

T H E D E S I G N

OF A

POWER AND LIGHTING AND HEATING PLANT

for the

UNIVERSITY OF TENNESSEE

Submitted by

P. I. TINKLER,

for the

D E G R E E

OF

BACHELOR OF SCIENCE

in

MECHANICAL ENGINEERING

Under the direction
of

DEAN CHARLES E. FERRIS

Professor of Mechanical Engineering

UNIVERSITY OF TENNESSEE

- - - - -

M A Y

1 9 2 1

3 4 - - - - -

THE PRESENT PLANT

Boilers

At present the power plant consists of 3 boilers, 2-80 H. P. and one 175 H.P. totaling 335 H. P. This plant supplies light for all buildings on the campus, and power for the machinery at Estabrook Hall.

In winter the exhaust steam is used to heat Estabrook Hall, all other buildings are heated by live steam from the boilers. At present this installation is large enough to supply the demand.

ENGINES & GENERATORS.

The generating units consist of one 50 K. W. D. C. generator, direct connected to a compound horizontal reciprocating engine, one 25K. W. D. C. generator direct connected to a simple slide valve engine. There is also an old belt connected generator, but this is to be considered more as a relic than an asset to the plant.

FEED WATER HEATER

The feed water is a closed heater and is heated by exhaust steam.

PROBLEM

It is desired to install a power plant that will furnish light heat and power for every building on the University campus at present and make provision for future extension. For present conditions about 1000 B.H.P. will be adequate to supply the demand, but there will be future extensions, and a probability that buildings not on the campus will be heated and lighted from this central plant

In this case, provision must be made for a larger installation. It is estimated that 1200 B.H.P. will be large enough to supply this extra demand. Instead of making provision for a future unit, it seems to be the best plan to make the initial installation large enough to supply this increased demand.

By having a plant that is more than sufficient to carry the load means that there will be no necessity for operating them above rated capacity.

Since this is to be a purely theoretical design and will not be used in event there is ever a new plant built, expense will not be considered as closely as if it was an actual design to be used, but will make an effort to design as near an ideal plant as possible.

LOCATION

The plant will be located on the west side of Estabrook hall, the front of the building being in line with the front of Estabrook. A fifty foot space will be left between buildings. In this location very little excavation will be necessary, since the hill slopes gradually back to the railroad. A spur track will be built from the Southern Railroad to supply coal. An entrance bridge will be built extending from the road to the main floor of the engine room.

SELECTION OF BOILERS

There is no doubt but what water tube boilers will be used. A water tube boiler has many decided advantages over a fire tube boiler such as:

1. Easy steamers
2. Contain less volume of water, this making them less dangerous in case of an explosion.

3. Easily accesable for cleaning.
4. Can be cooled off quickly.

After deciding to use water tube boilers, the question arises as to type and size to be used. The Babcock & Wilcox water tube boiler has been and is being used successfully and economically in many large central stations, so this type of boiler will be used.

Next the question as to size of units to install to obtain the required H.P. Six 200 H.P. units could be used, but a stoker fired boiler operates more efficiently in larger units and 6 boilers would require much more space than 4 units of larger size, so it seems best to use 4 units of 300 H.P. each. These boilers, with the builders rating of 10 square feet of heating surface for each boiler horse power will have 12000 square feet of heating surface. These boilers will be placed in batteries of two, making 2 batteries.

The boiler and engine room will be an end to end arrangement, so the steam from the boilers will be drawn off into a header and this header leads into a manifold, where the steam will be distributed to the turbines and auxiliaries.

A water tube boiler can not be operated continuously and be expected to last very long or operate with the greatest degree of efficiency, so provision must be made whereby one unit will always be idle for cleaning and repair.

A water tube boiler of this size can be operated at about 50% overload without lowering the efficiency to a very great degree, 3 boilers can furnish the desired power even at the maximum peak, for 3 boilers will, if operated at 50% overload, furnish 1350 H.P. This leaves one boiler idle at all times for overhaul.

During the summer months when the only demand will be

the lighting and power demand, 2 boilers can easily supply this, so the units can be used alternately, thus always having a clean boiler and an easy steamer. This will lower the coal consumption and increase the overall efficiency of the plant.

PRIME MOVER

In all present day power plants the tendency seems to be to select a rotary prime mover, and steam turbines are fast replacing the reciprocating engine.

When the advantages that a turbine has over a reciprocating engine all considered, it is quite evident that they should replace the straight line motion engine.

In a plant of this type the load varies to a great extent during the period of operation and a reciprocating engine is not as flexible as a turbine, and cannot adapt itself to these changes as readily as a turbine, and when they do there is a loss in efficiency much greater than that in case of a turbine. Some of the advantages of the turbine besides the one mentioned above are given below:

1. Do not require as large floor space.
2. Low maintenance and attendance.
3. Oil is not used internally, so feed water will be kept free from oil.
4. Free from vibration and has an even torque.

In the selection of turbines due consideration must be given to the heating demand. In the winter all the exhaust steam will be used for heating purposes so the turbines must be operated non-condensing and with a back pressure of about 10 pounds.

In summer when there is no heating demand the turbines can be operated condensing. Instead of using a turbine that operates

both condensing and non-condensing it is better to use a non-condensing turbine for winter use and a condensing turbine for summer use, by this plan a higher efficiency can be obtained.

This plant must be considered as a public service plant so provisions must be made for continuous operation. This would mean duplicate installations, so it would be better to install two turbines of the same capacity required on the heating system.

BOILER FEED PUMPS.

City pressure can not be used at any time to feed the boilers since the operating pressure will be about 150 pounds gage. This being the case, duplicate feed pumps will have to be installed, the question arises as to which, centrifugal, piston or plunger pumps will be used. Piston or plunger pumps are the most common type in use today. This class of pump is well adapted for boiler feeding purposes since it may be operated as slowly as is suitable for boiler feed purposes. The centrifugal pump has certain decided advantages over the piston pump, such as absence of valves and pistons, ease of operation, easy to repair, but the efficiency of such a pump is not as high as the best grade piston pumps, and the rate of flow cannot be as efficiently regulated for a wide range of operating conditions as can be in the piston or plunger type.

It seems best to use a piston pump. The problem is as follows: - Design

A pump that will supply water to 1200 H.P. of boilers operating at rated capacity. Gage pressure 150 pounds per square inch. Feed water temperature 175 deg. F.

Assume 15% loss due to fittings ect. -

$$150 \times .15 = 22.5 \quad 150 + 22.5 = 172.5 \text{ pounds per sq. in.}$$

total head, $172.5 \times 2.31 = 400$ feet, head to pump against 6" for P, =
 $1196 \text{ 6" for } 175^\circ = 142.9 \quad 1196 - 142.9 = 1053 \text{ B.T.U required per}$
 pound of steam

$$1 \text{ H.P.} = 33.480 \text{ B.T.Us per hour}$$

$$33,480 \div 1053 = 31.8 \text{ pounds of water evaporated per H.P. hr.}$$

$$31.8 \times 400 = 12,720 \text{ ft. lbs. of water per hour}$$

$12.720 \times 1200 = 15,280,000 \text{ ft. lbs of work per hour}$ $31.8 \times 1200 = 38,200$
 pounds of water per hour. Pumps are usually designed to supply about
 twice the required amount of water, so the above figure will be taken
 approximately as double value W 76,000 pounds per hour.

D Diameter of water cylinder in inches

W Weight of water delivered in pounds per hour

S Ratio of water actually delivered to piston displacement.

d diameter of steam cylinder in inches.

boiler pressure, pounds per sq. in.

length of stroke in inches

N Number of working strokes per minute

H Head in feet between suction and boiler water level

R Resistance in pounds per sq.in. between suction level
 and boiler water level, due to valves, fittings ect.

I Indicated horse power of pump at maximum capacity.

S=.8 assume N 100 feet per minute piston travel,

and L=10 inches.

$$\text{Therefore } L N = 10 \times 100 = 1000$$

$$D = 0.77 \sqrt{\frac{76000}{1000 \times 0.8}} = 0.77 \times \sqrt{93.7} = 7.45 \text{ inches}$$

use 8" as diameter of water end.

$$\text{assume } (0.433 H + R) = 0.25 p$$

$$0.25 \times 150 = 37.5 \quad \text{assume 60\% efficiency}$$

$$d = 8 \sqrt{\frac{150 \times 37.5}{0.60 \times 150}} = 8 \sqrt{\frac{187.5}{80}} = 12.2 \text{ inches}$$

use 12 inches as diameter of steam end

$$I = \frac{(38200 \times 2) (187.5 \times 2.3)}{33000 \times 60 \times 0.60} = 22.4 \text{ H.P.}$$

Use a piston pump direct acting, simplex with air

chamber and snifting valves attached, so as to give a steady flow and prohibit pounding.

These pumps should be installed in duplicate so that in case of accident one will be in reserve. These pumps will discharge through the feed water heater and have a suction both from the hot well and city mains. The pump is to be packed with metallic packing and must be able to pump water at a maximum temperature of 175 deg.F.

STOKERS

It is necessary to offer any argument in favor of mechanical stoking but the question that arises is, What type of stokes shall be installed? Whether underfed or over fed and what make of the type selected.

In making the selection the points to be considered are: kind of fuel to be burned, simplicity, prevention of smoke, maintenance, and efficiency.

In the plant, Brushy Mountain coal is to be used, this is relatively a cheap grade of soft coal and clinkers freely and also cokes very freely and will stick to bridge walls.

The principle of the underfed stoker is one of placing the fire on top of the fuel and blowing air up through the fuel, keeping the coal always pushed up under the fire, this method of firing will insure complete combustion, since the proper amount of air is put in with the fuel, complete combustion prevents smoke, and increases the heating value of the coal.

In the plant to be installed there will be only one fireman and this necessitates having a stoker that requires the least attention.

The Jones A.C. stoker seems to be the best fitted for all conditions that prevail in this plant; according to the ten points of superiority of the Jones A.C. stoker are:

1. Its simplicity - Few moving parts, simple dumping mechanism.
2. Low maintenance - No moving parts in the fire and green fuel is always over the tuyeres blocks, thus insuring them from burning out.
3. Its high efficiency - As shown by numerous tests.
4. Automatic cleaning - The fuel is burned complete, thus keeping a clean furnace.
5. Prevention of smoke - Combustion is so complete that there is no smoke.
6. The tuyeres are so designed and the movement is such that no clinkers can form.
7. Heavy overload capacity, results from the even fuel bed and control of air.
8. Minimum of moving parts, thus reducing maintenance.
9. Individual control of retorts thus producing an excellent fire at all times.
10. Automatic control of fuel and air insuring even steam pressure regardless of fluctuations.

It is evident that this stoker will suit the conditions from the above points and the company guarantees efficient operation. The Jones A.C. Stoker costs more than some other underfed stokers but the low operating cost and the fuel saved will eliminate this objection.

CONDENSING EQUIPMENT

The condensing problem is as follows:

A condenser is required that will condense the steam from a 250 K W turbine using 22 pounds of steam per K W hour or

5,500 pounds of saturated steam per hour and maintain about a 28" vacuum with cooling water at approximately 70 deg. F. It is doubted as to whether a 28" vacuum will be maintained, since the load factor will be very small, but if the equipment is designed for a 28 inch vacuum, it will be much easier to maintain a 27" or even 26.5" vacuum during operation.

All condensing water will be taken from second creek and during the summer this water will have rather a high temperature. In order that the creek will have sufficient depth to allow a suction pipe, a small dam will have to be constructed so as to make the depth about 5 feet where the suction pipe enters.

The power plant as installed should be such that it could be considered as a part of the steam engineering laboratory and in such event a condenser should be installed that will afford means of measuring the actual steam consumption of the turbine. A surface condenser is the only type that affords a reliable means of doing this because the exhaust steam from the turbine is kept entirely separate from the cooling water, hence it is a simple matter to measure the condensate in the hot well and avoid any error that might be made in calculating the steam consumption from the boiler feed pumps.

Then too, the surface condenser can be used with very impure cooling water and can use water with a large quantity of matter in suspension, since none of the cooling water comes into contact with the condensate. Such is not true in case of a barometric or jet condenser, for in both the steam is condensed by the water

coming into actual contact with the steam.

A screen must be placed over the injection pipe to keep any foreign matter from being drawn into the condenser that cannot be passed through the tubes.

A two pass condenser will be used, that is there will be a baffle placed in the intake and at the middle section, thus forcing the water through the lower half of the condenser first, and returning through the upper half to the discharge.

Condenser calculations:

250 K. W. turbine, water rate 22# per K.W. hour. Initial pressure 165 pounds absolute. Temperature of cooling water 70 deg. vacuum 28 inches referred to a 30 inch barometer. Assume a pressure drop from the turbine nozzle to the air pump suction of 0.2 inch.

H Heat content of the exhaust B.T.U.s per pound above 32 deg. F.

t_2 Temperature of discharge.

W Water rate per K. W. hour

t_0 Temperature of injection water.

H_i Initial heat content of steam entering the prime mover - B.T.U.s per pound above 32 deg. F.

H_r Heat lost by radiation from the prime mover and exhaust piping.

t_s Temperature of the steam.

ps 30-(28 .2) 1.8 inches Hg.

Corresponding to 1.8 inches Hg 97.5 deg. F.

$H = H_i - H_r - \frac{3412}{22}$ assume H_r .5% of H_i

$H = 1195 - .005 \times 1195 - \frac{3412}{22}$

$H = 1195 - 5.975 - 150$

$H = 1039$

Q Weight of circulating water in pounds per hour.

S Extent of cooling surface in sq. ft.

U Mean coefficient of heat transfer for the entire period of heat exchange.

d = Mean temperature difference.

$$Q = \frac{22 \times 250 (1039 - 97.5)}{85.5 - 70} = \frac{5350000}{15.5}$$

$Q = 346,000$ pounds

$$d = \frac{85.5 - 70}{\log \frac{97.5 - 70}{97.5 - 87.5}} = \frac{15.5}{0.8286} = 18.7 \text{ deg. F.}$$

With an assumed velocity of 8 ft. per second

$$V = 600$$

$$S = \frac{346,000 (85.5 - 70)}{600 \times 18.7} = \frac{5350000}{11200} = 480 \text{ sq. ft.}$$

A condenser with 480 square ft. of heating surface will be required. The tubes will be 5/8 inch No. 18 B.W.G. brass tubes.

AUXILIARIES

In the condensing equipment there shall be a rotary air pump of sufficient size to remove all air and condensate from the condenser necessary to maintain the required vacuum. This pump will be direct connected to a motor large enough to drive it at the required speed.

The circulating pump will be so designed as to deliver 346,000 pounds of circulating water per hour. A centrifugal pump will be used capable of making 600 R.P.M. A vertical reciprocating engine will be used for motive power. This engine will use steam at boiler pressure and be regulated by throttling.

DESIGN OF CHIMNEY

The problem is to design a chimney for a battery of Babcock Wilcox water tube boilers equipped with Jones A.C. stokers, to burn Brushy Mountain coal. The boilers are rated at 1200.H.P.

The material for the stack will be radial brick either red or light brown, to harmonize with the surrounding buildings.

Custodis radial brick will be used, perforated so as to reduce weight, and shaped so as to make the chimney proportion right. Ratio of heating surface to grate surface 50 to 1. Take flue approximately 60 feet long designed to carry 50% overload. Atmospheric temperature about 65 deg. F. Calorific value of coal 11000 B.T.U.s Temperature of flue gases 550 deg. F. A plant of the above size should have a combined efficiency of 65% operating at 50% overload. Maximum boiler H.P. $1200 \times 1.5 = 1800$ - Heat equivalent of 1 B.H.P. $34.5 \times 970.4 = 33,479$ B.T.U.s Coal per B.H.P. $33,479 \div (11000 \times .65) = 4.7$ (approx.). Total grate surface $(1200 \times 10) \div 50 = 240$ sq.ft. Total coal burned per hour $4.7 \times 1800 = 8,450$ #. Maximum rate of combustion $= 8,450 \div 240 = 35.2$ pounds per square foot.

Assume all pressure losses at maximum rating.

Loss through fuel and grate taken care of by forced draft.....	0.00
Loss in boilers (Furnace to stack side of daisyser).....	0.55
Loss in flue 60 ft. at 0.1 inch per 100 ft.....	.06
Loss in turns 2X.05.....	.10
Total loss or required effective pressure measured at flue entrance of stack.....	.71

D Maximum theoretical static draft inches of water

H Effective height of chimney in feet

Ta Absolute temperature of outside air degrees F.

Tc Absolute temperature of chimney gas degrees F.

$$D = H \left(\frac{7.64}{T_a} - \frac{7.95}{T_c} \right)$$

$$1.00 = H \left(\frac{7.64}{550} - \frac{7.95}{1010} \right)$$

$$H = 150'$$

To determine the diameter of stack at top use Kent's formula found in Gebhardt.

E Effective area, A=Inside area at top

$$E = \frac{.3 \text{ H P}}{\sqrt{H}} = \frac{.3 \times 1200}{\sqrt{150}} = \frac{360}{12.3} = 29.2 \text{ Effective area}$$

$$E = A - .6\sqrt{A}$$

$$29.2 = A - .6\sqrt{A}$$

$$29.2 + .09 = A - .6\sqrt{A} + .09$$

$$-\sqrt{A} = -0.3 - \sqrt{29.29} \quad -0.3 - 5.45$$

$$\sqrt{A} = 5.75 \quad A = 33.4$$

$$d^2 = \frac{33.4}{.7854} = 42.5$$

D=6.5 feet inside diameter at top.

FEED WATER HEATER

The question arises as to whether it will be profitable to use a feed water heater or not. In answer to this the situation must be considered. The auxiliary machinery driven by steam will be as follows: Circulating pump-Blower fan-Feed pumps and Stoker engine, all of these will be in use during the summer when the plant is operating condensing, but during the winter, only the feed pumps, stoker engine, and blower engine will be used.

In winter there will be approximately 5% of the steam required for the prime mover used for auxiliaries which will warrant the use of a heater, and in summer this figure will be increased because more auxiliaries will be in operation.

It is claimed that for every ten degrees rise in temperature of the feed water there will be a gain in heat of 1% and

a corresponding saving in fuel, that is if the steam that heats the water would be wasted. In this plant the steam would be wasted unless the auxiliaries were motor driven but motors do not seem to be a satisfactory solution where there is constant operation. In winter when there is condensate from the heating mains returning to the hot well, the temperature of the hot well will be approximately 150 deg.

The temperature rise due to the utilization of waste steam from the auxiliaries may be calculated as follows:

There is approximately 550 H.P. used during winter operation, assume 5% of this used by auxiliaries. $550 \times .05 = 27.5$ H.P. assuming a H.P. equal to 30# steam $27.5 \times 30 = 725$ pounds of steam per hour. The pressure will be atmospheric, therefore, $725 \times 970.4 = 705,000$ B.T.U.s per hour $550 \times 30 = 16500$

$705,000 \div 16500 = 42.5$ deg. temperature rise, during winter operation.

From the above calculations it would seem advisable to install a heater for the saving in fuel, also water is entering the boiler at a higher temperature which reduces the strains on the boiler to a considerable extent. So it seems advisable to use a heater. Now the question arises as to what type is best to be installed. Due to the fact that there will be considerable oil in the water, it seems the best plan to use a closed heater, also if an open heater is used the feed pumps will have to be located between the heater and boilers, this will necessitate the pumps having to pump water at a temperature of approximately 200 deg. F., this is

not desirable. So an exhaust steam, closed, atmospheric, vertical, water tube heater will be installed.

Feed water heater calculations.

Summer conditions.

Feed water for turbines.

22 X 250 = 5500# per hour

Feed water used by auxiliaries

6% of 5500 = 330 pounds per hour

Total feed 5500 + 330 = 5830 pounds per hour

A Required surface, Assume $V=300$

W Total weight of water heated.

V Mean coefficient of heat transfer for entire surface.

B.T.U.s per sq. ft. per degree difference in temperature per hour.

Final temperature of feed water, degrees F.

Initial temperature of feed water, degrees F.

Temperature at atmospheric pressure.

$$A = \frac{W}{U} \log_e \frac{t_s - t_o}{t_s - t_v} = \frac{5830}{300} \times \log_e \frac{212 - 110}{212 - 200}$$

$A = 19.4 \times 2.14 = 41.5$ square feet heating surfact. On the basis

of $1/3$ sq. ft. of surface per horse power the rating would be

$$\frac{41.5}{3} = 13.8 \text{ H. P.}$$

Now for winter operation the conditions will be different.

Winter conditions:

Feed water for turbine:

50 X 250 = 12500 pounds per hour

Feed water used by auxiliaries:

5% of 12500 = 625 pounds per hour

Total feed = 12500 + 625 = 13125 pounds per hour.

$$A = \frac{12500}{300} \times 2.14 = 41.6 \times 2.14 = 89 \text{ square feet.}$$

$$89 \div 3 = 29.6 \text{ H.P.}$$

It will be advisable to use the largest heater.

COAL HANDLING AND STORAGE

The problem is to transfer coal from the car to the bins without having to handle it by hand. In order to do this

it will be necessary to have a pit excavated beneath a spur track that will be run off the Sou. Railroad to the rear of the plant.

The cars will dump the coal into this pit and it will fall by gravity into the bin of the crusher, after passing through the crusher it will run down a chute to the conveyor.

The conveyor to be used will be a gravity discharge elevator conveyor as shown by the sketch. This conveyor will be run by a motor, and after elevating the case to the proper height will drag along the chute and deposit the case through the gates into the bunkers.

It is necessary to have at least 100 tons of coal stored so the problem that arises is where will be the best place to store this coal with the least additional expense. Since there is to be an excavation made beneath the fire room to take care of the case and ash handling apparatus, it will be very little more expense to use part of this space for storing coal. A bin can be made that will extend the length of the fire room or 48 feet, and 10 ft. wide and 10 feet high. This will allow a storage space of 4800 cubic feet or space for about 96 tons of coal in addition to the coal stored in the over head bin. The coal will be put in this bin by chutes leading from the bins similar to those used for feeding the stokes, and this bin will be filled before the overhead bins.

In order to remove this coal, sliding doors will be installed that discharge on a chute, similar to the one over the bins, provided for this purpose, coal will fall in this chute by gravity and will be conveyed into the overhead bins.

BLOWER ENGINE AND FAN.

Since the stokers require air under pressure for their operation, it will be necessary to install a blower engine and fan.

The engine will be a vertical, reciprocating engine, and direct connected to a a sirocco fan. Air ducts will lead to each air chamber under the stoker.

A balanced draft system will be used that is by adjusting the pressure created by the forced draft fan and the suction in the chimney so that the pressure in the combustion chamber becomes practically atmospheric.

METHOD OF ASH DISPOSAL.

Here the question arises as to whether it will be cheaper to use the coal handling equipment to handle the ashes also.

In order to use the coal elevator to dispose of the ashes, it will be necessary to construct an ash bin, to receive the ashes, This bin will have to be constructed so as to dump the ashes into a car run underneath. Then the ashes will have to be hauled away.

There will always be an expense connected with hauling the ashes away and will necessitate having a car placed to dump them into.

It seems advisable to install a steam jet conveyor and blow the ashes out on the bank behind the power plant, there is sufficient room there for all the ashes and the creek will wash them away when the pile reaches the creek.

The only disadvantages of this system are: (1) that the clinkers will have to be broken up before they will pass the intake opening but when stokers are used there will not be any large clinkers formed; (2) it cannot operate on less than 60# pressure, but this plant can furnish as high as 150# pressure so this objection will be eliminated; (3) a steam conveyor will not convey wet ashes. The ashes will not be purposely wet down, so there is no necessity in ever having wet ashes.

This system will not cost any more than the extra equipment that would have to be installed in connection with the coal handling equipment in order to handle the ashes.

The advantages of this system are:

- 1- Can be installed in any plant.
- 2- Easily installed.
- 3- Does not require any skilled labor to operate
- 4- There are no moving parts.
- 5- No bin is necessary.

The system will consist of a 6" pipe fitted with

extra fittings in front of each boiler to admit ashes.

There will be two right angle bands that will be made of specially designed elbows so that they can be easily replaced in case of too serious

Such a system seems to be the best method of disposing of ashes that can be used and by using it, the big problem of ash disposal is ended.

SOOT BLOWERS

In a modern plant every effort must be made to have the plant operate as efficiently as possible. Clean tubes and in saving of coal and increasing efficiency, so some method must be used to clean the tubes of soot. Probably the best method of cleaning the tubes is by use of the steam jet, this method not only cleans the tubes but lowers the temperature of the uptakes and stack.

A mechanical cleaner is best suited, one that can be operated without allowing any cold air to enter the boiler. The blower should also be a rotary type so that it can blow in any direction, thus cleaning all surfaces of the tubes. The nozzles must be set so that they do not allow the steam to be blown in direct contact with the tubes - thus eroding them. In a B & W Boiler as installed, the elements should be so placed as to discharge steam at an angle.

The vulcan soot cleaner seems to fulfil all requirements of an ideal installation so vulcan soot cleaner will be

installed in all boilers.

HOT WELL

A hot well will be installed to receive the condensate from the condenser during summer operation, and to receive the return condensate from heating mains in the winter, and drains from separators, and traps at all times. The boiler feed pumps will have a suction on the hot well, - The hot well will be large enough to hold all return condensate from the heating mains - for a period of one hour. This will require approximately

$$\frac{450 \times 30}{62.5} = 216 \text{ cubic feet.}$$

The hot well will be closed and strong enough to withstand the pressure placed on it by water backing up in the return pipe.

SEPARATORS

Since saturated steam is to be used, it is necessary to install separators on the pipe lines leading to the turbines. These separators must be able to take all water from the steam.

The Keystone steam separator, which is a centrifugal separator, will be used. This separator consists of a cast iron cylinder with a vertical pipe extending down about two-thirds the length. Around this pipe is a thread or screw wound spirally. Steam passing around this spiral causes the water to be thrown out against the walls where it drains down the side of the drain pipe and is carried away to the hot well.

The dry steam passes on to the turbines.

SUMMARY OF EQUIPMENT

Boilers - 4 units in batteries of 2 1200 H P Babcock & Wilson water tube, vertical header.

Turbines - 1- 250 K W condensing turbine steam consumption of 22.5#/ K W hr. 1-250 KW non condensing unit steam consumption 495#/ K W hr.

Pumps - 2 Worthington simples, direct acting horizontal diameter steam and 12" diameter water and B" 10" stroke. 22.5 H P

Chimney - 1 radial brick chimney 150' high 6.5' inside diameter at top.

Stokers - 8 Jones A. C. stoker retorts 2 for each boiler

Condensing equipment - 1- Wheeler condensing unit consisting of 1- surface condenser 5/8" tubes 18 B W G 480 ' heating surface

- 1- Centrifugal circulating pump
- 1- Rotary air and condensate pump
- 1- Reciprocating engine for driving circulating pump
- 1- Turbine for air pumps

Feed water heaters. 1 closed feed water heater, water, tube, vertical having 42 ' heating surface.

Coal handling equipment - 1 gravity discharge elevator conveyor, as manufactured by Link-Belt Co.

Ash handling equipment - 1-6" steam jet conveyor as manufactured by American Steam Conveyor Co.

Soot blowers, 4 sets of Vulcan soot cleaner.

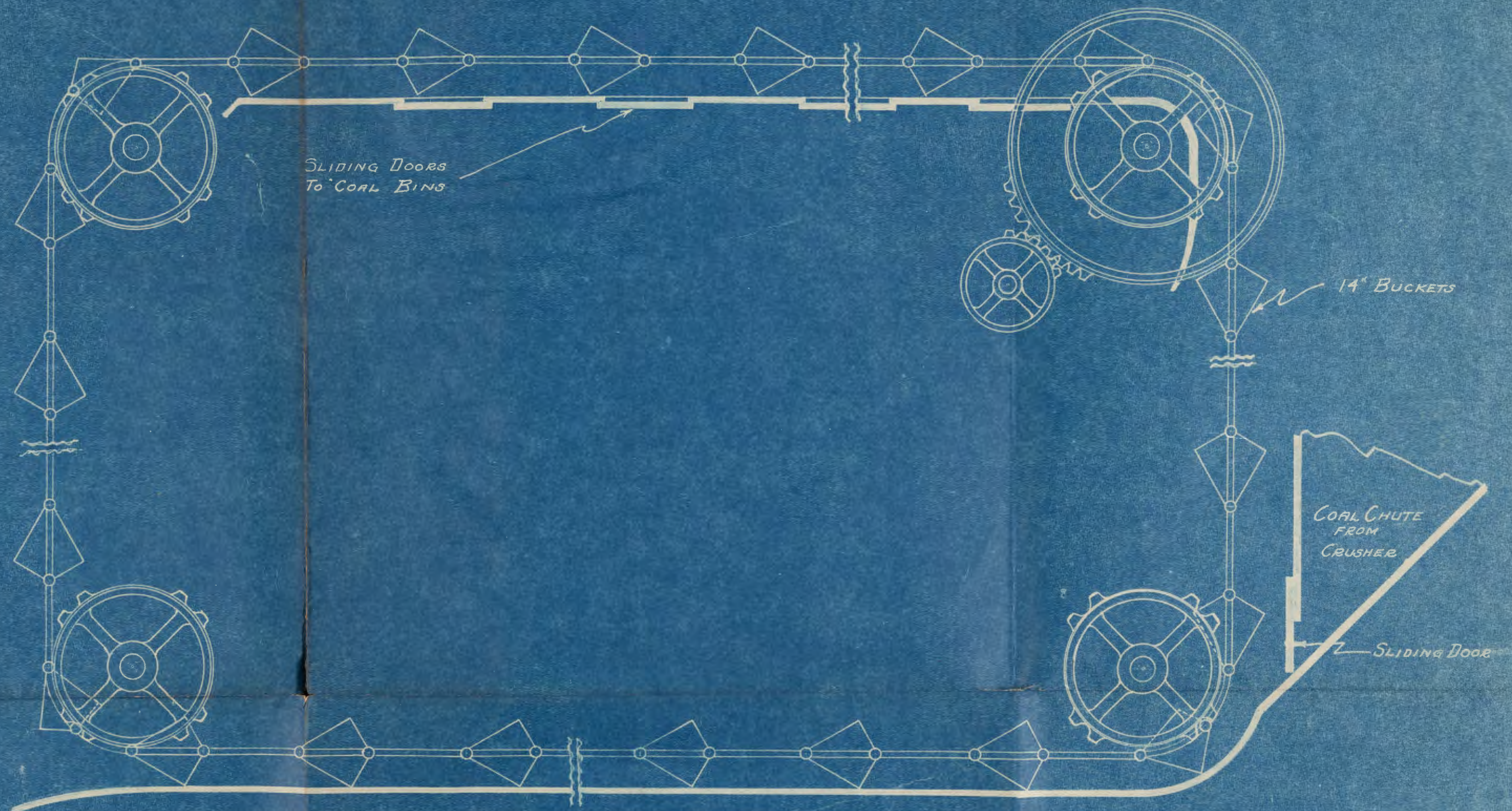
Separators-2 Keystone separator, installed on the steam lines to the prime movers.

Hot well. 1- Hot well with 216 cubic feet capacity.

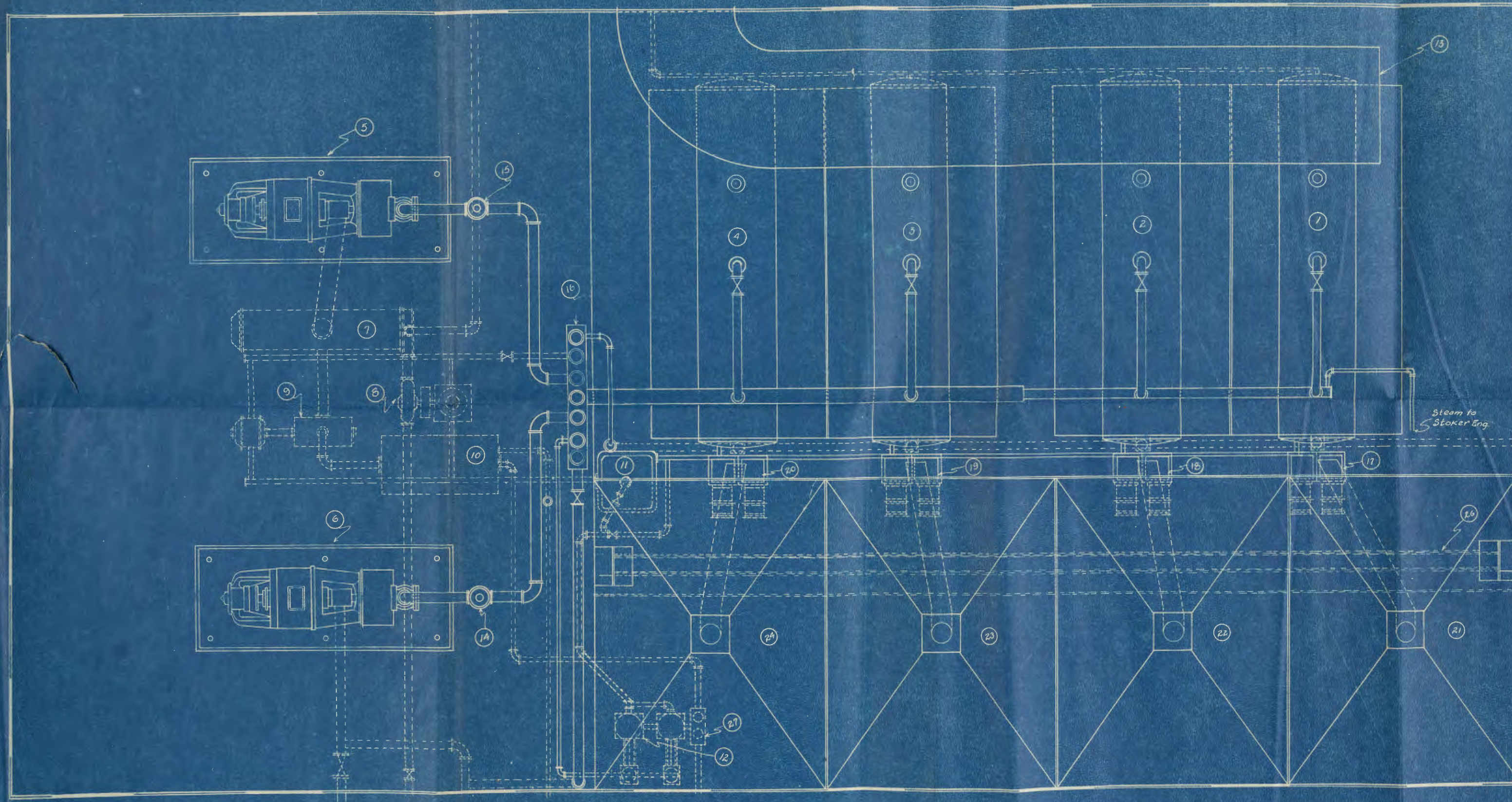
Blower Engine. 1, Vertical reciprocating engine to drive blower fan.

Fan. 1- Sirocco fan for mechanical draft for Jones A. C. Stokes.

Traps and drains where needed.

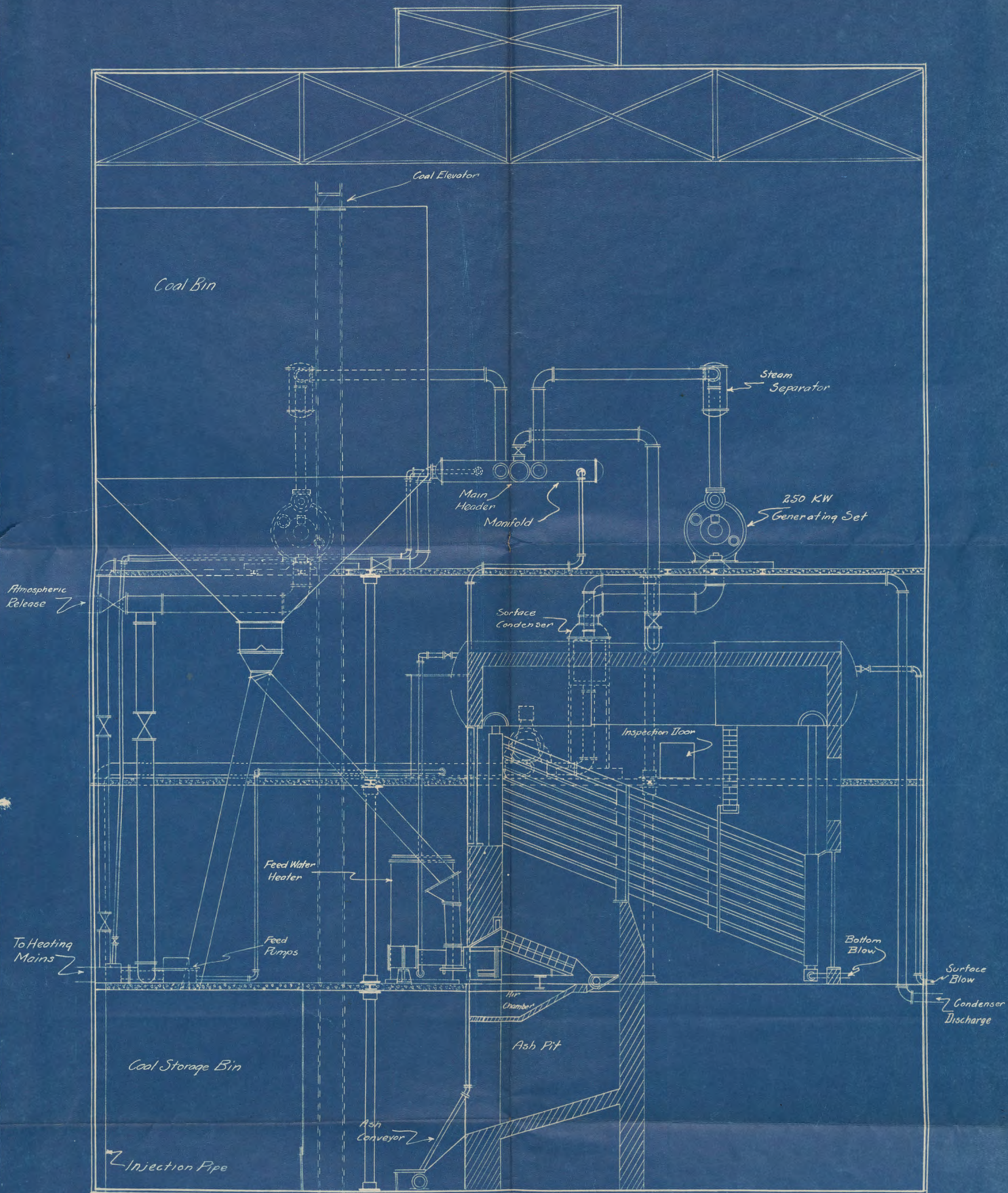


LINKBELT CONVEYOR
FOR
POWER PLANT
UNIVERSITY OF TENN
DO NOT SCALE THESIS
MAY 1921 R. S. Tucker

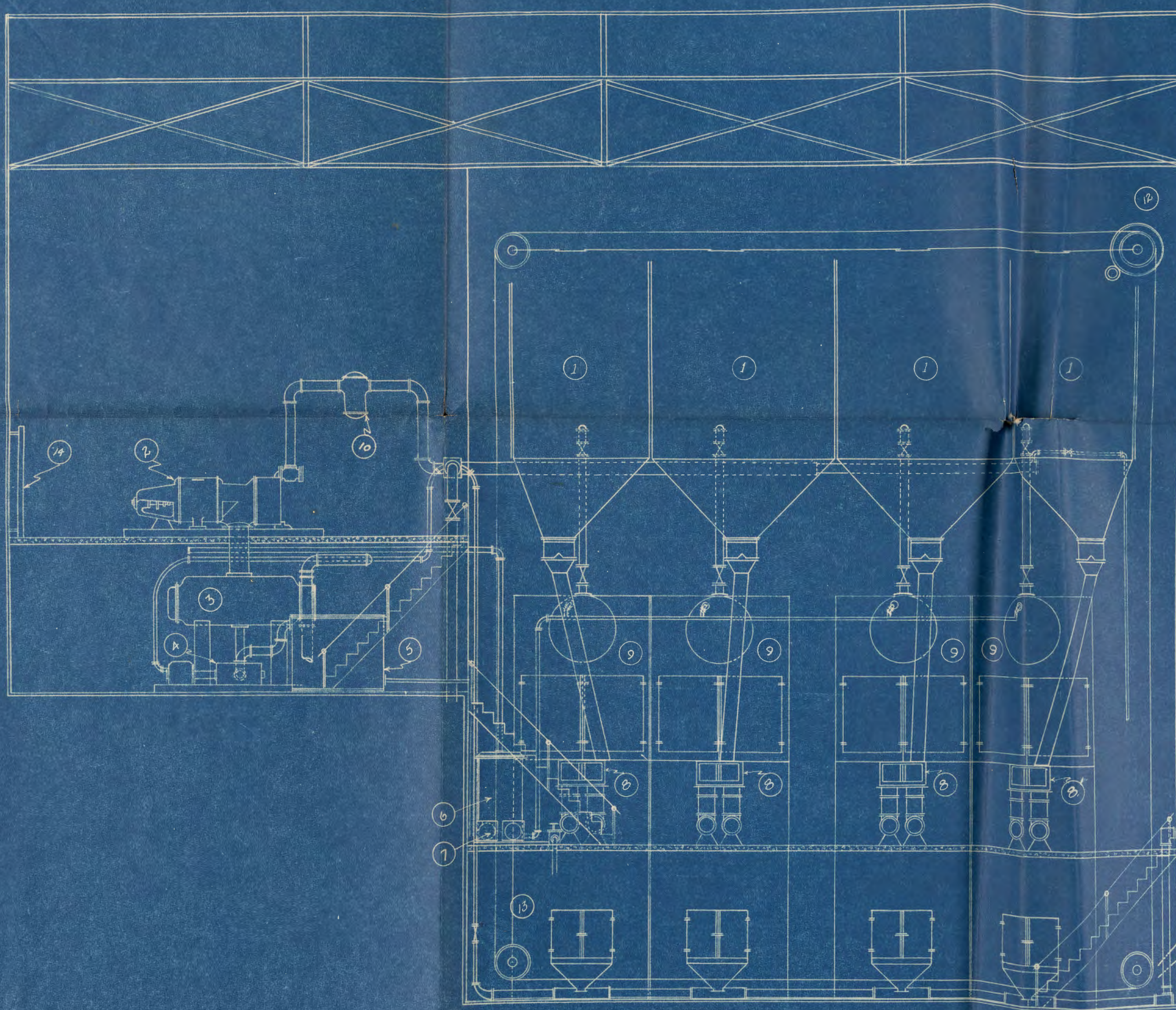


1	Babcock & Wilcox Boiler
2	" " "
3	" " "
4	" " "
5	250 K.W. Turbo-Generator
6	" " "
7	Surface Condenser
8	Circulating Pump
9	Flr. Pump
10	Hot Well
11	Feed Water Heater
12	Feed Pumps
13	Uptake to Chimney
14	Steam Separator
15	" " "
16	" " Manifold
17	Jones A.C. Stokers
18	" " "
19	" " "
20	" " "
21	Coal Bin
22	" " "
23	" " "
24	" " "
25	Crusher
26	" " Conveyor
27	Feed Pump Manifold

PLAN
 OF
 PLANT LAYOUT
 POWER PLANT
 UNIVERSITY OF TENN
 SCALE 3/8" = 1'
 MAY 1921
 THESIS
 B. J. Tucker

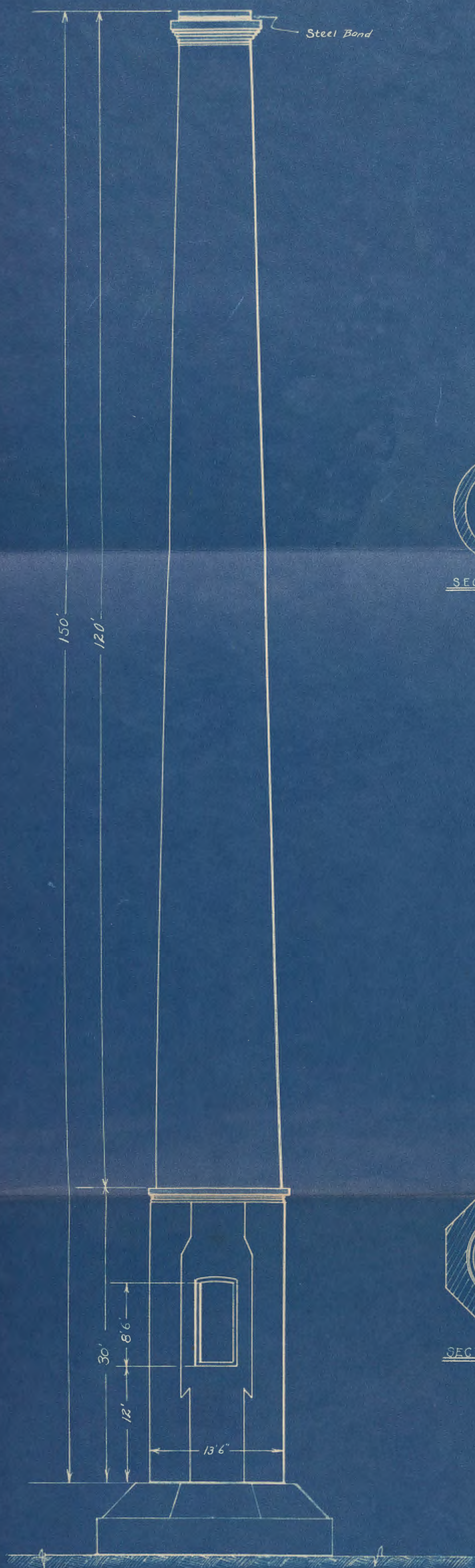


END ELEVATION
 OF
 PLANT LAYOUT
 POWER PLANT
 UNIVERSITY OF TENN
 THESIS
 SCALE $\frac{3}{8}'' = 1'$
 MAY 1921
 R. L. Tucker

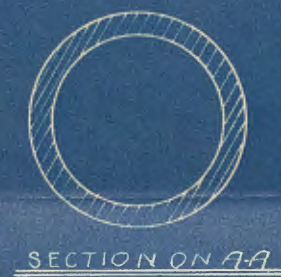


1	Coal Bins
2	Turbo-Generating Set
3	Surface Condenser
4	Rotary Air Pump
5	Hot Water
6	Feed Water Heater
7	Feed Pumps
8	Jones A.C. Stokers
9	Water Tube Boilers
10	Steam Separators
11	Coal Crusher
12	Coal Hoist
13	Ash Conveyor
14	Switch-board

SIDE ELEVATION
OF
PLANT LAYOUT
POWER PLANT
UNIVERSITY OF TENN
SCALE 1/4"=1'
MAY 1921
THESIS
R.J. Finkler



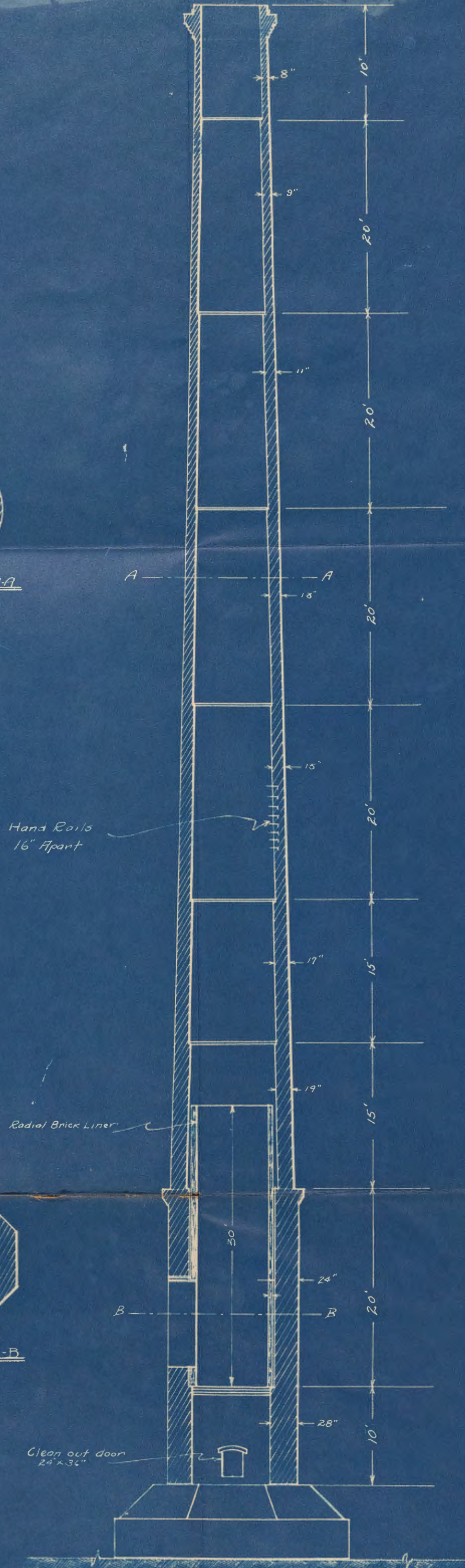
ELEVATION



SECTION ON A-A



SECTION ON B-B



SECTION V

RADIAL BRICK CHIMNEY
FOR
POWER PLANT
UNIVERSITY OF TENN
SCALE $\frac{1}{8}"=1'$
MAY 1921
THESIS
R.D. Tucker, Jr.

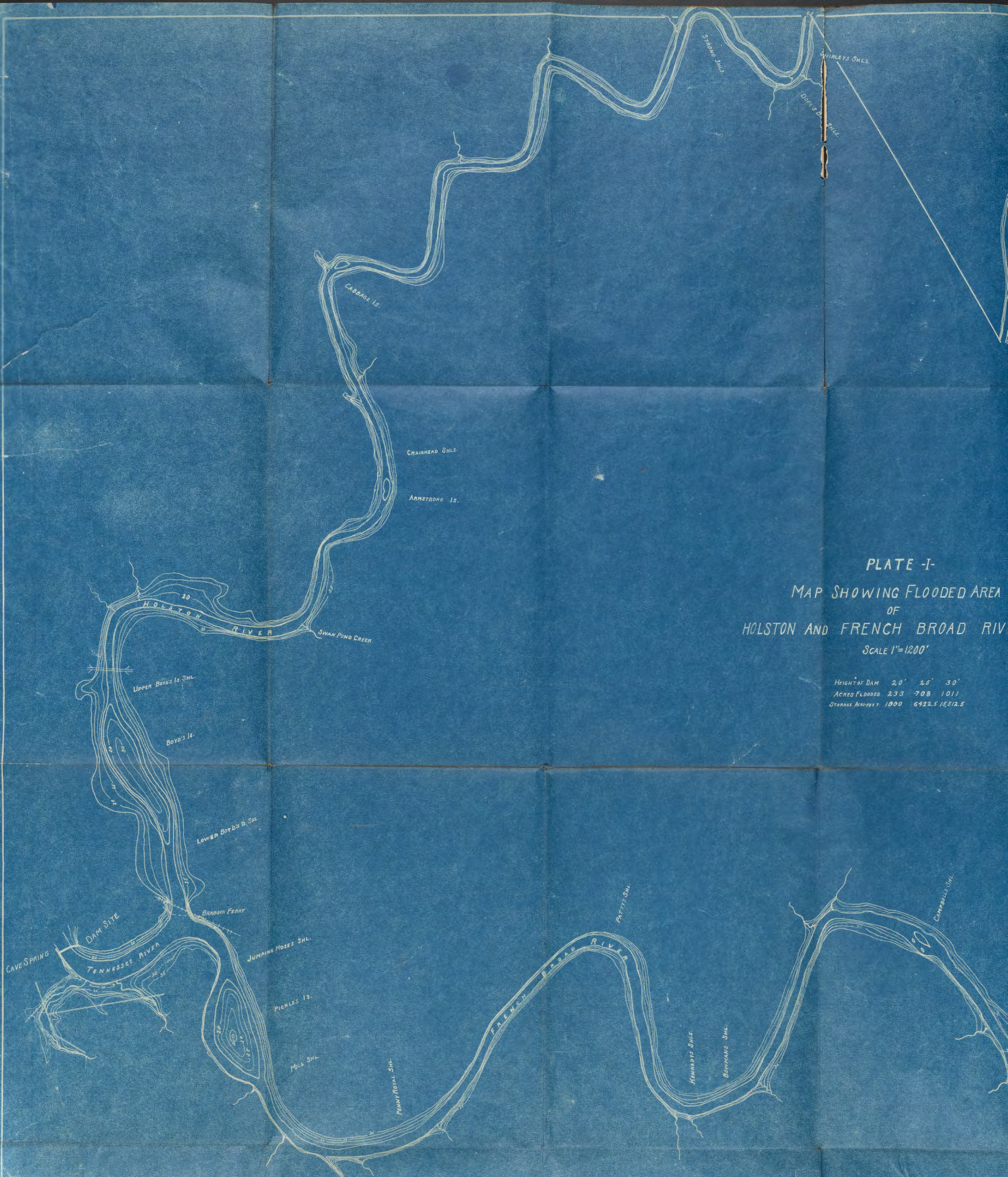


PLATE -I-
MAP SHOWING FLOODED AREA
OF
HOLSTON AND FRENCH BROAD RIV
SCALE 1"=1200'

HEIGHT OF DAM	20'	25'	30'
ACRES FLOODED	233	708	1011
STORAGE ACRES-FEET	1000	6422.5	15512.5

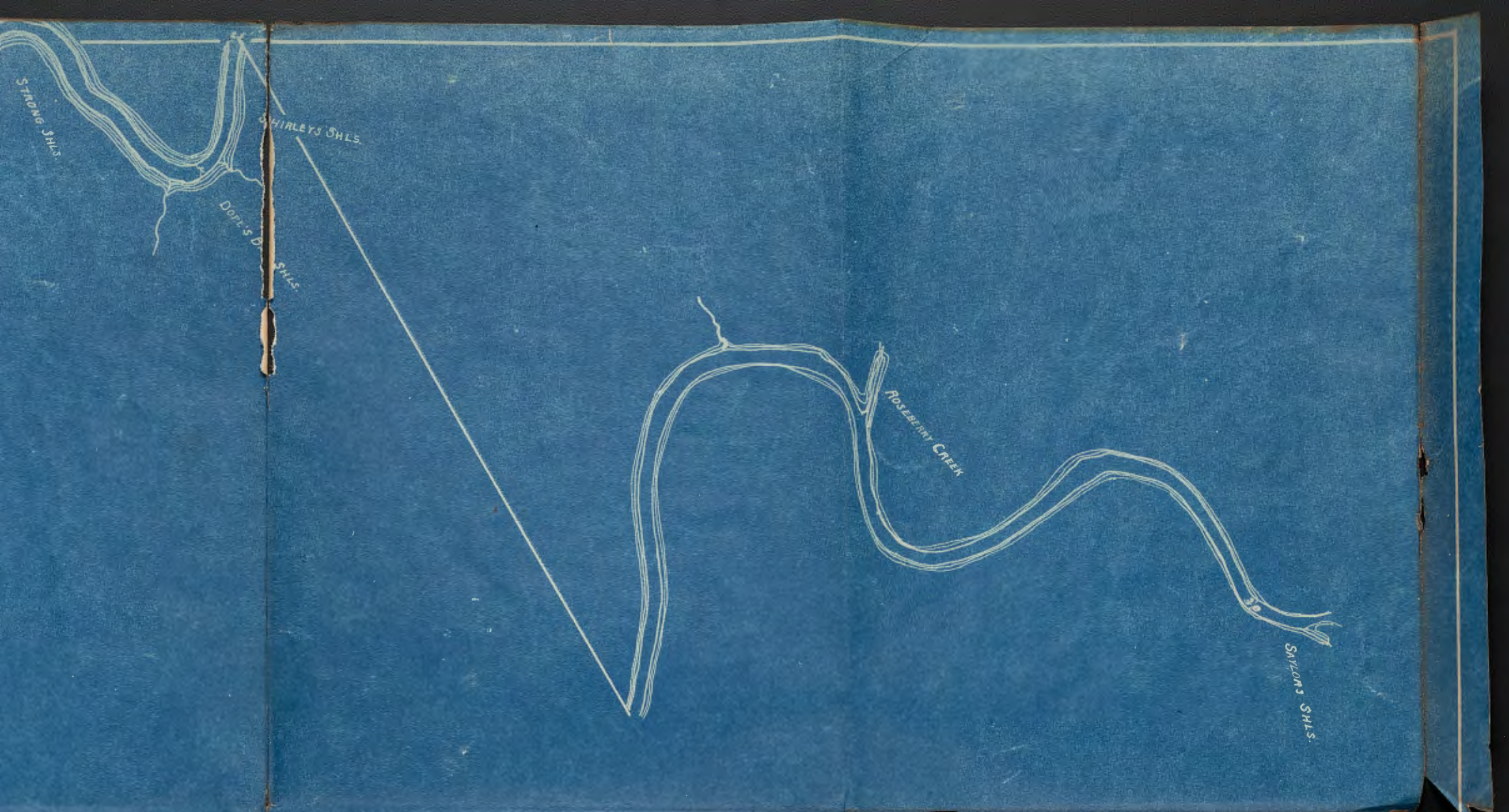


PLATE -I-
 MAP SHOWING FLOODED AREA
 OF
 HOLSTON AND FRENCH BROAD RIVERS
 SCALE 1"=1200'

HEIGHT OF DAM	20'	25'	30'
ACRES FLOODED	233	708	1011
STORAGE ACRE-Feet	1000	6422.5	15512.5



PLATE - II -
CROSS SECTION OF TENNESSEE RIVER
AT
SITE OF PROPOSED DAM

Scale - Vert - 1" = 10'
Horiz - 1" = 20'

CREST OF

CREST OF

CREST OF

10' DAM SPILLWAY-704'

25' DAM SPILLWAY 649'

10' DAM SPILLWAY 612'

WATER LEVEL KNOXVILLE GAUGE AT 2.4'

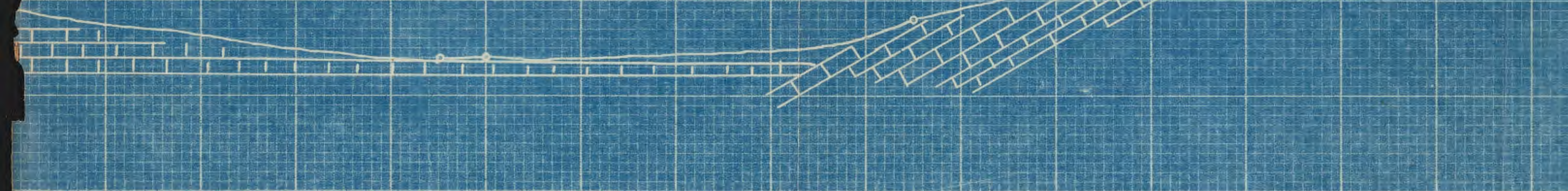


PLATE III
PROFILE OF FRENCH BROAD RIVER
 Transcribed from U.S. Survey under Major D.C. Mingsmaier 1899

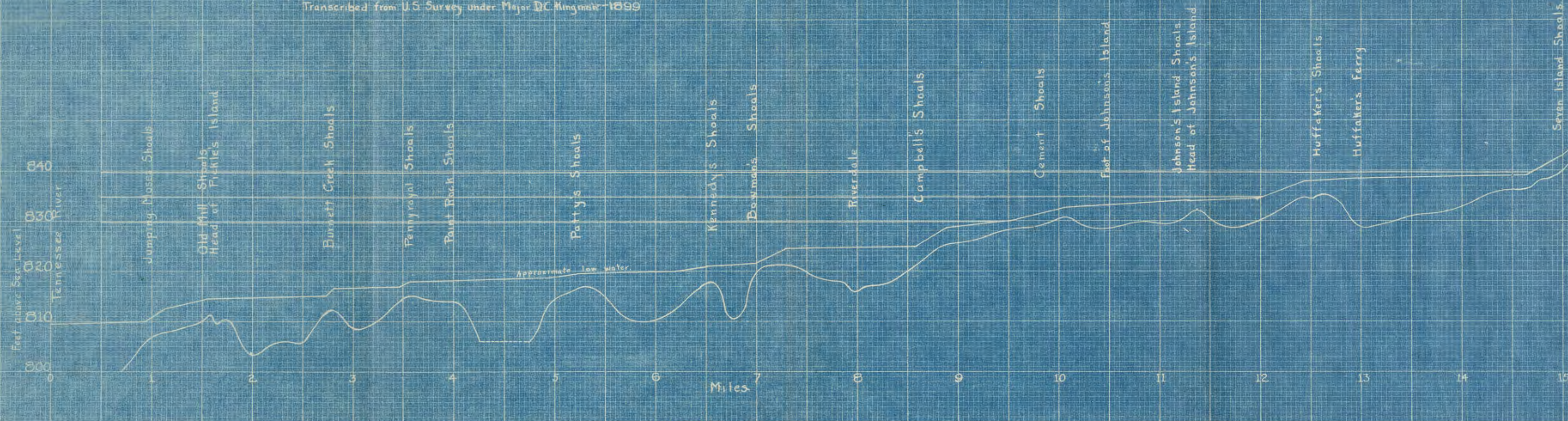


PLATE IV
PROFILE OF HOLSTON RIVER
Transcribed from U.S. Survey under Major D.C. Kingman
-1900-

