DESIGN OF FACTORY POWER PLANT

by

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Signature of Author ______________________

Signature of Professor
in Charge of Thesis ______________________

Signature of Chairman
of Department Committee
on Graduate Students ______________________

Department of Mechanical Engineering
February 11, 1946

M.I.T. Dormitories
Cambridge 39, Mass.
February 11, 1946

Professor George W. Swett
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge, Massachusetts

Dear Sir:

In partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering, I respectfully submit my thesis entitled, "Design of Factory Power Plant".

Respectfully submitted,

Aboudi Mowlem

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CONTENTS

I. INTRODUCTION

An industrial factory requires power and heating and manufacturing steam. The question arises before the designer of the choice between several plans of supplying the load. Should he install a plant to supply both the power and process steam, or should he purchase the power and install a low pressure boiler plant? Should he purchase both power and process steam, interchange with a utility company or purchase part of the power and generate the remainder and the process steam? His choice must be based on the most economical scheme, which must also be flexible and reliable as well as adaptable to the particular case under consideration, for no one scheme has all the advantages and no disadvantages.

Two methods of supplying the load are selected in this work for detailed investigation. One is the design of a steam power plant supplying both the power and the process steam. High plant thermal efficiency is sought but not to such an extent as to make an investment larger than for a plant with lower efficiency. The design is discussed in detail in section VI, while the calculations are found in the appendix. The equipment, in many cases, is selected from a number of designs according to economical performance and suitability. The other method is to purchase power and install a low pressure boiler plant. The rate used is the Block Hopkinson demand rate.

A cost comparison of the two methods is made and the annual saving realizable by the building of a steam power plant over the other method is found to be $33,310. which can pay for a new and complete steam plant in twenty-two years. It is, therefore, recommended that a new power plant be installed to handle the power and steam loads. The cost of the power is 1.935 cents per kw hr, a reasonable value for the size of the plant in question.

Drawings of overall plant layout have been made and can be found in section VIII.

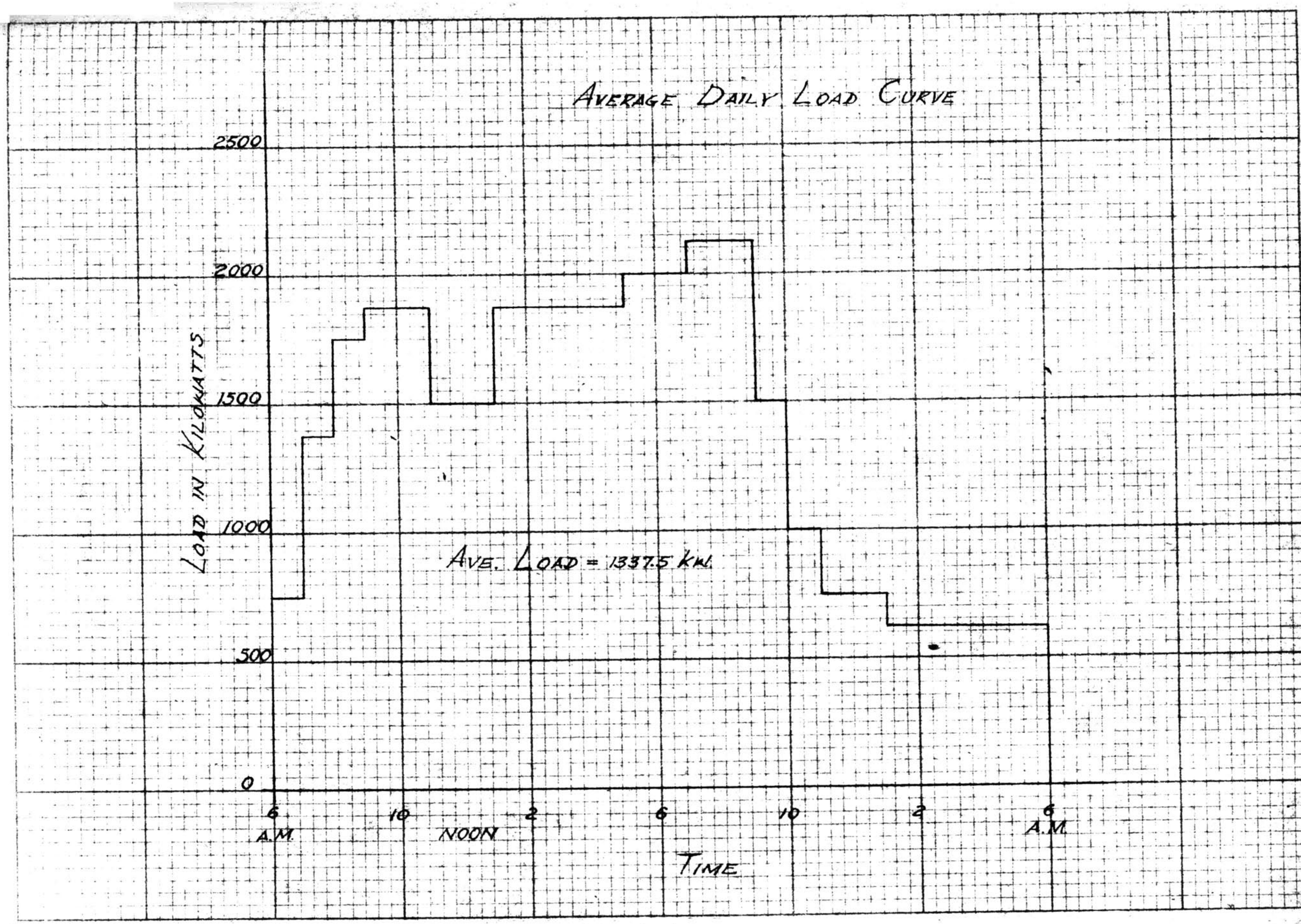
II. STATEMENT OF PROBLEM

A certain factory is to build a new plant and must decide whether or not they should generate their own power or purchase power from a neighboring central station and install low pressure boilers to supply steam for heating and manufacturing.

The plant engineer is requested to go over the plans and made a survey, design a power plant and decide whether or not to purcase power.

The plant survey was as follows:

1. The average daily load during the year would be as shown below. No load Sunday except steam for heating. Maximum estimated power load = 2250 kw.
2. Average yearly steam load curve for heating and manufacturing steam at 25 psi gage is also shown below. Condensate can be returned at a probable temperature of 140° F.
3. The plant is located on a river of fresh water which can be used for condensing purposes but must be filtered before used in boilers. Summer water temperature is 70° F.
4. There is plenty of space available near river and plant. Ground level about 6 feet above river. Tests show good gravel bottom.
5. Coal can be obtained at $7.00 per ton over crusher delivered by rail. Use New River coal.
6. Determine initial steam temperature to produce dry saturated steam at the bleeder point. Use 450 psia steam pressure.


AVERAGE DAILY LOAD CURVE
2500
2000
1500
1000
500
0
LOAD IN KILOWATTS
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6 A.M.
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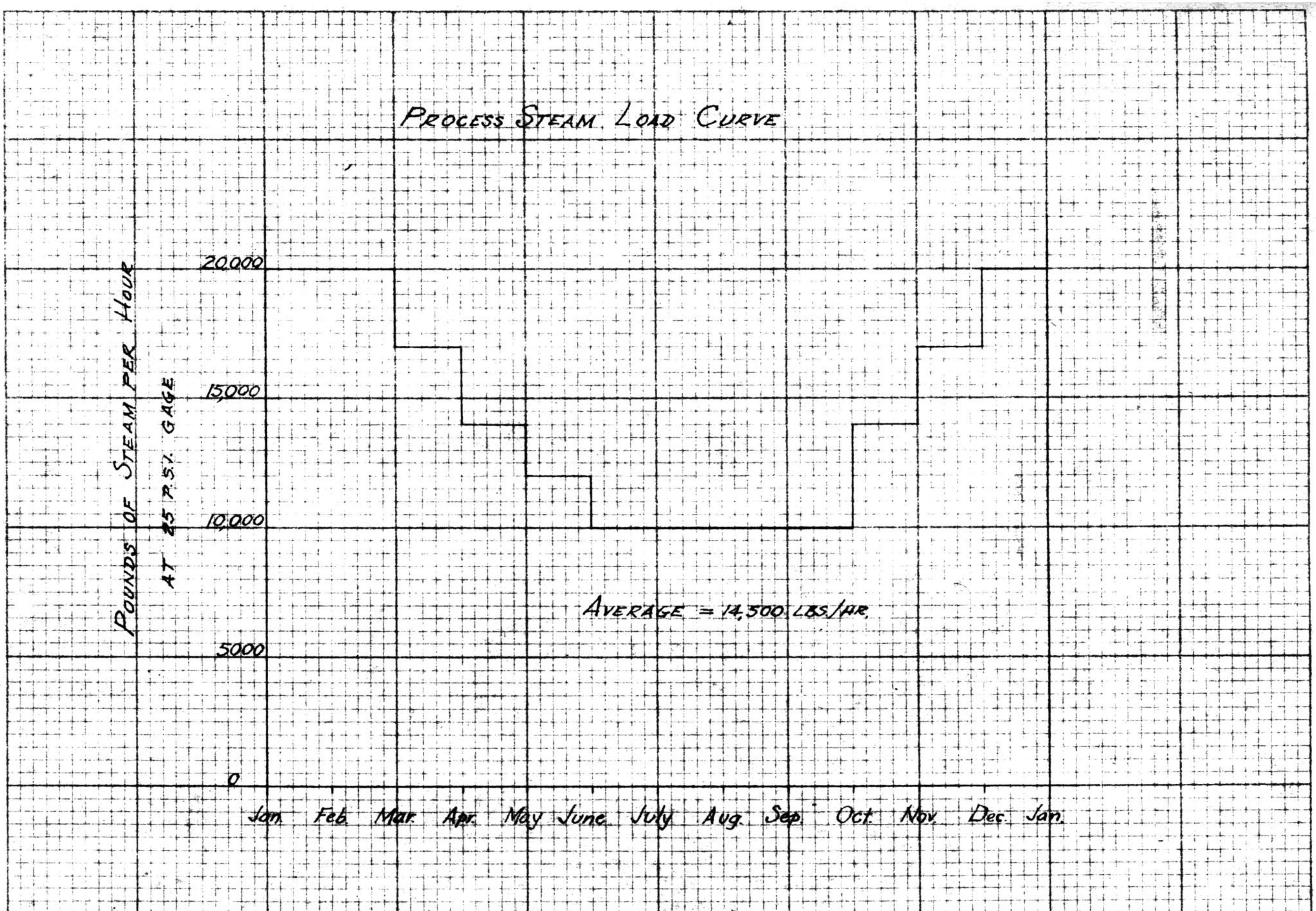

PROCESS STEAM LOAD CURVE
POUNDS OF STEAM PER HOUR
AT 25 P.S.I. GAGE
20,000
15,000
10,000
5000
0
AVERAGE = 14,500 LBS/HR.
Jan. Feb. Mar. Apr. May June July Aug. Sep. Oct. Nov. Dec. Jan.

III. BUY OR MAKE POWER?

The entire problem at hand is to decide whether to purchase power and install a process steam plant or to generate our own power to handle the load and the process steam required. Naturally, the ultimate basis of selection is the cheapest or the most economical plan possible. No definite policy or set of rates can be set down upon which plant owners can decide the question of making versus buying power. But engineering knowledge backed by experience make certain classifications possible.

It pays to purchase power

1. If power rates are right, together with cost of transmission and transformer loss—in plants of small size and moderate power demand with little or no need for process steam.
2. If business is growing so very rapidly that the risk of building own plant is unwarranted because of the changing size of generating units and type of equipment.
3. If by-product power is insufficient to meet requirements. The best economy may be served here by purchasing the balance of energy needed.
4. If exhaust steam can be used for heating during cold months only. Purchase of power at suitable rates in the warm months may result in better economy than operating non-condensing engines or turbines, or Diesel units.

It pays to make power

1. In large or small establishments, if all of the exhaust steam from the power units can be utilized for heating or for process work. This may also be true (as in the plant under consideration) when part of the exhaust is so utilized by bleeding from the turbine. The number of manufacturing plants which can profit by this method has largely increased--even those of very small capacity such as 100 kw plants. Where the process steam is one half of the power load, this method may cut down the fuel consumption by half.
2. Where there is a continuous moderate or large power demand throughout, even if there is no process steam load. Under these conditions the overhead cost per kw hr is small, so that with efficient design a saving can be made.
3. When a considerable boiler plant must in any case be installed for heating or process work. It may be possible here to install a power unit without adding boiler capacity so that the increase in overhead charge is only for the power unit.

Fuel is the greatest single item entering into the cost of power. It makes for the many possibilities for economy in the by-product power plant which are independent of the steam pressure required for the process. "An ordinary simple steam-engine unit whose exhaust is absorbed in building warming or heating liquids or drying processes is delivering by-product electrical energy for a small fraction of the fuel required per kilowatt-hour by the most 'super' of super power plants."*

Of course, fuel is not the only means with which to compare power costs. The following table illustrates the influence of the other factors.

Power Cost Items**

Item	Generated Power	Purchased Power
Fuel	May use more fuel. Modernizing old plant will reduce this quantity and cost.	May use less fuel, but fuel for heating and process must be continued.
Supplies	More supplies.	Less supplies, but supplies for boiler house still needed.
Labor	More labor, but modernizing may reduce amount.	Less labor, but steam still requires some.
Maintenance and repairs	More of these items which, however, modernizing may reduce.	Less of these items. For boiler plant may be the same.
Fixed charges	New fixed charges if plant is to be improved, otherwise none.	New fixed charges caused by installation of purchased power.
Purchased power	None if all the power is generated.	Cost of purchased power including transformer losses.
Total	Sum	Sum

Improvements in either scheme may often throw the decision in its favor. There are many situations even with plants operating continually for twenty-four hours a day with high power factors where

* D. Myers, Reducing Industrial Power Costs, p. 66, New York, 1935.
**D. Myers, op.cit., p. 66.

power can be generated more cheaply than by purchasing it, where it pays, nevertheless, to purchase power if the difference is not too great. This fact is illustrated in the case where the limited available capital can be turned over more rapidly by improvements in the methods of production than by investment in a private power plant.

There are compromise cases where power is purchased as well as made in an economical balance between the utility company and the industrial plant. Also when the load increases and power expansion is necessary, this expansion is often more easily covered by the purchase of power. The necessity for reserve capacity is sometimes relieved by cooperation with the utility company through the supply to the utility company of surplus power by the industrial plant or the latter being supplied with utility power in cases of peak and emergency loads.

For the plant under consideration we may choose one of several alternatives (as indicated above) on the most economical basis. The alternatives are:

1. Make own power and heating and process steam.
2. Purchase power and produce heating and process steam in a low pressure boiler plant.
3. Purchase power and process steam.
4. Interchange of power with a utility company.
5. Part purchase of power and generate remainder and the process steam.

From the above discussion, we note that only the first two alternatives can be suitable for the plant under consideration on a competitive basis since the fourth finds sway in large scale power generation, while the fifth is adaptable only to an old or existing plant whose load has slightly increased or in cases of emergency. The third method is of value when the demands of power and process steam are both small.

An investigation of the first two methods has been made and it is recommended to generate our own power and process steam. This method has the added advantage that the power plant can be located near the factory so that the voltage regulation is closer and the power transmission losses smaller. Often, steam cannot be purchased from a utility company, so that a boiler plant must be installed in any case and consequently it is more economical to generate the power as well as the steam.

Since process steam is required in the scheme adopted (generating own power), steam must be the working medium in the power plant, otherwise Diesel power could be efficiently and reliably used.

IV. ECONOMICS OF DESIGN

The examination of an industrial power plant should be based on suitability, the possibility of future enlargement, its proper and economical operation, availability of correct and continuous operating data and cost figures. An industrial plant differs from the central station in the fixity of load curve which is the main requirement the latter must meet. Changes in product, process or demand necessarily impose varying load conditions on the whole plant or its subdivisions not found in the central station. This difference is well illustrated in the matter of providing reserve capacity. "Some processes are continuous and any interruption in the flow of power or steam may cause a great loss in the semi-finished product. In some textile processes, for instance, a slight change in electrical frequency causes considerable loss"* In other cases, such as power supply to a mine, production can be stopped without much loss until service is resumed. In general, however, to be perfectly reliable, an industrial plant must have enough reserve capacity (usually equal to the largest unit) to meet any demand of power and process steam at any time. Reserve capacity must also have an eye for the future enlargement of the plant. This reserve must not be too large, but its amount must be based on an economical compromise between the increased investment, estimate of load growth and the worth of great reliability.

The cost of producing power is made up of two classes of costs—fixed and operating charges. The former deals with the plant itself, practically regardless of its use, and consists of interest on the investment, depreciation, taxes and insurance. The latter

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*Justin and Mervine, Power Supply Economics, p. 231, New York, 1934.

includes the cost of fuel, labor, oil, waste, supplies, and maintenance. The most desirable plant is that one with least total cost (sum of the two charges).

The conditions affecting investment are as follows:*

1. Location. The cost of power transmission is appreciably reduced if the power plant is located close to the manufacturing plant. It should also be located near an adequate supply of water for condensing purposes (a river in the plant under consideration); as much as possible on flat ground free from swamps, rock and gravel so that the foundation cost can be kept at a minimum level.
2. Manner in which fuel is received. If the coal is water borne, considerable investment must be made in docking and unloading facilities. Similar costs are not required where the coal is received by rail. If gas or oil is used, their handling is relatively unimportant because it is usually very cheap and simple.
3. Values of plant size. If the land is very expensive, as in metropolitan areas, it is necessary that the plant be built on the basis of maximum capacity per square foot of area, even at the expense of good design. This condition dictates the design of tall and often cumbersome buildings greatly crowded with equipment.

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*Justin and Mervine, op.cit., pp. 84-86.

4. Size of load. Low unit cost results if as large units as can be fitted to the load curve are installed.

5. Architectural treatment. This depends upon the preference of the management. Cost comparison would be superfluous in this case.

6. Fuel cost. Since this is the largest single item in the production cost of power, it is the controlling factor in plant investment. Where the fuel cost is high, equipment for improving plant efficiency is necessary. On the other hand, if the fuel cost is low, investment for efficiency improvement is unjustified.

7. Safety of operation. Investment required for this item is not based on an economical viewpoint, but <u>must</u> be made in any case, if the plant is to be safe and prosperous. It is an important item which the designer cannot afford to overlook.

V. CONDITIONS TO MEET PRIOR TO DESIGN

A load curve is useful and necessary for plant analysis, proper operation, prediction of magnitude and character of load on the power plant. No two load curves are ever identical, but they are similar in that they all have peaks and valleys. The ideal load curve is a flat rectangle, but this is never realized in practice. A lighting load is usually characterized by sharp peaks during the evening hours, thunderstorms or special events such as New Year's Eve at midnight, and long low valleys for the remainder of the time. On the other hand, the industrial power load has long, relatively flat, peaks during working hours, but almost disappears later. By flattening out the load curve, the load factor is increased resulting in the operation of boilers at a more uniform load, which is reflected in higher boiler and furnace efficiency. It is evident that the higher the load factor, the cheaper is the cost per kw hr, the fixed charges remaining the same regardless whether the plant is running at half load, full load, full time, half time or idle.

For the plant under consideration, the load curve is given and nothing can be done to improve or change it. In other words, the plant must be designed to meet <u>this</u> curve as economically as possible. The load is relatively steady and high as compared with the maximum during most of the day and then falls off to another steady demand. The morning is well started with 78% (at 8:00 a.m.) of maximum demand and continues for the remainder of the day with a steady average of about 83% (1875 kw) of maximum demand (2250 kw). From about 12 midnight until early morning a steady demand of about 28% (625 kw) of maximum is required. The load curve is therefore seen to be relatively favorable. The average load gives a dilay load factor of 59.5%, an unusually high value. This desirable condition is grestly impaired by the stoppage of power demand during Sundays. The boilers are kept warm during winter, however, because steam must be supplied for heating. During the summer months added cost is incurred in the necessity of purchasing extra coal for banking the boilers.

The process load is steady during the day and relatively so during the month, but the fluctuation per year is quite high (50% of maximum bleed). The average yearly bleed is nevertheless 73% of the maximum, which is about 50% of the total maximum turbine throttle. As was stated in a previous section, this ratio of bleed to throttle flow makes for greater economical coal consumption. The lowest bleed load continues for 33% of the year.

The power and bleed loads given must be fitted with the most suitable and economical power units. In general, the turbine has many advantages over the reciprocating steam engine. It requires only a fraction of the floor space. Its foundation bulk and cost are greatly reduced, not only because of the small weight and floor space of the turbine and generator, but because of the lack of vibration introduced by reciprocating parts. Besides, turbine speeds are more adaptable to generator drives, decreasing investment costs still further. Steam and exhaust piping are of smaller size owing to the continuous flow of steam to the turbine, which has no piston and cylinder to be lubricated thus saving the expense of cylinder oil and maintaining the exhaust steam and condensate oil-free. Non-condensing steam steam engines of ordinary sizes (500 kw), however, do require less steam per kw hr than a non-condensing turbine, but for larger capacities the opposite is true. Since the process steam demand does not coincide with that required for power, a non-condensing turbine cannot be used as the main unit and a bleeder turbine is necessarily imposed upon the designer for selection. There are many alternatives possible for choosing the number and sizes of the power units. These alternatives are listed below in accordance with standard sizes.

Item	Plan 1	Plan 2	Plan 3	Plan 4
Units Used:				
1. Bleeding turbine (kw)	2-1500	2-2500	3-1000	2-1250
2. Non-bleeding turbine (kw)	1-1000	0	1-1000	1-1000
Spare Unit (kw)	1-1500	1-2500	1-1000	1-1000
Total Number of Units	3	2	4	3
Installed Capacity (kw)	4000	5000	4000	3500
Maximum Load (kw)	2250	2250	2250	2250
Reserve Capacity (kw)	1750	2750	1750	1250
Average Load (kw)	1337.5	1337.5	1337.5	1337.5
Load Factor (%)	59.5	59.5	59.5	59.5
Capacity Factor (%)	33.4	26.7	33.4	38.2

The necessity for spare capacity, usually equal to the size of the largest unit, has already been pointed out and is seen to be common to all plans. The capacity factor varies little for all plans so that a reasonably equal use is made of the installed capacity.

The most economical selection is the largest unit that can be installed, for as turbine size increases, weight, floor area and consequently cost per unit of capacity decrease. By increasing steam pressures and temperatures, the station heat consumption can be decreased. Therefore, as units increase in size, steam pressures and temperatures also increase with a resultant lowering of steam consumption and increase in station efficiency. This consideration urges the rejection of plans 3 and 4. In plan 2, the capacity is very large and the unit the most efficient of all four plans, but for most of the time by far it would have to run much below rating, never developing its maximum power. This poor load factor on the turbine makes it run inefficiently. Besides, the installed capacity is highest and the investment cost likewise greatest. Plan 3 has the advantages of greater reliability and flexibility but these do not outweigh the disadvantages of poor efficiency and higher cost per unit of capacity. Plan 1 is more flexible than plan 2, as well as more economical. This is the plan to adopt, although it must be recognized that each plan has its advantages, and the selection of plan 1 is arrived at through the most economical compromise. Both the bleeding and straight condensing turbines in plan 1 operate at approximately three-quarter rating most of the time. The latter has a better engine efficiency than the non-condensing type.

The boiler, as well as other items of design, are considered in a later section.

VI. DESIGN OF THE STEAM PLANT

Having decided upon the means with which to supply the manufacturing plant with power and process steam and upon the plant location and site, the designer must next select the equipment which will produce the required amount of power with the greatest reliability at the lowest total cost. Where the load is variable, as in the proposed plant, equipment with a flat-topped efficiency curve is more desirable than with a peaked one.

For each combination of fuel cost, load factor and capacity factor, there is an optimum steam pressure and temperature which gives the lowest cost of power. With higher pressures and temperatures the thermal efficiency of a steam prime mover is increased thereby saving fuel cost. However, as pressures go up the investment cost per unit of capacity increases, so that an economical balance must be maintained between efficiency and investment. On this basis the steam pressure is selected. For the proposed plant a steam pressure of 450 psia is designated in the problem. This figure is in accordance with optimum conditions. The steam temperature is determined from the condition that the process steam is bled dry saturated at 25 psi gage. (See Appendix)

The vacuum which it is possible to attain depends upon the inlet temperature of the cooling water and the quantity of cooling water passing through the condenser. Summer water temperature is given as 70° F. This temperature does not vary greatly throughout the year. If it is lower than this value (as in winter) a better vacuum results in any case. Therefore, this temperature is assumed as the design condition. The selection of the optimum vacuum is one of balancing the saving due to decreased steam consumption by the turbine against the increased pumping costs and the increased investment cost to attain the high vacuum. Also as the vacuum increases, the volume of the air in the water to be handled by the air pump increases very rapidly. Taking all factors into account a 28" Hg vacuum is selected. The statement of the problem with regard to ground level and condensing water implies the adoption of a low level jet condenser. The feature of this condenser is that it can be operated with a 5° terminal difference between injection water temperature and steam temperature corresponding to the pressure in the condenser. A minimum quantity of injection water is handled. A multi-jet condenser has a more efficient removal pump but uses more injection water.

A low level jet condenser is finally designed under each turbine, together with all auxiliaries such as removal and injection pumps as well as the steam jet air pumps. The design also uses expansion joints to protect the turbine against upward thrust. The removal pumps discharge the mixture of water and steam back to the river downstream.

It was decided in an earlier section that two 1500 kw bleeding turbines (one spare) and one straight condensing 1000 kw turbine will be used as the power generating units in the plant. The most economical selection of boilers is subject to principles similar to the economical selection of turbines. The water tube boiler is the most suitable for large power plants because large powers can be obtained from single units due to the saving in floor space and because "high steam pressures in large units can be carried without any appreciable thickening of metal through which the heat of fire is transmitted."* Almost any type of water-tube boiler could fit a new installation equally well. The selection is one of floor space, investment and individual preference of the designer or owner. Overall plant efficiency is very little influenced by size and number, but the investment per unit of capacity is reduced by increasing the size and decreasing the number of units. The capacity increases with the width but setting costs do not increase proportionally so that the unit cost is less. The boiler must be able to meet not only the usual maximum load but also emergency demands. A spare unit should be added for greater reliability from the plant and continuity of operation.

Two alternatives are open in selecting the number of boilers. One is to have one unit per turbine and the other is to have one unit to handle the whole plant alone. With both schemes a spare unit of capacity equal to the largest unit is added. The two schemes are analysed in the appendix, and the first is adopted as the most economical scheme. The furnace volume is dependent upon the boiler and the rate of firing.

The statement of the problem specifies the use of coal as fuel so that the selection of the fuel burning equipment narrows down to the choice between mechanical stokers and pulverized coal firing. Efficiency is again of secondary importance in the selection of equipment. For plants of 1500 hp and over, mechanical stokers are undoubtedly the type of equipment to use because of the freedom from smoke troubles, insurance against labor troubles and the ability to push a boiler from a banked condition to 150 per cent rating in ten minutes. While pulver-

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*Miller and Holt, <u>Notes on Power Plant Design</u>, p. 25, 1930.

ized coal firing has the advantage of better combustion control, the use of less excess air, adaptability to burn any type of coal, it needs a larger furnace volume and is troubled with the great nuisance of fly ash. Underfeed stokers are best suited to the coals of lower ash content, as in the coal selected (3.5% ash content).

The object of the heat balance is to so correlate the apparatus and its functioning as to best conserve the utilization of energy under all rates of steam generation, and especially the average one. "..Although the use of more efficient equipment or the addition of heat-recovery apparatus, will always result in an improvement in plant performance, the inclusion of such equipment may not be economically sound.."* due to increased investment and operating costs. In a condensing plant, such as the one under consideration, a thermal gain results from the reduction in the back pressure through the use of a condenser; also by the use of steam-driven auxiliaries whose exhaust steam may be utilized in heating the condensate from the main condenser. By using a combination of electric as well as steam-driven auxiliaries a satisfactory energy balance can be obtained with varying loads by shifting from one to the other type of drive according to conditions. Motor drive from the main unit generates the power with the least consumption of steam but with a greater consumption of heat. Turbine drive swings to the other extreme and develops the power with the maximum steam consumption, but the exhaust is saved in heating the feedwater.

The regenerative cycle makes a far greater utilization of energy than is possible with the condensing or Rankine cycle mentioned above. Its use is accompanied, however, by a greater expense for the

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*Barnard, Ellenwood, Hirshfeld, Heat-Power Engineering, vol. III, p. 1028.

numerous heaters employed, together with the added piping, pumps and traps. As the prime movers in the proposed plant are relatively small, the cost of this added equipment is large. Nevertheless, the more complicated the plant, the less is the reliability.

Another problem in the economical heat balance is the selection of the type of equipment for treating the feedwater. The water available from the river is suitable for use in the boilers after it is filtered so that its treatment is not very complicated. From a thermal standpoint an evaporator gives very good performance because all the energy of the steam supplied to it, except for losses in blowdown and radiation, appears in the vapor produced or the condensate. The vapor should be condensed by the feedwater and the condensate enter the feedwater system at a suitable point. It, therefore, vaporizes raw water and acts as a heat-recovery apparatus, but the investment and operating charges are high. A zeolite softener, on the other hand, is very simple to operate and costs very much less than an evaporator.

The above considerations encourage the designer to investigate three plans for the proposed plant: (1) to use steam driven auxiliaries and a zeolite softener; (2) to use electric driven auxiliaries and a zeolite softener; (3) to use electric driven auxiliaries and an evaporator to treat the raw water. Plan 3 was found to be extremely uneconomical. Plan 1 was selected because the exhaust steam can be utilized for heating the feedwater although plans 1 and 2 were both equally feasible.

Since the temperature of the flue gases leaving the furnace is above 600° F, it is advantageous to install an economizer in the plant to recover the heat contained in the gases by heating the feedwater. Among the many requirements that the economizer must meet are: (1) the exit gas should not be cooled to the dew point of the sulphuric acid vapor contained in it; (2) the entering water should be at a temperature high enough to prevent internal corrosion of the tubes from dissolved or occluded gases.

Without fans, the high rates of heat transfer now possible together with the thick fuel beds of underfeed stokers could not have been realized. The forced draft fan ordinarily takes its air from the boiler room and delivers it through air ducts to the stoker plenum chamber overcoming the resistances of the stoker and fuel bed. Induced draft refers to gas movement produced as a result of a vacuum, and is created by chimneys and by fans located in the gas passages on the chimney side of the boiler and its auxiliaries. Either arrangement alone is undesirable. The most logical one is to employ both vacuum and plenum chamber in such proportions that the furnace is maintained at nearly atmospheric pressure so as to offset small leaks in setting and to make the opening of furnace doors or ports safe without an outflow into the boiler room. For the proposed plant a balanced draft system (with induced and forced draft fans) is adopted, and the controls are set to maintain 0.1 inch water vacuum over the fuel bed or in the furnace. The draft losses through equipment involved, such as economizer, boiler and stoker, are obtained from manufacturers' experimental data.

Bucket carriers are more economical for conveying in small capacities than belt conveyors. Other details of design are well considered in the appendix. These are ash cars, feed and other pumps and piping.

VII. DESCRIPTION OF THE NEW PLANT

The proposed plant is to be located on the river near the manufacturing plant in order to keep transmission costs at a minimum. The ground level is about six feet above the river, and the gravel bottom suitable for a sturdy foundation capable of bearing a load of 8 to 10 tons per sq. ft.* The coal will be handled by the siding connected to the main railway nearby. From the temporary open storage piles, the coal will be dumped from cars on trestles to the ground below and reclaimed by locomotive cranes.** It must not be piled higher than 30 to 40 feet owing to the danger of fire. The storage pile shall be spanned by a traveling bridge provided with power-driven self-filling grab bucket, forming a circular pile. There is ample space available for this storage both near the river and plant. For use the coal will be received by track hoppers, fed through an apron conveyor crusher and pivoted bucket carrier to the overhead bunker for 3-day storage. The ashes will be handled by steel ash hoppers of sufficient capacity for a 16-hour accumulation at maximum load. A 10 hp. motor will drive the carrier at a speed of 40 ft. per min. The equipment is of simple design for reliability of supply with minimum labor cost.

The building shall be of steel frame and hollow tile construction, stucco covered from firing floor up. The design should be attractive to the eye, and a pleasing effect can be produced by landscaping. The arrangement of units must permit work to be performed with as much freedom as though the building did not exist. In the basement will be the ash hoppers, and the ash will be removed by two men who handle all

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* Miller and Holt, <u>Notes on Power Plant Design</u>, p. 272.

** Marks, <u>Mechanical Engineers' Handbook</u>, pp. 1446-1448, (1941).

the coal and ashes from both shifts. Feed pumps and other auxiliaries will be located in the basement.

The boiler room will be 69 ft. long by 43 ft. wide and will house the two boilers each with a Riley underfeed stoker, Foster-Wheeler economizer, forced and induced draft fans. A Diamond soot blower will be supplied together with boiler meters and alarms, Copes feedwater regulator, safety valve, steam and draft gages for each boiler, a Brown electrical CO_2 recorder and other accessories. Advantage can be taken of the possibility of recovering heat lost by condensation and radiation from boiler settings by the use of hollow ventilated side walls. This is opposed to the more usual method of placing heat insulating bricks to retain the heat and prolong the life of the refractories by the cooling effect on the furnace lining. The Foster-Wheeler Convection Type Superheaters will be placed behind the front boiler tube end. The mud drums will be 8 ft. above the firing floor level.

In order to reduce the height of the boiler room, the economizers will be installed behind the boilers rather than on the top. The flue gases will make a 90° bend, therefore, and flow downward making another 90° bend at the bottom of the economizers before entering the induced draft fan, making it possible for an ash hopper to collect the fly ash carried by the gases as they round the bend. On the other hand, the feedwater will travel upward through the economizers to the boiler drums.

The forced draft fans to be installed below the firing floor level will take the air from the boiler room and deliver it through a common air duct, whose capacity is regulated by a damper, to the stoker plenum chamber. Flexibility is realized in cases of emergency where one fan may serve both boilers. The induced draft fans will take the flue gases from the bottom of the economizers and deliver them through a breeching to the superimposed steel stack which is designed for the two boilers. Again, capacity will be controlled by a damper. The drive for the fan, as for all auxiliaries, will be from steam turbines. Hagen control shall be installed to regulate fan speeds according to the steam pressure and to maintain draft over fires.

The piping arrangement is evident from the plant layout drawings. Auxiliary steam will be supplied from the main header through a reducing valve. Another such valve will be installed to take care of heating steam on Sundays at 40 psig. and the line shall be connected with the bleed line at the same pressure. The bleed pressure will be kept constant by a governor valve at the bleeder point. The steam from the superheater will be led to the main header in the boiler room, and from there an individual lead connects to each turbine. All piping is designed with long radius bends to prevent heat expansion from creating stresses at joints and distorting turbine alignment. The feedwater line will be covered with insulation and given good anchorage against vibration. The A.S.M.E. Boiler Construction Code states that the feed pipe shall be provided with a check valve near the boiler, a globe valve on the branch to each boiler, between the check valve and the source of supply. The branch lines will also contain feedwater regulator valves.

The turbine room will be 69 ft. long by 43 ft. wide and will house three turbines with a jet condenser directly under each. Each is to be directly coupled to a 2300 volt*, three-phase, 60 cycle gen-

*Morse, Power Plant Engineering and Design, p. 559.

erator and D.C. current exciter. All auxiliaries such as air ejectors, inter- and after-condensers, oil coolers and air coolers, removal and air pumps will be installed in the condenser floor. The circulating - water will be supplied from the river and then discharged back to the river.

A hot well pump of about 2500 gpm capacity will be built below the condenser floor. The feedwater will be pumped from here to the open heater together with the process and heating condensate. Before this water can be used in the boiler it must be filtered and treated in a zeolite water softener.

A switchboard will be installed in a separate control room next to the turbine room for the three generators and six feeders. A voltage regulator will be used to control generator voltages. All oil switches will be located in the basement and ample room will be provided for access to all parts.

The daily operation schedule consists of the continual operation of the 1500 kw turbine while the 1000 kw turbine remains idle from 10:00 P.M. to 7:00 A.M. On Sundays none of the turbines are running, but the boiler will be in continual operation.

Ventilation of the building is important and by the use of mechanical draft a system can be set up which withdraws the warm air from both the boiler and turbine rooms. Height of boiler and turbine rooms will be an aid to natural ventilation. Of course, the building will be equipped with doors and windows--these are to be made of steel to avoid the use of combustible materials. The plant must also be properly illuminated as well as have proper facilities for offices.

VIII. CONCLUSION

It was decided earlier that only two alternatives were to be investigated and a choice made on the most economical basis. The detailed analysis of these alternatives gives the following as the annual cost of supplying power, heat and manufacturing steam.

Item	Complete Plant Supplying Power and Heat	Purchase Power Boiler Plant Supplying Process Steam only
Total investment ($)	727,700	284,000
Annual fixed charges ($)	83,500	27,450
Annual operating charges ($)	109,460	78,560
Annual cost of original engineering supervision ($)	1,330	670
Annual cost of purchased power ($)		120,920
Total annual cost ($)	194,290	227,600
Annual saving ($)	33,310	
Cost per 1000 lb. steam (¢)	56.6	73.1
Cost in cents/kw hr	1.935	2.215

The above cost estimates are rather rough and many of them are outdated, but they serve as a method of comparing the two schemes. This comparison indicates that it is more economical to build a new steam plant to supply both power and process steam than to install a low pressure boiler plant and purchase the power. The annual saving is $33,310. which can pay for a new and complete steam plant in 22 years which is approximately the average life of a power plant. For a 2250 kw plant, 1.935 cents/kw hr is a reasonable and satisfactory cost.

It is recommended that a power plant be built to supply the power and steam required by the factory.

APPENDIX I

1. Turbine Calculations

To determine Steam Throttle Temperature

Given: 1. Steam throttle pressure = 450 psia.

2. Process steam is bled dry and saturated at 25 psi gage.

3. Condenser cooling water temperature = 70° F.

4. Select 28" Hg as the optimum vacuum* for the above temperature.

Bleeding Turbine:

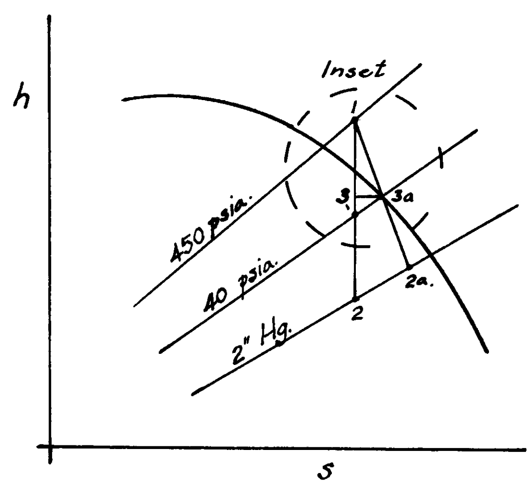
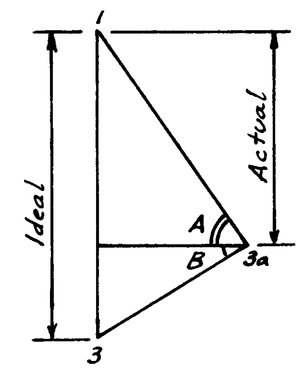


Fig. 1.



Inset.

Fig. 1a.

h_{3a} (40 psia and sat.) = 1169.7 Btu/lb.

$$\text{From Fig. 1a, } \tan B = \frac{h_{3a} - h_3}{s_{3a} - s_3}$$

For an increment of 0.1 in entropy on the Mollier chart:

$$\tan B = \frac{1170 - 1097}{1} = 730$$

---

*See condenser design and Miller and Holt, Notes on Power Plant Design, p. 218, 1930.

The turbine efficiencies for each stage are:

$$E_{T_1} = \frac{h_1 - h_{3a}}{h_1 - h_3}$$

$$E_{T_2} = \frac{h_1 - h_{2a}}{h_1 - h_2}$$

Assume $E_{T_1} = E_{T_2} = E_T$

Total power supply to the generator $= \dfrac{3413 \text{ (load)}}{E_{gen.}}$

Supply to turbine rotor $= \dfrac{3413 \text{ (load)}}{E_{gen.} \times E_{mech.}}$

The steam rates for a Curtis turbine of 1500 kw operating at 80% power factor, 175 psig., 100° F superheat and 28" Hg vacuum, are 16.2 lbs. per kw hr. at full load and 17.4 lbs. per kw hr. at half load.*

From the above, $E_T \, E_g \, E_m = \dfrac{3413}{(h_1 - h_2)}$

Generator efficiency is empirically given by

$$E_g = 0.98 - \frac{0.055}{\sqrt[3]{\frac{\text{kw rating}}{1000}}} \times \frac{\text{rating}}{\text{load}}$$

Similarly,

$$E_m = 1.00 - \frac{0.04}{\sqrt[2]{\frac{\text{kw rating}}{1000}}}$$

For the conditions of the above turbine, h_1 is 1258 Btu/lb. and h_2 is 903 Btu/lb.

Hence, at full load,

$$E_g \times E_T \times E_m = \frac{3413}{16.2\ (1258 - 903)} = 59.4\%$$

---

*Miller and Holt, Notes on Power Plant Design, p. 184.

$$E_g = 0.98 - \frac{0.055}{\sqrt[3]{\frac{1500}{1000}}} \times \frac{1500}{1500} = 93.2\%$$

$$E_m = 1.00 - \frac{0.04}{\sqrt{\frac{1500}{1000}}} = 96.7\%$$

Hence, $E_T = \dfrac{.594}{.967 \times .932} = 66\%$

Now $E_T = \dfrac{h_1 - h_{3a}}{h_1 - h_3} = \dfrac{(h_1 - h_{3a})}{(h_1 - h_{3a}) + (S_{3a} - S_3) \tan B}$

Or $\dfrac{h_1 - h_{3a}}{S_{3a} - S_3} = \dfrac{E_T \times \tan B}{(1 - E_T)} = \dfrac{.66 \times 730}{.34} = 1415$

Therefore $\tan A = 1415$

Drawing a line of the above slope (1415) through point 3a to intersect the 450 psia pressure line on the Mollier Chart, the steam temperature is found to be 608° F. Hence, h_1 is 1307, h_3 is 1098 and h_2 is 880 Btu/lb. This temperature is also used for the non-bleeding turbine. These values are true because the performance of the turbine to be designed is assumed to have the same performance as the model Curtis turbine above. To find the actual steam rates of the turbine to be designed,

$$\text{Steam rate at full load} = \frac{3413}{.594\ (1307 - 880)} = 13.45 \text{ lbs}$$

per kw hr.

Assume the ratio of full to half load steam rates of the two turbines are the same.

$$\text{Steam rate at half load} = 13.45 \times \frac{16.2}{17.4} = 14.4 \text{ lbs per kw hr.}$$

Willan's Line is therefore:

TABLE 1

Load	lbs.steam/kw hr	kw	lbs.steam/hr
Full	13.45	1500	20,200
Half	14.40	750	10,800

Let W_{nl} = no load throttle flow in lbs/hr

W_b = bleed in lbs/hr

W_t = turbine throttle flow in lbs/hr

From Figs. 1 and 1a,

$$\text{Supply to turbine rotor} = (W_t - W_b)\ (h_1 - h_{2a})$$

$$+ W_b(h_1 - h_{3a}) = (W_t - W_b)(h_1 - h_2)E_T + W_b(h_1 - h_3)\ E_T$$

Maximum throttle flow is when all steam is bled at full load.

Hence, $W_b = W_t$.

$$W_{b\ max}\ (h_1 - h_3)\ E_T = \frac{3413\ (\text{load})}{E_g\ E_m}$$

Maximum extraction occurs when all no load steam is bled

$$W_{nl}\ (h_1 - h_2)\ E_T = W_{nl\ max}\ (h_1 - h_3)\ E_T$$

$$W_{nl\ max} = W_{nl}\ \frac{h_1 - h_2}{h_1 - h_3}$$

From Fig. 2, W_{nl} = 1200 lbs/hr

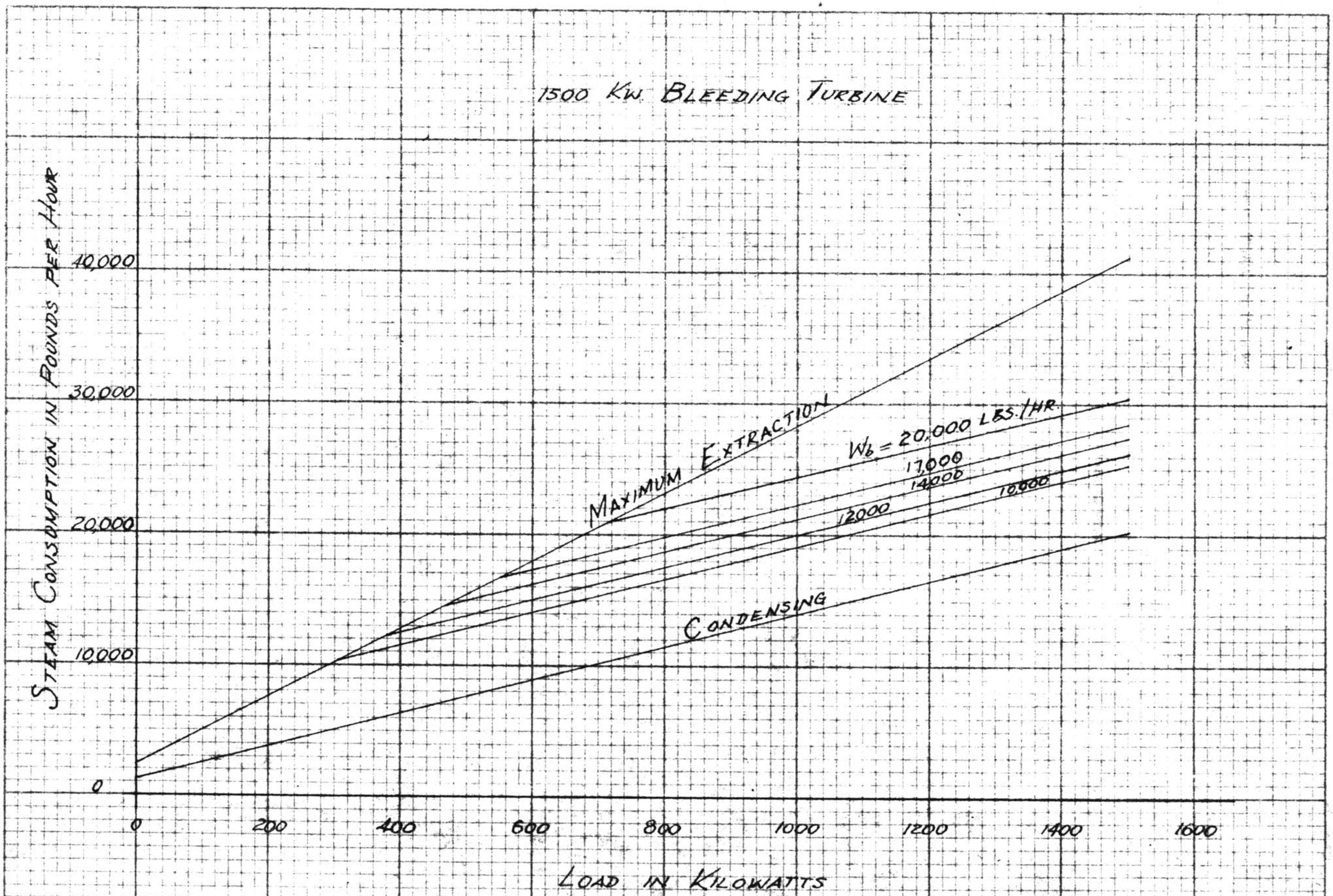
Therefore, $W_{nl\ max} = 1200 \times \frac{(1307 - 880)}{(1307 - 1098)} = 2450$ lbs/hr

For full load,

$$W_{b\ max} = \frac{3413 \times 1500}{.932 \times .967 \times .66\ (1307 - 1098)} = 41{,}100 \text{ lbs/hr}$$

$$(W_t - W_b)\ (h_1 - h_2)\ E_T + W_b\ (h_1 - h_3)\ E_T = \frac{3413\ (\text{load})}{E_g\ E_m}$$

The throttle flow for the different bleeds is calculated and listed in Table 2.



1500 KW. BLEEDING TURBINE
STEAM CONSUMPTION IN POUNDS PER HOUR
40,000
30,000
20,000
10,000
0
MAXIMUM EXTRACTION
W_b = 20,000 LBS./HR.
17,000
14,000
12000
10,000
CONDENSING
0
200
400
600
800
1000
1200
1400
1600
LOAD IN KILOWATTS

FIG. 2.

TABLE 2

W_b	W_t
10,000	25,300
12,000	26,200
14,000	27,400
17,000	28,600
20,000	30,400

To find exhaust enthalpy:

$$E_T = .66 = \frac{1307 - h_{2a}}{1307 - 880}$$

Therefore, $h_{2a} = 1025$ Btu/lb

Non-bleeding turbine (1000 kw):

At full load, $$E_g = 0.98 - \frac{0.055}{\sqrt[3]{\frac{1000}{1000}}} \times \frac{1000}{1000} = 92.5\%$$

$$E_m = 1.00 - \frac{0.04}{\sqrt{\frac{1000}{1000}}} = 96\%$$

Assume $E_T = .65$*

Therefore, steam rate $= \dfrac{3413}{.65 \times .925 \times .96\ (1307 - 880)} = 13.9$ lbs/kw hr.

Ratio of full to half load steam rates for 1000 kw non turbine is 0.886.**

Half load steam rate $= \dfrac{13.9}{.886} = 15.7$ lbs/kw hr.

Willan's Line is therefore:

TABLE 3

Load	lbs./kw hr	kw	lbs./hr
Full	13.9	1000	13,900
Half	15.7	500	7,750

Table 4 gives the load distribution and throttle flow at different times of the day for different bleeds.

* Kent, Mechanical Engineer's Handbook, p. 8-61, 1936.

**Barnard, Hirshfeld, Ellenwood, Heat Power Engineering, p. 103

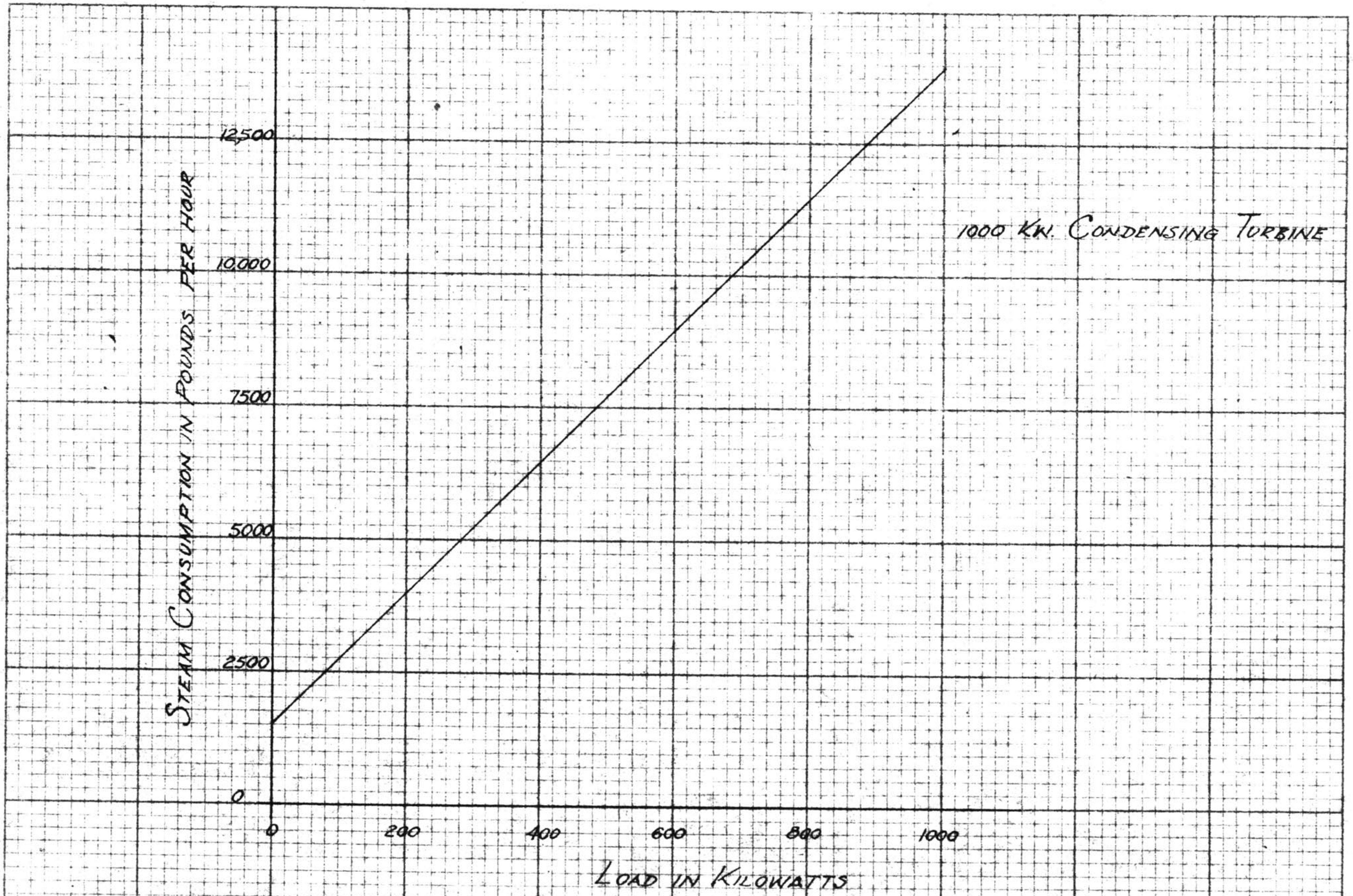

1000 KW. CONDENSING TURBINE
STEAM CONSUMPTION IN POUNDS PER HOUR
12,500
10,000
7500
5000
2500
0
0
200
400
600
800
1000
LOAD IN KILOWATTS
FIG. 3.

TABLE 4

Time	No.of Hours	Total Load (kw)	Load on Units (kw) 1500 kw	Load on Units (kw) 1000 kw	Total Turbine Throttle (lbs/hr) at Various Bleed Loads (lbs/hr) 20,000	17,000	14,000	12,000	10,000
1--6 a.m.	5	625	720 for 20,000 625 for others	-	21,000	17,600	16,500	15,200	14,400
6—7 a.m. 11—1 p.m.	3	750	750	-	21,200	19,100	18,000	16,900	16,000
10-11 a.m.	1	1000	1000	-	24,300	22,300	21,200	20,000	19,100
7—8 a.m.	1	1375	875	500	22,900	20,800	19,700	18,400	17,700
					7,750	7,750	7,750	7,750	7,750
					30,650	28,550	27,450	26,150	25,450
11—1 a.m. 9-10 p.m.	3	1500	750	750	21,200	19,100	18,000	16,900	16,000
					10,800	10,800	10,800	10,800	10,800
					32,000	29,900	28,800	27,700	26,800
8—9 a.m.	1	1750	875	875	22,900	20,800	19,700	18,400	17,700
					12,400	12,400	12,400	12,400	12,400
					35,300	33,200	32,100	30,800	30,100
9-11 a.m. 1—5 p.m.	6	1875	1000	875	24,300	22,300	21,200	20,000	19,100
					12,400	12,400	12,400	12,400	12,400
					36,700	34,700	33,600	32,400	31,500
5—7 p.m.	2	2000	1000	1000	24,300	22,300	21,200	20,000	19,100
					13,900	13,900	13,900	13,900	13,900
					38,200	36,200	35,100	33,900	33,000
7—9 p.m.	2	2125	1250	875	27,300	25,400	24,200	23,100	22,200
					12,400	12,400	12,400	12,400	12,400
					39,700	37,800	36,600	35,500	34,600

Average daily total throttle flow for different bleeds are listed in Table 5.

TABLE 5

W_b (lbs/hr)	Total Daily Ave. Throttle (lbs/hr)
20,000	30,400
17,000	28,100
14,000	27,000
12,000	25,800
10,000	25,000

The average yearly total throttle is computed according to the bleed load and is found to be 27,300 lbs/hr. Similarly, the annual average bleed flow is 14,500 lbs/hr.

The daily average throttle for each turbine is found as the sum for each, respectively, as obtained from Table 4 and then divided by 24 hours. The annual averages are easily computed.

Daily average throttle to 1000 kw turbine = 7600 lbs/hr.

Annual average throttle to 1000 kw turbine = 7600 lbs/hr.

Hence, annual average throttle to 1500 kw turbine is (27,300 - 7600) = 19,700 lbs/hr.

Daily average throttle for 1500 kw turbine is listed in Table 6.

TABLE 6

W_b (lbs/hr)	Daily Average Throttle (lbs/hr)
20,000	22,800
17,000	20,500
14,000	19,400
12,000	18,200
10,000	17,400

Max. steam flow to the two turbines = 39,700 lbs/hr.

Average annual steam from bleeder turbine to condenser = 19,500 - 14,500 = 5200 lbs/hr.

Average annual steam from non-bleeder unit to condenser = 7600 lbs/hr.

Average annual steam to the two condensers is 5200 + 7600 = 12,800 lbs/hr.

Maximum steam to the two condensers (from 7 to 9 p.m.) = (22,200 - 10,000) + 12,400 = 24,600 lbs/hr.

Maximum steam to bleeder turbine (not at maximum load) = 27,300 lbs/hr.

Maximum steam to 1000 kw turbine (not at maximum load) = 13,900 lbs/hr.

2. Condenser Calculations

The designs are based on maximum flow from the turbines with no steam bled. Average water temperature is assumed the same as that existing during the summer (70° F). This condition is on the safe side, because if the temperature is below 70° F, a better vacuum results. The statement of the problem with regard to ground level and condensing water implies the adoption of a low level jet condenser, whose features are a 5° terminal difference and the use of minimum injection water. For this type of condenser, Miller and Holt recommend 28" Hg as the optimum vacuum.*

Temperature of condensate (at 2" Hg) = 101° F

Injection water temperature = 70° F

Temperature of water leaving = 96° F

Steam comes in at h_{2a} = 1025 Btu/lb, condenses at 2" Hg.

h_{water} = 69 Btu/lb, cooling to 64 But/lb.

Condenser under 1500 kw turbine:

$$W_{steam} (h_{2a} - h_{water}) = W_{water} (96 - 70)$$

$$\text{Injection required} = \frac{20,000\ (1025 - 64)}{26} = 746,000 \text{ lbs/hr}$$

$$\text{Gpm required} = \frac{746,000}{60 \times 8.33} = 1495$$

Assume water required in air pump is 15% of that for injection.**

Water for air pump = .15 x 746,000 = 112,000 lbs/hr or 224 gpm.

From p. 218, we select No. 8 Westinghouse Leblanc Jet Condenser of 750,000 lbs/hr capacity with dimensions as given on p. 219.***

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* Miller and Holt, <u>Notes on Power Plant Design</u>, p. 218.
** <u>Ibid.</u>, p. 220.
*** Miller and Holt, <u>op. cit.</u>

Steam inlet diameter = 30 in.

$$\text{Area} = 0.785 \left(\frac{30}{12}\right)^2 = 4.9 \text{ sq. ft.}$$

Injection water inlet diameter = 9 in.

$$\text{Area} = 0.785 \left(\frac{9}{12}\right)^2 = .44 \text{ sq. ft.}$$

Quality at 2" Hg and 1032 Btu/lb is 0.922

Specific volume of steam at 2" Hg = 339.2 cu ft/lb

Specific volume of steam in condenser = 339.2 x .922 = 312 cu ft/lb

$$\text{Hence, steam velocity} = \frac{312 \times 20{,}200}{60 \times 4.90} = 21{,}200 \text{ fpm,}$$

which is a reasonable value.

$$\text{Injection water velocity} = \frac{746{,}000}{62.4 \times 60 \times .44} = 450 \text{ fpm., or } 7.50 \text{ fps}$$

Water through removal pump = 20,200 + 746,000 = 766,200 lbs/hr

or 1535 gpm

Condenser under 1000 kw turbine:

$$\text{Injection required} = \frac{13{,}900\ (1025 - 64)}{26} = 514{,}000 \text{ lbs/hr}$$

$$\text{Gpm required} = \frac{514{,}000}{60 \times 8.33} = 1030$$

As above, No. 7 Westinghouse Leblanc Jet Condenser of 600,000 lbs/hr capacity is selected.

Steam inlet diameter = 30 in.

$$\text{Area} = 0.785 \left(\frac{30}{12}\right)^2 = 4.9 \text{ sq. ft.}$$

Quality = 0.922 as before

$$\text{Steam velocity} = \frac{312 \times 13{,}900}{60 \times 4.9} = 14{,}750 \text{ fpm}$$

Injection water inlet diameter = 9 in.

$$\text{Water velocity} = \frac{514{,}000}{62.4 \times 60 \times .44} = 310 \text{ fpm or } 5.16 \text{ fps}$$

Condenser Auxiliaries:

Removal Pumps

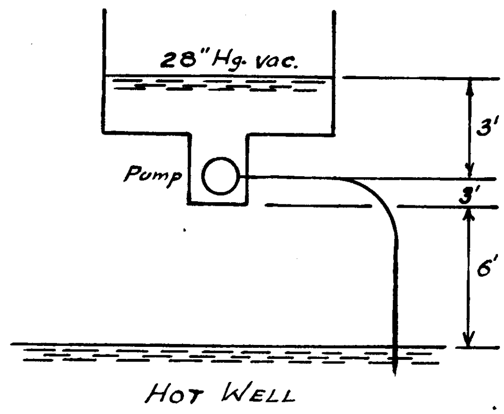

Fig. 4

1. Assume 4 ft. frictional resistance in the pipe.

2. To avoid air from being sucked in the suction pipe, sink pipe 3 ft. in the water.

3. Use dimensions and heads for pumps from condensers selected.

Removal pump head = (34 - 2.3 + 3) - (4 + 3 + 6) = 21.7 ft.

4. Assume maximum pump efficiency with no suction lift is 70%.

Suction factor for above head is found as 0.98.*

Hence, maximum efficiency = .98 x .70 = 68.5%

$$\text{Power drive for pump} = \frac{1535 \times 8.33 \times 21.7}{.685 \times 30{,}000} = 13.5 \text{ hp}$$

Use 15 hp drive for No. 8 condenser removal pump

For No. 7 condenser,

Suction head = 21.7 ft.

Suction factor = 0.98

$$\text{Power drive for pump} = \frac{1030 \times 8.33 \times 21.7}{.685 \times 30{,}000} = 9.06 \text{ hp}$$

Use 10 hp drive.

---

* Miller and Holt, Notes on Power Plant Design, p. 221.

Air Pumps:

For No. 8 condenser

Suction head = 3 + 6 + 1.3 = 10.3 ft.

Suction factor = 1.00

$$\text{Power drive} = \frac{224 \times 8.33 \times 10.3}{.70 \times 1.00 \times 30{,}000} = 0.935 \text{ hp}$$

Use 2 hp drive.

Assume 90° F air suction temperature from turbine steam.

Pressure of air at 90° F = (total pressure - pressure atmospheric vapor at 90° F.).

P_a for air = (30 - 28) x .491 - .696 = .286 psi

$$\text{Volume per pound} = \frac{R\ T_a}{P_a} = \frac{53.5 \times 550}{.286 \times 144} = 712 \text{ cu. ft.}$$

Assume pounds air per pound condensate as 0.0008.*

Steam in condenser = 20,200 lbs/hr and injection water = 746,000 lbs/hr.

Hence, volume air/lb condensate = .008 x 712 = .570 cu. ft. at 90°.

Total volume = .570 x 20,200 = 11,500 cu. ft./hr.

$$\text{Volume of injection water} = \frac{746{,}000}{62.4} = 12{,}000 \text{ cu. ft./hr.}$$

Solubility of air in water at 32° F and 29.92" Hg barometer is 1.88% by volume*

Volume of air at 32° F and 29.92" Barometer is

$$\frac{12{,}000 \times .0188 \times 29.92}{492} = \frac{.286V}{.491 \times 550}$$

$$\text{Volume, } V = \frac{12{,}000 \times .491 \times 550 \times .0188 \times 29.92}{492 \times .286}$$

$$= 13{,}000 \text{ cu. ft./hr. at } 90° \text{ F.}$$

*Miller and Holt, Notes on Power Plant Design, p. 227.

Total air to be handled by pump = 11,500 + 13,000 or 24,900 cu. ft. air at 90° F.

Steam turbine driving pump has a consumption of 8% of total steam condensed. While this seems a very high value, it is to be noted that this turbine runs non-condensing and the exhaust steam can be used for heating purposes.

For No. 7 condenser:

As above,

$$\text{Power drive} = \frac{154.5 \times 8.33 \times 10.3}{.70 \times 1.00 \times 30{,}000} = .645 \text{ hp}$$

Use 1 hp drive

Steam in condenser = 13,900 lbs/hr

Injection water = 514,000 lbs/hr

Total volume of air in condensate = .570 x 13,900 =

7930 cu.ft./hr

Volume of air in water at 90° F is

$$= \frac{514{,}000 \times .491 \times 550 \times .0188 \times 29.92}{62.4 \times 492 \times .286}$$

= 8,900 cu.ft./hr

Total volume of air handled = 8900 + 7930

= 16,830 cu.ft./hr at 90°

Total power of all pumps = 15 + 10 + 2 + 1 = 28 hp

Assume $0.01 per kw hr for operation of pumps

Operating days = 365 - 52 = 313

Cost of operation = 28 x .746 x 313 x 24 x .01

= $1580 per year

3. Heat Balance Calculations

Many plans must be taken into consideration before we can determine the most economical heat balance for the plant. For example, we may use steam or electrically driven auxiliaries; zeolite softener or evaporator for treatment of make-up water. Thus we shall investigate three plans and make selection on the basis of least cost.

Plan 1. All auxiliaries steam driven with zeolite softener for water treatment.

To find h_{2a} of exhaust from auxiliaries to open heater

$h_1 = 1307$ and

$h_2 = 1039$ Btu/lb

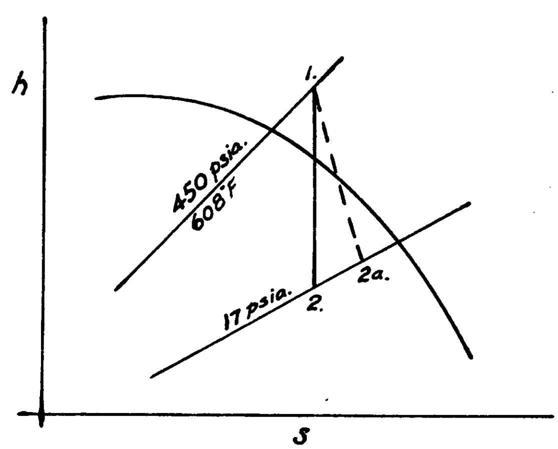

Fig. 5

1. Assume exhaust pressure is 17 psia so that the steam flows into the open heater.
2. Assume also average E_T for auxiliary is 40%.
3. Assume 10% of h_{2a} lost as heat loss from the piping, etc.

$$E_T = \frac{h_1 - h_{2a}}{h_1 - h_2} = .4$$

$$\text{or } \frac{1307 - h_{2a}}{1307 - 1039} = \frac{1307 - h_{2a}}{268} = .4$$

$$1307 - h_{2a} = 268 \times .4 = 107 \text{ Btu/lb}$$

$$h_{2a} = 1200 \text{ Btu/lb.}$$

$$\text{Heat loss} = .1 \times 1200 = 120 \text{ Btu/lb}$$

$$\text{Actual } h_{2a} = 1200 - 120 = 1080 \text{ out of aux.}$$

For maximum flow

Assume:

1. Auxiliaries use 10 - 16% steam from main turbine. Use 12% of turbine throttle flow.
2. 10% of steam supplied heating and manufacturing is lost by leakage.
3. Use 3% of turbine throttle flow for soot blowing.
4. Use 2% of turbine throttle flow for blow-down.

Therefore:

1. Total throttle flow to both turbines = 39,700 lbs/hr.
2. Auxiliary steam = .12 x 39,700 = 4760 lbs/hr.
3. Maximum bleed for heating, mfg. = 20,000 lbs/hr.

   Leakage steam = .1 x 20,000 = 2000 lbs/hr.
4. Soot blowing = .03 x 39,700 = 1190 lbs/hr.
5. Blow-down = .02 x 39,700 = 794 lbs/hr.

Heat balance is now made with emphasis on the plan feature shown below.

Calculation of steam through reducing valve:

Assume temperature of water leaving heater = 210° F.

Heat balance on open heater:

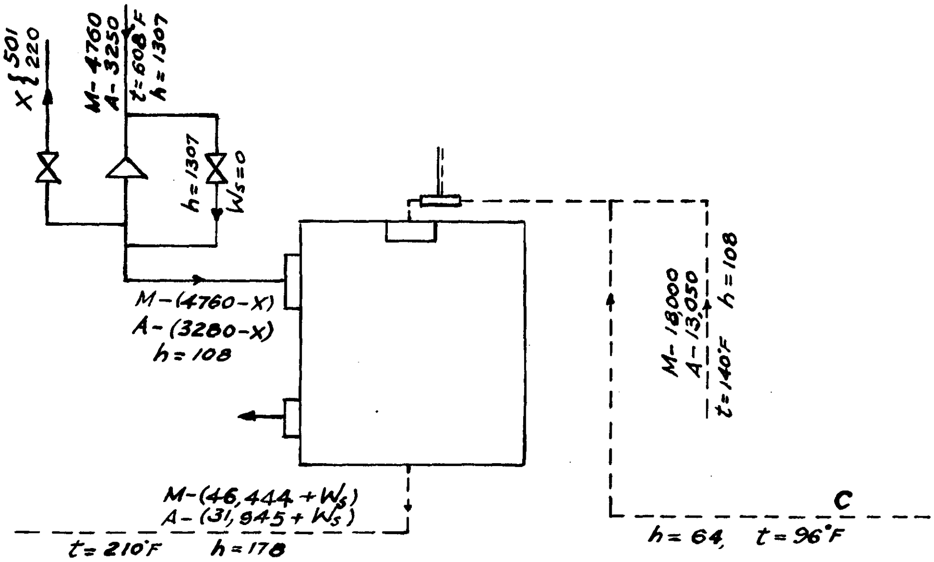

Fig. 6

Material balance:

Let X be weight of auxiliary exhaust not going to heater.

$$(39{,}700 + 1190 + 4760 + W_s + 794) = 18{,}000 + C + W_s + (4760 - X)$$

$$46{,}444 + W_s = 22{,}760 + C + W_s - X$$

$$C = 23{,}684 + X$$

Heat balance:

$$(46{,}444 + W_s)(178) = 18{,}000 \times 108 + 64\ C + 1307\ W_s + 4760 \times 1080 - 1080X \qquad \text{(A)}$$

$$8{,}260{,}000 + 178\ W_s = 1{,}943{,}000 + 64\ C + 1307\ W_s + 5{,}140{,}000 - 1080X$$

$$64\ C = 8{,}260{,}000 + W_s\ (178 - 1307) - 10^6\ (5.14 + 1.943) + 1080X$$

$$64\ C = 1.177 \times 10^6 - 1129\ W_s + 1080X$$

$$\underline{64\ C = 1.520 \times 10^6 \qquad\qquad + \ 64X}$$

$$0 = -.343 \times 10^6 - 1129\ W_s + 1016X$$

$$W_s = \frac{1016X - .343 \times 10^6}{1129}$$

If X = 0, or all auxiliary steam used for heating, W_s becomes - or 0. Substituting $W_s = 0$ and $X = 0$, the heat balance equation cannot

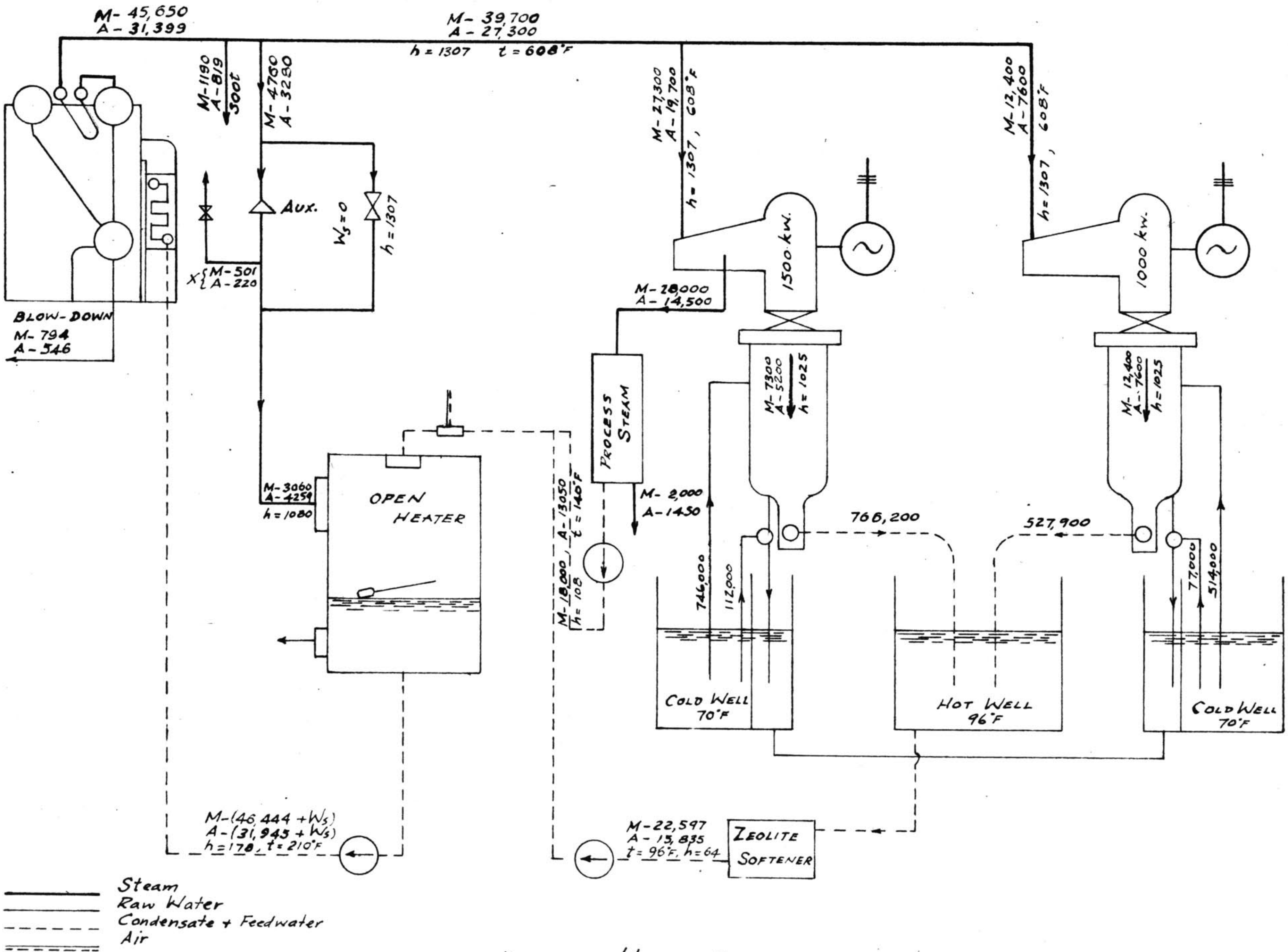

FIG. 7. HEAT BALANCE DIAGRAM

be balanced, even if the temperature of the water leaving the heater is raised to 212° F, which is the limit for an open heater. Two alternatives are open--to use a closed heater and raise the water to a higher temperature or to assume $W_s = 0$ and throw out some exhaust steam, i.e., X. Since X will be small in any case, we are not justified to use the more elaborate and more expensive closed heater. Therefore, we will use the second alternative.

$$1129\, W_s = 1016X - .343 \times 10^6$$

$$0 = 1016X - .343 \times 10^6$$

$$X = \frac{.343 \times 10^6}{1016} = 338 \text{ lbs/hr}$$

$$C = 23,684 + 338 = 24,022 \text{ lbs/hr}$$

$$\text{and } (4760 - X) = 4760 - 338 = 4422$$

$$44,856 + W_s = 44,856$$

Checking back into equation A, t is found to equal 210° F.

For average flow:

Average throttle flow = 27,300 lbs/hr

1. Auxiliary steam = .12 x 27,300 = 3280 lbs/hr

Average bleed for year = 14,500 lbs/hr

2. Leakage steam = .1 x 14,500 = 1450 lbs/hr
3. Soot blowing = .03 x 27,300 = 819 lbs/hr
4. Blow-down = .02 x 27,300 = 546 lbs/hr

Actual h_{2a} = 1080 Btu/lb

Assume t of water leaving heater = 210° F.

Material balance:

$$27,300 + 819 + 3280 + 546 = 13,050 + C + (3280 - X)$$

$$31,945 = 16,330 - X + C$$

$$C = 15,615 + X$$

Heat balance:

$$64\ C = 31,945 \times 178 - 1129\ W_s - 13,050 \times 108 - 3280 \times 1080 + 1080X$$
$$\underline{64\ C = .964 \times 10^6 \qquad\qquad\qquad\qquad\qquad\qquad\qquad\qquad + \ 64X}$$
$$0 = -.224 \times 10^6 - 1129\ W_s + 1016X$$
$$1129\ W_s = \underline{1016X - .224 \times 10^6}$$
$$0 = 1016X - .224 \times 10^6$$
$$X = \frac{.224 \times 10^6}{1016} = 220 \text{ lbs/hr}$$
$$C = 15,615 + 220 = 15,835 \text{ lbs/hr}$$
$$(3280-X) = 3280 - 220 = 3060 \text{ lbs/hr}$$

Plan 2. All auxiliaries electric driven with zeolite softener for water treatment.

For maximum flow:

1. Leakage steam = 2000 lbs/hr
2. Soot blowing = 1190 lbs/hr
3. Blow-down = 794 lbs/hr

Assume temperature of water leaving heater = 210° F.

Heat balance on open heater:

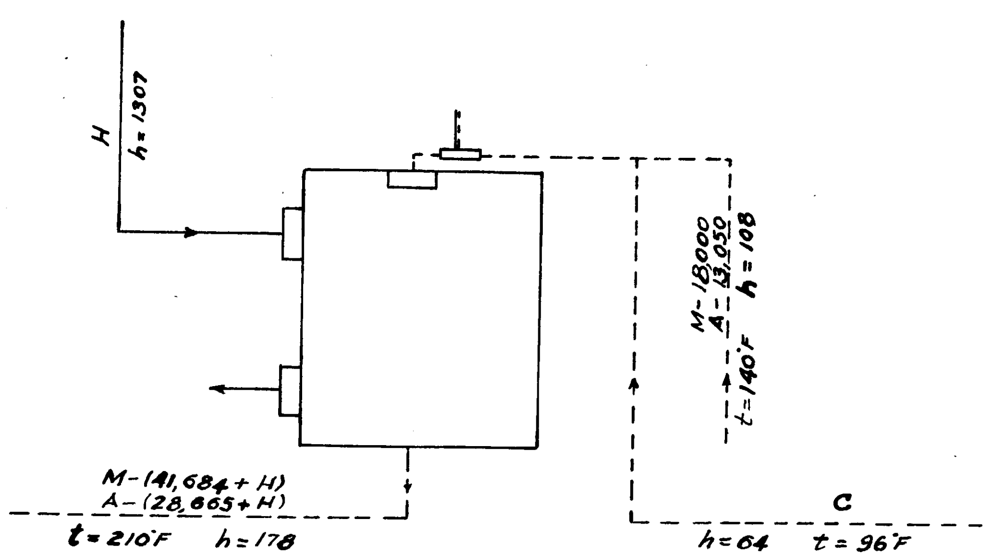

Fig. 8

PLAN 2.

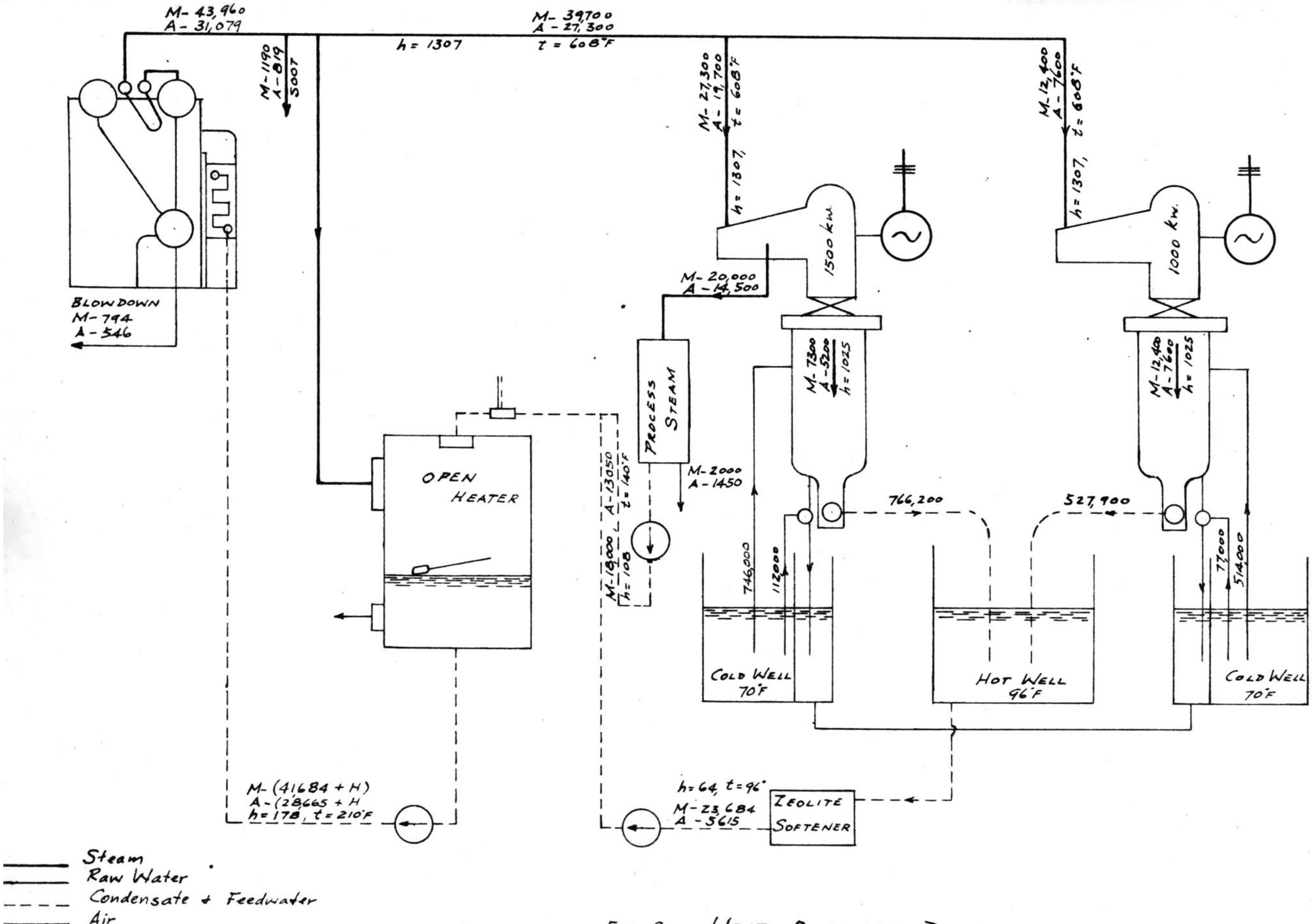

FIG. 9. HEAT BALANCE DIAGRAM

Material balance

$$41{,}684 + H = H + C + 18{,}000$$

$$C = 23{,}684$$

Heat balance:

$$1307H + 64C + 18{,}000(108) = (41{,}684 + H)\ 178$$

$$1307H + 64 \times 23{,}684 + 1.943 \times 10^6 = 7.42 \times 10^6 + 178H$$

$$1129H = 3.458 \times 10^6$$

$$H = 3070 \text{ lbs/hr}$$

For average flow:

1. Leakage steam = 1450 lbs/hr
2. Soot blowing = 819 lbs/hr
3. Blow-down = 546 lbs/hr

Material balance:

$$28{,}665 + H = H + C + 13{,}050$$

$$C = 5615 \text{ lbs/hr}$$

Heat balance:

$$1307H + 64C + 13{,}050(108) = (28{,}665 + H')\ 178$$

$$1307H + 64 \times 5615 + 1.41 \times 10^6 = 5.11 \times 10^6 + 178H'$$

$$H' = \frac{3.341 \times 10^6}{1129} = 2960 \text{ lbs/hr}$$

Plan 3. All auxiliaries are electric driven with evaporator to treat the raw water.

Heat balance on the evaporator:

Assume

1. Temperature of feedwater = 210° F.
2. Radiation loss 1% of heat supplied
3. 10% continuous blow-down

Heat loss = .01 x 1307 = 13.1 Btu/lb; hence h_m = 1294 Btu/lb.

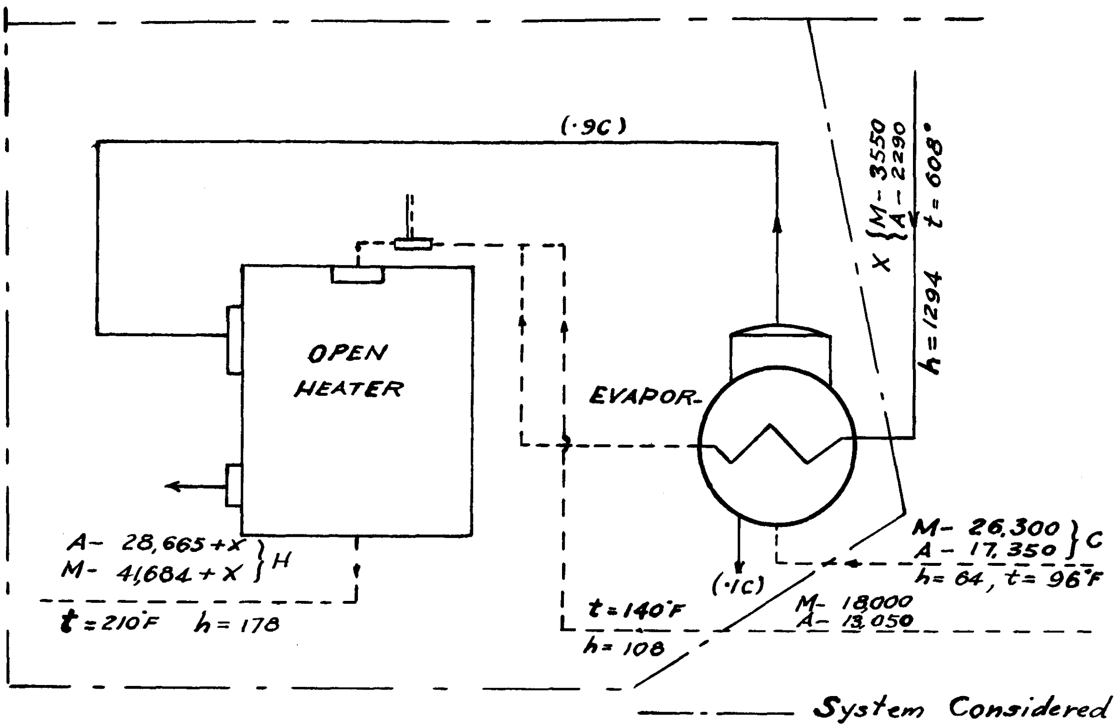

Fig. 10

For maximum flow:

Heat balance on open heater

$$18{,}000 + X + .9C = 41{,}684 + X$$

$$C = \frac{23{,}684}{.9} = 26{,}300 \text{ lbs/hr}$$

Heat balance on system indicated

$$1294X + (18{,}000)\ 108 + (26{,}300)\ 64 = (41{,}684 + X)\ 178$$

$$X\ (1294 - 178) = -\ 1.943 \times 10^6 - 1.\ 682 \times 10^6 + 7.42 \times 10^6$$

$$X = \frac{3.96 \times 10^6}{1116} = 3550 \text{ lbs/hr}$$

For average flow:

$$13{,}050 + X + .9C = 28{,}665 + X$$

$$C = \frac{15{,}615}{.9} = 17{,}350$$

$$1294X + (13{,}050)\ 108 + 17{,}350\ (64) = (28{,}665 + X)\ 178$$

$$X = \frac{2.559}{1116} = 2290 \text{ lbs/hr}$$

PLAN 3.

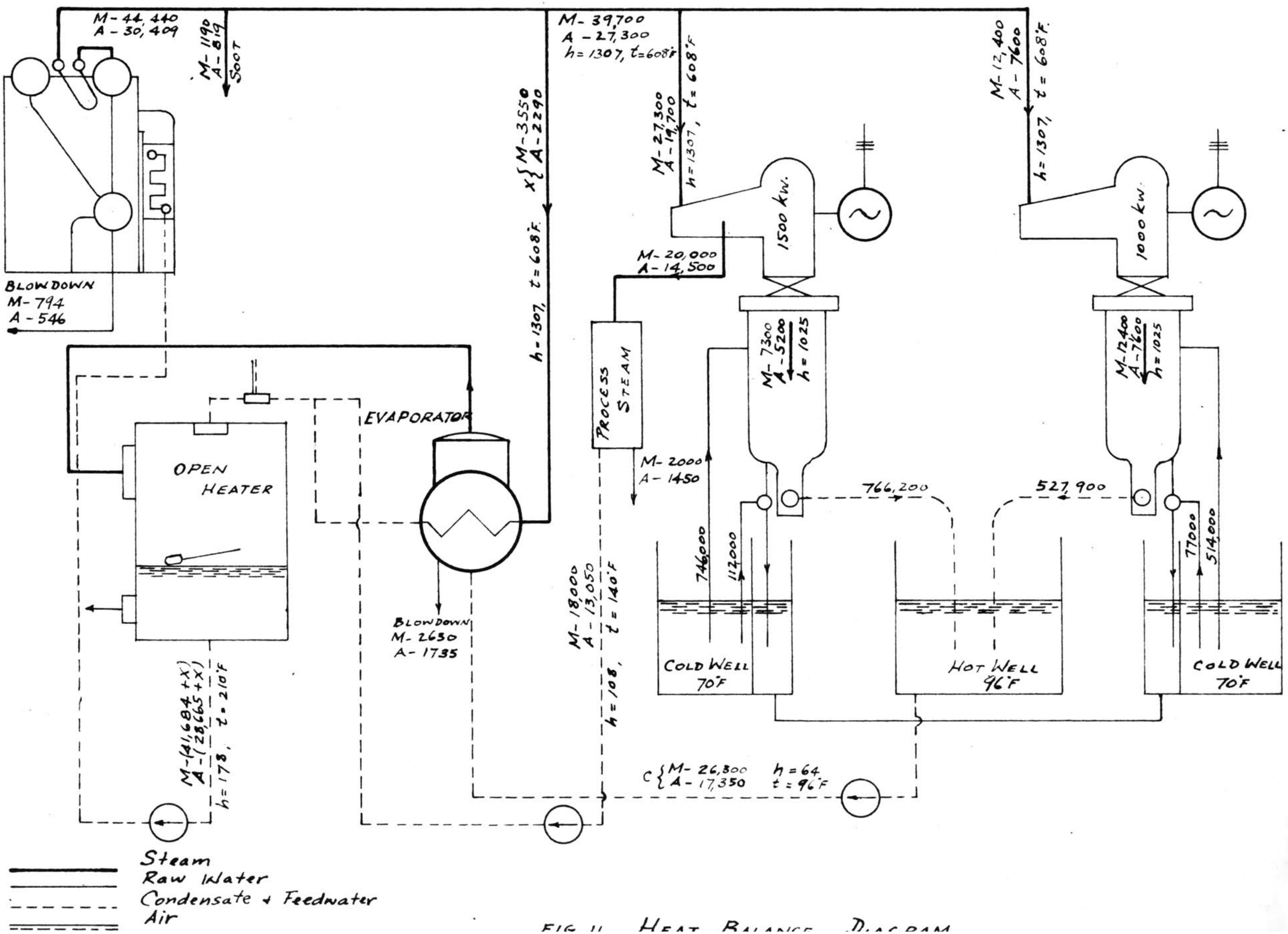

FIG 11 HEAT BALANCE DIAGRAM

Comparison of the Various Plans

For Plans 2 and 3, the power required from motors to drive auxiliaries is calculated on the basis of power developed by auxiliary steam turbine of Plan 1.

Maximum steam for auxiliaries = 4760 lbs/hr

Average steam for auxiliaries = 3280 lbs/hr

Assume $E_m = 90\%$; $E_T = 40\%$

Hence hp developed by turbine $= \dfrac{W_T\, E_T\, E_m\, (h_1 - h_2)}{2545}$

$$hp_{max} = \frac{4760 \times .9 \times .4\ (1307 - 1039)}{2545} = 180$$

$$hp_{ave} = \frac{3280 \times .9 \times .4\ (1307 - 1039)}{2545} = 124$$

Assuming motor efficiency = 85%, the power input required by the motors for electric driven auxiliaries is:

$$\text{Input kw} = \frac{hp \times .756}{.85}$$

$$\text{Max. kw} = \frac{180 \times .746}{.85} = 158$$

$$\text{Ave. kw} = \frac{124 \times .746}{.85} = 109$$

The following table is a complete comparison of plans showing the evaporator to be least economical. Plan 1 does not draw power for auxiliaries, while Plans 2 and 3 draw appreciable power. Since an open heater is used in all plans, this equipment is of no importance in the comparison. Therefore, on the basis of the above, we find Plan 1 to be the most economical. It also has the merit of being a simpler design and is simpler to operate. A zeolite softener requires almost no attention and little skill in operation as compared with the evaporator.

TABLE 7

Average Values

No.		Plan 1	Plan 2	Plan 3
1.	Boiler capacity (lbs/hr)	31,399	31,079	30,409
2.	Total feedwater to boiler (lbs/hr)	31,945	31,625	30,955
3.	Generating capacity (kw)	1337.5	1337.5	1337.5
4.	Power used by auxiliaries (kw)	0	109	109
5.	Net generating capacity (kw)	1337.5	1228.5	1228.5
6.	Btu received from boiler per kw hr at switchboard $= \frac{(\text{Item 1})(1307-178)+(546)(437)}{(\text{Item 3})}$	26,700	26,400	25,900
7.	[1] Thermal efficiency of turbine cycle $= \frac{3413\ (\text{Item 5})}{.96 \times .92 \quad (\text{Item 1})(1307)-(\text{Item 2})(178)}$	14.61	13.60	13.91
8.	[2] Btu from coal per kw hr generated $= \frac{(\text{Item 6})}{.80}$	33,400	33,000	32,400
9.	Per cent of power used by auxiliaries	0	8.15	8.15
10.	Btu from coal per net kw hr $= \frac{(\text{Item 8})}{(1-\text{Item 9})}$	33,400	36,000	35,300
11.	Cost of raw water treatment equipment including piping etc. (Plan 3 includes evaporator.)	$750	$750	$3000
12.	Fixed charges at 15% per year ($)	112.5	112.5	450
13.	Evaporator radiation loss (Btu/hr)	0	0	28,700
14.	Heat loss from evap. blow-down (Btu/hr)	0	0	302,000
15.	Total heat loss (Btu/hr)	0	0	330,700
16.	Using New River Coal, given as costing $7.00 per ton			
17.	[3] Cost due to heat loss per year $= \frac{330{,}700 \times 8760 \times 7.00}{14{,}582 \times 2000}$ ($)	0	0	695
18.	[4] Total yearly charge	112.5	112.5	1145

1. Assume average gen. eff. = 92%; average E_m = 96%
2. Assume boiler eff. = 80%
3. Assume heating value of coal (New River Coal) = 14,582 Btu/lb
4. Labor charge assumed same for all plans

4. Boilers

Certain boiler requirements must be met before a selection is made. "The general suitability of a boiler is determined by its design, size, and proportions, which should be such as to permit the element to be forced to the degree necessary to meet not only the usual maximum load, but also emergency demands."* For greater dependability from the plant and continuity of operation, a spare unit should be added. Accessibility of all parts, safety and durability, proper water circulation and finally cost are among the other factors for boiler selection.

Two alternatives are open for the determination of the number of units. The first is to use one unit to handle the total load of the plant, the other is to use one unit for each turbine. With both plans, spare units are necessary. The calculations below show that the first plan is the more economical, together with one spare unit of the same capacity. In order to make these calculations, it is necessary to assume the temperature rise in the economizer (usually 100°F) so that the feedwater is introduced into the boiler at 310° F. This temperature is chosen so that the flue gas temperature will not be below its dew-point—or in other words, cooling must not be below 200° F above dew-point temperature. If this requirement is not met, there is possibility of corrosion of economizer surfaces through condensation of moisture in the gases.

Determination of units and sizes:

Maximum boiler plant capacity = 45,650 lbs/hr

Average boiler plant capacity = 31,399 lbs/hr

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*Barnard, Ellenwood and Hirshfeld, Heat Power Engineering, vol. II, pp. 666-70.

Boiler capacity for 1500 kw turbine at maximum load

= 27,300 lbs/hr

Boiler capacity for 1000 kw turbine at maximum load

= 13,900 lbs/hr

TABLE 8

No.	Items	Plan 1 Unit System		Plan 2 Central System
1.	Number and capacities of units used (lbs/hr)			
	1. For 1500 kw turbine and auxiliaries	1-27,300		1-45,650
	2. For 1000 kw turbine		1-13,900	
	3. Spare unit	1-27,300		1-45,650
2.	Total boiler plant capacity (lbs/hr)	68,500		91,300
3.	Temperature of feedwater entering economizer (°F)	210	210	210
4.	Temperature of water leaving economizer (°F)	310	310	310
5.	Heat supplied by boiler alone = (1204.6 - 279.9)	924.7	924.7	924.7
6.	Boiler output = (Item 1) (Item 5) in (Btu/hr)	25.2×10^6	12.85×10^6	42.2×10^6
7.	Heat transfer rate (Btu/hr/sq ft)	10,000	10,000	10,000
8.	Required heating surface (sq ft)	2520	1285	4220
9.	Maximum per cent rating (%)	300	300	300
10.	Average steam output (lbs/hr)	19,700	7,600	31,399
11.	Annual steam output = (Item 10)(8760) in (Btu/hr)	172.5×10^6	66.5×10^6	275×10^6
12.	Annual coal consumption = $\frac{(\text{Item 11})(1064.1)}{(14{,}582)(2000)(\text{Boiler Eff.}=.80)}$ (tons)	7890	3040	11,800
13.	*Coal used in banking (tons)	0	200	0

*Barnard, Ellenwood and Hirshfeld, op.cit., pp. 605-606.

TABLE 8 (Continued)

No.	Items	Plan 1 Unit System		Plan 2 Central System
14.	Annual coal cost = 7.00 (Item 12 + Item 13)	55,200	22,700	82,600
15.	Annual labor cost ($)	16,000	7,000	16,000
16.	Maintenance at 4% of first cost ($)	15,100		14,000
17.	Total operating cost	116,000		112,600
18.	Cost of boiler plant including building, stoker, piping, etc. ($)	at $150. per kw 377,000		at $140. per kw 350,000
19.	Fixed charge at 15% per year ($)	56,500		52,500
20.	Total annual charge ($)	172,500		165,100

TABLE 9

% Boiler Rating	Flue Gas Temp. °F	Heat Transfer Rate Btu/sq.ft./hr.
100	470	3348
150	505	5030
200	550	6700
250	604	8370
300	680	10040
350	800	11720

Cost estimates were found from pp. 318-329, and the flue gas temperatures from p. 49 of Notes on Power Plants by E. Miller and Holt.

For the dimensions of the boiler, we refer to p. 277 of Power Plant Engineering and Design by F. Morse. We note that Erie City Standard Three-Drum, Type A, Water-Tube Boiler is recommended.

To determine the number of tubes, the above reference gives the following equation:

$$\text{Heating surface} = N\ (14.89L - 28.76)\ \text{sq.ft.},$$

where N is the number of tubes wide and L the length of the "front tube" in feet.

Assume L = 15 ft.

$$4220 = N\ (14.89 \times 15 - 28.76)$$

$$N = \frac{4220}{195.24} = 21.8$$

Use $N = 24$

Actual heating surface = 24 x 195.24 = 4680 sq.ft.

Therefore, the other dimensions are according to the reference:

A = 9' - 10 1/2"

B = 23' - 6 7/8"

C = 12' - 3 3/4"

$D = 13' - 9\ 1/4''$

$E = 12' - 10\ 7/8''$

$$\text{Heat transfer rate} = \frac{\text{Boiler Output}}{\text{Actual Heating Surface}}$$

At maximum flow:

$$\text{Heat transfer rate} = \frac{42.2 \times 10^6}{4680} = 9030 \text{ Btu/hr/sq.ft.}$$

$$\%\text{ rating} = \frac{9030}{3348} = 270\%$$

Flue gas temperature = 630° F

At average flow:

Average boiler output = 31,399 x 924.7 = 28.1 x 10^6 Btu/hr

$$\text{Heat transfer rate} = \frac{28.1 \times 10^6}{4680} = 6010 \text{ Btu/hr/ft}^2$$

$$\%\text{ rating} = \frac{6010}{3348} = 180\%$$

Flue gas temperature = 530° F

5. Combustion Calculation

Orsat analysis:

Use New River Coal, whose analysis is given in Barnard and Ellenwood, vol. II, p. 324. This type of coal is most suitable with a multiple retort underfed stoker.

Assume:

1. Air for combustion enters furnace at 70° F and 50% relative humidity.
2. 40% excess air.
3. 20% combustible left in ash.
4. No CO_2 in the flue gas.

Unburned carbon $= \frac{.2 \times .0385}{.8} = .0096$ lb/lb.

Hence carbon actually burned = .826 - .0096 = .8164 lb/lb.

High heating value of New River Coal = 14,582 Btu/lb.

Moisture in the coal = .0294 lb.

TABLE 10

	Per cent by Weight Dry	Fraction by Weight Dry (#/#)	O_2 Required	Air Required $= \frac{O_2}{.232}$	N_2 $=(Air-O_2)$	CO_2 $=(C+O_2)$	H_2O	SO_2
C	82.60	.8164	2.18	9.40	7.22	2.99		
H	5.04	.0504	.403	1.735	1.332		.454	
O	5.93	.0593						
N	1.46	.0416						
S	1.12	.0112	.011	.048	.037			.022
Ash	3.85	.0385						
Tot.	100.00	.9904	2.594	11.183	8.582	2.99	.454	.022

SO_2 is absorbed in CO_2 and appears as such.

Hence, $CO_2 = 2.99 + .022 = 3.012$ lb/lb of coal.

Necessary chemical equations:

$C + O_2 = CO_2$ or $12 + 32 = 44$

$2C + O_2 = 2CO$ or $24 + 32 = 56$

$2H_2 + O_2 = 2H_2O$ or $4 + 32 = 36$

$S + O_2 = SO_2$ or $32 + 32 = 64$

Of the total O_2 necessary for combustion, .0593 lb/lb of coal is obtained from the coal itself. Therefore, O_2 actually required $= 2.594 - .0593 = 2.535$ lb/lb and consequently the necessary air is $11.183 - \frac{.0593}{.232} = 11.183 - .254 = 10.93$ lb/lb coal.

Therefore,

$N_2 = 8.582 - (.254 - .195) = 8.387$ lb/lb coal

Excess air $= .4 \times 10.931 = 4.389$ lb/lb coal

Excess $O_2 = 4.389 \times .232 = 1.04$ lb/lb coal

Excess $N_2 = 4.389 - 1.04 = 3.349$ lb/lb coal

Total air $= 1.4 \times 10.931 = 15.32$ lb/lb coal

Total $O_2 = 1.04$ lb/lb coal

Total $N_2 = 8.387 + 3.349 = 11.736$ lb/lb coal

Flue gas analysis:

TABLE 11

Products of Combustion	For Perfect Combustion	With 40% Excess Air	Per cent by Weight	Per cent by Weight Dry	Fraction by Volume	Per cent by Volume Dry	Orsat Analysis
CO_2	3.012	3.012	18.55	19.00	.866	13.05	13.0
O_2	0	1.040	6.41	6.60	.413	6.20	6.1
N_2	8.387	11.736	72.40	74.40	5.310	80.80	80.9
H_2O	.454	.454	2.71				
Total	11.853	16.242	100.07	100.00	6.589	100.05	100.0

Moisture in air of 70° F and 50% relative humidity = .00787 lb/lb dry air

lbs. dry air per lb. dry coal = 15.32

Hence, total moisture going into the furnace with the air

= 15.32 x .00787 = .1206 lbs.

Material balance:

Materials entering furnace

(coal + dry air + moisture) = 1 + 15.32 + .1206 + .0294

= <u>16.4700</u> lbs.

Materials leaving furnace

(gas + moisture in air and coal + refuse) = 16.242 + .1206

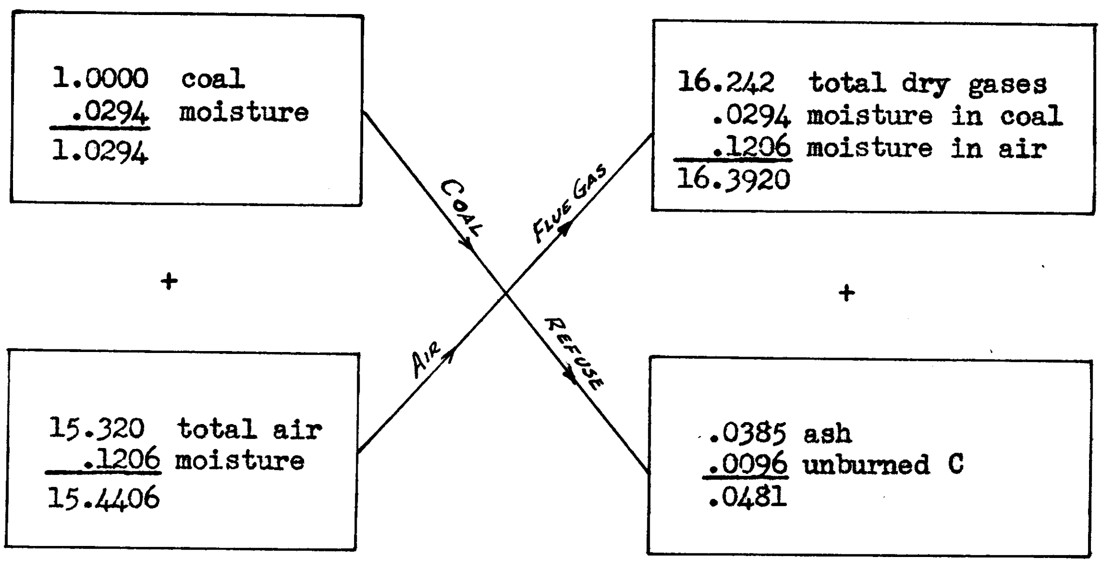
+ .0294 + .0385 + .0096 = <u>16.4401</u> lbs.

Total moisture in flue gas/lb coal = .0294 + .454 + .1206

= .6040 lbs.

TABLE 12

Values per Lb. Coal



= 16.4700 ← check → = 16.4401

Dew Point temperature:

$$\text{Moisture content} = \frac{.6040}{16.3920 - .6040} = \frac{.6040}{15.7880} = .0383 \text{ lb/lb of gases}$$

$$w = .0383 = \frac{.622\ P_s}{P - P_s} \quad ; \quad P_s = \text{saturation pressure}$$

$$P = 14.7 \text{ psi}$$

$$.0383P - .0383\ P_s = .622\ P_s$$

$$P_s = \frac{.0383 \times 14.7}{.6603} = .854 \text{ psi}$$

Hence T_s or dew point temperature = 96.3° F

Net available heat in the coal:

High heating value of coal = 14,582 Btu/lb (given)

Total moisture to be evaporated by coal = .6040 lbs/lb coal

Heat used to evaporate moisture at dew point temperature

$$= W_m \times h_{fg} = .604 \times 1039 = 606 \text{ Btu/lb}$$

Heating value of pure carbon = 14,150 Btu/lb

Heat of carbon in ashpit not available = .0096 x 14,150

= 136 Btu/lb

Net available heat in the coal = 14,582 - 606 - 136

= 13,840 Btu/lb

Mean specific heat of the flue gas:

$$\bar{C}_p = \frac{\sum W\ C_p}{\sum W}$$

* TABLE 13

Gas Constituents	Fraction by weight, lbs.	C_p (60° to 3000°)	W C_p
CO_2	3.012	.274	.825
O_2	1.040	.240	.250
N_2	11.736	.268	3.145
H_2O	.454	.548	.249
Total	16.242		4.469

$$\overline{C}_p = \frac{4.469}{16.242} = .276 \text{ Btu/\#/°F}$$

Theoretical flame temperature:

(Weight of dry gas/lb. of fuel) $\overline{C}_p$ $(t_{flame} - t_{air})$ = Net available heat of coal.

$$(16.242 - .454)\ (.276)\ (t_f - 70°) = 13{,}840$$

$$4.36t_f - 305 - 13{,}840$$

$$t_f = \frac{14145}{4.36} = 3240° \text{ F}$$

Efficiency of boiler and furnace at different loads:

$$\text{Efficiency} = \frac{\text{Btu absorbed by water}}{\text{Btu supplied}}$$

$$\text{Efficiency} = \frac{(\text{Wt. gases/lb.fuel})\overline{C}_p(t_f - t_{flue})(\text{loss for setting})}{\text{Btu supplied}}$$

Loss due to the setting is assumed = 6%

$$\text{Hence Efficiency} = \frac{(16.3920)(.276)(3240 - t_{flue})(1 - .06)}{14{,}582}$$

$$\text{Boiler Efficiency} = 2.92 \times 10^{-4}\ (3240 - t_{flue})$$

* Morse, <u>op.cit.</u>, p. 363.

TABLE 14

% Boiler Rating	*Flue Gas Temp. °F	Boiler Efficiency
100	470	80.9
150	505	79.8
200	550	78.5
250	604	76.9
300	680	74.7
350	800	71.4

Coal consumption:

Assume rise in feedwater temperature through economizer is 100° F; h_{310} = 280 Btu/lb.

For maximum flow

% rating = 270

Hence, boiler efficiency = 76.0%

$$\text{Heat output} = 45{,}650\ (1307-280) + 794\ (437-280)$$
$$= 46.8 \times 10^6 + .1296 \times 10^6 = 46.93 \times 10^6$$

$$\text{Coal consumption} = \frac{46.93 \times 10^6}{14{,}582 \times .760} = 4230 \text{ lbs/hr.}$$

For average flow

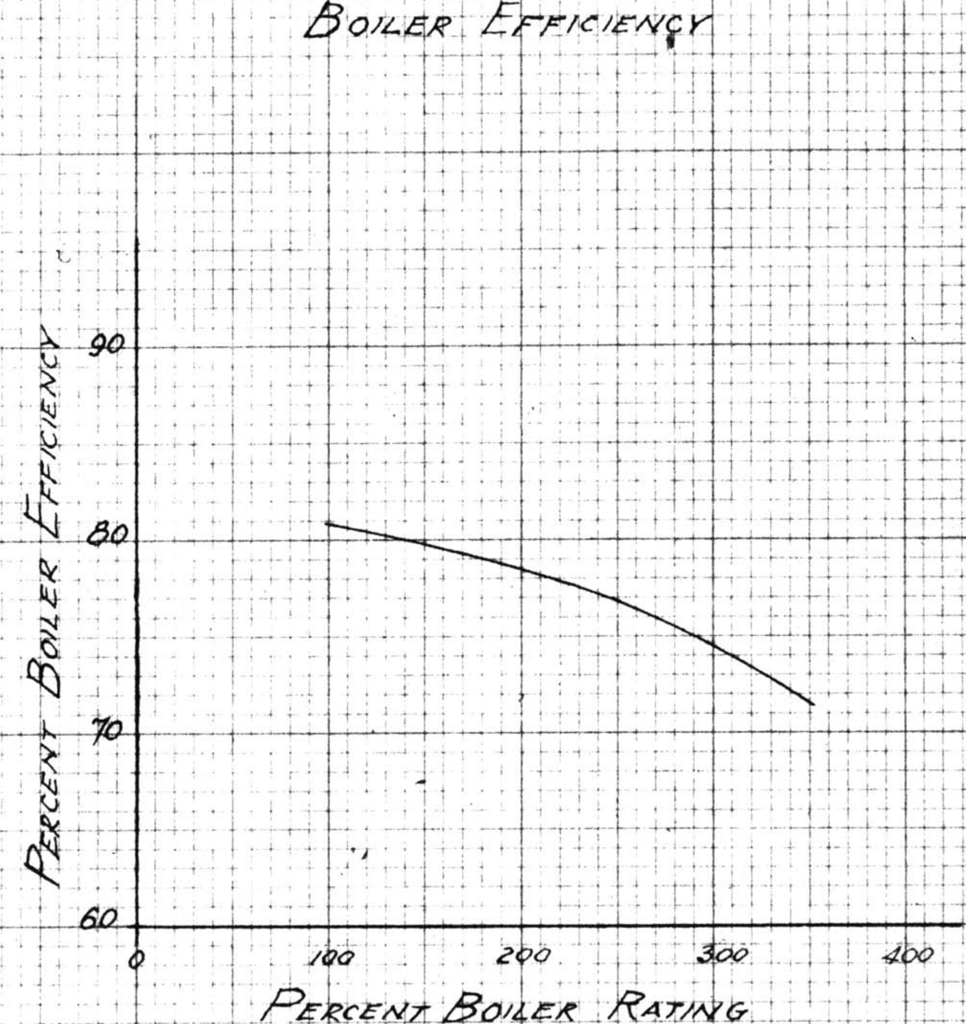
% rating = 180

Hence, boiler efficiency = 79.0%

$$\text{Heat output} = 31{,}399\ (1307-280) + 546\ (437-280)$$
$$= 32.2 \times 10^6 + .0856 \times 10^6 = 32.29 \times 10^6$$

$$\text{Coal consumption} = \frac{32.29 \times 10^6}{14{,}582 \times .790} = 2810 \text{ lbs/hr.}$$

*Miller and Holt, op.cit., p. 49.



BOILER EFFICIENCY
PERCENT BOILER EFFICIENCY
90
80
70
60
0
100
200
300
400
PERCENT BOILER RATING

6. Stokers.

"There is no question about the desirability of mechanical stokers in a plant of 1500 hp. While there may not be any saving in the cost of labor on a plant of 1500 hp, the protection against labor trouble which a stoker affords warrants its use on a plant of this size."* The loads for our proposed plant are quite steady and peaks are of short duration. Since the boiler rating is not over 300%, we see from p. 80 of Notes on Power Plants that an underfeed stoker capable of turning 700 to 800 lbs. of coal per retort per hour continuously is the one most suitable for our plant. We may, therefore, use a $\frac{4230}{700} = 6$ retort Riley self-dumping underfeed stoker.

Hence, furnace width of stoker = 9' - 6 3/4"

Furnace width of boiler = 12' - 3 3/4"

Clearance = 2' - 9"

Standard depth of furnace = 9' - 2"

"Projected" depth of grates = 6' - 8"

Hence, projected grate area = (6'-8") (9'-6 3/4") = 63.9 sq.ft.

For maximum load, lbs.coal/sq.ft./hr. = $\frac{4230}{63.9} = 66.2$

For average load, lbs.coal/sq.ft./hr. = $\frac{2810}{63.9} = 44.0$

The above rates of firing are in keeping with usual values (40 lbs./sq.ft./hr. for average load).

Stoker power**

"The power required to operate underfeed stokers is used: (1) to feed and distribute the coal, (2) to operate the ash dumps or clinker grinders, and (3) to supply the required quantity of air at the necessary

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* Miller and Holt, op.cit., p. 80.

**Barnard, Ellenwood and Hirshfeld, op.cit., vol. II, pp. 504-506.

pressure.... Roughly from 3/4 to 1 hp are installed per retort burning from 700 to 1100 lb. of coal per hour."

The maximum coal burning rate per retort $= \frac{4230}{6} = 706$ lb./hr.

Hence, power required from stoker engine $= 3/4 \times 6 = 4.5$ hp

Also from a curve on page 505, for maximum steam generation of 45,650 lbs. steam/hr., the power of stoker engine is found to be 4.5 hp (check)

Use 5 hp engine

Maximum height of stoker = 5 ft.

7. Furnace Calculations.*

Standard depth of furnace = 9' - 2"

Height of center of mud drum above boiler room floor = 8 ft.

E = 12' - 10 7/8"

Maximum furnace height = 20' - 10 7/8"

Use "effective" height = 15 ft.

Hence, "effective" furnace volume = (15) (63.9) = 958 cu.ft.

Maximum heat liberated = $\frac{14,582 \times 4230}{958}$ = 64,400 Btu/cu.ft./hr.

Average heat liberated = $\frac{14,582 \times 2810}{958}$ = 42,800 Btu/cu.ft./hr.

These values of heat liberation are reasonable.

*Morse, op.cit., p. 277.

COAL AND ASH HANDLING

8. Coal Storage Area*

The cost of fuel storage is very inexpensive insurance against the loss occasioned by complete cessation of a manufacturing process, even for a short time, caused by lack of fuel.

Assume storage capacity = 10% of annual consumption

Approximate weight of coal = 50 lb./cu.ft.

$$\text{Volume of coal storage} = \frac{2810 \times .1 \times 8760}{50} = 49{,}300 \text{ cu.ft.}$$

$$\text{Total weight of storage} = \frac{2810 \times 8760 \times .1}{2000} = 1230 \text{ tons}$$

Assume storage depth = 8 ft.

$$\text{Hence, storage area} = \frac{49{,}300}{8 \times 43{,}560} = .1415 \text{ acre}$$

* Morse, op.cit., p. 370.

9. Coal Bunker.*

The suspension bunker, designed with a cross-section such that tension is the only stress produced in the envelope, is a very economical type, since stiffeners are required only on end or interior bulk-heads and on the girders which support the bag bottom.

Length of each boiler is approximately 14 ft. so that allowing 10 ft. between each and some for walls, bunker length may be taken = 40 ft. along the whole length of the boiler room. Design of bunker is for three day storage.

3 days' storage at average condition = 3 x 24 x 2810 = 202,000 lbs.coal

Volume required at 50 lb.coal per cu.ft. = $\frac{202{,}000}{50}$ = 4050 cu.ft.

Cross-section area required = $\frac{4050}{40}$ = 101 sq.ft.

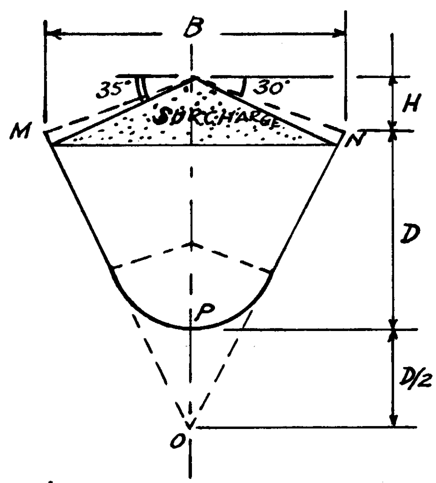


Fig. 12

If B x D = 14'x 11' (no surcharge)

= 154 sq.ft.

For no surcharge, bunker capacity,

$$C = \frac{5}{8} \times 14 \times 12 = 105 \text{ sq.ft.}$$

or cu.ft. per foot length.

Thus, no surcharge is required.

Hence, use Berquist suspension bunker with:

Cross-section area = 105 sq.ft.

B = 14 ft.

D = 11 ft.

H = 0

*Morse, op.cit., pp. 370-377.

10. Coal Conveyor.*

Use "rectangular" Peck pivot-bucket conveyor. Coal discharged from a car or from a cart falls into a crusher where the large lumps are broken up. From the crusher, the coal is taken directly into the conveyor or into the feeding mechanism which fills the conveyor. The carrier discharges the coal directly into the overhead bunker.

Hence, Speed of the conveyor . . . . 40 ft.per min.

Pitch of chain . . . . 24 in.

Average coal consumption . . . . 1.41 tons/hr.

Maximum coal consumption . . . . 2.12 tons/hr.

Assume capacity of conveyor . . . . . . 30 tons/hr.

Hence, Bucket size is approximately 18" x 18"

Approximate elevation . . . . 80 ft.

Power required = 0.000085 x tons/hr. x speed in fpm x elevation in ft.

Power required = .000085 x 30 x 40 x 80 . 8.17 hp.

Use 10 hp drive

Crusher capacity . . . . 30 tons/hr.

Three day storage in bunker = 202,000 lb. coal = 101 tons.

Time required to fill 3 day supply of coal in bunker = $\frac{101}{30}$ = 3.37 hours or 3 1/2 hours approximately.

A 30 ton/hr. coal crusher requires a floor space of 7' x 4'- 6" and height of 3 feet overall when set on a cast iron base. It requires 5 hp. to drive it.

Ash Handling.

Use small ash car delivering to an outside storage. In selecting the hopper, assume capacity such as will handle the ashes accumulated

*Miller and Holt, op.cit., pp. 253-272.

during 16 hours of operation at maximum load.

Maximum weight of coal burned = 4230 lbs./hr.

Hence, maximum weight of ash = (.0385 + .0096) 4230 = 203 lbs./hr.

Weight of ash per cu.ft. = 40 lbs.

Capacity of hopper $= \frac{203 \times 16}{40} = 81.4$ cu.ft.

Use Beaumont Copper-Steel Ash Hopper, 6'- 6" x 11'- 0".

11. Economizers.

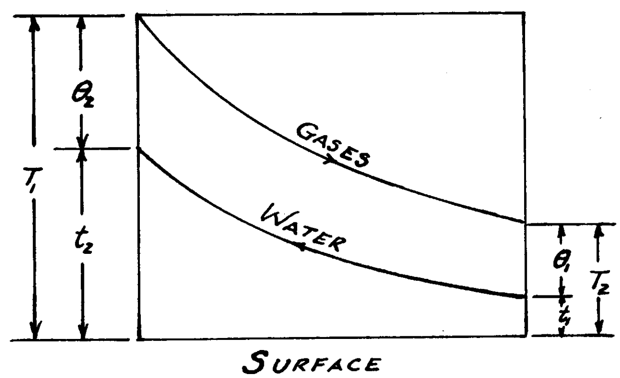
To find the required heating surface

Assume 100° F temperature rise of the feedwater through the economizer. This gives the temperature of the water leaving as 310° F, chosen so that the flue gas temperature will not be below its dew point—or in other words, cooling must not be below 200° F above dew point temperature, protecting economizer surfaces from corrosion.

Weight of flue gas flowing = 16.392 lbs. per lb. of coal

Maximum weight of flue gas = 16.392 x 4230 = 69,300 lbs./hr.

Average weight of flue gas = 16.392 x 2810 = 46,000 lbs./hr.



COUNTER FLOW

Fig. 13

$P = \dfrac{W_w\ C_w}{W_g\ C_g}$, where W_w, C_w are the weight and the specific heat of water; W_g and C_g are the weight and specific heat of the flue gases.

$$(t_2 - t_1)\ P = (T_1 - T_2)$$

$\log_{10} \dfrac{T_1 - t_2}{T_2 - t_1} = \dfrac{SU\ (P - 1)}{W_w\ C_w \times 2.303} = n$, where S is the surface, ft.^{2} and U is the heat transfer coefficient.

Assume minimum flue gas temperature leaving (T_2) at <u>average</u> load is 325°F (dew-point) consideration).

For maximum flow:

$$P = \frac{46,444 \times 1}{69,300 \times .276} = 2.43$$

From experimental data for a cast iron economizer, we have

$U = 5.313$ Btu./hr./sq.ft./°F for a gas flow of 69,300 lbs./hr.

$$T_2 = T_1 - P\ (t_2 - t_1) = 630 - 2.43\ (310 - 210) = 387$$

$$10^n = \frac{T_1 - t_2}{T_2 - t_1} = \frac{630 - 310}{387 - 210} = 1.81$$

$$n = \frac{SU\ (P - 1)}{W_w\ C_w \times 2.303} = \frac{5.313\ S\ (2.43 - 1)}{46,444 \times 1 \times 2.303} = 7.09 \times 10^{-5}S$$

$$\log_{10} 1.81 = 7.09 \times 10^{-5}S$$

$$.258 = 7.09 \times 10^{-5}S$$

$$S = \frac{.258}{7.09 \times 10^{-5}} = 3640 \text{ sq.ft.}$$

Use two economizers, one spare.

For average flow:

$$P = \frac{31,945 \times 1}{46,000 \times .276} = 2.52$$

For $W_g = 46,000$ lbs./hr., $U = 4.089$ Btu./hr./sq.ft./°F

$$n = \frac{4.089 \times 3640\ (252 - 1)}{31,945 \times 1 \times 2.303} = .307$$

Assume $T_2 = 325°$ F $\quad$ ($t_2 < 310°$ F; $T_2 > 300°$ F)

or $T_2 = t_1 + \frac{T_2 - T_2}{P} = 210 + \frac{530 - 325}{2.52} = 291.4°$ F

Checking for T_2, using S = 3640 sq.ft.,

$$\frac{T_1 - t_2}{T_2 - t_1} = 10^n = 10^{.307} = 2.02$$

$$\frac{530 - 291.4}{T_2 - 210} = 2.02$$

$$T_2 = \frac{238.4 + 2.02 \times 210}{2.02} = 327° \text{ F}$$

This value checks closely with assumed value.

Size of the economizer:

Since furnace width of boiler is 12' - 3 3/4", economizer tube length can be taken as 12 feet. Use standard Foster Wheeler economizer, 8 tubes wide, with 3 sq. ft. surface per linear foot of tube.

Hence, number of rows $= \dfrac{3640}{3 \times 8 \times 12} = 12.65$ or 13

The height of the economizer is approximately 12 ft., which is within other furnace and boiler dimensions.

Draft loss through economizer:

Test data is shown in Table 15.

w = gas flow in lbs./hr.

W = gas flow in lbs./hr. per sq.ft. of free area

L = draft loss in inches of water

TABLE 15

w	W	L
50,000	2400	.52
75,000	3600	1.11
100,000	4800	1.74
125,000	6000	2.45
150,000	7000	3.32

From test data:

For w = 69,300 lbs./hr., W = 3300 lbs./hr./sq.ft.

For w = 46,000 lbs./hr., W = 2130 lbs./hr./sq.ft.

The corresponding draft losses are:

Maximum load, L = .93 in. water

Average load, L = .46 in. water

The above values can also be obtained from an empirical equation:

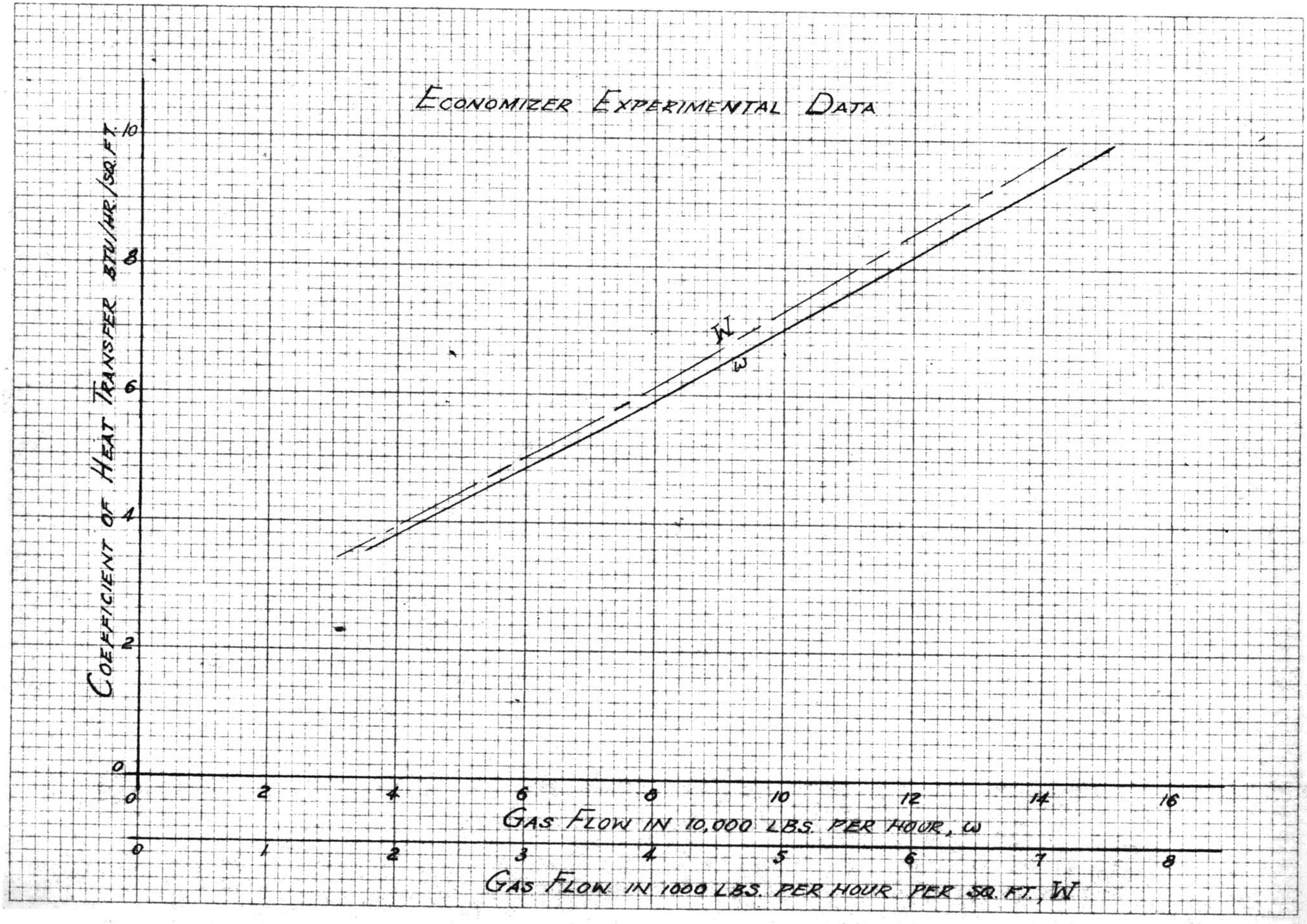
$$L = \frac{5.55\ W^2}{10^8}$$

At maximum load, $L = \frac{5.55\ (3300)^2}{10^8} = .596$ in. water

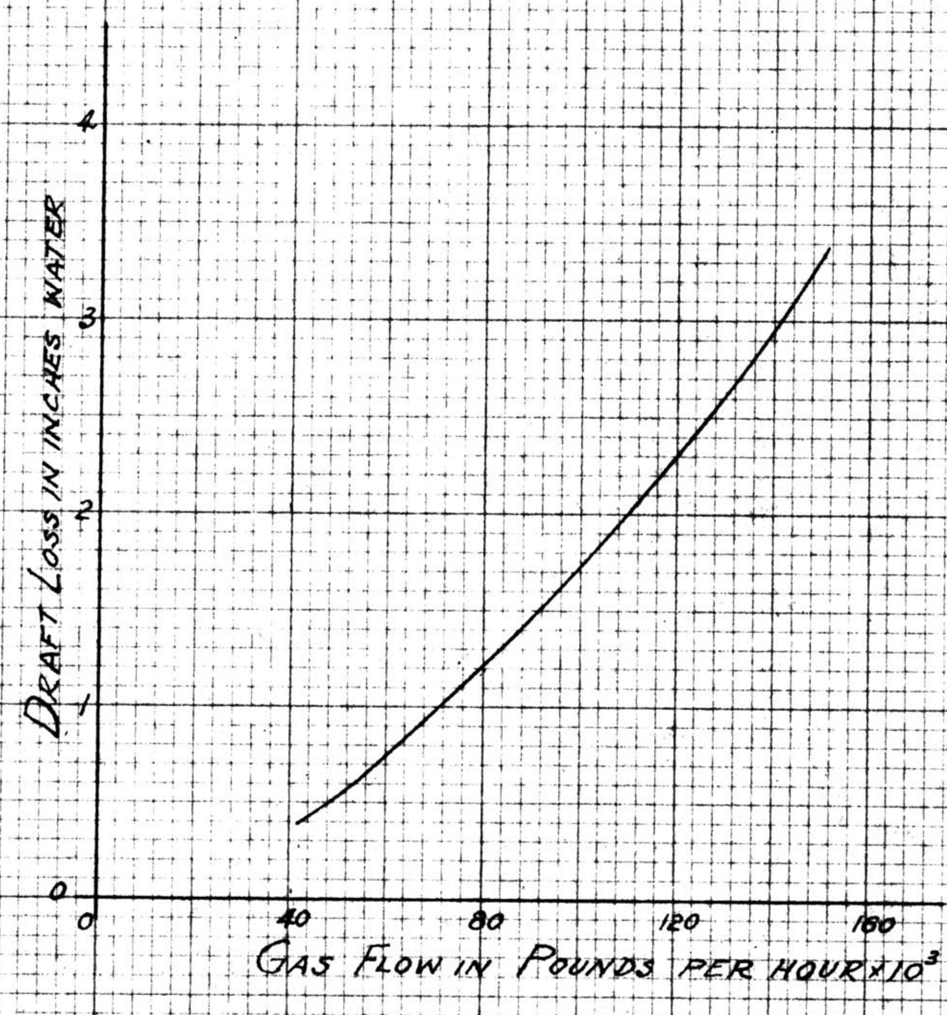
At average load, $L = \frac{5.55\ (2130)^2}{10^8} = .251$ in. water

The test values are the ones to use.

$$\left.\begin{array}{l} \text{Free area} = \dfrac{69,300}{3300} = 21 \text{ sq.ft.} \\ \quad \text{or} \\ \text{Free area} = \dfrac{46,000}{2130} = 21.5 \text{ sq.ft.} \end{array}\right\} \ 21 \text{ sq.ft.}$$



ECONOMIZER EXPERIMENTAL DATA
COEFFICIENT OF HEAT TRANSFER BTU/HR./SQ.FT.
0
2
4
6
8
10
W
w
0 2 4 6 8 10 12 14 16
GAS FLOW IN 10,000 LBS. PER HOUR, w
0 1 2 3 4 5 6 7 8
GAS FLOW IN 1000 LBS. PER HOUR PER SQ. FT., W



ECONOMIZER DRAFT LOSS

12. The Draft System

We will use a balanced draft system in which forced draft is used in combination with natural or induced draft in such a manner that the pressure in the combustion space of the furnace is nominally atmospheric but actually slightly negative in order to keep a small current of air passing into the furnace from the room. This will prevent overheating the stoker fronts and doors, also dangerous gases or flames from blowing into the boiler room. If the furnace pressure is much less than atmospheric, stack loss may become too high for good economy.

We will maintain a balanced draft of 0.1 in. water vacuum over the fuel bed. For emergency, assume 60% excess air as basis of design and one draft plant for each boiler.

Forced draft fans:

Use two, one connected to each boiler, and both to the main air duct.

Actual air required (40% excess air) = 10.931 x 1.4 = 15.32 lbs./lb.coal.

Design air required (60% excess air) = 10.931 x 1.6 = 17.50 lbs./lb.coal.

Air duct:

Assume length of duct = 40 ft.

Density of air at 70° F = .0752 lb./cu.ft.

$$\text{Maximum capacity} = \frac{17.500 \times 4230}{60 \times .0752} = 16{,}400 \text{ cfm.}$$

Velocities in air ducts range from 20 to 50 ft./sec.

For maximum load, firing rate of 66 lb./sq.ft. grate area/hr., (from test data curves) we find draft loss through fuel bed = 5.75 in.

water, equivalent to 3050 ft./min.*

Specific volume is calculated by the approximate pressure in the duct.

$$\text{Specific volume} = \frac{53.4\ (460 + 70)}{14.7 \times 144} = 13.4 \text{ cu.ft./lb}$$

$$\text{Required duct area} = \frac{13.4 \times 4230 \times 17.50}{60 \times 3050} = 5.42 \text{ sq.ft.}$$

Duct dimensions selected are 35" x 23" = 5.59 sq.ft.

$$\text{Velocity head} = \frac{\left(\frac{3050}{60}\right)^2}{64.4} = 40.0 \text{ ft.}$$

$$\text{Specific gravity of air} = \frac{1}{13.4 \times 62.5} = .00119$$

Velocity head = 40.0 x .0119 x 12 = .571 inches water.

The friction draft loss of gas flowing through a conduit is given by the equation:

$$D = \frac{fV^2H}{64.4R} \text{ ft. of air;}$$

or $D' = 12bD$ inches water

where H = duct height, ft.

R = hydraulic radius of cross-section, ft.

V = air velocity, ft./sec.

b = specific gravity of air referred to water

f = coefficient of friction

For air, f = .005, multiplied by 1.15 for rectangular duct.

$$D = 1.15 \times \frac{.005\ \left(\frac{3050}{60}\right)^2 \times 40}{64.4 \times \frac{35 \times 23}{12(70+46)}} = 15.9 \text{ ft.air}$$

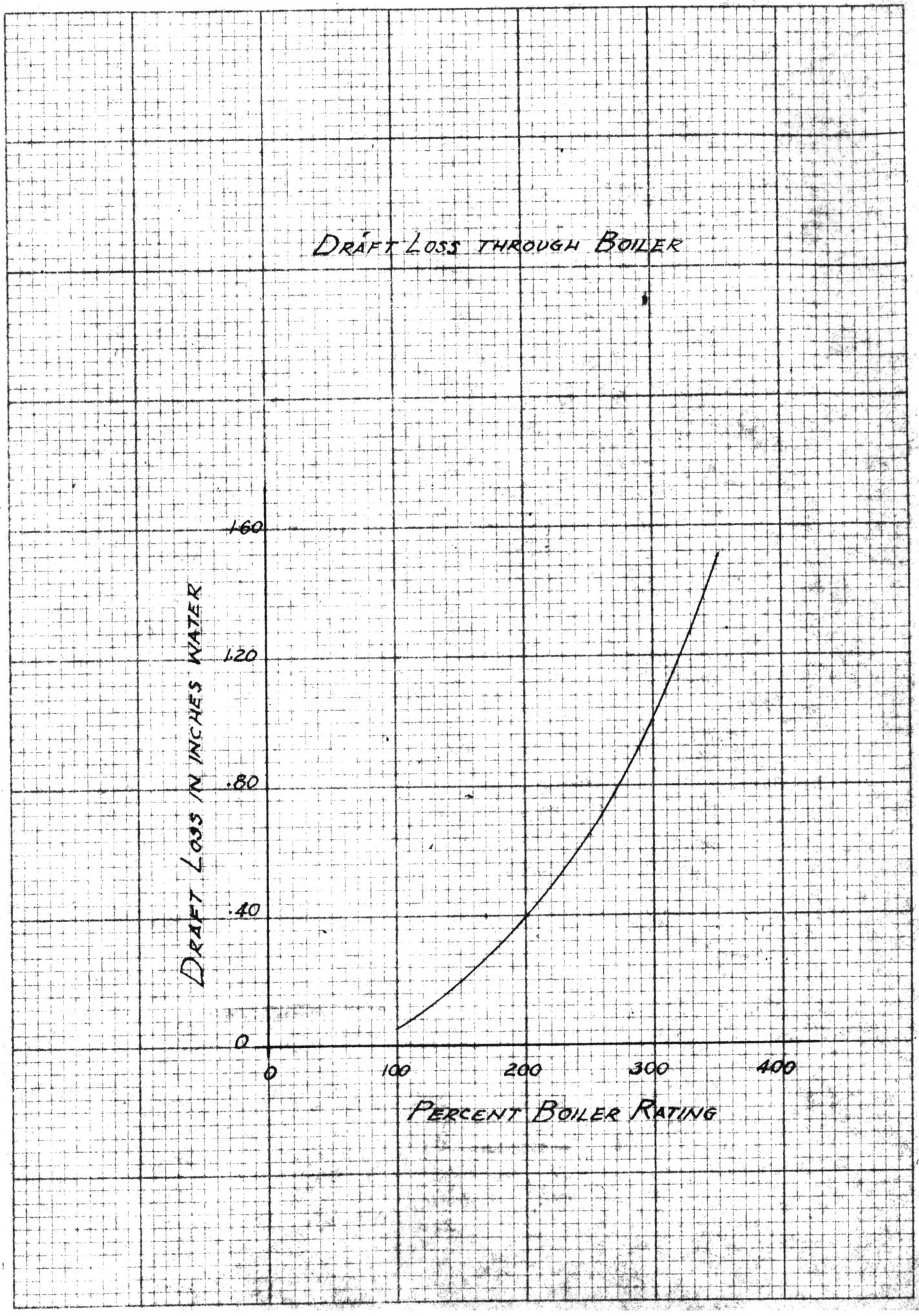
Friction loss = 12 x .00119 x 15.9 = .208 in. water

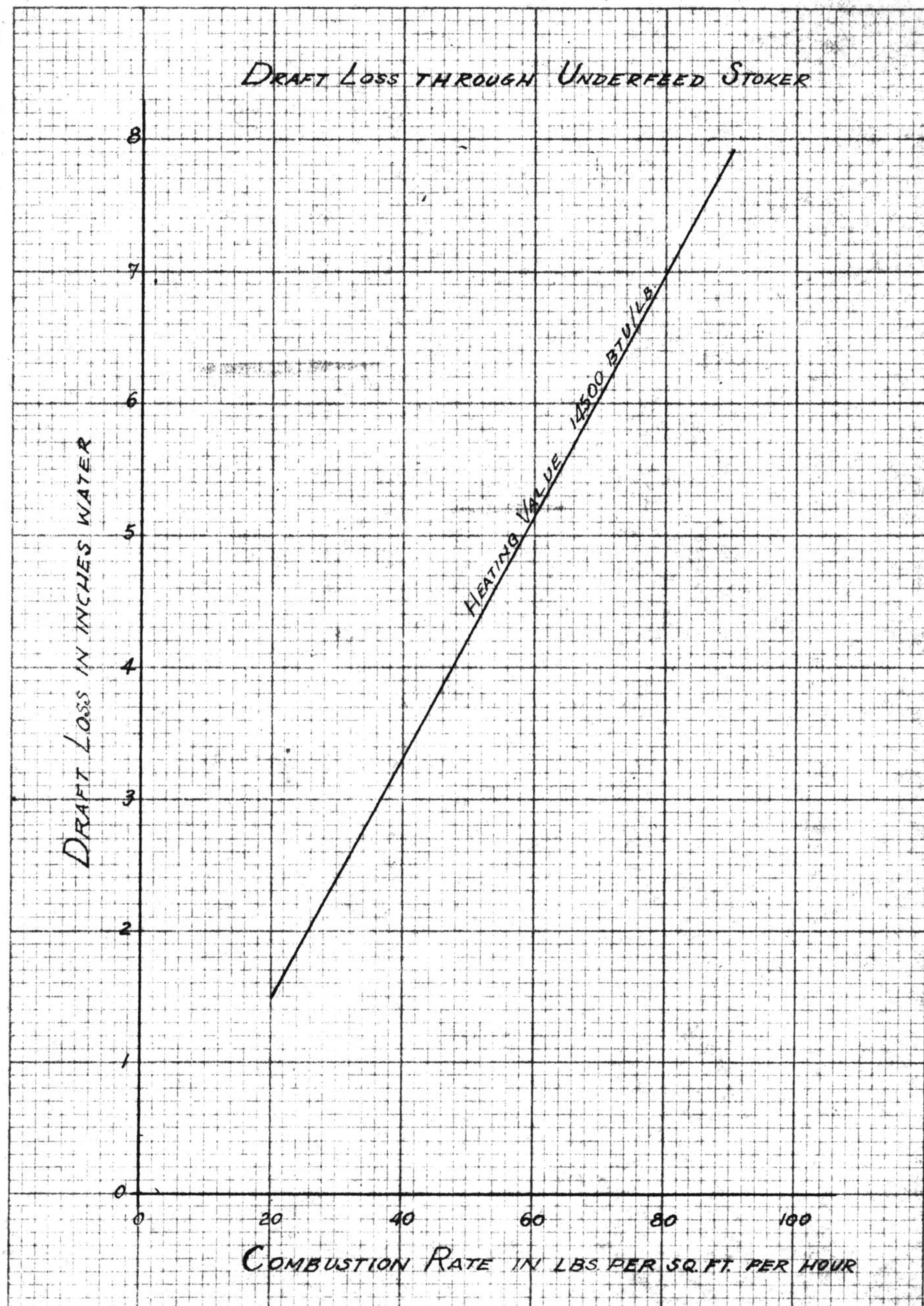
Loss in 3 bends at .05" per bend = .15 in. water

Total draft = 5.75 + .571 + .208 + .15 = 6.679 in. water

Total draft loss = 6.679 - 5.75 = .93 in. water

* Morse, op.cit., p. 429


DRAFT LOSS THROUGH BOILER
DRAFT LOSS IN INCHES WATER
1.60
1.20
.80
.40
0
0
100
200
300
400
PERCENT BOILER RATING



DRAFT LOSS THROUGH UNDERFEED STOKER
DRAFT LOSS IN INCHES WATER
0
1
2
3
4
5
6
7
8
0
20
40
60
80
100
COMBUSTION RATE IN LBS PER SQ. FT. PER HOUR
HEATING VALUE 14500 BTU/LB.

TABLE 16

Item	Max. Flow 40% XS Air	Ave. Flow 60% XS Air	Ave. Flow 40% XS Air
Capacity (cfm)	13,650	10,900	9,540
Velocity (ft/min)	*2,440	1,950	1,700
Velocity head ("H_2O)	.256	.1650	.125
Friction loss in duct ("H_2O)	.1460	.0937	.0714
Friction loss in bends ("H_2O)	.15	.15	.15
Total draft loss in duct ("H_2O)	.552	.4087	.3464

*Agreement with 3050 fpm is close enough, divergence being due to 60% XS.

TABLE 17

	Max. Flow 60% XS Air	Max. Flow 40% XS Air	Ave. Flow 60% XS Air	Ave. Flow 40% XS Air
Coal burned (lbs/hr)	4,230	4,230	2,810	2,810
Capacity of fan (cfm)	16,400	13,650	10,900	9,540
Static pressure ("H_2O)	5.75	5.75	3.90	3.90
Balanced draft in furnace ("H_2O)	-.10	-.10	-.10	-.10
Total friction loss in duct ("H_2O)	.358	.296	.2437	.2214
Total static pressure ("H_2O)	6.008	5.946	4.0437	4.0214
Tip speed (fpm)	13,900	12,600	10,300	9,900
Rpm = tip speed x .159	2210	2000	1640	1570
Static efficiency of fan (%)	60	63.5	63.5	64
Required hp = $\frac{.000158\ (\text{Volume})(\text{Static Press.})}{\text{Static Efficiency}}$	25.9	20.2	11.0	9.46

Use Design 4 Turbovane fan No. 70 with 30 hp drive.

Before selecting the induced draft fans, we must determine the volume of flue gas that must be handled by the fans, the actual design again being based on 60% excess air.

TABLE 18

Item	CO_2	O_2	N_2	H_2O	Total
Wt. of constituents per lb. coal at 40% XS air (lb)	3.012	1.040	11.736	.604	16.440
Wt. of constituents per lb. coal at 60% XS air (lb)	3.012	1.52	13.407	.6204	18.559
Density at 68° F (lb/cu ft)	.114	.083	.073		
Density at 387° F (lb/cu ft)	.071	.052	.046		
Density at 325° F (lb/cu ft)	.077	.056	.049		
Volume in cu ft per lb coal at 387° F and 40% XS air	42.4	20	255	20.7	338.1
Volume in cu ft per lb coal at 387° F and 60% XS air	42.4	29.2	292	21.5	385.1
Volume in cu ft per lb coal at 325° F and 40% XS air	39.1	18.6	240	19.3	317.0
Volume in cu ft per lb coal at 325° F and 60% XS air	39.1	27.2	274	19.8	360.1

Induced Draft Fans:

The Cindervane fan is a special design straight blade fan designed not only to furnish induced draft but to act as a cinder catcher as well. Design is based on maximum flow and 60% excess air.

TABLE 19

	Maximum Flow		Average Flow	
	60% XS Air	40% XS Air	60% XS Air	40% XS Air
Coal burned (lbs/hr)	4230	4230	2810	2810
Volume of flue gas (cu ft/lb)	385.1	338.1	360.1	317.0
Capacity of fan (cfm)	27,200	23,800	16,900	14,850
Balanced draft in furnace ("H_2O)	.10	.10	.10	.10
*Draft loss in boiler (")	1.30	.79	.403	.31
*Draft loss in economizer (")	1.15	.93	.60	.46
*Draft loss in breeching (")	.05	.05	.05	.05
Draft loss for 3 bends at .05" H_2O per bend (")	.15	.15	.15	.15
Draft created by chimney (")	-.2	-.2	-.2	-.2
Total static pressure at running condition or high temperature	2.55	1.82	1.103	.87
Weight of flue gas (lbs/hr)	78,500	69,300	52,400	46,000
Density of flue gas (lbs/cu ft)	.0481	.0485	.0516	.0516
Temperature of flue gas (°F)	387	387	325	325

*Boiler draft losses were obtained from test data, while those for 60% excess air were assumed to be proportional to the square of the velocities. The losses in the economizer were also obtained from manufacturers' test data. As for the breeching, it was assumed that the losses were 0.001 in. water per ft. and the length 50 feet.

TABLE 19 (Continued)

	Maximum Flow		Average Flow	
	60% XS Air	40% XS Air	60% XS Air	40% XS Air
Equivalent static pressure at 65° F ("H_2O)	4.16	2.94	1.65	1.31
Static efficiency of fan (%)	57	57	57	58
Use 7 ft. Cindervane type fan K = .0442				
Tip speed (fpm)	8000	6700	5100	4500
Rpm = (16) x (K)	353	294	225	199
Hp required = .000158(Volume)(Static Press. at 65°F)/ Static Efficiency)	31.0	19.4	7.75	5.40
Use 35 hp drive				

13. Chimney and Breeching.*

Chimney proportions should be such as will meet the draft and capacity requirements with least cost. It is provided mainly to develop available draft at the entrance of the flue gases. Of course, the draft must also take care of the chimney frictional losses, which are proportional to the square of the velocity which is itself proportional to the cross-section area for a given mass flow. It has been found that a combination whose diameter multiplied by its height was the least of all workable combinations would be the most economical. Velocities usually range between 20 - 50 feet per second. Use superimposed steel stacks.

Draft at bottom of stack = .2" at maximum load.

Maximum volume of gas through stack = 27,200 cfm.

Corresponding gas temperature = 387° F.

Outside air at 70° F and 29.92" Hg with density = .075 lb./cu.ft.

Flue gas density = .0481 lb./cu.ft.

Gas specific gravity = $\frac{.0481}{62.4}$ = .00077

If D = effective draft per 100 ft. of stack

K = 15.4 for steel stacks

d_a = density of air, lb./cu.ft.

d_f = density of flue gas, lb./cu.ft.

V = gas velocity in the stack, fps.

F = gas flow, cfs.

D_s = stack diameter, ft.

Then Morse gives on page 433* the following equation for estimating the draft through a chimney or stack:

---

* Morse, op.cit., pp. 433-437.

$$D = K\ (d_a - d_f) - .0148\ d_f \sqrt{\frac{V^5}{F}}$$, based on the assumption that the friction factor is .0148.

Therefore, $F = \frac{27,200}{60} = 454$ cfs

$$D = 15.4\ (.075 - .0481) - .0148 \times .0481 \sqrt{\frac{V^5}{454}}$$

or $D = .399 - .0000333\ V^{2.5}$ .... Eq. (1)

Since, $$V = \frac{27,200}{\frac{\pi\ D_s^2}{4} \times 60} = \frac{577}{D_s^2}$$

$D_s = 24.0/\sqrt{V}$ .... Eq. (2)

Velocity head loss $= \frac{V^2}{64.4} \times .00077 \times 12 = .000143\ V^2$ in. water

Required effective draft of stack = loss in the flues and bends + velocity head loss + draft at bottom of stack + draft loss in breeching.

Hence, required effective draft $= 3 \times .05 + .000143\ V^2 + .2 + 50 \times .001$.

Effective draft $= .40 + .000143\ V^2$ .... Eq. (3)

Now, by selecting various values of V, the corresponding chimney heights and diameters are quickly and easily computed.

TABLE 20

Velocity of gas trial, V, fps.	20	25	30	35	15
$V^{.5}$	4.48	5.00	5.49	5.92	3.88
V^2	400	625	900	1225	225
$V^{2.5}$	1785	3135	4944	7279	870
D, effective draft per 100 ft.	.340	.295	.235	.157	.370
Required effective draft	.457	.489	.528	.575	.433
Height, H, ft.	134	165	224	366	118
Diameter, D_s, ft.	5.45	4.80	4.38	4.06	6.20
H x D_s	730	790	982	1480	733

From the above it is seen that of the velocities selected, 20 fps results in the minimum product of height by diameter. This value is within practice range of velocities. The implication is that the chimney dimensions thus indicated are the most economic for the job and should be selected, provided the resulting structure is stable (depending on taper of chimney and wind pressure). Hence, select chimney height of 134 feet, with an internal diamter at the bottom of the chimney equal to 5'-5.5" and 20 fps velocity. As a result of the shrinkage in volume of the gases while traveling from bottom to top, the top diameter may be smaller.

The performance of the stack at different flow conditions is indicated below.

TABLE 21

Item	Maximum Flow 60% XS	Maximum Flow 40% XS	Average Flow 60% XS	Average Flow 40% XS
Gas flow, F (cfs)	454	396	282	248
Velocity, V (fps)	19.55	17.1	12.2	10.7
Density of gas (lb/cu.ft.)	.0481	.0485	.0516	.0516
Effective draft (in.H_2O)	.455	.477	.336	.342
Velocity head loss (in.H_2O)	.057	.042	.021	.016
Total draft (in.H_2O)	.398	.435	.315	.326

Breeching:

The purpose of the breeching, which is a duct in effect, is to direct the flue gases from the economizer to the stack. Here again, the velocity should range between 20 and 50 ft./sec., but the same as that in the economizer.

Economizer free area = 21.0 sq.ft.

Maximum mass rate at 60% excess air $= \frac{18.559 \times 4230}{21} = 3770$ lb./hr. per sq.ft. free area.

Density of the flue gas at maximum load = .0481 lb./cu.ft.

Hence, gas velocity $= \frac{3770}{.0481 \times 3600} = 21.8$ ft./sec., which is good enough.

Maximum gas flow at 60% XS air = 454 cfs.

Area of breeching $= \frac{454}{21.8} = 20.8$ sq.ft.

Assume breeching at top is 15% greater.

Area at outlet = 20.8 x 1.15 = 24 sq.ft.

Assuming the ratio of height to width is 2, then the width of a rectangular breeching $= \sqrt{\frac{24}{2}} = 3.47$ ft.

Hence, use 3' - 6" x 7' - 0".

14. Piping.

Morse gives on page 531 a table of average practice in flow velocities for the determination of pipe sizes.

TABLE 22

Average Practice in Flow Velocities, fpm	
Water	300 - 600
High pressure sat. steam	5,000 - 10,000
High pressure sup. steam	10,000 - 15,000
Atmospheric exhaust steam	8,000 - 12,000
Low pressure exhaust steam	20,000 - 24,000

Steam line from superheater outlet to main header:

Specific volume at 450 psia and 608° F is 1.314 cu.ft./lb.

Assume steam velocity = 12,000 fpm.

Maximum steam flow = 45,650 lbs./hr.

$$\text{Pipe cross-section} = \frac{45{,}650 \times 1.314 \times 144}{60 \times 12{,}000} = 12 \text{ sq.in.}$$

$$\text{Diameter} = \sqrt{\frac{12}{.785}} = 3.92''$$

Use D = 4.00"

Steam line from main header to 1500 kw turbine:

Specific volume = 1.314 cu.ft./lb.

Steam velocity = 12,000 fpm.

Maximum steam flow = 27,300 lbs./hr.

$$\text{Pipe cross-section} = \frac{27{,}300 \times 1.314 \times 144}{60 \times 12{,}000} = 7.16 \text{ sq.in.}$$

$$\text{Diameter} = \sqrt{\frac{7.16}{.785}} = 3.03''$$

Use D = 3 1/4"

Steam line from main header to 1000 kw turbine:

Specific volume = 1.314 cu.ft./lb.

Steam velocity = 12,000 fpm.

Maximum steam flow = 13,900 lbs./hr.

$$\text{Pipe cross-section} = \frac{13,900 \times 1.314 \times 144}{60 \times 12,000} = 3.64 \text{ sq.in.}$$

$$\text{Diameter} = \sqrt{\frac{3.64}{.785}} = 2.13"$$

Use D = 2 1/2"

Steam line to auxiliaries:

Specific volume = 1.314 cu.ft./lb.

Steam velocity = 12,000 fpm.

Maximum steam flow = 4760 lbs./hr.

$$\text{Pipe cross-section} = \frac{4760 \times 1.314 \times 144}{60 \times 12,000} = 1.25 \text{ sq.in.}$$

$$\text{Diameter} = \sqrt{\frac{1.25}{.785}} = 1.26"$$

Use D = 1 1/2"

Line for bled steam:

Specific volume at 40 psia sat. = 10.5 cu.ft./lb.

Steam velocity = 12,000 fpm.

Maximum steam flow = 20,000 lbs./hr.

$$\text{Pipe cross-section} = \frac{20,000 \times 10.5 \times 144}{60 \times 12,000} = 42 \text{ sq.in.}$$

$$\text{Diameter} = \sqrt{\frac{42}{.785}} = 7.31"$$

Use D = 7 1/2"

Exhaust steam line from auxiliaries to open heater:

Quality at 17 psia and h = 1080 = .924

Specific volume = 23.4 x .924 = 21.6 cu.ft./lb.

Assume steam velocity = 10,000 fpm

Maximum steam flow = 4259 lbs./hr.

Pipe section $= \frac{4259 \times 21.6 \times 144}{60 \times 10{,}000} = 22.1$ sq.in.

Diameter $= \sqrt{\frac{22.1}{.785}} = 5.31''$

Use D = 5 1/2"

Exhaust steam line from auxiliaries to atmosphere:

Specific volume = 21.6 cu.ft./lb.

Steam velocity = 10,000 fpm.

Maximum steam flow = 501 lbs./hr.

Pipe section $= \frac{501 \times 21.6 \times 144}{60 \times 10{,}000} = 2.6$ sq.in.

Diameter $= \sqrt{\frac{2.6}{1785}} = 1.82$ in.

Use D = 2"

From condenser calculations:

<u>No. 8 Leblanc Jet Condenser</u>

Condensate pipe from 1500 kw turbine, D = 7"

Air pump suction, D = 6"

Air pump discharge, D = 5"

Injection pipe, D = 9"

<u>No. 7 Leblanc Jet Condenser</u>

Condensate pipe from 1000 kw turbine, D = 7"

Air pump suction, D = 6"

Air pump discharge, D = 4"

Injection pipe, D = 9"

Feed water from hot well:

Maximum water flow = 22,597 lbs./hr.

Assume velocity of flow = 400 ft./min.

$$\text{Pipe section} = \frac{22{,}597 \times 144}{62.5 \times 60 \times 400} = 2.17 \text{ sq.in.}$$

$$\text{Diameter} = \sqrt{\frac{2.17}{.785}} = 1.66''$$

Use D = 2"

Manufacturing condensate line:

Maximum flow = 18,000 lbs./hr.

$$\text{Pipe section} = \frac{18{,}000 \times 144}{62.5 \times 60 \times 400} = 1.73 \text{ sq.in.}$$

$$\text{Diameter} = \sqrt{\frac{1.73}{.785}} = 1.48''$$

Use D = 1 1/2"

Total feed entering open heater:

Maximum flow = 40,597 lbs./hr.

$$\text{Pipe section} = \frac{40{,}597 \times 144}{62.5 \times 60 \times 400} = 3.9 \text{ sq.in.}$$

$$\text{Diameter} = \sqrt{\frac{3.9}{.785}} = 2.23''$$

Use D = 2 1/2"

Feed water line leaving open heater:

Maximum flow = 46,444 lbs./hr.

$$\text{Pipe section} = \frac{46{,}444 \times 144}{62.5 \times 60 \times 400} = 4.45 \text{ sq.in.}$$

$$\text{Diameter} = \sqrt{\frac{4.45}{.785}} = 2.38''$$

Use D = 2 1/2"

Main steam header:

Use D = 4.00" (superheater outlet)

15. Pumps.

Boiler feed pumps:*

TABLE 23

Item	Max.Flow	Ave.Flow
Flow (lbs/hr)	46,444	31,945
Capacity (gpm)	92.8	63.8
Feed water temperature (°F)	210	210
Boiler pressure (psi)	450	450
Boiler head at 60° F equivalent temp. = 2.4 x 450 (ft)	1080	1080
Static head assumed (ft)	30	30
Velocity and friction head (ft)	10	10
Pump operating head (ft)	1120	1120
Assume pump efficiency (%)	50	40
Drive horsepower $= \frac{\text{(gpm) (head) 8.33}}{\text{33,000 x efficiency}}$	50.3	45

Use 100 gpm Westinghouse centrifugal pump with 55 hp. drive.

*Morse, op.cit., p. 476.

Pump for manufacturing condensate:

TABLE 24

Item	Max.Flow	Ave.Flow
Flow (lbs/hr)	18,000	13,050
Capacity (gpm)	25	18.1
Temperature of condensate (°F)	140	140
Density at 140° referred to 60° standard	.985	.985
Static head assumed (ft)	50	50
Velocity and friction head (ft)	30	30
Pump operating head (ft)	80	80
Assume pump efficiency (%)	40	30
Drive horsepower $= \frac{(\text{gpm})\ (\text{head})\ 8.33 \times .985}{33{,}000 \times \text{efficiency}}$	1.23	1.20

Use 30 gpm Westinghouse centrifugal pump with 2 hp. drive.

Open heater pump:

TABLE 25

Item	Max.Flow	Ave.Flow
Flow (lbs/hr)	22,597	15,835
Capacity (gpm)	31.4	22
Feed water temperature (°F)	96	96
Density referred to 60° standard	.994	.994
Static head assumed (ft)	50	50
Velocity and friction head (ft)	10	10
Suction head assumed (ft)	-10	-10
Pump operating head (ft)	50	50
Assume pump efficiency (%)	45	35
Drive horsepower $= \frac{(\text{gpm})\ (\text{head})\ 8.33 \times .994}{33{,}000 \times \text{efficiency}}$	.875	.785

Use 35 gpm Westinghouse centrifugal pump with 2 hp. drive.

16. Power Required by Auxiliaries.

All auxiliaries are steam driven according to the most economical heat balance. The following table is to check the assumption previously made as to the power required by the auxiliaries.

TABLE 26

Item	Design Condition Units Required	Design Condition Power Consumption (hp)
Air pump for No. 8 condenser	1	2.0
Air pump for No. 7 condenser	1	1.0
No. 8 condenser removal pump	1	15.0
No. 7 condenser removal pump	1	10.0
Forced draft fan	1	20.0
Induced draft fan	1	35.0
Boiler feed pump	1	55.0
Manufacturing pump	1	2.0
Heater pump	1	2.0
Coal conveyer	1	10.0
Coal crusher	1	5.0
Stoker	1	5.0
Total power drive		172.0

This compares favorably with the assumption for auxiliary steam which required 180 hp. max., and is on the safe side.

$$\text{Auxiliary steam consumption} = \frac{172 \times 2545}{.9 \times .4\ (1307 - 1039)} = 4550 \text{ lbs./hr.}$$

Assumed steam consumption = 4760 lbs./hr.

17. Cost Estimates.

The following estimates were obtained from Notes on Power Plant Design, applying to the year 1924; Heat Power Engineering, vols. 2 and 3.

Estimated Cost of Equipment

Real estate	$ 8,000.
Building	200,000.
Railroad siding	3,200.
Boilers, settings, and superheaters	52,000.
Stoker	20,800.
Condensers	43,000.
Piping (including covering)	90,000.
Switchboard	24,000.
Turbine, generator and air piping	110,000.
Wiring (motor and light)	16,000.
Chimney and flue	28,000.
Forced draft fans and ducts	10,000.
Ash and coal handling equipment	52,000.
Feed water heater	3,900.
Feed pump	5,600.
Equipment foundations	12,000.
Water softener	2,000.
Generator ventilating ducts	6,000.
Engine room crane	2,400.
Machine shop equipment	4,000.
Oil filters and tanks	2,800.
Economizer	12,000.
Total cost of equipment	707,700.
Engineering supervision	20,000.
Total	$727,700.

Real estate	$ 8,000.
Building	200,000.
Railroad siding	3,200.
Chimney and flue	28,000.
Total building	239,200.
Cost of machinery	468,500.
	707,700.
Engineering supervision	20,000.
Total cost	727,700.

Cost of operation:

A. Fixed Charges

These estimates are based on charging 14% on machinery and 7.5% on building.

Fixed charges = 468,500 x .14 + 239,200 x .075 = $83,500.

B. Operating Charges

In order to allow for Sunday heating, assume coal consumption to range over entire year instead of 313 days. This allowance will be on the conservative side, since heating consumption is not as high as operating consumption.

Annual coal consumption previously determined on the above basis is 11,800 tons, costing $7./ton.

Assume labor charges as 20% of coal cost, which is a reasonable figure for engineer's salary and other labor.

Hence,	Cost of coal = 11,800 x 7 =	$ 82,600.
	Cost of labor = .2 x 82,600 =	16,600.
	Oil, waste and supplies =	2,000.
	Maintenance at 10% of coal cost =	8,260.
	Total operating charges	$109,460.

Assume life of plant is 15 years so that engineering supervision is distributed over this period.

$$\text{Engineering supervision} = \frac{20{,}000}{15} = \$1330.$$

$$\text{Total cost of operation} = 83{,}500 + 109{,}460 + 1330 = \$194{,}290.$$

18. Investigation for Purchasing Power and Producing Manufacturing Steam by the Installation of a Low Pressure Boiler.

A. Low Pressure Boiler Calculations:

Maximum steam demand = 20,000 lbs./hr.

Average steam demand = 14,500 lbs./hr.

To supply this steam, use a 115 psia boiler generating saturated steam. Assume 15% of the steam demand is required to drive the auxiliaries on the basis of the most economical heat balance.

Hence, h at 115 psia and saturation = 1189.7 Btu/lb.

Boiler capacity = 1.15 x 20,000 = 23,000 lbs./hr.

Also assume the condensate returns to an open heater leaving it at 210° F while receiving all the exhaust steam from the auxiliaries. Let the temperature rise through the economizer be 90° F.

Temperature of feed water leaving economizer = 300° F.

Therefore, h of water entering boiler = 269.6 Btu./lb.

Heat supplied by boiler = 1189.7 - 269.6 = 920.1 Btu./lb.

At maximum load, assume heat transfer rate = 10,000 Btu/hr./sq.ft.

$$\text{Required boiler surface} = \frac{920.1 \times 23{,}000}{10{,}000} = 2120 \text{ sq.ft.}$$

Use Erie City standard three-drum water tube boiler with one spare unit.

Assume length of "front tube", L = 15 ft.

Heating surface = N (14.89L - 28.76)

2120 = N (14.89 x 15 - 28.76)

No. of tubes, N = 10.8, Use N = 12.

Therefore, surface = 12 x 195.24 = 2340 sq.ft.

Dimensions of boiler:

A = 9' - 10 1/2"

B = 23' - 6 7/8"

C = 6' - 2 1/4"

D = 6' - 10 1/4"

E = 12' - 10 7/8"

Annual steam output = $1.15 \times 14{,}500 \times 8760 = 145 \times 10^6$ lbs.

Average boiler rating = $\frac{1.15 \times 14{,}500 \times 920.1}{2340 \times 3348} = 195\%$

Using boiler efficiency-rating curve, corresponding efficiency is 78.6%.

Hence, annual coal consumption = $\frac{145 \times 10^6 \times 920.1}{14{,}582 \times 2000 \times .786}$

$= 5820$ tons.

Coal used in banking and starting assumed = 150 tons.

Total annual coal consumption = 5970 tons.

Cost Estimate of the Low Pressure Boiler Plant

Real estate	$ 5,000.
Building	100,000.
Railroad siding	3,000.
Boilers, settings	20,000.
Stoker	11,000.
Economizer	7,000.
Piping (including covering)	45,000.
Chimney and flue	15,000.
Forced draft fan and ducts	5,000.
Ash and coal handling equipment	25,000.
Feedwater heater	20,000.
Feed pump	3,000.
Equipment foundations	5,000.
Water softener	2,000.
Machine shop equipment	2,000.
Oil filters and tanks	2,000.
Added wiring	4,000.
Total cost of equipment	274,000.
Engineering supervision	10,000.
Total cost	$284,000.

Real estate	$ 5,000.
Building	100,000.
Railroad siding	3,000.
Chimney and flue	15,000.
Total cost of building	168,000.
Cost of machinery	106,000.
Total cost	$274,000.

Cost of operation:

A. Fixed charges = 106,000 x .14 + 168,000 x .075 = $27,450.

B. Operating charges

Cost of fuel = 5970 x 7 =	$ 59,600.
Cost of labor = .2 x 59,600 =	12,000.
Oil, waste and supplies =	1,000.
Maintenance = .1 x 59,600 =	5,960.
Total operating charges	$ 78,560

Assume life of plant is 15 years so that engineering supervision is distributed over this period.

$$\text{Engineering supervision} = \frac{10{,}000}{15} = \$670.$$

Total cost of operation = 27,450 + 78,560 + 670 = $106,680.

19. Cost of Purchasing Power.

Average amount of energy purchased $= 313 \times 24 \times 1.02 \times 1337.5$

$$= 10.25 \times 10^6 \text{ kw hr},$$

where transformer loss is assumed = 2%.

Use Block Hopkinson Demand Rate.*

A. Demand Charge:

1. $1.50 per kw of demand per month for first 500 kw and
2. $1.00 per kw for all additional per month.

B. Energy Charge:

1.	3.5	cents	per	kw	hr	for	first 5,000	kw	hr	per	month.
2.	2.0	"	"	"	"	"	next 10,000	"	"	"	".
3.	1.50	"	"	"	"	"	next 15,000	"	"	"	".
4.	1.25	"	"	"	"	"	next 20,000	"	"	"	".
5.	1.00	"	"	"	"	"	next 100,000	"	"	"	".
6.	0.9	"	"	"	"	"	next 350,000	"	"	"	".
7.	0.8	"	"	"	"	"	all excess	"	"	"	".

C. Prompt payment discount: A discount of 7 per cent will be allowed for payment of the bill within 10 days of the date rendered where the net bill exceeds $5000.

D. Primary discount: A discount of 10 per cent will be allowed if energy is purchased at 11,000 to 22,000 volts and the customer furnishes all transformer and substation equipment.

Applying the rate:

A. Demand Charge:

1.	Annual for 500 kw	$ 9,000.
2.	Annual for 1750 kw	21,000.
		$30,000.

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*Justin and Mervine, <u>Power Supply Economics</u>, p. 245.

B. Energy Charge:

1. Annual at 3.5 cents	$ 2,100.
2. Annual at 2.0 cents	2,400.
3. Annual at 1.5 cents	4,500.
4. Annual at 1.25 cents	3,000.
5. Annual at 1.00 cents	12,000.
6. Annual at 0.9 cents	37,800.
7. Annual at 0.8 cents (excess)	34,000.
Total energy charge	95,800.
Demand charge	30,000.
Total demand and energy charge	$125,800.

C. Prompt payment discount:

7% of $125,800. $ 8,800.

D. Primary discount:

Assume transformer, switchboard and other equipment supplied by the factory, but since the voltage is not 11,000 or 22,000 volts, assume discount only of 5%.

5% of $125,800. $ 6,280.

E. Cost of switchboard, transformer, etc. = $60,000.

Assume annual fixed charge and operating cost for this item at 17% so that annual cost is .17 x 60,000 = $10,200.

Total annual cost of purchasing power = (125,800 - 8,800 - 6,280 + 10,200) = $120,920.

Annual cost of boiler plant	$106,680.
Annual cost of purchasing power	120,920.
Total annual cost	227,600.
Annual cost of steam plant	194,290.
Annual saving	$ 33,310.

This annual saving will pay for a new steam plant after 22 years which is approximately the average life of a new plant.

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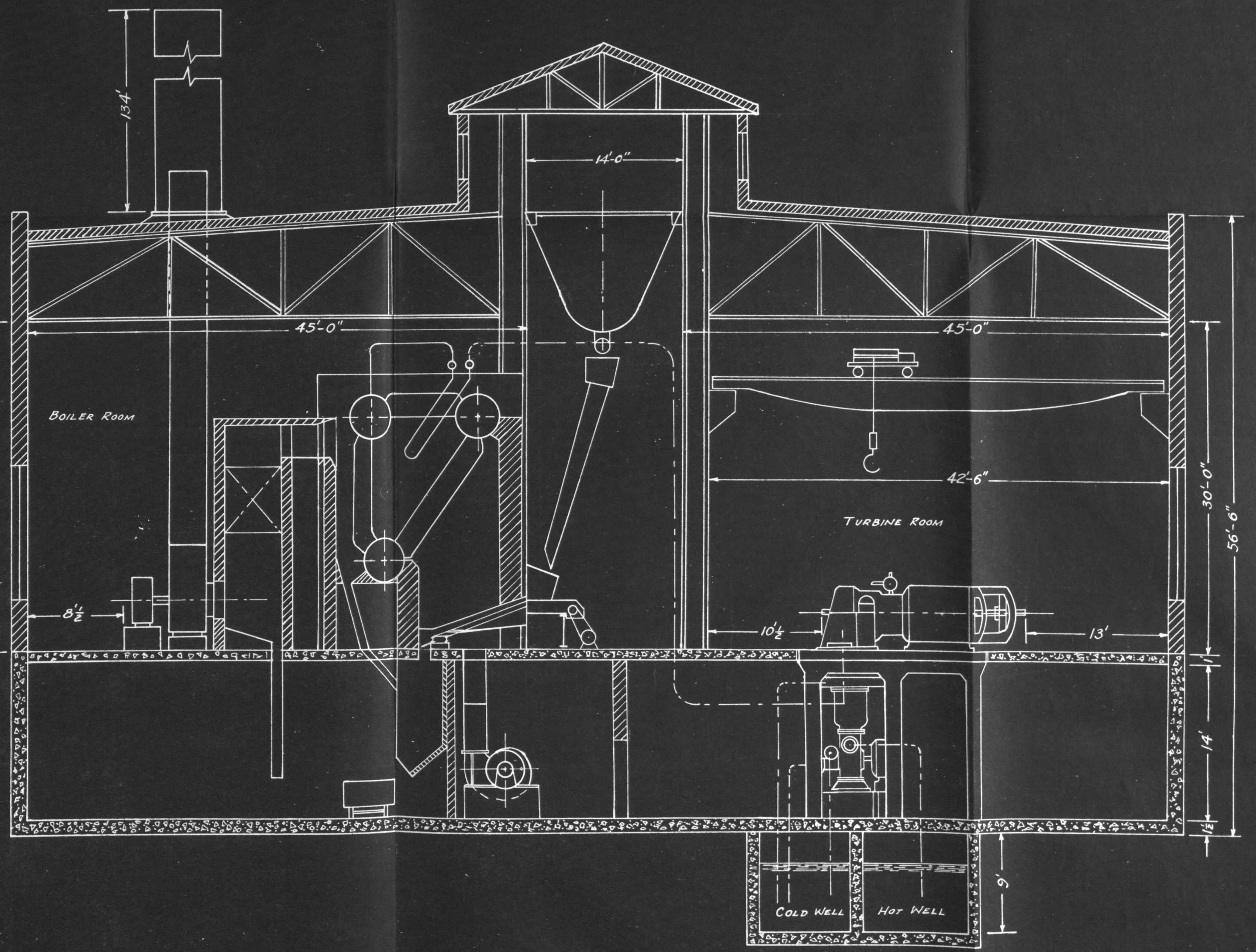
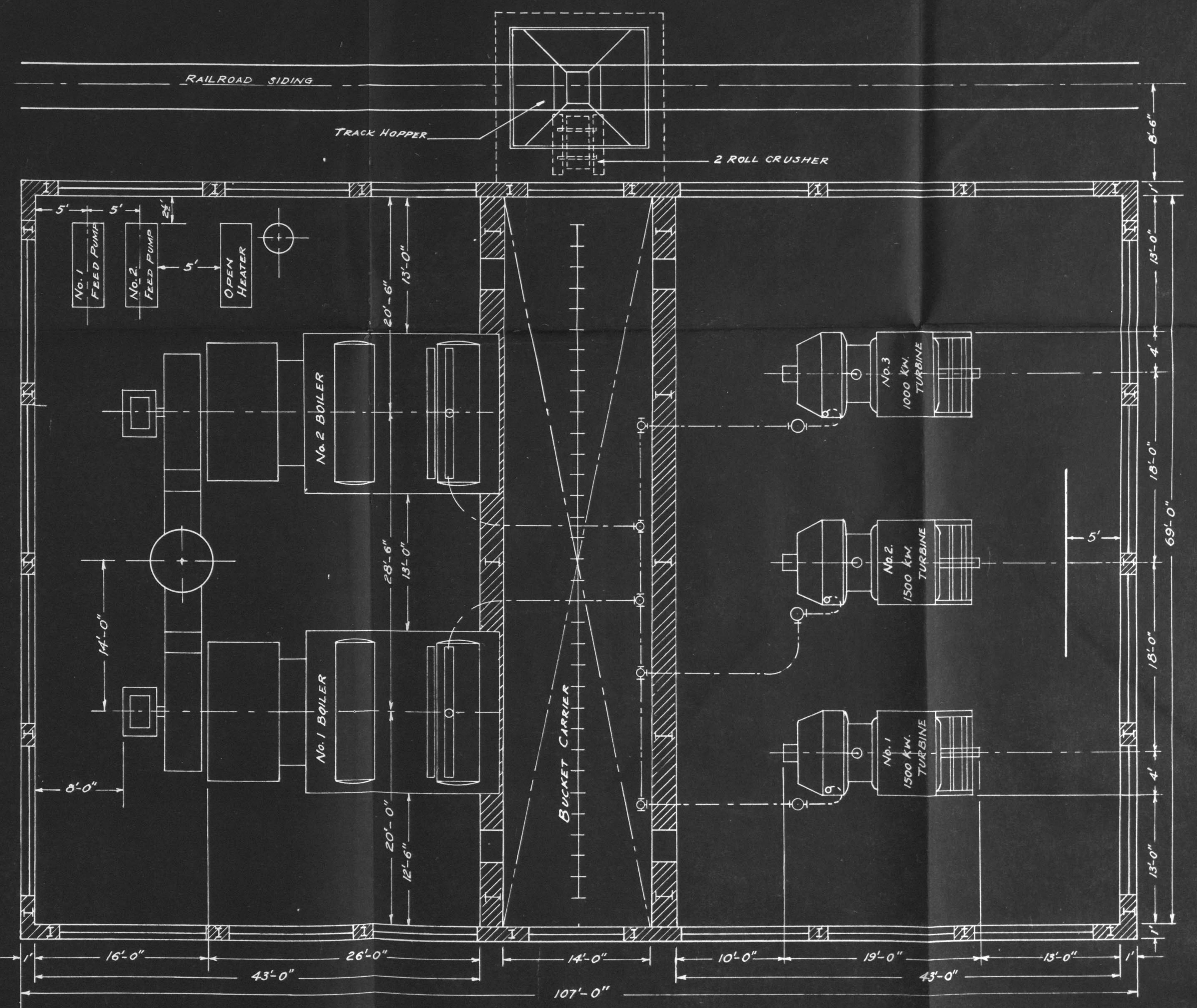


4000 Kw. POWER PLANT LAYOUT

SCALE 1/8" = 1'



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