Westinghouse

Steam Condensers and Auxiliaries

INSTRUCTION BOOK

In the event that it becomes necessary to return any part of this equipment to the South Philadelphia Works, it should be tagged with the Sender's name and address and the serial number of the unit. Shipments by freight, express or parcel post should be addressed to:

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PREFACE

THE purpose of this instruction book with its supplements is to present to those interested in steam condensers and their auxiliaries a brief description of Westinghouse equipment and its operation.

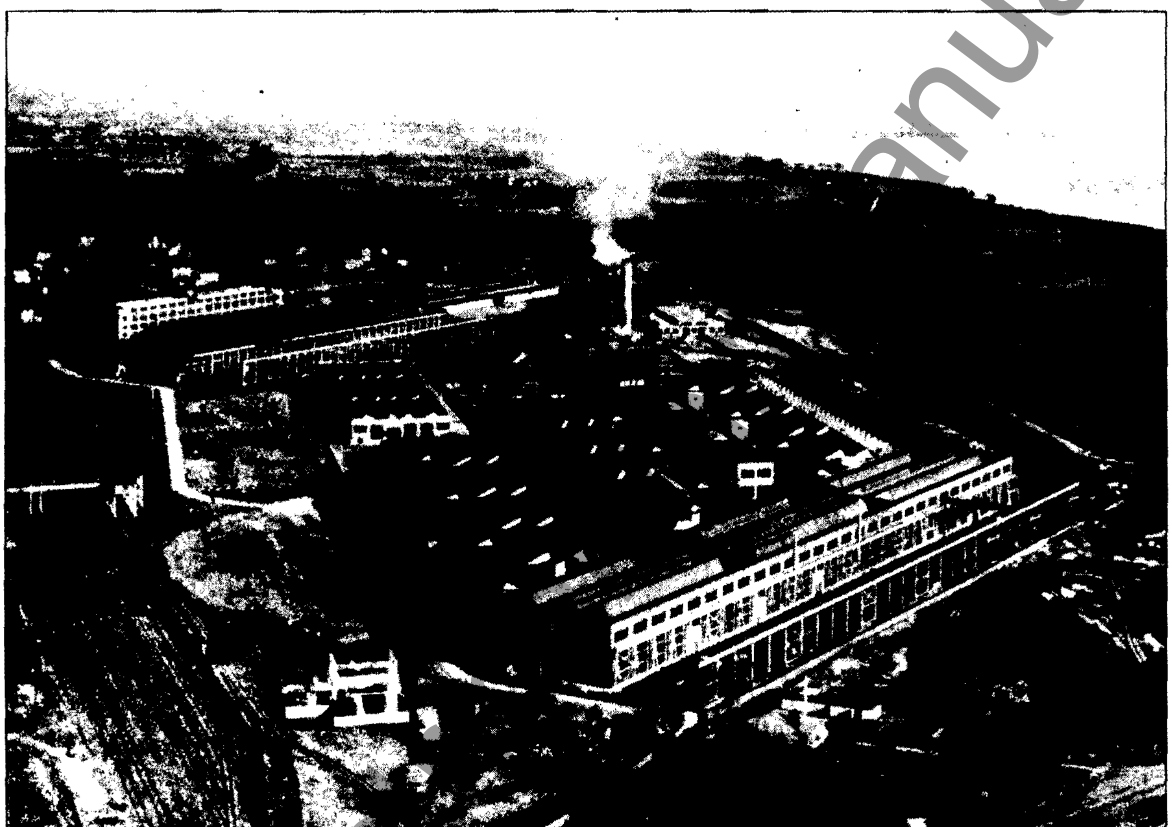
The tables, factors, formulae and calculations are presented in a simple form which requires only moderate knowledge of mathematics for their use in solving condenser problems. The concentration of information necessary for condenser calculations into a convenient form for use by the practical operator should meet with approval.

Since the information is intended for practical use the mathematical development of the formulae and factors have been omitted and only established facts are presented.

The operation and care of equipment is more or less a station problem requiring the best judgment to meet the occasion, therefore, that which is covered in this paper is to be considered in a broad and general sense. The experience of our engineers is that no two installations present the same problem, therefore, no one solution can be given.

The modern condenser may be considered a comparatively new development and it would be impracticable to enumerate the very latest of experimental features. The reader must consider that the Westinghouse equipment as described will, from time to time, incorporate the very latest features of modern practice.

It is hoped that the information herein presented will assist in the care and operation of condenser equipment and that familiarity and economy of the equipment will be gained.



THE SOUTH PHILADELPHIA WORKS

Surface Condensers

General Description

The progress of the turbine and the age of greater power plant efficiency, has pressed development of the high vacuum surface condenser. The improvement in turbine design has made it possible to utilize efficiently the high vacuum that is now practical to maintain; therefore, vacua that were originally considered high are now entirely inadequate. The development of condensers followed in line with that of the prime mover and it is of interest to note that the Company with undivided responsibilities in the complete unit desires to supply to the field the most economical equipment possible. Experiments and tests are always being conducted on extensive scales to prove the latest developments in keeping with good engineering practice.

The Westinghouse Company, builders of practically all types of condensing apparatus, have developed and perfected to a high degree the two principal types of surface condensers, the straight down flow (conventional design), and the radial flow.

STRAIGHT DOWN FLOW TYPE SURFACE CONDENSER

The straight down flow type of surface condenser is the development of our earlier design of simply applied elementary principles. The steam enters at the top and passes downward through the tubes, the condensation taking place as the steam flows through the tube nest from top to bottom. The steam passages between the tubes are proportioned to minimize the resistance and the pressure drop from the steam entrance opening to the air offtake opening. The condenser shell is usually made of cast iron. One or more steam openings are provided as the case may demand. The steam distribution to the tube nest is provided by omitting tubes from the upper segment of the tube sheet, or by a special dome construction. When it is impractical to locate the atmospheric exhaust opening in the exhaust connection, an opening is provided in the upper part of the condenser shell. In the lower part of the condenser a definite proportion of tubes are separated from the main condensing surface by a baffle of such material that will resist erosion. This section of the tube surface under the baffle completes condensation and then cools the air and non-condensable vapors. The reduction in temperature of the saturated air mixture causes a large part of the water vapor to precipitate, which reduces the volume of mixture to be handled by the air pump or air ejector. This point will be covered more thoroughly under performance.

To remove the air from the condenser an opening is provided at the highest point under the baffle on the side that will best suit plant conditions. The internal design of the tube section under baffle is such that an ample passage is provided for air flow from either end of the shell toward the point of air offtake. The internal passage reduces the external fittings that are subjected to vacuum and air leaks and also eliminates the number of pipe fittings on the side of the shell.

A condensate well of liberal proportions equipped with a water glass, for observing the submergence, is provided. The passages in the condensate well are large, permitting unrestricted flow and drainage of the condenser without submerging the lower tubes. The straight down-flow type of condenser is designed for one, two, or more water passes; however, two pass is more common. The water passes through the lower section and returns in the upper section, which is the contra-flow principle, the hottest water being the cooling medium for the area subjected to the hottest steam. The water box design is such that the water inlet and discharge openings may be placed in most any position, however, the discharge is generally located in a position to insure that the upper tubes will always be filled with water.

In many installations, especially those of larger sizes, the water boxes are made sufficiently deep so that it is possible to carry out the periodic cleaning by the use of mechanical cleaners without removing the water box covers.

UNIT TYPE SURFACE CONDENSER

The general design of the standard down flow type of condenser is used in a combination of condenser and pumps known as the unit type and is built in the smaller sizes. The circulating and condensate pumps are provided with vertically split casings, this design permitting removal of the covers and rotating parts for inspection and repair without removing the body, which in the case of the circulating pump, may be used as the support for the inlet end of the condenser. The air removal equipment may consist of a Westinghouse air ejector or hydraulic air pump. The original design of this unit contained the three pumps in one unit, which, due to structure, made it necessary to remove the entire unit from its position to conduct inspection and repairs.

THE RADIAL FLOW TYPE SURFACE CONDENSER

The Westinghouse radial flow condenser embodies many unique and ideal principles that cannot be equalled. The feature of this design is the steam belt around the tube nest, the path of the steam toward the cold central part and point of lowest pressure or highest vacuum where the air is withdrawn. The center of the tube nest is located eccentric with respect to the horizontal center line of the shell, providing a steam belt that feeds steam to the entire outer periphery of the tube nest. The path of the steam from the outer belt to the center, the point from which the air is removed, is very short; therefore, the resistance through the condenser is negligible. By removal of the air from the cold central part through specially constructed baffles, the condensate drains to the bottom through the surrounding belt of steam, which is at a temperature and pressure equivalent to that of the steam space at the top of the condenser. The condensate temperature depression is negligible due to this design, while in the conventional design it must be equivalent to the existing vacuum drop, plus the cooling due to surface conduction as it passes from the tubes on its way to the hot well.

The removal of air from the central part of the condenser establishes flow from all outward points. This flow will scavenge the air and moisture from the tubes. The baffle structure is simple in principle, however, particular attention must be paid to assembling.

The hot well is designed to meet the requirements of this type and is provided with a gauge glass to indicate the submergence level above the pump center line. Water seal pockets for taking care of pressure differences are provided, as well as baffles to prevent direct flow of air into the hot well. A thorough description will be given under condenser performance.

Earlier designs of the radial flow condenser incorporated a radial flow water principle, the colder water passing through the center of the tube nest and the warmer water through the outer tubes. The air was removed from the center with offtake from either the top or bottom of the shell. In the later type the air removal principle still exists, but the point of offtake is at the bottom only.

Special designs of divided water boxes in the larger sizes for thorough tube cleaning while operating, have been deveolped, and are in successful use. This arrangement is suggested for stations operating units continuously for long periods, or where possibility of rapid fouling exists.

The water box covers are provided with an ample number of manholes that will permit entrance for inspection and cleaning.

Openings are provided in the top or highest point of the water box for primer connections and at the lowest point for draining the boxes when not in use or at the time of cleaning.

TUBE PLATES

The tube plates of high grade rolled Muntz metal and the cast iron support plates are carefully stayed so as to maintain uniformity of shape under all operating conditions. All stay bolts are covered with suitable metal sleeves to withstand corrosive action. All bolts and parts used when neccessary, are made of non-corrosive metal composition, which insures life equivalent to that of all other parts.

TUBES

The tubes are selected with the greatest of care and are required to meet rigid specifications. The purchaser is consulted in an effort to have the tubes best suited to meet the actual plant conditions. The tube lengths are selected to give correct proportions for efficient operation; however, in stations of more than one unit an effort is made to maintain standard tube lengths so the purchaser may reduce his spare parts stock to the minimum.

The tubes are usually packed at each end with fibre packing rings tightened by Muntz metal ferrules. A shoulder is provided in the ferrules to prevent the tube from creeping out at either end. Tubes are occasionally rolled on one end and packed on the other, to allow for expansion. The water entrance of the ferrule is proportioned to reduce the entrance head loss to a minimum, which in turn reduces the pumping head.

To prevent sagging and vibration of the tubes while operating, a number of cast iron tube supports are provided. Generally two support plates are sufficient; however, the number of supports is dependent upon the tube length.

Operation and Performance

The performance of a surface condensing unit is dependent not only upon the surface condenser, but also upon the auxiliary units serving the condenser. In a brief way, an attempt will be made to cover the operation of each element of a condensing unit.

The condenser, if of the radial flow type with its characteristic steam belt supplying steam to practically the entire outer periphery of the tubes, will condense the steam and maintain a condensate temperature practically equivalent to the steam surrounding the tubes. This unique principle can easily be understood in view of the fact that the drop of pressure from outer to inner part of tube nest is very little and that the condensate must pass through the steam belt and zone of highest pressure before entering the hot well, where the pressure is the same as that of the exhaust space.

If the condenser is of the straight down flow type the steam passes from top to bottom through the tubes with a slight loss of pressure. The condensate falling through the tube nest and zone of lower pressure, decreases in temperature due to conduction of heat and decrease in pressure.

The loss in pressure through the condenser is dependent not only upon design, but upon the amount of steam, air leakage, and the vacuum. By reference to curve Fig. 1, page 7, the specific volume of steam for the different pressures can be easily determined, and it will be noted that for a change from 28 to 29 inches of vacuum the volume practically doubles. If a change of condition from that of design takes place a change in the pressure drop can be expected. With increased pressure drop a depression of condensate temperature can be expected in a downflow condenser.

The performance of the condenser may be expressed in terms of heat transfer, for this unit is the basis of design. The rate of heat transfer is dependent upon the water velocity in the tubes, the scavenging effect of the steam through the spaces between the tubes, the air removal, and the condition of the tube surfaces.

The heat transfer varies approximately as the square root of the water velocity with clean tubes, therefore, for general calculations and all practical purposes the following value, can be used with safety. As an illustration: if the heat transfer is 590 B.t.u. with 6 ft. water velocity, then with 5 ft. water velocity the heat transfer will be 538 B.t.u. When designing a condensing unit a reasonable water velocity should be considered, for this will enable a smaller unit to accomplish the work intended. This water velocity will also help to maintain cleaner tubes, due to the greater turbulence and scouring effect, which will prevent precipitation of fouling matters. It is often possible to maintain higher velocities without marked increase of power by selecting the circulating pump with the proper efficiency characteristic under normal operating conditions.

The water film with its numerous bubbles of air and non-condensable vapor surrounding the tube as condensation takes place affects the heat transfer. Therefore, the manufacturer must design the tube arrangement so that the

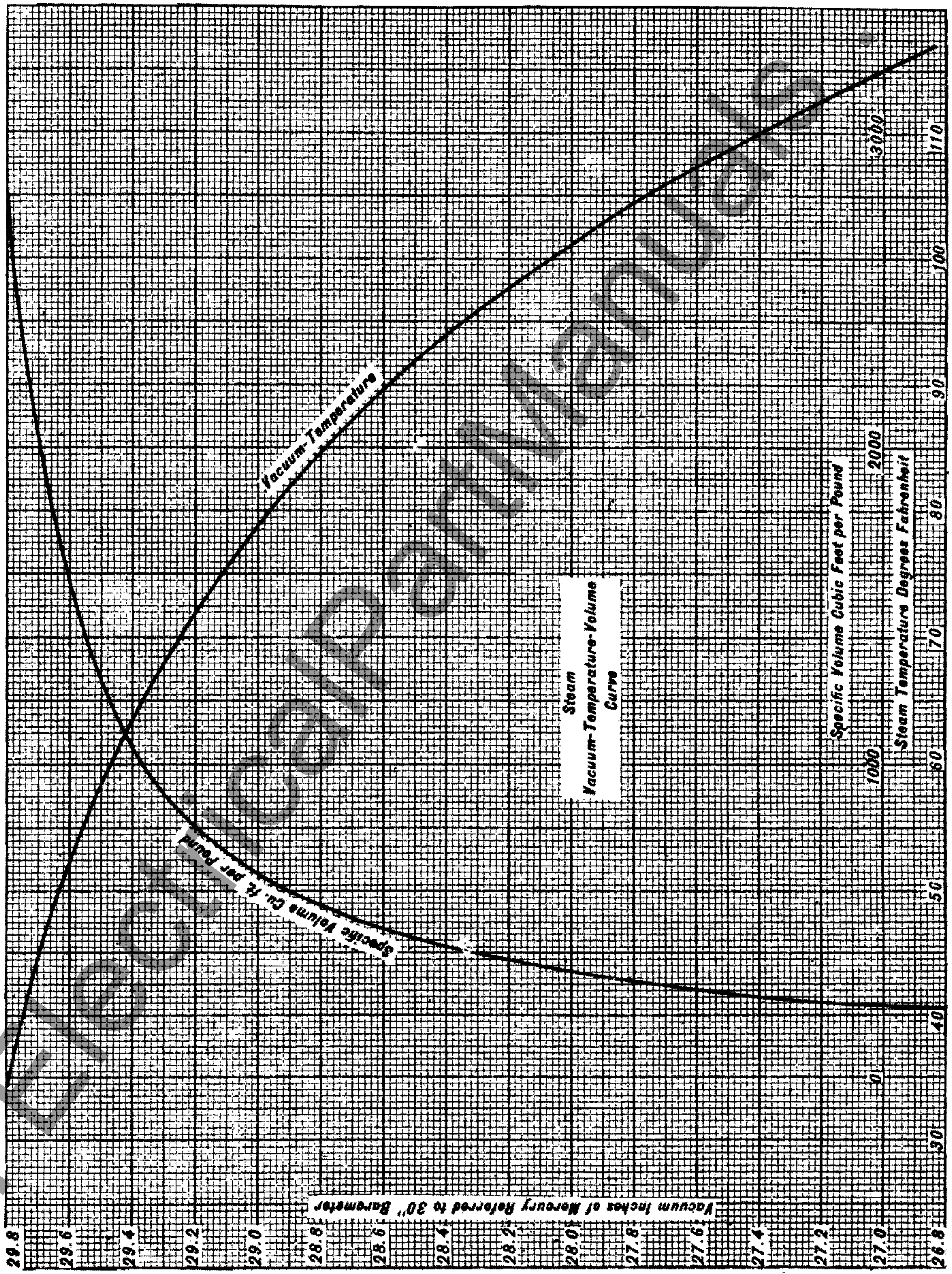


Fig. 1—Steam Vacuum-Temperature-Volume Curve

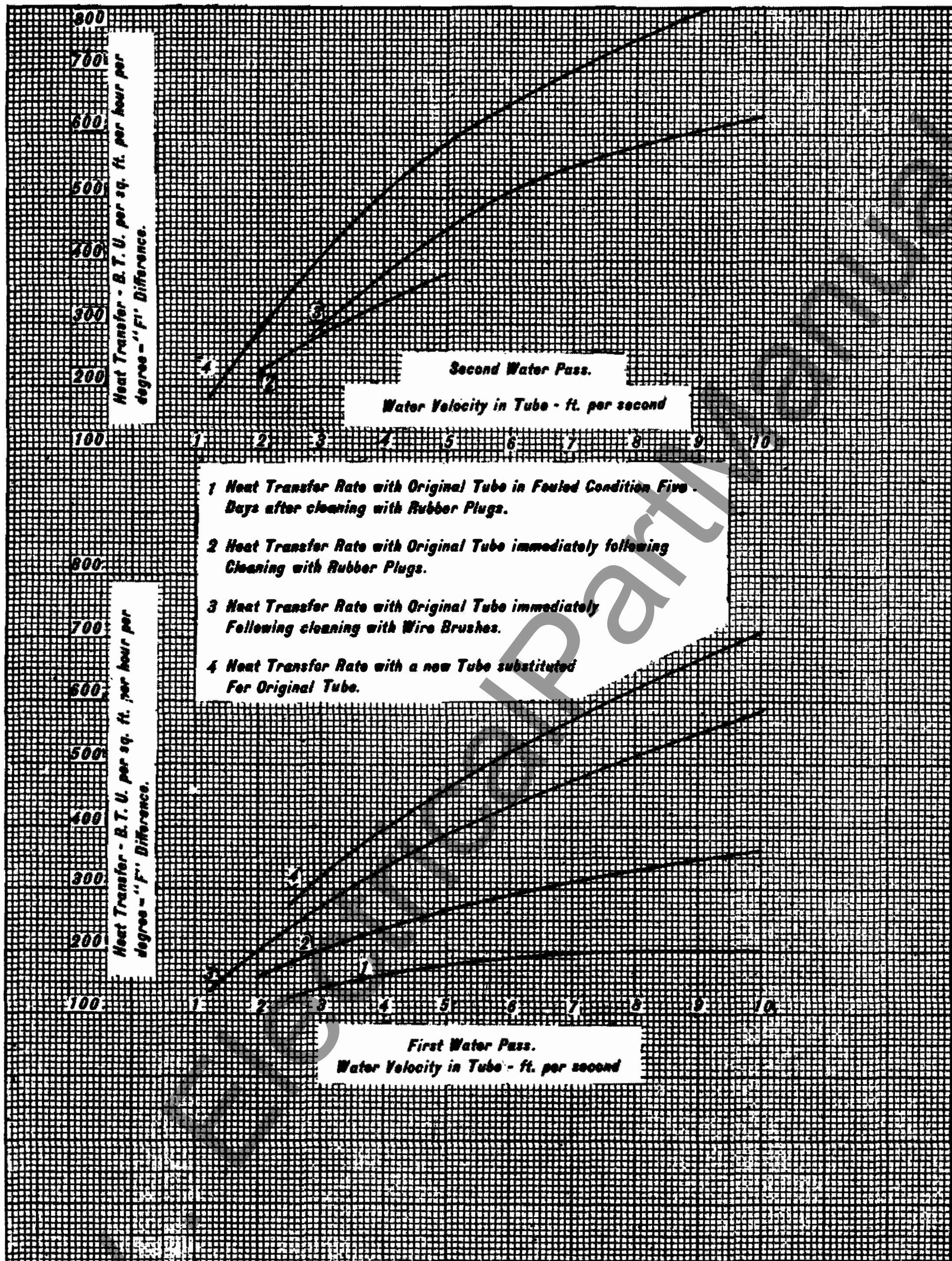


FIG. 2—CURVES SHOWING THE EFFECT OF CLEANLINESS ON THE RATE OF HEAT TRANSFER THROUGH SURFACE CONDENSER TUBES

passing of steam through the tube nest will not be restricted and at the same time have sufficient velocity to scavenge the condensing surface and keep the film reduced to a minimum. The problem of correct tube spacing, for all vacua and load conditions, presents quite a problem. Ordinarily the best condition is sought for that of normal operation.

The highest rate of heat transfer can only be obtained by the complete removal of air and non-condensable vapors from the condenser shell, a feature which was early realized by this Company and was accomplished by the use of the Westinghouse LeBlanc hydraulic air pump. Later developments also proved the steam jet air ejector a very satisfactory air removal apparatus. Air removal is imperative, for air collected near the point of air offtake forms a blanket preventing steam vapors from entering the affected section. This section then becomes inactive and causes a reduction of overall heat transfer.

One of the most important conditions affecting the heat transfer is that of tube cleanliness, which so often seems to be disregarded. It is an economic necessity to maintain both the interior and exterior surface of the tube in a condition of cleanliness as near to that of a new tube as operating conditions will permit. Reference to curve Page 8 (Fig. 2) will show the results of a test conducted under normal operating conditions on a single tube specially arranged in a large condenser for determination of the effect of cleanliness and water velocity upon the heat transfer.

The change of relative results is so surprising that one cannot but realize the advantage of the clean tube. Taking 6 ft. water velocity as a base, it will be noted that the heat transfer obtained is 190 B.t.u. for the fouled tube, 437 B.t.u. for the same tube after thorough internal cleaning with wire brushes, and 520 B.t.u. for a new tube, a percentage increase of 230 and 274% respectively. Such a relative comparison could be established on units in operation at little cost for labor and material, and with such information the operator would have some idea of what could be expected by cleaning the tubes.

A condition which very often affects the performance and which seems least to be considered is that of excessive free air leakage into the condensing system. The methods used in search for leaks are often so crude that the time spent could be considered an absolute waste. For instance, the candle flame method should be associated with the days of the reciprocating engine and 25″ of vacuum, and not with the present day turbine and 29″ of vacuum. Practice has dictated that a water or air test be conducted with the greatest of care to make the condensing systems tight.

In very large units where it is impractical to subject the shell and foundation to the weight and pressure of the water a compressed air test is used. The condenser and turbine or the vacuum system is subjected to an air pressure of approximately 5 to 10 lb. gauge, and all exterior surfaces painted with a concentrated soap solution. Observation at the time of painting for leaks in the form of bubbles or escaping air jets should be made. This method, carried out properly, has given very satisfactory results.

In making tests for air leaks it is not only necessary to see that the condenser shell is tight, but to be assured that all parts of the entire system that may be under vacuum at all load conditions are tight. Air leaks are very common in poorly packed valve stems that may withstand water tests due to swelling of the packing at the time of tests. Leaks are also common in the turbine casing, glands, drains, equilibrium pipes, etc. Many of the modern plants have provided gasometers to enable the operators to determine the air leakage and assist their organization in maintaining leakage below a set standard. Much could be said regarding air leakage and methods of detection, but each unit presents a different problem. Therefore, it is necessary for the operator to establish a correct procedure of search. Without any doubt the statement that eternal vigilance is the price of freedom from air leakage expresses the efforts necessary to maintain a tight condensing system.

Jet Condensers

General Description

Three types of jet condenser are manufactured by the Westinghouse Company and are known as the low level jet, barometric and ejector condenser. The low level jet is in greatest demand. The condensation of the steam in any of the above types is accomplished by direct contact with the cooling water. The jet type of condenser is generally installed in small central stations and industrial plants where reasonably good boiler feed water can be obtained at small expense. The first cost and maintainance is usually less than the surface type of condenser.

LOW LEVEL JET CONDENSERS

The low level jet condenser construction consists of inlet spray chamber condenser body, air and water remova pumps. The air is evacuated by either an air ejector or a hydraulic air pump, each of which is described in a separate supplement. Either a condensing or a non-condensing type ejector may be used depending upon the amount of steam required for heating purposes. The condensing type ejector is built with either jet or surface type intermediate and after condenser.

The elevation at which the condenser is located is governed by the intake water level, for, it is well understood, the water is admitted to the condenser by virtue of the difference of external and internal pressure. The location must be such that the pipe and nozzle friction plus the static lift of the water shall not exceed the minimum vacuum that may normally exist. To standardize the injection lift, this company has established 18 ft. as the maximum static lift measured from the water line to the centerline of the condenser injection opening, based on 30" barometer, and 2 ft. pipe friction.

The removal pump is designed to remove the condensed steam and the cooling medium admitted to the condenser. Common practice is to design the pump for a specified external head, the builder making allowance for the suction head imposed by the vacuum in the condensing space. It is very important that correct pumping head be determined at time of purchase for capacity-head characteristic largely determines the performance.

THE BAROMETRIC CONDENSER

The barometric condenser in principle is nothing other than the low level jet condenser so arranged that the water flows from the vacuum space to the atmosphere by gravity, thereby eliminating the use of the water removal pump. The spray chamber is usually arranged so that the water will entrain the air with it as it flows from the condenser.

The barometric condenser is usually applied to installations where a natural water supply with sufficient head is available or in industrial plants where a large quantity of water is pumped for process work.

In order to obtain performance equal to that obtained by a low level jet condenser, an air pump must be used. The air pump can be either the hydraulic, or air ejector type as outlined for the jet condensers.

The installation requires a great amount of piping, but little upkeep or operating expense. The condenser is usually located outside of the building, thereby occupying little space in the plant proper.

THE EJECTOR CONDENSER

The ejector condenser built by the Westinghouse Company is a combination of the hydraulic air pump and condenser, which condenses the steam and removes the condensation from the vacuum space. The unit is built only in the small sizes and applies to installations where the small jet condenser cannot be used economically.

Performance of Jet Condensers

An approximate determination of jet condenser performance is quite simple in comparison to that of the surface condenser and with comparatively few readings a very complete analysis can be made.

It is necessary that the water levels, head, temperature, load, etc., be as specified. These can be easily established by observation and then compared with the contract specifications.

The heat units (B.t.u.) in the steam to be absorbed by the cooling water are determined in the same manner as in surface condenser installations.

The ratio of water to steam is determined by dividing the B.t.u. per pound of steam by the temperature rise of the water, degrees Fahrenheit.

The steam consumption of the turbine cannot be accurately measured after leaving the turbine as in a surface condenser installation. Therefore, it is necessary, when accurate determination is desired, to isolate the necessary boilers so as to supply steam during the test to the turbine only. The feed water supplied to the test boilers during the test period should be accurately weighed or measured. A boiler leakage test should be made both before and after each turbine test and the average quantity of these two leakage tests should be deducted from the steam measured during the turbine tests.

Precautions such as ascertaining that valves are absolutely tight, etc., must be taken to assure that steam or water cannot escape from or enter the isolated boilers or steam system, other than what is being supplied for the turbine under test.

Where accurate dimensions of the nozzles of the impulse elements are known, the steam consumption can be calculated providing accurate pressure and temperature of the steam at the inlet of the nozzle is determined. A nozzle flow co-efficient of 95% should be used in calculations.

Approximate methods such as flow meters and corrected guarantee water rates can be used, but are not recommended where accuracy is desired.

The quantity of condensing water in pounds per hour is determined by multiplying the pounds of steam per hour by the ratio. Methods of measuring the discharge by weighing, venturi meter, calibrated nozzles, wiers, Pitot tubes, etc., can be carried out, but up to the present this does not seem practical. Very few installations could adopt such methods for measuring the water due to short piping and bends.

If the analysis shows the proper amount of water introduced into the condenser then it is only necessary to arrive at the air removal capacity of the pump and the proper spraying of the water.

Usually if the air pump by test is stable its capacity will be satisfactory. However, an orifice test will determine the approximate capacity. Generally the air handling capacity is quite liberal and no troubles are experienced provided the condensing and piping systems subjected to vacuum are reasonably tight.

The spraying or mixing of water can be easily checked, for it is only necessary to make a thorough inspection and cleaning of the spray nozzles and the annular water chamber for debris and corrosion. A check of the pressure difference across the nozzles when operating may supply sufficient information to form a definite conclusion.

The three major factors needed to make a correct analysis in order to obtain satisfactory operation are: quantity of steam, quantity of water, and capacity of air removal equipment.

The quantity of steam is dependent upon the pressure and quality of the steam, the vacuum, load, and the condition of the nozzles and blading of the prime mover.

The quantity of water may be limited by two conditions in a jet installation: first, flow to the condenser, either by excessive lift or by obstructed or incorrect nozzles; and second, improper removal, due to excessive head, incorrect speed of pump, obstructed or eroded impeller, or increased seal ring clearances.

The air removal is dependent upon the mechanical and operating conditions of the apparatus. If the air pump is used, its capacity is dependent upon water temperature, suction lift, discharge head, speed, and condition of parts. If an ejector is used, its air removal capacity is dependent upon pressure and quality of steam, back pressure, nozzle condition, quality of vapor mixture handled, and the temperature and quantity of water used in the intermediate condenser, if the ejector is of the condensing type.

The air removal apparatus is designed for a given capacity, therefore, it is absolutely essential that the vacuum system be tight. Excessive air leaks often mislead the investigator; therefore, a safe method of procedure is to subject the apparatus to water tests for air leakage.

A simple method often used for analyzing condenser performance, after the steam and water have been considered, is the air pressure in the condenser which in satisfactory operation is approximately three tenths (3/10) of an inch of mercury. It must be considered that air removal equipment has definite rated capacities at various vacua; therefore, unless the correct capacity is provided it cannot be expected to obtain correct performance. This condition is not serious with the hydraulic air pump for the characteristic is such that it will take care of the change in vacuum. Steam jet ejectors and displacement pump capacities decrease very rapidly at the higher vacua; therefore, the relative performance will not be the same for all vacua.

It is impossible to go into detail giving all of the possible troubles, symptoms, and corrections in connection with jet type condensers in such an article. With the fundamental ideas as covered, it is assumed that the locating and correcting of discrepancies can be carried out with success.

Location of Gauges and Method of Taking Condenser Readings

In taking readings to determine the condenser performance care must be taken that gauges and thermometers are in the proper place.

Referring to the jet condenser performance sheet for the following reference figures, pages 29 and 30 these precautions should be observed.

The main injection gauge "10" should be placed between the condenser and the injection valve. The gauge "10" should be placed as close to the condenser as practicable, and the thermometer "7" as close to the gauge as possible.

In general it is desirable to have the thermometer and the gauge as close as possible for any one reading.

The air pump suction gauge should be placed between the air pump suction valve and the air pump, or on the air pump body where a tap is provided for it. Again the air pump suction gauge "17" and the thermometer "16" should be as close as possible. The discharge pressure gauge "15" should be placed as close to the condenser discharge pump as possible and to get accurate readings this gauge should not read to more than 30 lbs. The vacuum in the condenser should not be measured by a gauge as it is not sufficiently reliable or sensitive for accurate work. A mercury column should always be used and should be tapped into the condenser head. The thermometer "5" for measuring the temperature of the exhaust steam should be placed on the top of the inlet chamber of the condenser, diametrically opposite the exhaust steam entrance so that no errors will be introduced by heat being conducted from the turbine. The thermometer should be placed as close to the mercury column connection as possible.

The barometer reading should be taken in the engine room or from the nearest barometer that is available. The elevation must also be taken into account, as a difference of altitude of 100 feet makes a difference of approximately 1/10 of an inch of mercury in the pressure of the atmosphere.

In taking readings on a condenser it is desirable that they be taken simultaneously particularly in the case of the fluctuating load.

It is possible to get the vacuum at the condenser steam inlet by either the mercury column or the absolute pressure gauge. The most satisfactory and reliable method is by a mercury column.

Submergence is the height of the level of the water in the condenser body in inches above the centerline of the water pump while running. It is necessary to carry the water in the condenser body up to full submergence in order to keep the pump runner passages full of water and to get full capacity out of the water pump. **Always take the performance readings with full submergence.**

Vapor tension and air tension are two terms that are often confused. Vapor tension is the absolute steam pressure in the condenser measured in inches of mercury. Air tension is the term used to express the absolute pressure in the condenser due to the presence of air. The following example shows how the values may be obtained.

Example: Discharge water temperature 96°F. In the steam table we find 96° gives a vacuum of 28.2892″. Subtracting 28.2892″ from 30″ gives 1.7108″ vapor tension. Note—that this value may be obtained from the table directly in column headed INCHES of MERCURY at 58.4°F. Exhaust steam temperature is 101°. In the steam table 101° gives a vacuum of 28.0095″, and subtracting this value from 30″ gives 1.9905″ combined air vapor tension. Subtracting the vapor tension 1.7108″ from the combined air and vapor tension 1.9905″, gives the air tension .2797″.

Comparing the air tension for different conditions, such as different loads or different quantities of water from the same load is a good way to check the performance of a condenser with regard to how well the air is being removed. Air tension in excess of .3″ is due to air leaks or low air pump capacity.

The speed of the pump measured in revolutions per minute is a very valuable reading to be taken, since the pumps are designed for a certain operating speed, their efficiency will be decreased if they are not operated at the contract speed. Hence this speed should be checked and maintained throughout the test.

Starting The Jet Condenser

Case 1—When there is an exhaust valve between the turbine and the condenser, but when there is no supply of water under pressure for priming the condenser.

Case 2—When there is no exhaust valve between the turbine and the condenser, and when there is no supply of water under pressure for priming the condenser.

Case 3—When there is no exhaust valve, but when there is a supply of water under pressure for priming the condenser.

Hydraulic Air Pump Equipment

Case 1

With the exhaust valve between the turbine and condenser closed, open the main injection valve to the condenser a few turns and open the air pump suction valve wide. When the pumps are up to speed, turn on the water seal to the condenser pump shaft glands and open the steam primer. The vacuum created by the primer will immediately bring water to the air pump, the primer should be opened wide and kept so for one-half a minute even though the air pump has got its water. This primer will quickly produce a high vacuum in the whole installation.

After the air pump has once got its water, it will easily pull the vacuum up the rest of the way, but not quite as rapidly as with the use of the primer.

When the vacuum has risen to about 25″ or 26″ the main injection valve which was only slightly opened, should be opened until the condenser is circulating the proper amount of water. The exhaust valve can now be opened and the main unit which has been warming up or running non-condensing may exhaust into the condenser. The atmospheric relief valve, of course, should be closed and its water seal turned on. When the vacuum has been obtained, the air pump suction may be throttled down to such a point that the needle on the air pump suction gauge stands at the red mark on the dial. This dial is marked at the proper point as indicated by the test of the condenser made at the shop.

In installations where there is an excessive air leak, it may be necessary to open the air pump suction, further to maintain the maximum vacuum, thus giving a lower reading on the air pump suction gauge. The power taken by the air pump is proportional to the amount of water allowed to pass through it. It is, therefore, desirable to reduce the air leakage as much as possible, thereby reducing the amount of water required by the air pump and reducing the horsepower required for its operation.

Case 2

There are some installations in which no valve is provided between the turbine and the condenser.

In such cases the condenser cannot be started when the turbine is in operation, since the condenser would be filled with exhaust steam. Since water cannot be drawn into the condenser until a partial vacuum exists, and the vacuum cannot be produced until water is present to condense the incoming steam, it can be seen that it is impossible to produce a vacuum in the condenser when the turbine is exhausting steam to the condenser. Therefore, it is necessary that the turbine be shut down when the condenser is being started and furthermore that the glands on the shaft must be tight against air leakage. The procedure of starting will then be the same as for Case 1.

Case 3

When there is no exhaust valve but when there is a supply of water under pressure for starting the condenser. With no gate valve between the main unit and the condenser and the main unit operating non-condensing, the condenser would be full of steam as described in Case 2. However, with the injection water under pressure a vacuum may be created quickly.

The injection water which is under pressure is usually supplied for a few minutes only when starting the condenser, for after the vacuum is once established the main injection line may be opened and the pressure line closed. The main injection line valve is closed in starting up until the vacuum is established and then opened before the pressure line is closed.

In case the air pump discharges into a pit and the end of the air pump diffuser is not submerged, it will be necessary to open the steam primer before opening the injection, as otherwise air would immediately travel back to the air pipe and no initial vacuum could be established by the injection water.

Jet Condensing Air Ejector Equipment

Case 1

With the exhaust valve between the turbine and condenser closed, open the main injection valve to the condenser a few turns.

The turbine is started exhausting to the atmosphere through the relief valve.

The cooling water is circulated through the precooler and the after condenser. If a valve is provided in the exhaust line between the second stage ejector and the after condenser it must be opened before steam is admitted to the second stage ejector.

The ejector creates a vacuum in the condenser shell and the main injection valve which was only slightly opened has allowed some water to be drawn into the condenser.

The water seal is turned on the condenser pump shaft glands and the pump brought up to speed when the main injection valve is opened sufficient to circulate the proper amount of water.

The exhaust valve between the turbine and the condenser may be opened when the second stage ejector has obtained the highest vacuum it is capable of carrying. The atmospheric relief valve is closed and its water seal turned on.

The cooling water is circulated through the intercooler and the first stage ejector is started by opening the steam inlet valve.

Case 2

See Case 2 for Hydraulic Air Pump Equipment.

Case 3

With no exhaust valve between the main unit and the condenser and the main unit operating non-condensing, the condenser would be full of steam as described in Case 2. However, with the injection water supplied under pressure a vacuum may be created quickly.

The injection water which is under pressure is usually supplied for a few minutes only when starting the condenser, for after the vacuum is once established the main injection line may be opened and the pressure line closed. The main injection valve is closed in starting up until the vacuum is established and then opened before the pressure line is closed.

The procedure of starting the air ejectors is the same as described in Case 1.

Information for Jet Condenser Installations

To obtain satisfactory operation and performance with the jet type condenser, the following suggestions should be carried out:

Injection:

Injection lift including pipe friction should not exceed 20 ft. with barometer at 30 inches. Approximately 1.1 ft. reduction should be made for every 1 inch decrease of the barometer. With long injection lines or lifts, less than 8 ft. static, an automatic shut-off valve should be installed to prevent flooding of the condenser and damage to the turbine blading. In all cases a gate valve is necessary for regulating the injection water supply.

Injection Water Piping:

The injection water piping should be of cast iron with bolted flanges, rather than bell and spigot lead caulked pipe, which in many cases has proven unsatisfactory. This pipe line should be tested for air leaks and made absolutely tight, for any leakage into the line while under vacuum will be a direct leakage into the vacuum system, and will be the cause of reduced vacuum. The injection line should be as short as possible, tunnels being most satisfactory. However, where long lines cannot be avoided, they should be installed at a very low level, even below the water line to reduce the air leakage to the minimum.

Forced Injection:

When valves are not provided between the turbine and condenser, forced injection is necessary for priming. The amount of water needed for forced injection is as follows:

Per Cent Load on Prime Mover	Per Cent of Normal Amount of Condensing Water
100%	50%
50%	25%
No Load	10%

Discharge Water Piping

The discharge water piping should be sufficiently large to prevent excessive friction head brought about by corrosion. A check valve should be provided to prevent return flow in case of shut down of pumps; and when the discharge water level is above the pump, a gate valve should also be provided for complete isolation of the machine in case of repair to check valve or main unit. The discharge line should be free from pockets and unnecessary short bends. When bends are necessary nothing shorter than long radius ells should be used.

Automatic Shut-off Valves

The automatic shut-off valve should be installed in the injection line when the lift is less than 8 ft. or where the pipe line is of great length. The valve is controlled by the vacuum breaker and its purpose is to stop the flow to the condenser. Auxiliary to the shut-off valve is installed a relief valve or air cushion chamber for relieving or absorbing the shock caused by the sudden stopping of flow.

Circulating Pump

Description and Operation

The circulating pumps serving condensers are designed and developed for the purpose of obtaining the best efficiencies that are economically possible. The selection of pumps is of great importance therefore, a design with characteristics meeting the plant conditions is always to be considered.

Westinghouse circulating pumps are single stage with either one or two impellers as the capacity requires. The suction and discharge openings are in the lower half or base permitting the removal of the cover without interference.

The vacuum of a condenser is affected by the water temperature, the temperature rise, and the water velocity through the tubes. Therefore, it is necessary to make proper selection of the circulating pump and the number of water passes in the condenser.

The capacity of a circulating pump is dependent upon the head, which is approximately 20 ft. in a well designed syphon system. It has been quite necessary to have a head characteristic that will not be affected by slight varia-

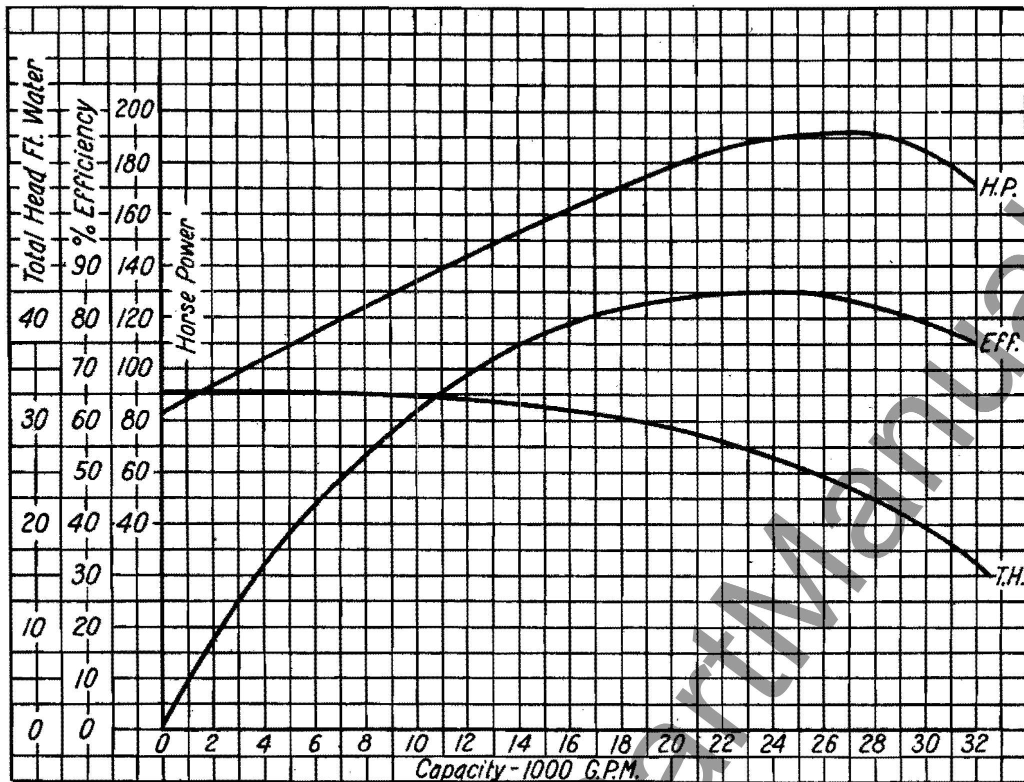


FIG. 3 —CHARACTERISTIC CURVE OF CIRCULATING PUMP

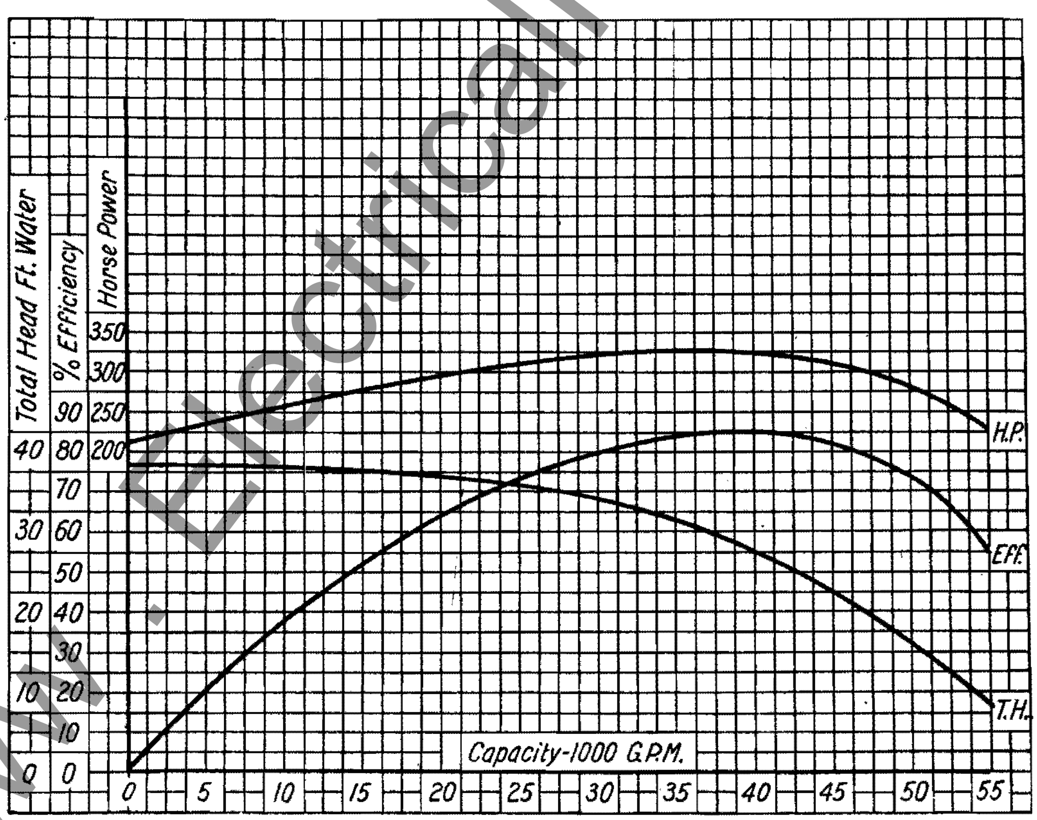


FIG. 4—CHARACTERISTIC CURVE OF CIRCULATING PUMP

tions caused by collection of foreign material on the tube ends, or a change of head due to increased friction.

From curves Figs. 3 and 4, the characteristics for two of the large size pumps usually furnished with large condensers, it will be noted that the efficiency obtained is well within that of general practice and covers a wide range of head conditions. The characteristics of the smaller pumping units are selected to meet the requirements of each particular installation.

The operating principle of the circulating pump, being one of displacement aided by atmospheric pressure or the force of gravity, is so well understood that no description should be necessary. The successful operation of the pump is greatly dependent upon the location with reference to water level and the tightness of the piping system, particularly where the syphon system is employed. The pump should, in all cases, be located so that the suction lift will be the minimum, a condition that will give the greatest efficiency and capacity with the least amount of operating difficulties. General practice has dictated 12 ft. as the maximum lift, however, some special cases have exceeded this amount, but are not recommended. For the best operation, it is desirable to have the greater proportion of the total head on the discharge side of the pump.

The suction and discharge lines should be flanged pipe free from air leaks with ends liberally submerged and of bell shape, in order to obtain the full syphon effect and reduce the entrance and exit losses to the minimum. Bell and spigot joint piping with lead caulked joints should not be used on account of the detrimental effect of air leakage upon capacity, efficiency, head, and operation. The saving in first cost with bell and spigot joint piping often will be counteracted by slightly poorer performance over a long period.

The troubles experienced with circulating pumps are usually of a very minor nature and easily corrected, unless such troubles are due to defective suction and discharge pipe lines as just explained. Very often foreign material finds its way into the inner openings of the impellers or into the suction piping, causing reduced capacity or so affecting the water flow and the efficiency that the power consumed becomes excessive. In view of the detrimental effects of foreign material to impellers, performance, and operation, the cleaning of tunnels cannot be too carefully carried out when starting new stations. The tunnels should be so covered that after once in operation no foreign material can enter, and by repeated inspection of the pump whatever material that does find its way into the impellers is removed.

The smaller particles that pass through the circulating pump and lodge in the tubes should be removed at first opportunity and as frequently as possible, for the additional head imposed will automatically decrease the water capacity, which results in poorer condenser performance.

It is very important that the correct heads be determined before installation, for if not correct the replacement of the rotating element with one for greater head may later be necessary. When doubtful of pump capacity the first procedure, after inspection is made for foreign material, is that of determining the actual pumping heads and water levels, which should then be compared with the conditions contemplated. The head should be accurately determined by calibrated gauges and corrected to the pump centerline. For low head determination the use of separate mercury tubes, or a differential mercury tube which requires no correction is more desirable.

The approximate laws of a circulating pump are that the capacity varies directly as the speed; the head as the square of the speed; and the power as the cube of the speed, for small variations. Therefore, it is very imperative to maintain the correct speed and report same at time of tests or investigations.

In small installations where foot valves and screens are attached to the submerged end of the suction pipe, frequent inspections of the valves and screens should be made. Inspection of the suction well should also be made for often the material that surrounds the screen falls back when the unit is shut down.

Frequent cases have developed where, after cleaning the strainers of the foot valves, no better results were obtained. However, after cleaning the suction well of such material that would usually fall back when the unit was shut down, the trouble was entirely eliminated.

Condensate Pump

Description and Operation

The varying conditions of operation that are met in removing the condensate from the condenser has brought about the development of a complete line of pumps that will suit most any case of power plant operation. For pumping heads generally experienced the single or multiple impeller single stage is recommended. For heads in excess of 150 feet multiple stages are used. Practice dictates that the condensate pump be designed for capacities somewhat in excess of that of maximum load; however, when two pumps are supplied to each unit the capacity need not be as liberal, for in cases of emergency the second pump may be operated.

The suction chamber of the pump is vented to the condenser at a point where the vacuum is equal or exceeds that of the part of the condenser from which the condensate is being removed. Previous practice of venting to the air line is satisfactory only in certain cases, therefore, precaution should be taken.

The condensate is usually removed from the condenser shell by a centrifugal pump known as the condensate pump. For the large size condenser either a single or multiple stage condensate pump can be used; the choice depending entirely upon the conditions of application.

The single stage pump, either of single or multiple impeller design, is capable of handling desired capacities, but at moderate heads only.

The multi-stage pumps are designed for high head conditions that may be due to plant arrangement or to the varied systems of heating that are now being adopted. Either type of pump has characteristics that are quite suitable to the application. Curve Fig. 5 shows the characteristics of a two stage pump intended for high head conditions. The condensate pump is usually designed to operate at a constant speed to meet definite head conditions that seldom vary. The speed should be the first and the head the second point to be checked when operation is in question. When capacities are questionable, the inspection of rotating parts for wear and foreign material should be made before conducting or submitting information for comment.

A thorough test for tube leakage should also be conducted when con-

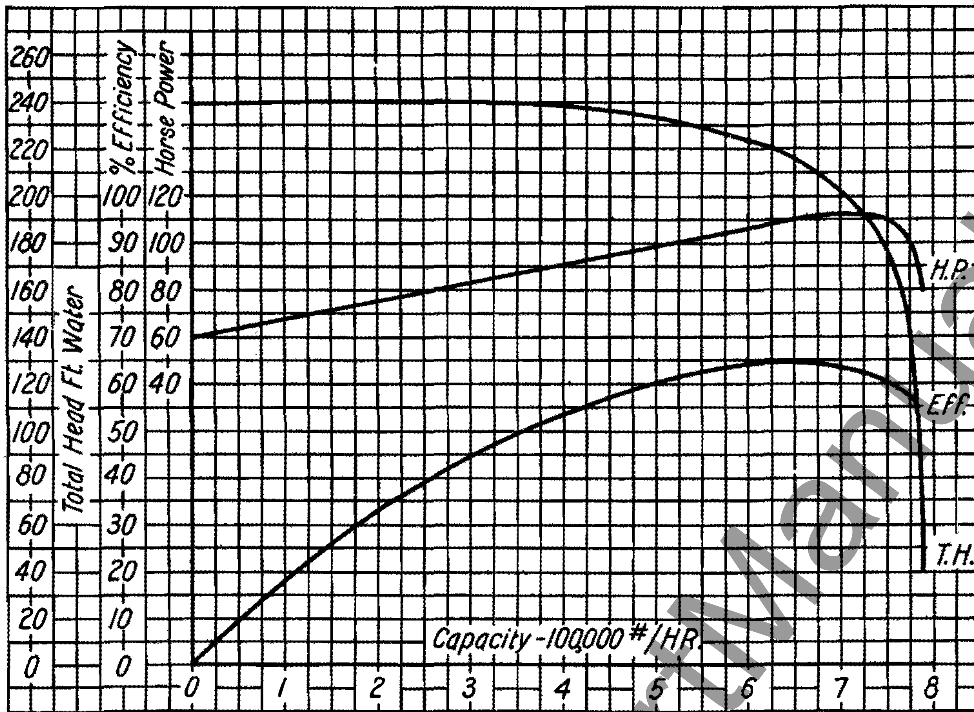


FIG. 5—CHARACTERISTIC CURVE OF A TWO-STAGE CONDENSATE PUMP

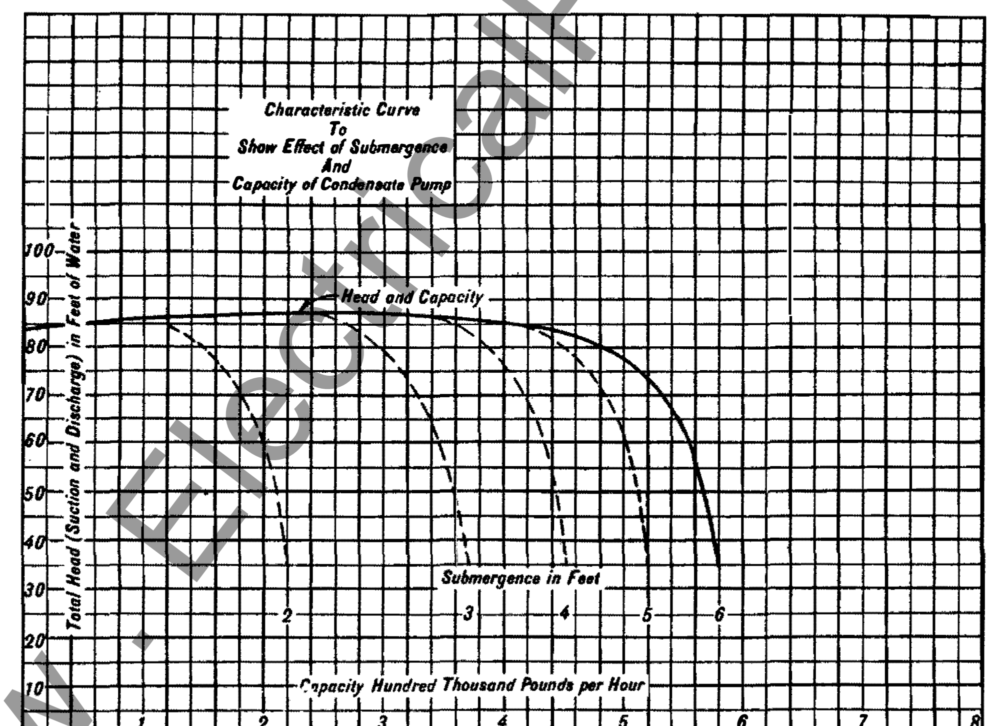


FIG. 6—CHARACTERISTIC CURVE SHOWING EFFECT OF SUBMERGENCE TO CAPACITY OF CONDENSATE PUMP

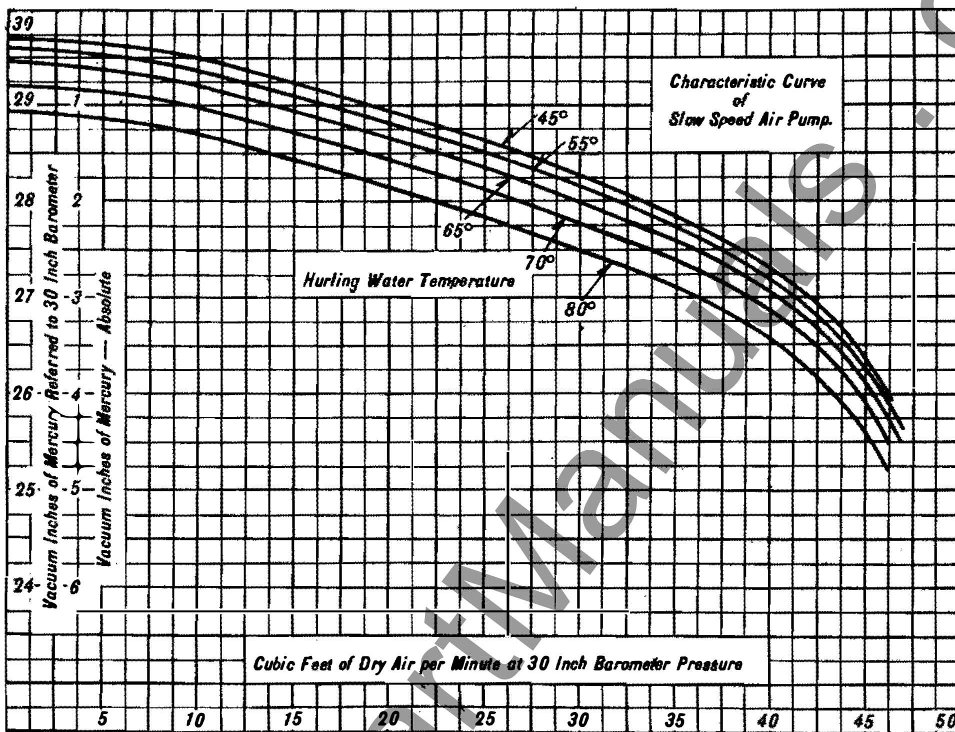


FIG. 7—CHARACTERISTIC CURVE OF SLOW SPEED AIR PUMP

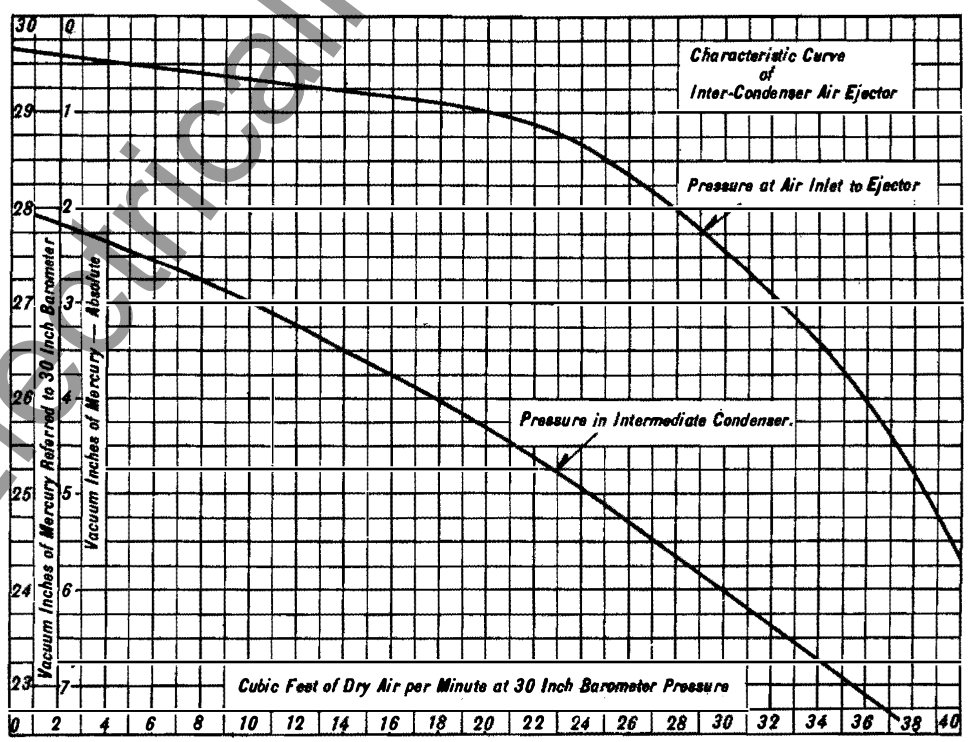


FIG. 8—CHARACTERISTIC CURVE OF CONDENSING AIR EJECTOR

densate pump capacity is in question, for it has often been found that packing or tubes have developed leaks.

Practice has demonstrated that air leaks into a condensate pump are quite detrimental not only to the pump capacity, but also to the performance of the main condenser. The air admitted through leaks or poorly packed glands also assists in erosion of pump parts as well as the system beyond the pump. Therefore, after making inspection or repairs a careful water test should be made.

For demonstrating how submergence (the water level in the condenser above centerline of condensate pump), affect the capacity of a condensate pump, the curve Fig. 6, which is the result of actual test, has been constructed. Each type of pump, of course, will have characteristics peculiar to that class. From the curve it can be immediately realized why at least 3 ft. of submergence is necessary and more desired if a reasonable size pump frame is to be utilized. The theory of operation is similar to other pumps of the centrifugal type, depending upon the absolute pressure to force the water into the pump impeller. Where extremely high vacuum exists in the condenser, the pressure is obtained by utilizing the weight or gravity due to height of water above the centerline.

AIR PUMP
Description and Operation

Good condenser performance is greatly dependent upon proper air removal. The Westinghouse Company, by experience with condensing apparatus, has established as a standard the Westinghouse hydraulic air pump and the Westinghouse air ejector.

The Westinghouse air pump is of simple structure and correct in principle, operating at a constant rotative speed, with no sliding parts or complex port openings or the like that require care and accurate adjustment. In the hydraulic air pump the air is trapped by intermittent pistons of water from a rotating wheel directed into a properly designed cone from which the mixture of air and water discharges to the atmosphere through a diffuser. The principle of the pump is simple and it is readily understood how theoretical vacuum can easily be obtained in a case where there are no clearance spaces nor changes of pressure within the pump itself.

The air pump by virtue of its design, is intended to operate at a constant speed. However, a two per cent variation of speed in either direction seems to have no marked effect. The water suction should not exceed that indicated on the gauge furnished with the apparatus. An increase or decrease of water will affect the air handling capacity and the power in respective order. It is often found necessary to increase the quantity of water as the pump parts become worn, a condition which indicates that an inspection of the apparatus should be made.

A very serious operating problem often arises due to excessive head on the air pump discharge, caused either by obstruction, long pipe lines, unnecessary rises of pipe, or corrosion. At the first indication of trouble the pressure should be determined and a thorough investigation of check valves, sharp bends, etc., carried out. The ideal conditions for air pump operations are a very low suction lift and free discharge.

The proper operation or functioning of the "Westinghouse Hydraulic Air Pump" can be easily detected by a simple test of pressure determination in the chamber surrounding the discharge end of the collector cone. When removing small quantities of air the pressure, (vacuum) in this chamber very closely approaches that of the condenser. When removing large volumes of air the pressure difference may be several inches of vacuum. If the pressure difference is great and fluctuating over a wide range, or from vacuum to positive pressure, it can be concluded that the pump is not operating properly and should be investigated.

The hurling water used in the pump, which is transformed into pistons that entrain the air between them and carry it through to the discharge, should be of a temperature corresponding closely to that of the main injection, for if the temperature is high the vapor pressure may be such as to limit the vacuum obtainable. This point is very easily understood if it is considered that it is impossible to obtain a vacuum higher than that corresponding to the water temperature present. In many stations where special tank arrangements for recirculating the hurling water are provided, the temperature of the water used in the air pump is disregarded with the consequent reduction of air handling capacity and a decrease of vacuum. By referring to curve Fig. 7, which indicates the characteristic of a large size air pump with varying water temperatures, it can readily be seen how the temperature affects the capacity at a given vacuum, or the vacuum at given capacity. A characteristic feature of the air pump little understood, is that the air handling capacity is not affected by the moisture content of the vapors to be removed, as is the case in many other types of air removal equipment. In other words, the free air handling capacity is the same whether handling dry air from the atmosphere by orifice test, or saturated air from the condenser.

Air Pump Suction

The air pump suction pipe should be of liberal size to prevent excessive lift. A strainer, regulating valve, and vacuum gauge should be provided. In cases where the water supply is above the pump, a valve on each side of the strainer is necessary in order to clean same without shutdown.

Air Pump Discharge

When the air pump is used, the discharge pipe should be of sufficient size free from short bends and pockets. Atmospheric discharge is ideal for operation and life of the parts. However, the pump is capable of discharging against 12 ft. maximum discharge head. Should the discharge head exceed 12 ft. a re-circulating system in the form of a tank or well should be installed. To maintain proper water temperature in the re-circulating system, make up water amounting to 2% of the main injection should be supplied.

The amount of water circulated through the air pump is approximately 15% of that required by the condenser. Therefore, for proper liberation of the entrained air a lineal travel of 12 ft., with a velocity not exceeding ½ ft. per second between intake and discharge, should be provided. Proper installation of baffles will materially reduce the size of tank or well required.

When the discharge pipe is submerged a check valve should be installed to prevent return flow.

When the discharge pipe line is long and submerged, a vent for releasing the air and steam of the steam primer should be provided. Where pockets in the discharge line cannot be avoided, a vent should be installed.

AIR EJECTOR
Description and Operation

The Westinghouse air ejectors are divided into two classes, condensing and non-condensing, each to serve the purpose intended. The major point of difference is that in the condensing type the steam used by the first stage is condensed before entering the second stage, thus reducing the volume of vapor or steam mixture entering the second stage. The steam consumption of the condensing type ejector due to the condensing of the first stage steam as above described is approximately 40% of that required by the non-condensing type ejector. The air ejector is a static piece of apparatus, the work being wholly dependent upon the friction of steam jets entraining the gases from the condenser.

The operation of the air ejector is quite simple; however, the conditions of the design must be adhered to. As a general rule, the steam pressure must not fall below that given for the design nor should the back pressure exceed that indicated in the contract. Very often trouble is experienced due to either low steam pressure or fouled steam strainers. It is absolutely necessary that the nozzles be free from scale or any obstruction that will reduce the steam flow. Such a condition is often detected by the fact that ejectors require higher pressure than usual for stable operation. The condensing type ejector is less susceptible to scale or obstruction due to the use of single instead of multiple nozzles, and consequently greater throat areas.

The intermediate condenser of the jet type should be periodically inspected for obstructed water spray nozzles and the surface type for obstructed or defective tubes. In the early operation of most units, conditions exist that cause trouble. Therefore, an inspection should be made a short time after the units have been placed in continuous operation. The water supply to the intermediate condenser should be sufficient so that proper condensation and cooling will take place, for otherwise the second stage will be required to handle vapors in excess of that contemplated and a reduction of overall air handling capacity will result.

The characteristic curve of a large size intermediate condenser type air ejector, Curve Fig. 8, by orifice test gives the vacuum that will be maintained when handling free dry air.

The free air handling capacity of an ejector when removing air from a condenser is reduced due to the quantity of water vapor of saturation that has to be removed with the air; therefore, approximately twice the ejector capacity is necessary for extracting the amount indicated by a free dry air orifice test. Thus, for handling 10 cu. ft. of air per minute from a condenser it would take an ejector with an orifice test capacity of approximately 20 cu. ft. of free dry air per minute.

By the use of curve Fig. 9, the vapor content per pound of air to be removed from the condenser can be determined if the temperature and pressure of the mixture are known. For example, if the pressure is 28″ of vacuum or 2″ of mercury absolute, and the temperature of the vapors from the condenser is 86°F., then for every pound of air it is necessary to handle 1.02 lbs. of water vapor. Therefore, it is necessary to have ejector capacity to take care of the vapor as well as the air. As previously stated, the air pump automatically takes care of the vapor content, which need not be considered when selecting size, as is necessary in air ejectors and other types of pumps.

Due to the characteristic of the air ejector—that the capacity decreases at high vacuum, it becomes necessary to supply large capacities when high vacuum operation is desired.

Very often the question arises—Why is it necessary to supply large air removal equipment for high vacuum? This question can best be answered by referring to curve Fig. 10, which shows the volume of one pound of air at various pressures. In pumps of the displacement type, where the small clearance space is filled with air or water at atmospheric pressure, it can readily be seen that, for a great part of the stroke, the existing air is rarefied and the water, vaporized before the air is drawn from the condenser, while for the remainder of the stroke, a very large volume has to be removed to actually remove a reasonable quantity of free air. The hydraulic pump may be called a continuous piston pump removing air at a constant rate without intermittent action.

The general operating staffs often treat the vacuum apparatus, especially air removal equipment, as a mysterious device that is placed in the station to assume the entire blame for improper operation or faults of other equipment. Often the air ejector or the air pump is criticized for low vacuum when the cause is entirely due to air leaks, fouled tubes, lack of circulating water, etc., that arises due to negligent operators or to change of conditions on other units of the equipment. When discrepancies exist, a thorough investigation should be conducted on all equipment and, by process of elimination, the trouble will be detected.

Steam Supply to Ejectors

The ejector is designed for a given steam pressure and quality. Therefore, correct operation can be obtained only when such conditions are maintained. Where pressure fluctations exist, a reducing valve is necessary. The steam nozzle should be designed for the lowest pressure experienced under normal conditions. Increase in steam pressure will not increase the air capacity and when pressure increase is excessive, a reduction in capacity will take place due to the restriction caused by the increased volumes passing through the diffuser throats and the decreased performance of the inter-condenser caused by the increased quantity of steam.

Ejector Steam Piping

Ejector steam piping should be of liberal size with sufficient supports to prevent all strains caused by weight, expansion and contraction. The lead-off from the header should be from the top or side of the pipe so that condensation, scale and sediment will not be carried to the ejector nozzles. Drains and traps should be properly located; for correct operation depends upon dry steam. When a reducing valve is used, a bypass should be installed to prevent interruption of service, if the reducing valve becomes inoperative.

Ejector Exhaust Piping

To obtain the maximum economy and the best operating condition, a full size short pipe without restrictions should be used for the exhaust, in order that the back pressure will be reduced to a minimum. When exhausting into open heaters with the end of the exhaust pipe submerged, a non-return check valve or a loop with an air vent should be provided.

Ejector Air Vent Lines

The air vent from the after condenser should be piped to the atmosphere; quite often this can be easily accomp-

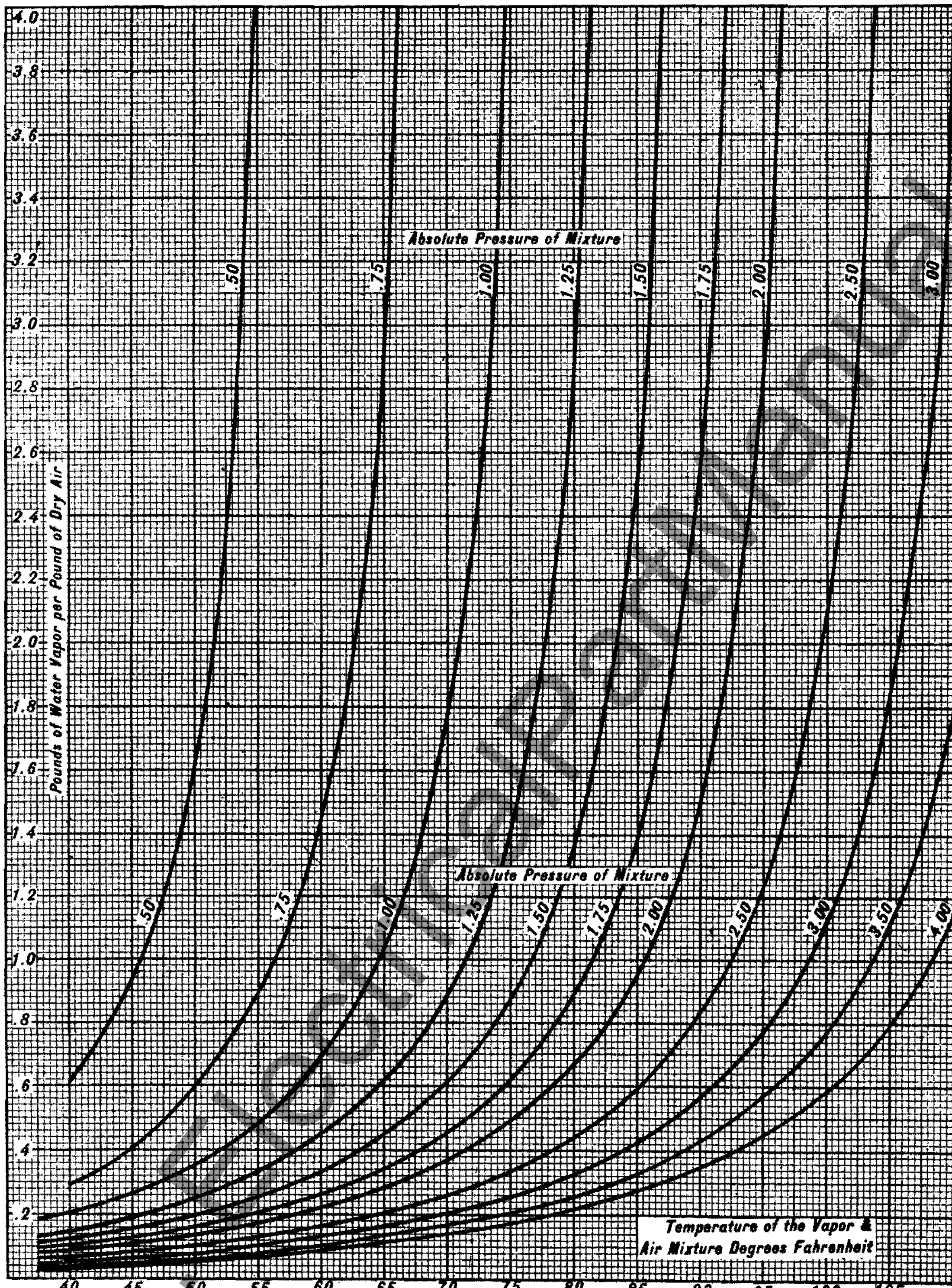


FIG. 9—CURVE SHOWING THE WEIGHT OF WATER VAPOR THAT MUST BE REMOVED FROM A CONDENSER IN REMOVING ONE POUND OF FREE DRY AIR

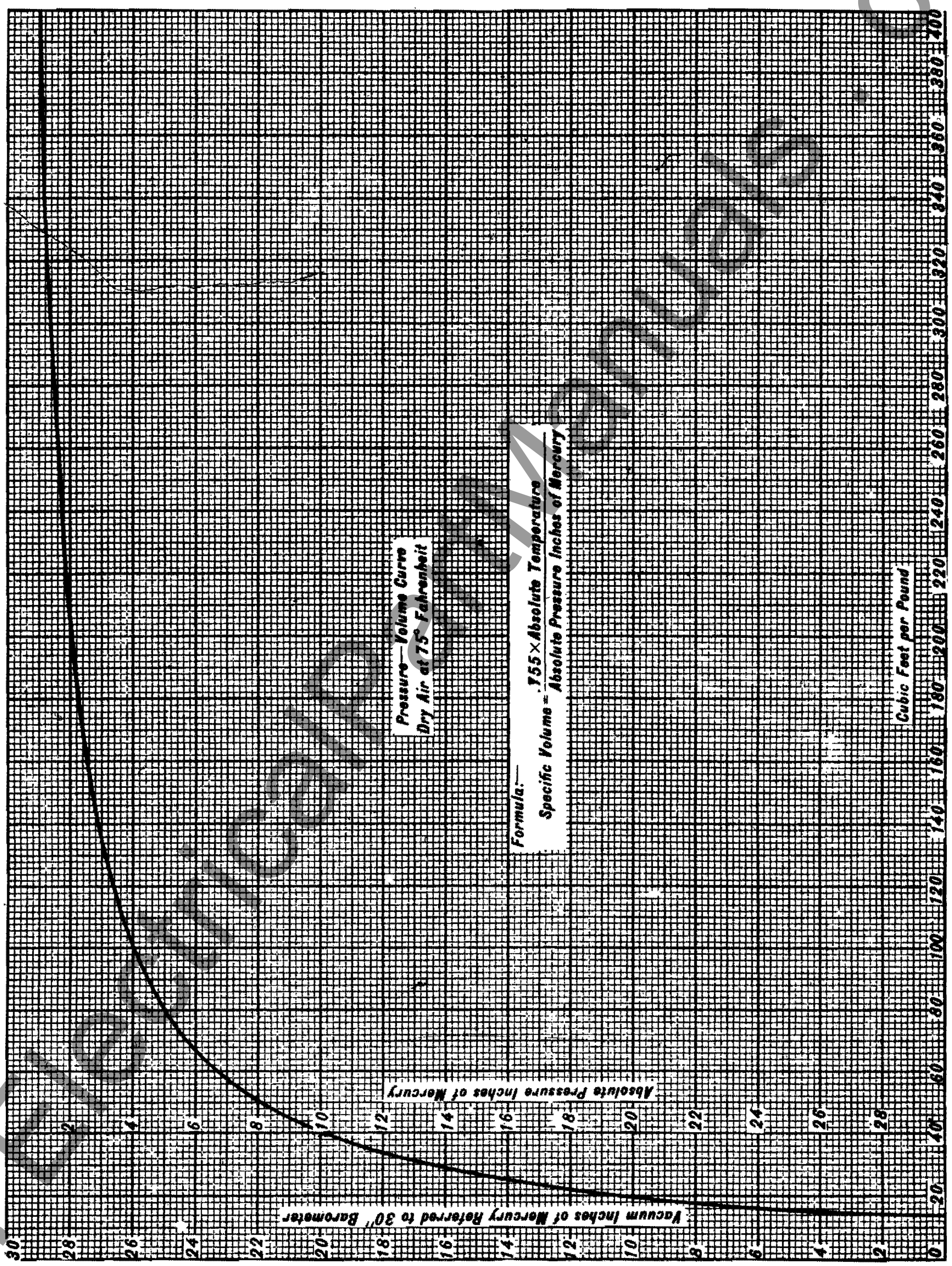


FIG. 10—PRESSURE VACUUM CURVE FOR AIR

lished by connecting to the main turbine atmospheric relief exhaust. This line should be full size in order that no back pressure will be imposed upon the ejector. Occasionally it may be necessary to operate for short periods non-condensing. Therefore, unless correct sizes are installed excessive back pressure will exist.

Water Seal Between Ejector Stages

The two-stage condensing ejector, when each element is working most efficiently, will require a water seal loop of 7 ft. between the intermediate condenser and the main condenser, to which the condensate is drained, to prevent the re-circulation of air from the intermediate condenser to the main condenser. If it is desired to recover the condensate from the surface intermediate condenser a suitable pump or trap must be provided. The water seal loop should be without globe or check valves and so arranged that when the first stage is not operating the intermediate condenser will drain by gravity.

Cooling Water for Condensing Type Ejector

When ejectors with the jet type of intermediate condenser are used with the low level jet condenser, either raw water taken from the injection line or from a separate source is used in the intermediate condenser and drained into the main condenser. If boiler feed water of normal temperature is used in the intermediate condenser, then it will require a small pump to remove the water from the intermediate condenser which is under vacuum.

When ejectors with surface intermediate and after condensers are used with a jet condenser installation, the cooling water is usually the boiler feed supply. The quantity of water governs the number of passes in the condenser. In cases where only a small amount of water is available a raw water section is necessary in order that correct performance may be obtained. The section of the intermediate condenser supplied with raw water completes condensation and then reduces the temperature of the air and vapor mixture below the dew point, causing further condensation. The cooling brought about by the raw water section, as explained, greatly reduces the proportion of water vapor to the second stage, with the result that the air handling capacity is greatly increased. The effect of temperature and pressure change, of a saturated air and water vapor mixture, is clearly illustrated in Fig. 1 of the condenser section.

Priming Ejector

Description and Operation

To reduce the time required to prime the circulating system as well as the vacuum system, a large capacity air ejector has been developed, known as a "priming ejector". The steam consumption is greater, but this is not important as it is only used a comparatively short time. This ejector is of the single stage type with manganese copper nozzles.

In large installations an auxiliary priming ejector of liberal air handling capacity may be used to advantage for quick evacuation of the steam space. This priming ejector is an auxiliary, and does not substitute for other methods of starting the unit.

The operation of the priming ejector is similar to that of any steam jet ejector which removes air by high velocity steam jets flowing through the media to be removed. This ejector is of the single stage type, designed for removing large volumes of air in a very short time, as is often desired in priming the circulating system, or removing air from a very large condenser and exhaust fitting. Often this type of ejector is so connected that after priming the circulating system it can be temporarily used to create vacuum in connection with regular efficient air removal apparatus usually supplied with the units.

Atmospheric Relief Valves

To safeguard the condenser from excessive pressure, should the vacuum for some reason be destroyed, a relief valve is installed. This valve is so constructed that when it is closed there will be a channel of water entirely around the edge to make a seal and prevent air leakage. The opening is positive and should the pressure become higher than that of the atmosphere the valve would be easily raised. Near the end of the upward travel a cushion chamber becomes active, thereby preventing excessive hammering.

General Information

Operating Difficulties That May Be Encountered

Surface Condensers:

Air leakage into the condenser and vacuum system.

Air leakage through improperly packed valve stems.

Leaky steam and air baffles.

Steam surface of the tubes coated with oil, sediment or scale.

Water surface of the tubes fouled with debris, slime or scale.

Insufficient amount of circulating water.

Low water velocity in the tubes.

Water pockets in the air line.

Excess amount of steam from turbine due to mechanical or operating conditions.

Connecting of vents and drains to air line not recommended.

Circulating Pumps:

Reduced Capacity.

Excessive total pumping head.

Excessive suction lift.

Incorrect speed.

Eroded Impellers.

Excessive liner ring clearances.

Debris in impeller or suction piping.

Air leakage through the glands.

Air leakage into pump body or suction piping.

Obstructed gland water piping.

Incorrect position of gland spacer ring.

Worn shaft sleeves.

Obstructed or inoperative foot valves.
NOTE—Foot valves not recommended

Excessive screen and tunnel losses.

Excessive syphon head loss in discharge pipe.
(Caused by air collection)

Insufficient submergence of suction and discharge pipe.

Incorrect gauge location.

Improper corrections for gauge location.

Error of gauges.

Excessive debris collected on tubes causing excessive pumping head.

Condensate Pumps:

Air leakage into pump or suction pipe.

Air leakage through the glands.

Improper or obstructed vapor vent.

Debris in impeller and suction piping.

Excessive liner ring clearances.

Eroded impeller.

Excessive discharge head.

Incorrect speed.

Excessive friction head in discharge pipe.

Obstructed or defective discharge check valve.

Obstructed gland water piping.

Incorrect position of gland spacer ring

Defective packing.

Defective or incorrectly installed shaft sleeves.

Steam Jet Air Ejectors:

Air leakage of valve stems and interconnecting piping.

Steam pressure other than design.

Superheat other than design.

Wet steam.

Entrained water or condensate.

Restricted steam pipe after pressure gauge.

Improper operation of reducing valve.

Sediment and scale obstructing the nozzle.

Corrosion and scale formation on nozzle surface.

Eroded nozzles and diffusers.

Incorrect nozzle position.

Excessive exhaust back pressure.

Restricted air vents (causing cessive exhaust back pressure).

Obstructed water seal loop between intermediate condenser and the main condenser.

Insufficient height for gravity drain from intermediate condenser when operating second stage only.

Incorrect height of water seal loop between intermediate and main condenser.

Leaky tubes causing flooding of the intermediate and after condenser.

Insufficient cooling of the vapors to the ejector.

Excessive temperature of the water to intermediate and after condenser.

Insufficient water for intermediate and after condenser.

Incorrect number of water passes in the intermediate and after condenser.

Hydraulic Air Pump:

Eroded runner, distributor, steam or collector cone.

Incorrect distributor and cone openings.

Incorrect distributor setting.

Misalignment of runner and cones.

Air leakage into suction pipe and strainer.

Air leakage through the glands.

Air leakage of expansion joints.

Improper air separation when recirculating tank is used.

Excessive temperature of hurling water, especially when recirculating tank is used.

Excessive discharge head.

Defective discharge check valve.

Excessive water suction lift.

Incorrect speed.

Excessive water leakage from water pump to air pump.

Leakage of water from seal pockets of the jet condenser combining cone into the air pipe.

Obstructed gland water piping.

Incorrect position of gland spacer ring.

Defective packing.

Defective or incorrectly installed sleeves.

Priming Ejector:

Incorrect steam pressure.

Wet steam.

Excessive exhaust back pressure.

Corroded or scaled nozzles.

Eroded nozzles or diffuser throat.

Obstructed or clogged nozzles.

Jet Condensers:

Air leakage into the condenser or vacuum system.

Air leakage through the glands.

Air leakage along the shaft due to defects or incorrect installation of sleeves.

Insufficient water.

Excessive injection water lift.

Excessive friction losses in injection pipe line.

Obstructed strainers or foot valves.

Obstructed nozzles.

Eroded nozzles.

Excessive discharge head on circulating pump.

Defective check valves.

Incorrect impeller diameter.

Incorrect speed.

Excessive liner ring clearance.

Eroded impeller.

Worn shaft sleeves.

Improper packing of the glands.

Incorrect position of gland spacer ring.

Obstructed water piping to the glands.

Incorrect submergence.

Water leakage along the shaft from water pump through diaphragm into the air pump.

Water leakage from seal pockets of condenser combining cone into the air pipe.

Insufficient water to precooler.

Excessive water to precooler causing pressure drop.

Obstructed precooler water nozzle.

Obstructed precooler drain to condenser.

Obstructed drain from upper section of precooler.

Excessive submergence flooding precooler.

For air pump difficulties, refer to items under air pump.

For air ejector difficulties, refer to items under air ejectors.

Vacuum Breaker:

Improper water seal and drain.

Improper seating of valves.

Insufficient guides causing incorrect seating.

Tilting of float, causing side strain on the valve, due to wear of arm and fulcrum pin.

Incorrect counterweight.

Water logged float.

Corroded float.

Obstructed screen or drain with old type.

Adjusting vacuum breaker counterweight without submerging.

Useful Information

The information that is generally used in connection with selecting, installing, testing, and operating condensers can be found in hand books, texts, etc., but the subject matter is so treated that often it cannot be entirely understood. The development of formulas incorporating complex mathematics often discourage the practical operator with the result that the performance of the equipment receives no correct analysis. The subject matter pertaining to the condenser is widely scattered in the reference books and unless one is quite familiar with what is desired it is almost impossible to correlate the information needed, therefore, with the practical operator in mind the following useful information in the form of conversion factors, tables, formulas, correction factors, etc., is presented.

CALCULATIONS AND DETERMINATIONS OF PERFORMANCE

The determination of results obtained is often considered quite technical by operators, however, such determination is quite simple, if it were possible to find the information necessary to make calculations in some simple and compact form. The following determination of results will be concisely and clearly given step by step so that by the simplest calculations every point can be understood.

Determination of Heat Absorbed by Condensing Water

The amount of heat absorbed by the condensing water to effect condensation within a surface condenser is equal to the (latent) heat ordinarily required in evaporating water at the same temperature and pressure. The heat unit generally used is known as the British Thermal Unit (B.t.u.) which is the quantity of heat necessary to raise one pound of water 1°Fahrenheit (from 63° to 64°F.)

In determining the amount of heat expelled to a condenser three factors must be considered.

FIRST—The heat equivalent to work performed.

SECOND—The heat expelled in the condensate.

THIRD—The heat losses in generator, radiation and other mechanical losses.

The determination of losses under the third factor is quite difficult and therefore often assumed from general practice which gives a very close approximation.

For illustration: Consider a turbine supplied with steam at a pressure of 200# absolute and 150°F. superheat expanding to 28.5" of vacuum or 1½" absolute, and a water rate of 13# per Kw., generating through a 95% efficient generator. Assume 1% radiation, bearing, and mechanical losses.

The total heat of steam at 200# absolute, 150° superheat	= 1285.3
Heat transformed into work in the turbine per lb. of steam $\dfrac{\text{3415 B.t.u. per Kw.}}{\text{13\# Water Rate x 95\% generator eff. x 99\% mechanical Eff.}}$	= 279.1
Heat B.t.u. per lb. of steam entering condenser	= 1006.2
Heat expelled in condensate (Condensate Temp. 92–32)	= 60.0
Heat B.t.u. per Lb. of steam absorbed by Circulating Water	= 946.2

For all practical purposes and approximate checks the following heat B.t.u. per pound of steam absorbed by the condensing water can be assumed as:

945 B.t.u. for 29" vacuum or 1" Absolute.

940 B.t.u. for 28" vacuum or 2" Absolute.

935 B.t.u. for 27" vacuum or 3" Absolute.

For uniformity of commercial calculations the heat content of 950 B.t.u. is customarily used.

Determination of Circulation Water by Temperature Rise

A very accurate determination of the quantity of circulating water can be made when the quality of the generated steam, load and water rate of the turbine are known. By the determination of heat units per pound as explained previously and knowing the rise of the condenser water, the quantity of condenser water is determined as follows:

$$\text{Circ. Water Lbs./Hr.} = \frac{\overset{\text{(Load}\times\text{Water rate)}}{\text{Lbs. of steam per hr.}} \times \overset{\text{(Approximately 950)}}{\text{Heat units B.t.u. per \# of steam}}}{\text{Rise °F. of circ. water}}$$

$$\text{Circ. Water Gallons per minute (GPM)} = \frac{\text{Circ. water \# per hr.}}{8\tfrac{1}{3}\ \text{\# per gal.} \times 60\ \text{min. per hr.}}$$

$$= \frac{\text{Circ. water \# per hr.}}{500}$$

$$\text{Circulating Water G. P. M.} = \frac{\text{Steam \# per hr.} \times \text{Heat units}}{\text{Rise} \times 500}$$

Determination Water Pass Tube Area

$$\text{Water pass area sq. ft.} = \frac{(D)^2 \times .7854}{144} \times \text{No. of tubes per pass}$$

$$= C \times \text{Number of tubes per pass}$$

Assume:
- D = Inside diameter of tube.
- C = Constant for size of tube.

For values see following table:

Bwg	Thickness in Inches	Outside Diameter of Tubes in Inches — 1"		7/8"		3/4"		5/8"	
		D Internal Diameter Inches	C Constant for Size of Tube	D Internal Diameter Inches	C Constant for Size of Tube	D Internal Diameter Inches	C Constant for Size of Tube	D Internal Diameter Inches	C Constant for Size of Tube
16	.065	.870	.00413	.745	.00302	.620	.00210	.495	.001338
17	.058	.884	.00426	.759	.00314	.634	.00219	.509	.001413
18	.049	.902	.00444	.777	.00329	.652	.00232	.527	.001518
20	.035	.930	.00471	.805	.00353	.680	.002525	.555	.001680

Determination of Circulation Water Velocity in Tubes

$$\text{Velocity Ft. per Second} = \frac{\text{Pounds of circ. water per hour}}{3600 \text{ sec. per hr.} \times 62.4 \,\#\, \text{per cu. ft.} \times \text{area (in sq. ft.) of tubes in pass}}$$

$$= \frac{\text{Pounds of circ. water per hour}}{224{,}640 \times \text{area (in sq. ft.) of tubes in pass}}$$

Calculation of Condenser Surface

$$\text{Sq. Ft. of Surface} = \frac{D \times 3.1416 \times L \times \text{No. of tubes}}{12}$$

$$= D \times L \times .2618 \times \text{No. of tubes.}$$

L = Active length of tubes in feet

D = Outside diameter of tubes in inches

Determination Mean Temperature Difference

Mean temperature difference (M. T. D.) is determined by two methods. The logarithmic mean temperature difference and the arithmetical mean temperature difference.

The logarithmic determination is considered the most accurate for all cases, while the arithmetical is considered approximate, however, for cases where the temperature rise of the circulating water is 12°F. or less, either method is sufficiently accurate for general condenser calculations. The arithmetical mean is more simple and easily obtained.

The logarithmic mean temperature difference (M. T. D.) is expressed.

$$M.\,T.\,D. = \frac{T_2 - T_1}{Log_e \dfrac{Ts - T_1}{Ts - T_2}}$$

The arithmetical mean temperature difference (M. T. D.) is expressed:

$$M.\,T.\,D. = Ts - \frac{T^2 + T^1}{2}$$

Ts = Steam temperature degrees Fahrenheit.

T_1 = Circulating water inlet temperature degrees Fahrenheit.

T_2 = Circulating water discharge temperature degrees Fahrenheit.

Log_e = Common log base (10) × 2.3026.

Determination of Heat Transfer

Due to the fact that two methods are employed in finding the mean temperature difference (M. T. D.) the heat transfer per square foot per hour per degree Fahrenheit, difference of temperature can be determined by either method.

$$\text{Heat transfer} = \frac{\text{Lbs. of steam per hr.} \times B.t.u. \text{ per lb.}}{\text{Surface Sq. Ft.} \times M.\,T.\,D.}$$

Where the temperature rise of the circulating water is approximately 12°F. or less, the arithmetical mean temperature difference can be used with reasonable accuracy.

Determination of Correct Pressure

Barometer

The U. S. Weather Bureau usually gives barometer readings reduced to some standard elevation and to 32°F. temperature therefore, readings obtained should be clearly understood and corrections made.

The barometer reading changes at the rate of 0.00115" of mercury per foot of elevation, therefore, when ascending the correction should be subtracted, and when descending the correction should be added.

For every degree change in temperature or difference of temperature a correction of 0.00278 inches should be made. Example: Barometer 29.400 inches of mercury at 500 ft. elevation and 32°F. when corrected to sea level and 80°F. becomes:

Barometer at 500 ft. and 32°F.	= 29.400
Altitude Correction = 500 × 0.00115	= .575
Barometer Corrected to sea level	= 29.975
Temp. Correction = 48°F. × 0.00278	= .133
Barometer Corrected to sea level & Temp.	= 30.108

Mercury Column

When the corrected barometer reads more than 30 inches the excess amount should be subtracted from the mercury column reading, and when the corrected barometer reads less the discrepancy should be added to the mercury column reading.

EXAMPLE

Vacuum by mercury column reading 29.6 corrected barometer 30.2. Corrected vacuum 29.4 inches mercury.

Vacuum by mercury column reading 29.6 corrected barometer 29.8. Corrected vacuum 29.8 inches mercury.

Determination of Flow of Air Through Orifice

For conditions when discharge pressure is less than 53% of the inlet pressure:

W = Weight of air, pounds per second.

A = Area of orifice in square inches.

P = Pressure in pounds per square inch.

T = Absolute temp. of air (460 + Thermometer Reading °F.).

N = Nozzle co-efficient = 97% for well rounded orifice.

$$W = .53 \times \frac{A \times P}{\sqrt{T}} \times N$$

NOTE: When the air is flowing from the atmosphere to a vacuum P = Barometer reading × .491 = Atmospheric pressure, pounds per square inch.

Determination of Specific Volume of Air at given conditions and Flow in cubic feet per unit of time

The weight of air in pounds per unit time should be mutiplied by the specific volume.

V = Specific volume cubic feet.

T = Absolute Temp. (460 + Thermometer Reading °F.).

P = Absolute Pressure inches of mercury.

W = Weight of air per unit of time.

Q = Volume cubic feet per unit of time.

$$V = \frac{.755 \times T}{P}$$

$$Q = W \times V.$$

Readings for Determination of Performance

The determination of condenser performance requires certain definite readings that very often are omitted. Unless the information is complete assumptions are necessary and conclusions are quite indefinite. The readings that are necessary for surface condensers, are shown on pages 27 and 28, and for jet condensers on pages 29 and 30.

Additional readings can be added if certain specific information is to be transmitted.

READINGS OF SURFACE CONDENSER PERFORMANCE

Customer..Serial No...

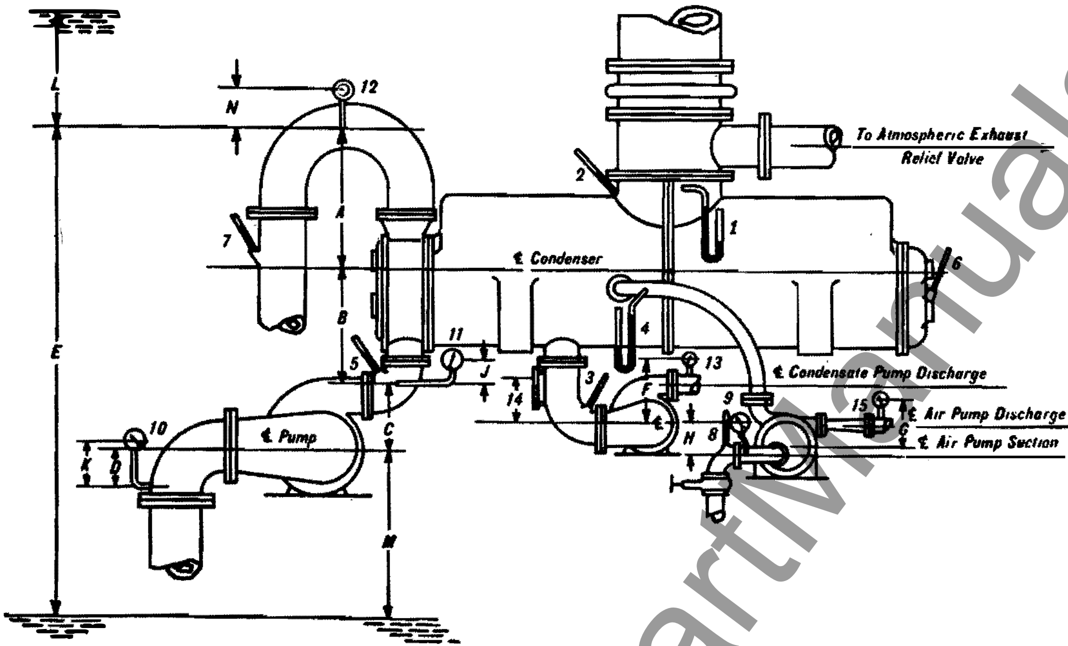
Location...Connected to {Engine / Turbine}.....................................{Hp. / Kw.}

Condenser Size...Sq. Ft. Maker...

		TIME				
Date..............................		....A.M. P.M.	....A.M. P.M.	....A.M. P.M.	....A.M. P.M.	...A.M. P.M.
Load in kw. on engine or turbine						
Total steam flow, lbs. per hr.						
Steam pressure at turbine (engine) throttle lbs. per sq. in.						
Steam temperature at turbine (engine) throttle °F.						
Main unit primary inlet pressure lbs. per sq. in.						
Main unit secondary inlet pressure lbs. per sq. in.						
Barometer {Mercurial / Aneroid} Located..............feet {Above / Below} Condenser						
Vacuum at top of condenser by mercury column	1					
Temperature at top of condenser °F.	2					
Temperature condensate pump water °F.	3					
Vacuum at air connection at condenser by mercury column	4					
Temperature of air in air line at condenser						
Temp. injection water inlet °F.	5					
Temp. injection water end of first pass °F.	6					
Temp. injection water discharge °F.	7					
Temp. air pump water °F.	8					
Vac. air pump injection inlet	9					
Circulating pump suction {inches vac. / lbs. press.}	10					
Condenser injection inlet {inches vac. / lbs. press.}	11					
Condenser discharge water press. {inches vac. / lbs. press.}	12					
Condensate pump discharge press.—lbs. per sq. in.	13					
Level of condensate water above centre line of condensate pump	14					
Air pump discharge pressure {inches vac. / lbs. press.}	15					
Rpm. air pump						
Rpm. circulating pump						
Rpm. condensate pump						
Inlet steam pressure for condenser pump drives						

Signature...

READINGS OF SURFACE CONDENSER PERFORMANCE—(Continued)



OBSERVERS SHOULD FILL IN DIMENSIONS A. B. C. D., ETC. ON FIRST REPORT	FT.	INS.
A—Distance from center line of condenser to point of attachment of circulating water discharge gauge = ...		
B—Distance from center line of condenser to point of attachment of circulating water inlet gauge = ...		
C—Distance from center line of circulating pump to point of attachment of circulating water inlet gauge = ...		
D—Distance from center line of circulating pump to point of attachment of circulating pump suction gauge = ...		
E—Distance from circulating water level to point of attachment of circulating water discharge gauge = ...		
F—Distance from center line of condensate pump to center line of condensate discharge gauge = ...		

OBSERVERS SHOULD FILL IN DIMENSIONS A. B. C. D., ETC. ON FIRST REPORT	FT.	INS.
G—Distance from center line of air pump to center line of air pump discharge gauge = ...		
H—Distance from center line of air pump suction to center line of suction gauge =		
J—Distance from point of attachment of circulating water inlet gauge to center line of gauge = ...		
K—Distance from point of attachment of circulating pump suction gauge to center line of gauge = ...		
L—Distance from point of attachment of circulating water discharge gauge to circulating water discharge level =		
M—Distance from circulating water level to center line of circulating pumps =		
N—Distance from point of attachment of circulating water discharge gauge to center line of gauge = ...		

JET CONDENSER PERFORMANCE

Customer..

Location..Altitude..................................

Condenser size...Serial Number.. ....Type.....................................

Condenser Pumps driven by { Motor Turbine Direct Geared } No..............................Frame..............................Speed..............................

Gear No..............................Motor Voltage..............................Frequency..............................Phase..............................

Kind of Water for Air Pump..............................Kind of Water for Main Injection...

The Service is { Continuous Intermittent } ....Average continuous run.......................... { Days Hours

The Load is { Steady Swinging } and the average peak swing is...................................... { Kw. Hp. } Variation .... { Kw. Hp.

Main Unit Kind..............................Make.............................. { Frame Size } .... { Kw. Hp.

Main Unit Serial Number...Main Unit Drives..

REMARKS:..

....

....

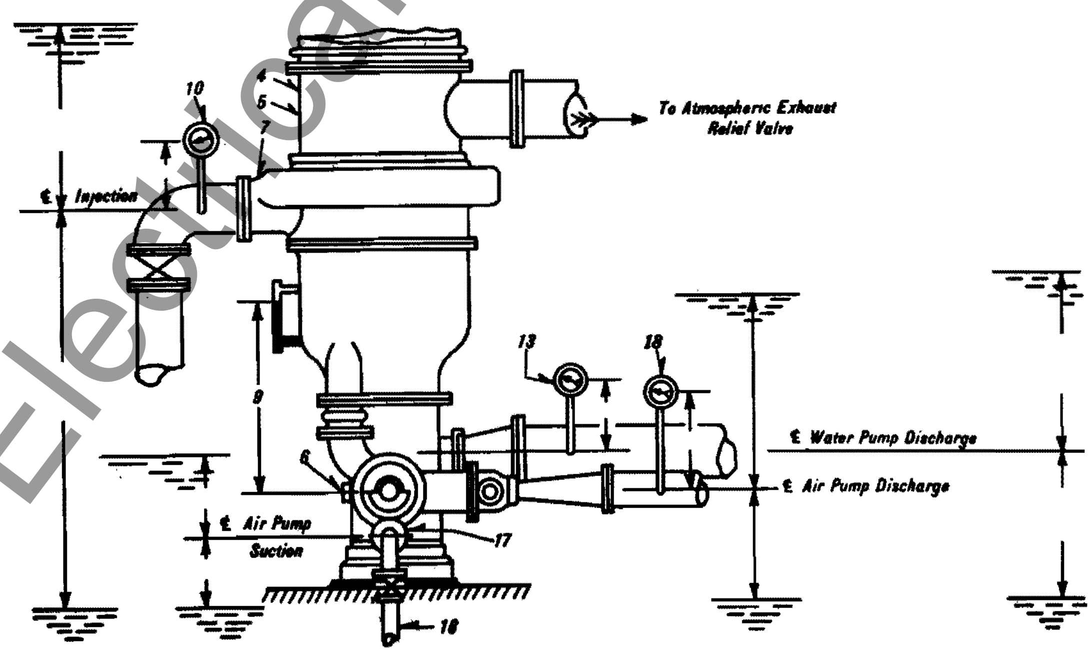
....

....

Data below heavy rule is necessary on first sheet only.

WATER LEVEL DATA

NOTE:—Always specify low water level dimensions, and give variations from low to high level.



Signature..

JET CONDENSER PERFORMANCE—(Continued)

Readings for Determination of Jet Condenser Performance

DATE TIME	A.M. P.M.	A.M. P.M.	A.M. P.M.	A.M. P.M.	A.M. P.M.
1. **Load in {K.W. / Hp.} on Main Engine**					
2. Total Steam entering Condenser Pounds per Hour					
3. **Barometer {Mercurial / Aneroid} Located feet {Above / Below} Condenser**					
4. **Vacuum at Exhaust Steam Inlet to Condenser by Mercury Column**					
5. **Exhaust Steam Temperature at Condeuser In et °F**					
6. **Discharge Water Temperature °F**					
7. **Injection Water Temoerature °F**					
8. Rise °F					
9. **Submergence**					
10. **Vacuum at Condenser Injection Inlet**					
11. Ratio—lbs. of Water per lb. of Steam					
12. Quantity of Injection Water lbs. per hour					
13. **Water Pump Discharge lbs. per sq. in. back pressure**					
14. Vapor Tension—Inches Mercury					
15. Air Tension—Inches Merc r					
16. **Air Pump Suction Water Temperature °F**					
17. **Vacuum Air Pump Suction In et**					
18. **Air Pump Discharge back pressure lbs. per sq. in.**					
19. **Speed of Pumps—Rpm.**					
20. **Drive Turbine Thrott e lbs. per sq. in. Gauge Steam Pressure**					
21. **Drive Turbine In et lbs. per sq. in. Gauge Steam Pressure**					
22. **Drive Turbine Exhaust back pressure lbs. per sq. in. gauge**					
23. **Main Turbine Throttle Pressure lbs. per sq. in. gauge**					
24. **Main Turbine In et Steam Pressure lbs. per sq. in. gauge**					
25. **Main Turbine Thrott e Steam Temperature °F**					
26. Main Turbine Throttle Superheat °F					
27.					
28.					
29.					

NOTE:—Heavy faced type are readings; light f ced type are results worked out from readings.

PRESSURE-TEMPERATURE CONVERSION TABLE

Absolute Pressure Ins. Hg.	Saturation Tempera-ture °F	Absolute Pressure Ins. Hg.	Saturation Tempera-ture °F	Absolute Pressure Ins. Hg.	Saturation Tempera-ture °F	Absolute Pressure Ins. Hg.	Saturation Tempera-ture °F
0.20	34.57	0.60	63.96	1.00	79.03	1.40	89.51
0.21	35.78	0.61	64.43	1.01	79.33	1.41	89.73
0.22	36.95	0.62	64.90	1.02	79.63	1.42	89.95
0.23	38.08	0.63	65.36	1.03	79.93	1.43	90.19
0.24	39.17	0.64	65.82	1.04	80.23	1.44	90.42
0.25	40.23	0.65	66.26	1.05	80.53	1.45	90.64
0.26	41.24	0.66	66.69	1.06	80.82	1.46	90.85
0.27	42.22	0.67	67.13	1.07	81.11	1.47	91.07
0.28	43.16	0.68	67.57	1.08	81.39	1.48	91.28
0.29	44.07	0.69	67.99	1.09	81.68	1.49	91.50
0.30	44.96	0.70	68.41	1.10	81.96	1.50	91.72
0.31	45.82	0.71	68.82	1.11	82.24	1.51	91.93
0.32	46.66	0.72	69.23	1.12	82.50	1.52	92.14
0.33	47.48	0.73	69.63	1.13	82.78	1.53	92.35
0.34	48.28	0.74	70.03	1.14	83.06	1.54	92.56
0.35	49.06	0.75	70.43	1.15	83.33	1.55	92.77
0.36	49.80	0.76	70.82	1.16	83.60	1.56	92.97
0.37	50.54	0.77	71.20	1.17	83.87	1.57	93.19
0.38	51.26	0.78	71.58	1.18	84.13	1.58	93.40
0.39	51.95	0.79	71.95	1.19	84.39	1.59	93.61
0.40	52.64	0.80	72.32	1.20	84.64	1.60	93.81
0.41	53.32	0.81	72.69	1.21	84.91	1.61	94.01
0.42	53.98	0.82	73.06	1.22	85.17	1.62	94.21
0.43	54.63	0.83	73.42	1.23	85.43	1.63	94.41
0.44	55.27	0.84	73.78	1.24	85.68	1.64	94.61
0.45	55.89	0.85	74.13	1.25	85.93	1.65	94.80
0.46	56.49	0.86	74.48	1.26	86.18	1.66	95.00
0.47	57.08	0.87	74.83	1.27	86.44	1.67	95.20
0.48	57.67	0.88	75.17	1.28	86.68	1.68	95.39
0.49	58.24	0.89	75.51	1.29	86.93	1.69	95.59
0.50	58.80	0.90	75.84	1.30	87.17	1.70	95.78
0.51	59.37	0.91	76.18	1.31	87.41	1.71	95.97
0.52	59.91	0.92	76.51	1.32	87.65	1.72	96.16
0.53	60.44	0.93	76.83	1.33	87.88	1.73	96.35
0.54	60.97	0.94	77.15	1.34	88.12	1.74	96.54
0.55	61.48	0.95	77.48	1.35	88.36	1.75	96.73
0.56	61.98	0.96	77.80	1.36	88.59	1.76	96.92
0.57	62.48	0.97	78.12	1.37	88.82	1.77	97.10
0.58	62.98	0.98	78.43	1.38	89.05	1.78	97.29
0.59	63.48	0.99	78.73	1.39	89.28	1.79	97.47

NOTE: Absolute Pressures are given at Standard Conditions:

Mercury at 32° F. Sea Level, acceleration of gravity 32.174 ft. per second, per second.

Values obtained directly or by interpolation from "Thermodynamic Properties of Steam", by Keenan and Keyes, 1936, by permission.

PRESSURE-TEMPERATURE CONVERSION TABLE

Absolute Pressure Ins. Hg.	Saturation Temperature °F	Absolute Pressure Ins. Hg.	Saturation Temperature °F	Absolute Pressure Ins. Hg.	Saturation Temperature °F	Absolute Pressure Ins. Hg.	Saturation Temperature °F
1.80	97.65	2.22	104.64	2.64	110.58	3.50	120.56
1.81	97.83	2.23	104.79	2.65	110.72	3.60	121.57
1.82	98.02	2.24	104.94	2.66	110.85	3.70	122.57
1.83	98.20	2.25	105.09	2.67	110.98	3.80	123.53
1.84	98.38	2.26	105.25	2.68	111.11	3.90	124.49
1.85	98.56	2.27	105.40	2.69	111.24	4.00	125.43
1.86	98.73	2.28	105.55	2.70	111.37	4.10	126.32
1.87	98.91	2.29	105.70	2.71	111.50	4.20	127.21
1.88	99.08	2.30	105.85	2.72	111.62	4.30	128.09
1.89	99.26	2.31	106.00	2.73	111.75	4.40	128.94
1.90	99.43	2.32	106.15	2.74	111.88	4.50	129.78
1.91	99.60	2.33	106.29	2.75	112.01	4.60	130.61
1.92	99.78	2.34	106.43	2.76	112.13	4.70	131.41
1.93	99.95	2.35	106.58	2.77	112.26	4.80	132.20
1.94	100.13	2.36	106.73	2.78	112.38	4.90	132.98
1.95	100.30	2.37	106.87	2.79	112.51	5.00	133.76
1.96	100.47	2.38	107.02	2.80	112.63	6.00	140.78
1.97	100.64	2.39	107.17	2.81	112.76	7.00	146.86
1.98	100.80	2.40	107.30	2.82	112.88	8.00	152.24
1.99	100.97	2.41	107.44	2.83	113.01	9.00	157.09
2.00	101.14	2.42	107.58	2.84	113.13	10.00	161.49
2.01	101.29	2.43	107.73	2.85	113.25	11.00	165.54
2.02	101.45	2.44	107.87	2.86	113.37	12.00	169.28
2.03	101.62	2.45	108.01	2.87	113.49	13.00	172.78
2.04	101.79	2.46	108.15	2.88	113.62	14.00	176.05
2.05	101.96	2.47	108.28	2.89	113.74	15.00	179.14
2.06	102.12	2.48	108.42	2.90	113.86	16.00	182.05
2.07	102.28	2.49	108.56	2.91	113.98	17.00	184.82
2.08	102.45	2.50	108.71	2.92	114.10	18.00	187.45
2.09	102.62	2.51	108.84	2.93	114.22	19.00	189.96
2.10	102.77	2.52	108.97	2.94	114.33	20.00	192.37
2.11	102.93	2.53	109.11	2.95	114.45	21.00	194.68
2.12	103.09	2.54	109.24	2.96	114.58	22.00	196.90
2.13	103.25	2.55	109.38	2.97	114.70	23.00	199.03
2.14	103.40	2.56	109.52	2.98	114.82	24.00	201.09
2.15	103.56	2.57	109.65	2.99	114.94	25.00	203.08
2.16	103.72	2.58	109.78	3.00	115.06	26.00	205.00
2.17	103.87	2.59	109.92	3.10	116.22	27.00	206.87
2.18	104.02	2.60	110.06	3.20	117.35	28.00	208.67
2.19	104.18	2.61	110.19	3.30	118.44	29.00	210.43
2.20	104.33	2.62	110.32	3.40	119.51	29.921	212
2.21	104.49	2.63	110.46				

TEMPERATURE CONVERSION TABLES

Fahrenheit—Centigrade

Fahr.	Cent.	Fahr.	Cent.	Fahr.	Cent.	Fahr.	Cent.	Fahr.	Cent.	Fahr.	Cent.	Fahr.	Cent.	Fahr.	Cent.
40	4.4	50	10.0	60	15.6	70	21.1	80	26.7	90	32.2	100	37.8	110	43.3
41	5.0	51	10.6	61	16.1	71	21.7	81	27.2	91	32.8	101	38.3	111	43.9
42	5.6	52	11.1	62	16.7	72	22.2	82	27.8	92	33.3	102	38.9	112	44.4
43	6.1	53	11.7	63	17.2	73	22.8	83	28.3	93	33.9	103	39.4	113	45.0
44	6.7	54	12.2	64	17.8	74	23.3	84	28.9	94	34.4	104	40.0	114	45.6
45	7.2	55	12.8	65	18.3	75	23.9	85	29.4	95	35.0	105	40.6	115	46.1
46	7.8	56	13.3	66	18.9	76	24.4	86	30.0	96	35.6	106	41.1	116	46.7
47	8.3	57	13.9	67	19.4	77	25.0	87	30.6	97	36.1	107	41.7	117	47.2
48	8.9	58	14.4	68	20.0	78	25.6	88	31.1	98	36.7	108	42.2	118	47.8
49	9.4	59	15.0	69	20.6	79	26.1	89	31.7	99	37.2	109	42.8	119	48.3

Degrees Centigrade = 5/9 (Degrees Fahr.—32)
Degrees Fahrenheit = 9/5 × Degrees Cent. + 32

CONVERSION FACTORS

U. S. gallon = 231 cubic inches = 0.1337 cubic feet.
= 8.3356 lbs. of water = 3.785 liters = 0.833 Imperial gals.
Cubic foot = 1,728 cu. ins. = 7.481 U. S. gals. = 62.355 lbs. water @ 62°F.
Horse power = 33,000 ft. lbs. per min. = 2546 B.t.u. per hr. = 0.7457 kilowatts
Kilowatt = 1.341 Horse Power = 3415 B.t.u. per hour.
British thermal unit (B.t.u.) = 777.5 Foot lbs.
1 Inch of Mercury = 1.133 ft. of water = 0.491 lbs. gauge = 13.6 ins. of water.
1 Lb. Gauge pressure = 2.31 feet of water = 2.033 inches of mercury.
1 Lb. of dry air at 30″ barometer 75°F. = 13.44 cubic feet.
1 Lb. of Saturated air at 30″ barometer, 75°F. = 13.59 cubic feet.
Pounds of dry air per hour at 30″ barometer and 75°F. × 0.224 = cu. ft. per minute.

MEASURE

1 Centimeter = 0.3937 inches.
1 Inch = 2.54 centimeters.
1 Meter = 39.37 inches = 1.0936 yards
1 Yard = 0.9144 meters.

1 Kilometer = 0.62137 miles.
1 Mile = 1.6093 kilometers.
1 Knot = 6080.26 feet = 1.15156 miles.
1 Mile = 5280 feet.

MENSURATION

Area of circle = Diameter × Diameter × .7854
Circumference of circle = Diameter × 3.1416
Area of sphere (Surface) = Diameter × Diameter × 3.1416
Volume of sphere = Diameter × Diameter × Diameter × 0.5236

POWER MEASUREMENT

Alternating-Current Motor

$$\text{Single phase, horse power} = \frac{\text{Amperes} \times \text{volts} \times \text{Power factor} \times \text{Motor efficiency}}{746}$$

$$\text{Two phase, horse power} = \frac{2 \times \text{Amperes} \times \text{volts} \times \text{Power factor} \times \text{Motor efficiency}}{746}$$

$$\text{Three phase, horse power} = \frac{1.73 \times \text{Amperes} \times \text{volts} \times \text{Power factor} \times \text{Motor efficiency}}{746}$$

The efficiency and power factor may be assumed as 90% for general purposes; for accurate results correct percentage should be obtained.

Direct-Current Motor

$$\text{Horse Power} = \frac{\text{Amperes} \times \text{Volts} \times \text{Motor efficiency}}{746}$$

Pony Brake

$$\text{Brake Horse Power} = \frac{2 \times 3.1416 \times L \times W \times Rpm.}{33{,}000}$$

L = Length of brake arm from center of shaft to bearing point.
W = Net weight at end of brake arm.
Rpm.-- Speed in revolutions per minute of the brake shaft.

Pumps

$$\text{Water Horse Power} = \frac{G.p.m. \times 8\tfrac{1}{3}\# \text{ per gal.} \times \text{Total head in ft. of water}}{33{,}000}$$

G.*p.m.* = Gallons per minute

$$\text{Pump efficiency} = \frac{\text{Water horse power}}{\text{Brake horse power}}$$

Total head = Suction lift + discharge head.

$$\text{Velocity head} = \frac{V^2}{2g} = \frac{V^2}{2 \times 32.16} = \frac{V^2}{64.32}$$

V^2 = Velocity × Velocity

$$\text{Velocity} = \sqrt{2gh} = 8.02\sqrt{h}$$

or

$$\text{Velocity} \times \text{Velocity} = 2gh = 64.32h$$

ESTABLISHED VALUES

Specific gravity of water at 39.2°F. = 1 at normal atmospheric pressure.
Specific gravity of sea water at 39.2°F. = 1.02 to 1.03 at normal atmospheric pressure.
Specific gravity of mercury (*Hg*) 13.6; Boiling point 680°F.
Atmospheric pressure = 29.92 inches of mercury @ 32°F. = 30 inches of mercury @ 58.4°F.
= 33.93 feet of water. = 14.7 pounds per square inch.
Naparian logarithm (Log_e) = Common logarithm (Log_{10}) × 2.3026.
Absolute temperature = (460 + Temperature °F. above zero).

MERCURY COLUMN AND BAROMETER CORRECTIONS

0.00115 inch barometer change per foot altitude.
0.00278 inch barometer change per degree Fahrenheit.

DALTON'S LAW OF GASES

Every portion of a mass of gas enclosed in a vessel contributes to the pressure against the sides of the vessel the same amount that it would have exerted by itself had no other gas been present. That is the total pressure in the vessel is the sum of the partial pressures of each of the constituent gases.

CHARACTERISTIC EQUATION OF PERFECT GAS

$PV = WRT$

Let V = Volume of mixture cubic feet.
W = Weight of constituent gas in pounds.
R = The constant for corresponding gas (Air constant = 53.34)
(Steam Constant Approx. 85.5)
T = ABSOLUTE temperature of mixture.
P = Pressure in lbs. per SQ. FT.

WESTINGHOUSE ELECTRIC & MANUFACTURING COMPANY

Headquarters—306 4th Ave., Pittsburgh, Pa. P.O. Box 1017

*AKRON, OHIO, 106 South Main St.
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*ALLENTOWN, PA., 522 Maple St.
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†APPLETON, WISC., 1029 So. Outagamie St.
#†*ATLANTA, GA., 426 Marietta St., N. W.
xATTICA, N. Y.
*†AUGUSTA, MAINE, 9 Bowman St.
*BAKERSFIELD, CALIF., 2224 San Emedio St.
*BALTIMORE, MD., 118 E. Lombard St.
†BALTIMORE, MD., 501 East Preston St.
xBALTIMORE, MD., 2519 Wilkens Ave.
①*†BATON ROUGE, LA., 128-134 So. Sixteenth St.
*BINGHAMTON, N. Y., Suite 704, Marine Midland Bldg., 86 Court St.
*BIRMINGHAM, ALA., 1407 Comer Bldg.
*BLUEFIELD, W. VA., 208 Bluefield Avenue
*BOSTON, MASS., 10 High St.
†#BOSTON, MASS., 235 Old Colony Ave., So. Boston, Mass.
①*†BRIDGEPORT, CONN., 540 Grant St.
① *BUFFALO, N. Y., 814 Ellicott Square
†#BUFFALO, N. Y., 1132 Seneca St.
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#BUTTE, MONTANA, Iron & Wyoming Sts.
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*CEDAR RAPIDS, IOWA, 361 21st St., S.E., P. O. Box 148
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① *CHATTANOOGA, TENN., Volunteer State Life Bldg., Georgia Ave. & East Ninth St.
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†#CHICAGO, ILL., 2211 W. Pershing Road, P.O. Box 1103
*†#CINCINNATI, OHIO, 207 West Third St.
*†x#CLEVELAND, OHIO, 1216 W. Fifty-Eighth St.
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#DALLAS, TEXAS, 1712 Laws St.
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#EL PASO, TEXAS, 450 Canal St.
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#†EMERYVILLE, CALIF., 1466 Powell St.
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#HOUSTON, TEXAS, 2313 Commerce Ave.
†HOUSTON, TEXAS, 2315 Commerce Ave.
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*†JOHNSTOWN, PA., 107 Station St.
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#†KANSAS CITY, MO., 2124 Wyandotte St.
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xNEWARK, N. J., Plane & Orange St.
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#NEW ORLEANS, LA., 527 Poydras St.
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†NEW YORK, N. Y., 150 Varick St.
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#PHOENIX, ARIZONA, 425 Jackson St.
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†#ST. LOUIS, MO., 717 South Twelfth St.
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*SAN FRANCISCO, CALIF., 1 Montgomery St.
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†#SEATTLE, WASH., 3451 East Marginal Way
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°SOUTH PHILA. WKS., P.O. Box 7348, Philadelphia, Pa.
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*†SPRINGFIELD, MASS., 395 Liberty St.
xSPRINGFIELD, MASS., 653 Page Boulevard
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*TULSA, OKLA., 303 East Brady St.
*†#UTICA, N. Y., 113 N. Genesee St.
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*†WILKES-BARRE, PA., 267 N. Pennsylvania Ave.
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*YOUNGSTOWN, OHIO, 25 E. Boardman St.

Where address and P. O. box are both given, send mail to P. O. box, telegrams to address indicated.

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Westinghouse Electric Supply Company—Headquarters—150 Varick St., New York, N. Y.

Fully equipped sales offices and warehouses are maintained at all addresses.

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①Changed or added since previous issue. **October, 1940**

HP DOP. SEP BA Spl.

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‡DENVER, COL., The Mine & Smelter Supply Co.
‡EL PASO, TEX., Mine & Smelter Supply Co.
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SCRANTON, PA., Penn Elect'l Engineering Co.
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xBELLEVILLE, N. J., 720 Washington Ave.
zxBLOOMFIELD, N. J., Clearfield Ave.
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♯BOSTON, MASS., 235 Old Colony Ave., S. Boston, Mass.
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yNEW YORK, N. Y., 150 Broadway
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Headquarters—150 Broadway, New York, N. Y., U. S. A.

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*AUSTRALIA, SYDNEY, Box 2634-EE G.P.O.
*BRAZIL, RIO DE JANEIRO, Caixa Postal 1320
*BRAZIL, SAO PAULO, Caixa Postal 4191
① *COLOMBIA, MEDELLIN, Apartado Nacional 43
① *CHILE, SANTIAGO, c/o Wessel Duval y Cia. S.A.C., Casilla 86-D
*CUBA, HAVANA, Apartado 2289
*ENGLAND, LONDON, W.C. 2, 2 Norfolk St., Strand
① *INDIA, BOMBAY, Westinghouse Electric Co. of India Ltd., 294A Bazargate St.
*ITALY, MILANO, Piazza Crispi 3
*MEXICO, D. F. Mexico, Cia. Westinghouse Electric Internacional, Edificio la Nacional, Apartado 78 Bis.
*PANAMA, REPUBLIC, Panama, Apartado 742
*PERU, LIMA, Casilla 1685
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♯*DETROIT, MICH., 5757 Trumbull Ave.
*KANSAS CITY, MO., 410 Professional Bldg.
zxLONG ISLAND CITY, N. Y., 21-16 43rd Ave.
*LOS ANGELES, CALIF., 420 S. San Pedro St.
*MILWAUKEE, WISC., 534 North Broadway
*NEW ORLEANS, LA., 427 Baronne St.
*NEW YORK, N. Y., 173 E. Eighty-Seventh St.
*OMAHA, NEB., 117 N. Thirteenth St.
*PHILADELPHIA, PA., 3001 Walnut St.
*PITTSBURGH, PA., 3702 Fifth Ave.
*PORTLAND, OREGON, 1220 S. W. Morrison St.
① *ROCHESTER, N. Y., 1048 University Ave.
*SAN FRANCISCO, CALIF., 870 Market St.
*SEATTLE, WASH., 3451 E. Marginal Way

BRYANT ELECTRIC COMPANY
Headquarters—1421 State St., Bridgeport, Conn.

*BOSTON, MASS.
zxBRIDGEPORT, CONN., Main Plant, 1421 State St.
xBRIDGEPORT, CONN., Plastics Division Plant, 1105 Railroad Ave.
*♯CHICAGO, ILL., 844 West Adams St.
*♯LOS ANGELES, CALIF., 420 S. San Pedro St.
*NEW YORK, N. Y., 101 Park Ave.
*♯SAN FRANCISCO, CALIF., 325 Ninth St.

WESTINGHOUSE RADIO STATIONS
Headquarters—2519 Wilkins Ave., Baltimore, Md.

STATION KDKA, 310 Grant St., Pittsburgh, Pa.
STATION WBZ, 271 Tremont St., Boston, Mass.
STATION KYW, 1619 Walnut St., Philadelphia, Pa.
STATION WBZA, Hotel Kimball, Springfield, Mass.
STATION WOWO, 925 So. Harrison St., Fort Wayne, Ind.
STATION WGL, 925 So. Harrison St., Fort Wayne, Ind.

CANADIAN WESTINGHOUSE COMPANY, LIMITED
Headquarters—Hamilton, Ontario, Canada

*†♯CALGARY, 320 Eighth Avenue West, Calgary, Alberta, Can.
♯*EDMONTON, 10127, 104th St., Armstrong Block, Edmonton, Alberta, Can.
♯*FORT WILLIAM, 112 McVicar St., Fort William, Ontario, Can.
♯*HALIFAX, 158 Granville St., Halifax, Nova Scotia, Can., P.O. Box 204
xz*HAMILTON, Hamilton, Ontario, Can.
*LONDON, 504 Huron & Erie Bldg., London, Ontario, Can.
*MONTREAL, 1135 Beaver Hall Hill, Montreal, Quebec, Can.
§MONTREAL, 400 McGill St., Montreal, Quebec, Can.
♯†MONTREAL, 1844 William St., Montreal, Quebec, Can.
*NELSON, B. C. Can., P. O. Box 70
♯*OTTAWA, Ahearn & Soper Limited, P.O. Box 794, Ottawa, Ontario, Can.
*†♯REGINA, 2408 Eleventh Ave., Regina, Saskatchewan, Can.
*SASKATOON, 238 First Ave. N., Saskatchewan, Can.
†♯*SWASTIKA, Swastika, Ontario, Can.
*†♯TORONTO, 355 King St., West, Toronto, Ontario, Can.
*VANCOUVER, 1418 Marine Bldg., Vancouver, B. C., Can.
†♯VANCOUVER, 1090 Homer St., Vancouver, B. C., Can.
♯*†WINNIPEG, 158 Portage Ave. East, Winnipeg, Manitoba, Can.

①**Changed or added since previous issue.**
* Sales Office   † Service Shop   x Works   ♯ Warehouse   z Headquarters   y Executive Office   § Merchandising Products Only   ‡ Apparatus Products Only
◆ Service Office Only