of the fuel bed where the resistance is the least. As the combustion is practically complete by the time the fuel reaches the back side of the grate, there will be very little slagging in this area, and the uprushing air will cause blowholes to form in the fire bed. The air which passes up through these holes escapes up the stack before it has had an opportunity to thoroughly mix with the gases distilled from the coal.

CHAPTER III

OBSERVATIONS

The steam generating units in the boiler room of the University of Tennessee central power plant consist of a battery of four boilers equipped with underfeed stokers. Boilers number one and two are of the cross-drum, water-tube type. These boilers were manufactured by the Walsh-Weidner Boiler Company of Chattanooga, Tennessee. Each boiler has a heating surface of 3370 square feet, a combustion space of 597 cubic feet, and is built to operate at a pressure not to exceed 200 pounds per square inch.

Each of these boilers is fired by a Taylor underfeed multiple retort stoker, manufactured by the American Engineering Corporation of Philadelphia, Pennsylvania. These stokers, having four retorts each, were designed to operate with a furnace having an active grate area of 64.7 square feet. They are guaranteed, by the manufacturers, to burn sufficient coal of specified analysis, to develop 275% normal boiler rating. These stokers are further guaranteed to so operate that when burning coal of not less than 12,500 B.T.U. per pound heat value and operating at 200% of rated capacity or lower, they will be able to maintain a steam pressure of within five pounds of the desired working pressure.

Boilers number three and four are of the multi-pass, water-tube type. These vessels are the product of Casey-Hedges Boiler Company of Chattanooga, Tennessee. Each boiler is limited by the Traveler's Indemnity Company to an operating pressure not to exceed 200 pounds per square inch. These boilers have a capacity of 350 horsepower based upon a rating of one horsepower per ten square feet of heating surface.

The tubes of this boiler are made of #10 hot drawn seamless steel and are 266 in number with an outside diameter of 3¼ inches.

The Westinghouse multiple-retort, underfeed stokers which fire the Casey-Hedges boilers are designed to fire an actual area of fuel supporting surface of 72.82 square feet. The auxiliary equipment used with these stokers consists of a steam operated agitator and dump grates, Stoker Turbo-Gearbox to operate at 175#, gage steam pressure, and an American blower designed to supply 20,000 cubic feet of air per minute at 70°F.

The specifications on this stoker call for the capability of efficiently burning sufficient coal to produce, over a period of 24 hours, an overload capacity of 250% on a water-tube boiler having a heating surface of 3500 square feet.

In this work the number "one" Walsh-Weidner boiler with its Taylor stoker was the only unit tested. There was no day during the entire duration of this test during which the demands for steam were high enough to even approach the capacity of this one unit. During the winter months all of the four units are sometimes taxed to supply heat and electricity to the several buildings belonging to the University, but as practically no heat and very little electricity is required during the summer months, there was no occasion for bringing the other steam generating units into use.

ANALYSIS OF FLUE GASES AND CONDITIONS

Flue Damper Half Closed

Time	$\%CO_2$	%O	%CO	Furnace Temp. °F	% Rating
10:00 A.M.	7.2	12.2	Negligible	2300	100
10:25 A.M.	7.0	11.2	"	2315	103
10:50 A.M.	7.3	12.0	"	2340	120
11:15 A.M.	6.8	11.0	"	2310	95
11:40 A.M.	7.0	11.3	"	2375	113

Temperature of Flue Gases in Last Pass 485-495°F

There was a considerable amount of clinker on the rear edge of the grate area. Many small blow-holes in the rear side of the fire bed. Furnace walls at a dull, white heat.

Flue Damper Open

Time	$\%CO_2$	%O	%CO	Furnace Temp. °F	% Rating
1:00 P.M.	6.4	13.2	0.3	2375	97
1:25 P.M.	5.5	14.1	0.5	2410	116
1:50 P.M.	6.2	13.3	0.3	2396	110
2:15 P.M.	6.4	12.9	0.2	2380	113
2:40 P.M.	6.2	13.2	0.4	2400	106

Temperature of flue gases in same range as above.

There were large holes approximately 4" to 12" in diameter in the rear side of the fuel bed. Flame about 30" long appearing above fire bed midway its length. Walls of furnace dull white hot.

Average temperature of feed-water - 210°F.

Average temperature of air entering furnace - 85°F.

CHAPTER IV

CALCULATIONS

Actual evaporation in steam per pound coal = 8.15

Steam is saturated at 175# gage = 190# absolute pressure.

Latent heat of steam at 190# absolute pressure = 847,9 B.T.U.

Heat of the liquid at 190# absolute pressure = 350.0 B.T.U.

For feed water at 210°F it will require 210-32 = 178 B.T.U. less heat than water raised from 32°F.

Then 350-178 = 172 B.T.U. required to raise water from 210°F to boiling temperature at 190# absolute pressure.

Each pound of water delivered to the boiler and made into steam will require 172 - 847.9 = 1019.9 B.T.U.

Therefore the factor of evaporation = $\frac{1019.9}{971.7}$ = 1.049

At 100% rating the equivalent evaporation for 350 boiler horsepower 350 X 34.5 = 11,975 pounds.

Equivalent evaporation in pounds of steam per pound of coal = 8.15 X 1.049 = 8.55

Coal burned per hour on the grates = 11,975/8.55 = 1400 pounds.

To find the air required for the combustion of 1400 lb. coal per hour.

Coal burned per hour per square foot grate area = 1400/66 = 21.2 lb.

By tables, page 312, "Fan Engineering", air required for combusion of 13,500 B.T.U. coal at rate of 21.2 lb./sq. ft./ hr. = 248 cu. ft. = 15.1 lb.

Total air required + 100% excess = 15.1 X 1400 X 2 = 42,280 lb.

Boiler room temperature = 85oF

Heat entering furnace with air per hour = 42,280 X 85 X 0.2374(S.H.) = 853,528 B.T.U.

Heat liberated by combustion of coal per hour = 13,500 X 1400 = 18,900,000 B.T.U.

Total heat liberated = 18,900,000 + 853,528 = 19,753,528 B.T.U.

Heat liberated per cubic foot furnace volume = 19,753,528/597 = 33,087 B.T.U.

To operate this boiler at 200% rating with steam at 190# absolute pressure and an equivalent evaporation of 8.55 pounds steam per pound of coal the calculations for heat liberated per cubic foot of furnace volume are as follows:

Equivalent evaporation per hour = 350 X 34.5 X 2.00 = 23,950 lb. steam.

Coal burned per hour on the grates = 23,950/8.55 = 2800 lb.

Coal burned per hour per square foot grate area = 2800/66 = 42.4 lb.

By tables, page 312, "Fan Engineering", air required for combustion of 13,500 B.T.U. coal at rate of 42.4 lb./sq.ft./hr. = 224 cubic foot = 13.6 lb.

Total air required plus 100% excess = 13.6 X 2800 X 2 = 76,160 lb.

Heat entering furnace with air per hour = 76,160 X 85 X 0.2374 = 1,536,833 B.T.U.

Heat liberated by combustion of coal per hour = 13,500 X 2800 = 37,800,000 B.T.U.

Total heat liberated per hour = 1,536,833 + 37,800,000 = 39,336,833 B.T.U.

Heat liberated per cubic foot of furnace volume per hour = 39,336,833/597 = 69,890 B.T.U.

NOTE: As the furnace walls were at a dull white heat while operation was at 100% rating, it is obvious that in operating at 200% rating the furnace refractories would be unable to withstand the excessive heat. (In common practice a heat release of 20,000 to 25,000 B.T.U. per cubic foot of furnace volume is considered good practice while a release of 50,000 heat units per cubic foot of combustion chamber is encountered in the larger furnaces having water cooled walls in rare instances).*

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*Indirectly quoted from "Fan Engineering", page 316.

PROXIMATE ANALYSIS OF COAL

	As fired	Moisture and ash free
Fired carbon	- 54.76%	64.88%
Volatile matter	- 29.64%	35.12%
Ash Content	- 8.29%	-----
Moisture Content	- 7.31%	-----

Coal, moisture and ash free weight of moisture — Coal as fired — 1 minus proportioned ash.

Fixed carbon $54.76/1-(0.829+0.731) = 64.88\%$

Volatile matter $29.64/1-(0.829+0.731) = 35.12\%$

Calculation of Ultimate Analysis From Proximate Analysis.

$$H = V\left(\frac{7.35}{v+10} - 0.013\right) = 35.12\left(\frac{7.35}{35.12+10} - 0.013\right) = 5.11\%$$

$$N = 2.10 - 0.012\,V = 2.10 - 0.012 \times 35.12 = 1.68\%$$

$$C = F + 0.9\,(V-14) = 64.88 + 0.9(35.12-14) = 83.89\%$$

In these equations H = percentage of hydrogen in the combustible.
V = " " volatile matter in the combustible.
N = " " nitrogen in the combustible.
C = " " fixed volatile carbon in the " .

H =	5.11 X 0.844	= 4.31%
N =	1.68 X 0.844	= 1.42%
C =	83.89 X 0.844-1.97	= 68.83%
S =		= 1.97%
Ash =		8.29%
Moisture =		7.31%
O =	by difference	7.87%
		100.00%

NOTE: Formulae from "Steam Power Plant Engineering" by Gebhardt.

Calculation of Excess Air in Combustion.

To find the minimum weight of dry air necessary to completely burn the specified coal.

$$A = 11.58\,C + 34.8(H-O/8) = 4.355$$

$11.58 \times 0.6883 + 34.8(0.043 - 0.079/8) + 4.35 \times 01.97 =$

9.22 lb. air per pound coal.

To determine weight of dry air supplied per pound of coal as fired if flue gas resulting from combustion is composed of

CO_2	6.4	O_2	13.2
CO	0.3	N_2	80.1 percent by volume.

$$G = \frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)}$$

$$= \frac{4(6.4) + 13.2 + 700}{3(6.4 + 0.3)} = 36.75 \text{ lb. dry gas per pound of carbon actually burned.}$$

Dry gas per pound of coal = 36.75 X 0.6883 = 25.29

Assuming 5% of coal passes through furnace unburned.

Total weight dry air actually supplied per pound of coal is

25.29 - 0.6883 5(0.043 - 0.079/8) = 24.44

Percent of excess air = $100\left(\frac{24.44 - 9.22}{9.22}\right) = \underline{165}$

If it is assumed that 10% of coal passes through furnace u

Total weight dry air actually supplied per pound of coal is

25.29 - 0.6883 + 10(0.043 - 0.079/8) = 24.27

Percent excess air = $100\left(\frac{24.27 - 9.22}{9.22}\right) = \underline{163}$

CHAPTER V

CONCLUSIONS

As a result of these observations and calculations it is to be seen that the essential cause for the limited boiler horsepower rating obtainable with this unit may be traced directly to the size of the grate area. When operating at 100% boiler hoursepower rating and burning 21.2 pounds of coal per square foot of heating surface per hour, the heat released amounts to over 33,000 heat units per cubic foot of furnace volume. This amount of heat is rather high for the ordinary type of refractory which is designed to withstand a heat release of from 18,000 to 25,000 B.T.U. per cubic foot without water-cooled walls. The obtaining of 200% of boiler horsepower rating would necessitate the release of slightly less than 70,000 B.T.U. per cubic foot of furnace volume with a coal rate of 42.4 lb. per square foot of grate area. Obviously, the refractory in this furnace would burn out long before this figure could be reached. It is probable that in firing this furnace at a rate consistent with the obtaining of a boiler horsepower rating of 200% a large quantity of coal would be pushed across the grate and into the ash dump without having time for complete combustion.

This unit was taken out of operation on Sunday, August 12, 1934 to be repaired. It was found that there was a large quantity of fused ash and clinker in the grates. This condition was brought about by the burning of coal, the ash of which fused at a lower temperature than that usually maintained in the furnace. This ash has a fusing temperature of 2240°F as compared with a fuel bed temperature which seldom falls below 2350°F. This formation of ash on the grates will tend to give the hot gases rising

from the fuel bed directly above an eddying or swirling motion instead of allowing them to rise directly to the surface to be heated. This will cause a dissipation of some of the heat carried by the gases.

The fuel bed on the grates was seen to be excessively thick. This thick fuel bed required the incoming air from the blower to have a high velocity in order to penetrate it. The result of this condition is that near the back edge of the fuel bed, where the fuel is practically burned out, it is impossible to prevent the formation of blowholes. The presence of these blowholes allows a stream of air to pass through the fuel bed and on up the stack without having taken part in the combustion process. This air has a cooling effect upon the furnace.

The conclusion I have drawn from this investigation is that with a larger grate area it would be possible to obtain more efficient operation with this unit; assuming that Brushy Mountain coal with high heat value and low point of ash fusibility is used. A larger grate area would permit the combustion of the same amount of fuel in a thinner bed. This would eliminate the requirement of an excessively high velocity of air from the blower with the result that no blowholes would be formed in the fuel bed and no cool air would pass over the surface to be heated.

A larger grate area with the same depth of furnace would increase the volume of the combustion chamber thereby decreasing the amount of heat released per cubic foot of furnace volume. In this way the danger of injuring the furnace wall refractory by excessive heat would be lessened.

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APPROVAL

The foregoing thesis is hereby approved as a creditable study of an engineering subject, carried out and presented in a manner sufficiently satisfactory to warrant its acceptance as a prerequisite to the degree for which it has been submitted. It is to be understood that by this approval the undersigned does not necessarily endorse or approve any statement made, opinions expressed, or conclusions drawn therein, but approves the thesis only for the purpose for which it is submitted.

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