A Digest of the Rules for the Design of Condensers and Air Pumps

Mechanical Engineering

B. S.

1912
A DIGEST OF THE RULES
FOR THE
DESIGN OF CONDENSERS AND AIR PUMPS

BY

HERMAN CHARLES KRANNERT

FOR THE

DEGREE OF BACHELOR OF SCIENCE

IN

MECHANICAL ENGINEERING

COLLEGE OF ENGINEERING

UNIVERSITY OF ILLINOIS

1912
UNIVERSITY OF ILLINOIS

June 1, 1912

THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

HERMAN CHARLES KRANNERT

ENTITLED A DIGEST OF THE RULES FOR THE DESIGN OF

CONDENSERS AND AIR PUMPS

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

DEGREE OF BACHELOR OF SCIENCE IN MECHANICAL ENGINEERING

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HEAD OF DEPARTMENT OF MECHANICAL ENGINEERING
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A DIGEST OF THE RULES GOVERNING
THE
DESIGN OF CONDENSERS AND AIR PUMPS

INTRODUCTION

A great many experiments have been conducted for the purpose of arriving at the constants which govern the design of condensers. On the other hand, considerable theory has also been evolved by designers based fundamentally upon the results obtained from practical experience and installations now in operation. These deductions embody to some degree the results of experimental investigations but are more or less empirical in nature.

The object of this thesis is to prepare a digest of the rules which govern the design of condensers. The author has proposed a method of design which includes the salient factors of the methods reviewed. Various problems are presented and analyzed showing the application of the method to any problem which presents itself. Due credit is here given to the works of R. M. Neilson, F. R. Low, and Professor Weighton, the text books of Professor G. F. Gebhardt, and Koerster as well as the catalogs of prominent manufacturers. Personal interviews with Mr. C. E. Anderson, of the Wheeler Condenser and Engineering Company, Mr. C. H. Wheeler, Jr., of the C. H. Wheeler Manufacturing Company, and Mr. Montgomery, the Chicago agent of the Alberger Condenser Company, were of invaluable assistance.
CHAPTER I

GENERAL DISCUSSION OF CONDENSERS

1. The function of a condenser in connection with a steam engine or turbine is primarily the reduction of the back pressure, though in some classes of work, such as marine, the recovery of the steam is of equal importance. A condenser is a device in which the process of condensation and subsequent removal of the air and condensed steam is continuous. The degree of vacuum depends upon the amount of air taken in with the steam, the tightness of the system, and the temperature to which the steam is reduced. A pound of dry steam at atmospheric pressure (30 inches of mercury) occupies a volume of 26.79 cubic feet. If this steam were condensed in a closed vessel, and its temperature lowered to 110° Fahr., it would then have a volume of approximately \( \frac{1}{1700} \) of the original volume and the vapor pressure would be 2.60 inches of mercury. With a temperature of 100° Fahr., the pressure would be practically 2 inches of mercury; the lower the pressure, the higher would be the degree of vacuum.

If air is mixed with the steam, the vacuum will be more imperfect. Considering that one pound of steam and one-tenth pound of air be mixed in the same vessel under atmospheric pressure, the volume would be 26.79 + 1.71 = 28.50 cubic feet. After the steam has been condensed and the pressure reduced to 110° Fahr., the absolute pressure would be 2.60 + 1.53 = 4.13 inches of mercury or a vacuum of 25.87 inches. The purpose of the condenser is thus to reduce the volume and temperature of the steam, and the function of the pumps is to remove the condensate, air, and other vapors at such a rate as to maintain the required degree of vacuum.

2. In general, condensers are classified into two groups: jet and surface. A jet condenser is of such design that the steam and water are brought into intimate contact thus bringing about the condensation. A surface condenser
is that type in which the steam is exhausted into a large closed shell where it comes into contact with tubes which are cooled by means of circulating water; this contact condenses the steam and brings about the desired vacuum. Suitable pumps are required to handle the condensate, air, and vapors, and the cooling water in both types.

Jet Condensers

Parallel flow
Ordinary
Siphon
Ejector
Counter flow
Prometric

3. The classification of jet condensers is made first according to the direction of flow of the cooling water. The parallel flow type are further subdivided as shown. In the "ordinary type" the steam exhausts directly into a tube which converges into the cylinder of the pump. The pump creates a vacuum in the tube and thus water enters at a point somewhat above the exhaust steam inlet and mingles with the steam, cooling and condensing it on its downward path. The pump here must be of such a capacity to handle the water, condensed steam, vapors, and air. With the "siphon type" the water is passed into a converging and then diverging tube, giving it a high velocity. The exhaust steam is brought in over a cone which spreads it out and causes it to mix intimately with the entering cooling water. The jet of water having a high velocity is exhausted into a pipe which is about 34 feet in length above the level of the well. The length of tail pipe is made 34 feet which is equivalent to about a 30 inch mercury column. In practice 24 to 25 inches vacuum can be maintained with this type of condenser though some instances have shown a higher value. When the syphon condenser is once put into operation by forcing the water through it, the suction action of the condenser will lift the water about fifteen feet. The supply pump thus has only to raise the water to this level. The advantage of this type is that muddy water can be used without seriously affecting the condenser action in any way.

The ejector type, as the name indicates, operates on the ejector principle. A vacuum is created and this suction along with the velocity of the exhaust steam imparts a high velocity to the water, the steam being condensed upon coming into
4. Surface condensers are classified into divisions depending upon the method of cooling and further depending upon the number of passages of the cooling fluid. The evaporative condenser is used where there is a scarcity of cooling water. This type will not be discussed in this thesis.
CHAPTER II

COOLING WATER FOR JET CONDENSER

1. In a jet condenser the cooling water and exhaust steam mingle and the degree of vacuum is a function of the final or discharge temperature of the mixture. The quantity of cooling water depends upon its initial temperature, the temperature of the discharge water and the total heat in the steam entering the condenser. In calculations for general use, it is accurate enough to consider the heat of the entering steam at the pressure corresponding to that of the condenser, although considerable moisture is sometimes exhausted into the condenser.

Let \( \Delta = \) total heat of steam at condenser pressure above 32 degrees Fahr.

\[ T_1 = \text{the temperature of initial water} \]

\[ T_2 = \text{the temperature of the discharge water} \]

\[ W = \text{weight of cooling water in pounds necessary to condense and cool one pound of steam to the required discharge temperature} \]

\[ W = \frac{\Delta - T_2 + 32}{T_2 - T_1} \]  \( \text{(1)} \)

The range of temperatures found in practice are shown in Table 1 and \( W \), the weight of water, may be found from Fig. 1, 2, 3, or 4.

2. It might be assumed since a high vacuum is desirable, that it would be possible to increase the vacuum by exhausting the aqueous vapor and other gases. The following problem will illustrate this point. Consider a 100 horse power steam turbine with a steam consumption of 20 pounds per horse power hour; a vacuum of 26 inches is produced by the condenser with a 30 inch barometer, the discharge water leaves at 125 degrees and 25 pounds of cooling water per pound of steam condensed is required. The total amount of cooling water handled per minute in the above plant is \( \frac{100 \times 20 \times 25}{60} = 833 \) pounds. To increase the
vacuum one pound requires an approximate increase of two inches of mercury. The water must then be lowered to a temperature of 101.15 degrees or for the problem call it 102 degrees. This means that $833 \times (125 - 102) = 19,159$ B. t. u., that must be abstracted from the water in one minute, or $19,159 \div 1050 = 18.6$ pounds of water to be evaporated in a minute. One pound of vapor at from 102 to 125 degrees has an average volume of 270 cubic feet. $18.6 \times 270 = 5022$ cubic feet of vapor must be exhausted per minute to increase the vacuum from 26 to 28 inches which apparently is impracticable.

To increase the degree of vacuum it is seen from (1) that the temperature of the entering cooling water should be low so as to have the temperature within the condenser as low as possible. Further the circulation should be arranged so that the air and other vapors are cooled to as great a degree as possible thus decreasing their volume.
CHAPTER III

COOLING WATER FOR SURFACE CONDENSERS

1. In the surface condenser the rate of condensation is dependent upon:

1. Inlet temperature of circulating water
2. Velocity of circulating water
3. Diameter of tubes
4. Length of tubes
5. Number of passes made by water
6. Cleanness of tubes
7. Vacuum
8. Temperature and wetness of steam when entering the condenser
9. Amount of air entering the condenser with the steam
10. General design of condenser as regards flow of steam
11. Capacity of the air pump.

The inlet temperature of the water is usually fixed by local conditions and varies only from season to season. Insofar as the difference between the temperature at the pressure of the condenser and that of the circulating water determines the amount of water that must be pumped, it is essential to have the initial temperature as low as possible. Special means, such as sprays, are used to reduce the temperature. The limiting factor will be the expense incurred by cooling the water and that of pumping the additional circulation water.

Various experiments have been performed showing that the rate of transmission is a function of the velocity of the steam and water, the diameter, and the length of the tubes. It has been shown that the rate of transmission is not directly proportional to the temperature difference but that other peculiarities of the surface enter into the final relation. The cleanness of the tubes which is dependent upon the water used is also a vital factor in the rate
of transmission. It has been found that the value is increased by having the water make several passes through the condenser; the explanation for this is that the mixing that occurs when the water leaves a tube tends to break up the core or central column of water which is often colder than that in contact with the sides of the tube. It is known that the rate of transmission varies with the density of a gaseous fluid although no definite data are available as to how the variation takes place with the degree of vacuum. It is obvious that the condensation rate may be lowered if the steam carries a large amount of water with it into the condenser.

The amount of air in the mixture is of rather great importance as it must be withdrawn from the system and is not altered in volume to any great degree. The percentage is one of greater or less speculation depending to a great extent upon whether the unit is air tight. For calculation from two to five per cent of the mixture entering the condenser is considered as air. In larger units the percentage of air may be taken even less than the two per cent depending entirely upon the installation. The general design of the condenser proper, the tubes, turns and bends, all affect the velocity of the steam and water and hence are determining factors of the rate of condensation. The facility with which the air and vapors can be handled affect, to some extent, the degree of vacuum. Sufficient excess capacity is allowed in the selection of the air pump to handle the volume necessary and an increase in the speed of the pump will readily affect the desired displacement if necessary.

2. Let \( T_1 \) = temperature of entering water 
   \( T_2 \) = temperature of discharge water 
   \( T_3 \) = temperature of steam at condenser pressure 
   \( T_4 \) = temperature of condensed steam.

The range of temperatures found common in practice for surface condensers is: \((T_2 - T_1)\) 15 to 25 degrees with 15 as a minimum for usual practice. \((T_3 - T_2)\) 6 to 20 degrees with an average of 15 degrees. \((T_3 - T_4)\) 6 to 15 degrees.

The same ranges of temperatures hold for \((T_2 - T_1)\) and \((T_3 - T_2)\) for jet condensers.
TABLE I

<table>
<thead>
<tr>
<th>Vacuum</th>
<th>Temperatures</th>
</tr>
</thead>
<tbody>
<tr>
<td>In. of Hg.</td>
<td>$T_1$</td>
</tr>
<tr>
<td>25 inches</td>
<td>65 to 75</td>
</tr>
<tr>
<td>26 inches</td>
<td>65 to 75</td>
</tr>
<tr>
<td>27 inches</td>
<td>65 to 75</td>
</tr>
<tr>
<td>28 inches</td>
<td>60 to 75</td>
</tr>
<tr>
<td>29 inches</td>
<td>55 to 70</td>
</tr>
</tbody>
</table>

It will be seen that the temperature drops are limited by the temperature of the steam at condenser pressure and the temperature of the entering cooling water. It is common to consider the lower temperature as 70 degrees although this may be somewhat lower in the winter. As most steam turbine installations are figured on a 28 inch vacuum the drops mentioned above are found to be applicable.

3. The amount of cooling water required per pound of steam in a surface condenser is dependent upon the vacuum, the temperature of the condensed steam, and the rise in the temperature of the cooling water.

Let $\lambda = \text{total heat of exhaust steam above 32 degrees Fahr.}$

$T_4 = \text{temperature of the condensed steam}$

$T_1 = \text{temperature of injection water}$

$T_2 = \text{temperature of discharge water}$

$W = \text{weight in pounds of injection water necessary to condense one pound of steam.}$

Then approximately

$$W = \frac{\lambda - T_4 + 32}{T_2 - T_1}$$
Figure 1

Value of \( W \)

25 inch vacuum

1. \( T_2 - T_1 = 0 \)
2. \( T_2 - T_1 = 10 \)
3. \( T_2 - T_1 = 20 \)

---

Value of \( W \) - Gallons of pounds:

---

\( T_2 - T_1 \)
FIGURE 2

VALUE OF W
26 INCH VACUUM

I $T_3 - T_4 = 0$
II $T_3 - T_4 = 10$
III $T_3 - T_4 = 20$

- VALUE OF W

- 0

- POUNDS

- GALLONS

- $T_2 - T_1$
Figure 3

Value of $W$

27 Inch Vacuum

1. $T_2 - T_4 = 0$
2. $T_2 - T_4 = 10$
3. $T_2 - T_4 = 20$
Figure 1

Value of W
28 inch vacuum

\[ W = \frac{F}{T} \]

I. \( T_2 - T_1 = 0 \)
II. \( T_2 - T_1 = 10 \)
III. \( T_2 - T_1 = 20 \)
The range of temperatures found common in surface condenser practice are shown in Table I. Table II shows the weight of cooling water necessary to condense one pound of steam under various vacua and varying temperatures of water. These values have been plotted in Fig. 1, 2, 3, and 4.

### TABLE II

<table>
<thead>
<tr>
<th>Vacuum</th>
<th>$T_2 - T_1$</th>
<th>$T_3 - T_4$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0</td>
<td>5</td>
</tr>
<tr>
<td>25 inches</td>
<td>10</td>
<td>101.65</td>
</tr>
<tr>
<td></td>
<td>15</td>
<td>67.77</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>50.62</td>
</tr>
<tr>
<td></td>
<td>25</td>
<td>40.64</td>
</tr>
<tr>
<td>26 inches</td>
<td>10</td>
<td>102.13</td>
</tr>
<tr>
<td></td>
<td>15</td>
<td>68.10</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>51.06</td>
</tr>
<tr>
<td></td>
<td>25</td>
<td>40.85</td>
</tr>
<tr>
<td>27 inches</td>
<td>10</td>
<td>102.71</td>
</tr>
<tr>
<td></td>
<td>15</td>
<td>68.47</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>51.36</td>
</tr>
<tr>
<td></td>
<td>25</td>
<td>41.08</td>
</tr>
<tr>
<td>28 inches</td>
<td>10</td>
<td>103.49</td>
</tr>
<tr>
<td></td>
<td>15</td>
<td>68.99</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>51.75</td>
</tr>
<tr>
<td></td>
<td>25</td>
<td>41.39</td>
</tr>
</tbody>
</table>

By comparing (2) with (1), it is seen that when $T_4 = T_2$, which is the case in jet condensers, the same formula will apply to either type.
Letting $T_3$ represent the temperature of the steam at condenser pressure, then

$$(T_3 - T_2)$$

gives the drop in temperature of the steam in the jet condenser, while

$$(T_3 - T_4)$$

gives the drop in temperature of steam in the surface condenser. Fig. 1, 2, 3, and 4 thus give the value of $W$ for both surface and jet condensers.

The variation of $W$ is not great for either variation in the temperature ranges $(T_2 - T_1)$, or $(T_3 - T_2$ or $T_4)$, nor for a variation in the degree of vacuum. In these calculations the quality of steam was considered 100 per cent, although it is common practice to consider a quality of 97 per cent. This gives a net heat content of approximately 1000 B. t. u., which is commonly used in condenser calculations.

An average curve may be used taking a heat content of 1000 B. t. u. for any particular degree of vacuum, and this will give a value of $W$ which is sufficiently accurate for practical purposes.

4. The manufacturers of condensers have determined from experience certain constants which they apply to ascertain the amount of circulating water necessary. These factors have been found to vary from 40 to 60 according to the design. Consider a turbine using 20,000 pounds of steam an hour or 833 pounds per minute. Considering $(T_2 - T_1)$ as 20 degrees, Fig. 4 shows $W = 6.2$ gallons per pound of condensate, and a total of $833 \times 6.2 = 2065$ gallons per minute for the entire unit.

A condensate of 20,000 pounds is equivalent to $\frac{20,000}{60 \times 8.3} = 40$ gallons per minute. Using a ratio of 50 to 1 we get $W = 50 \times 40 = 2,000$ gallons per minute.

The factor used is determined by taking the ratio of the total heat ($A = 1000$) to the rise in temperature $(T_2 - T_1)$. Thus $\frac{1000}{90 - 70} = 50$ as in the above example. With a rise of $(T_2 - T_1) = 17.5$ the ratio $\frac{1000}{17.5}$ is 57.3 or 60 in round numbers. This gives a value of $W = 60 \times 40 = 2400$ gallons per minute for the above unit. From Fig. 4 $W$ for $(T_2 - T_1) = 7.1$ gallons.

$$7.1 \times 333 = 2365$$

gallons per minute for total $W$. This shows a close agreement of the two methods used.
5. A pump is selected having the desired capacity allowing for any future overload, and which is capable of pumping this capacity against the desired head. The head will be determined by the particular design of the plant including the bends and water levels.

Let \( h \) = head in feet

\( g \) = gallons per minute of circulating water

\( \eta \) = efficiency of steam to water transmission

\( H \) = horse power of pump

Then \( H = \frac{h \times g \times 8.3}{33,000 \times \eta} \) \( (3) \)

\( \eta \) is taken usually as 60 per cent, although the range for different pumps is from 50 to 70 per cent.
CHAPTER IV

THEORETICAL COOLING SURFACE FOR SURFACE CONDENSERS

1. Theoretically the cooling within the condenser is divided into two periods, (1) that during which the heat of vaporization of the fluid is withdrawn, and (2) that in which the temperature of the liquid is reduced. As the processes are continuous it is sufficiently accurate to consider factors which eliminate the more minute analysis of the double process.

Let 

\[ S = \text{cooling surface in square feet} \]

\[ \Lambda = \text{total heat above 32 degrees of exhaust steam at condenser pressure} \]

\[ T_1 = \text{temperature of entering water} \]

\[ T_2 = \text{temperature of discharge water} \]

\[ T_3 = \text{temperature of steam at condenser pressure} \]

\[ T_4 = \text{temperature of condensed steam} \]

\[ U = \text{coefficient of heat transmission B. t. u. per hour per degree difference in temperature per square foot of cooling surface} \]

\[ T_m = \text{mean difference in temperature between } T_1, T_2, \text{ and } T \]

\[ W = \text{weight of steam condensed per hour} \]

\[ T_m = \frac{T_3 - \frac{T_1 + T_2}{2}}{2} \text{ arithmetic mean} \]  

\[ T_m = \frac{T_2 - T_1}{\log_e \frac{T_3 - T_2}{T_3 - T_2}} \text{ logarithmic mean} \]  

Assuming that the heat absorbed by the water is equal to the heat in the steam we have

\[ S U T_m = W (\Lambda - T + 32) \]  

\[ S = \frac{W (\Lambda - T + 32)}{U T_m} \]
2. The value of \( U \) varies with the material of the tube, character of the surface, and the velocity of the fluids. The velocity of the water in the tubes is on the average about 50 feet per minute with 150 feet maximum and 30 feet minimum. Ordinarily \( U \) is taken as 300 although in special designs with high velocity of water and numerous masses the value is pushed from 400 to 600.

3. For the arithmetic mean

\[
T_m = T_3 - \frac{T_2 + T_1}{2}
\]

\[
= \frac{2T_3 - T_2 - T_1 + T_2 - T_1}{2}
\]

\[
= \frac{2(T_3 - T_2) + (T_2 - T_1)}{2}
\]

\[
= (T_3 - T_2) + \frac{T_2 - T_1}{2}
\]

\( (4a) \)

Table III gives the values of \( T_m \) for temperatures within a practical range shown in Table I for (4).

**TABLE III**

<table>
<thead>
<tr>
<th>( T_3 - T_2 )</th>
<th>( T_2 - T_1 )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10</td>
</tr>
<tr>
<td>15</td>
<td>20</td>
</tr>
<tr>
<td>12</td>
<td>17</td>
</tr>
<tr>
<td>9</td>
<td>14</td>
</tr>
<tr>
<td>6</td>
<td>11</td>
</tr>
<tr>
<td>4</td>
<td>9</td>
</tr>
</tbody>
</table>

For the logarithmic mean

\[
T_m = \frac{T_2 - T_1}{\log_e \frac{T_3 - T_1}{T_3 - T_2}} = \frac{T_2 - T_1}{\log_e \left( 1 + \frac{T_2 - T_1}{T_3 - T_2} \right)}
\]

\( (5a) \)
Figure 5
VALUES OF $T_m$
Table IV gives the values of $T_m$ for temperatures within the range shown in Table I for (5).

<table>
<thead>
<tr>
<th>$T_3 - T_2$</th>
<th>$T_2 - T_1$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10</td>
</tr>
<tr>
<td>15</td>
<td>19.5</td>
</tr>
<tr>
<td>12</td>
<td>16.5</td>
</tr>
<tr>
<td>9</td>
<td>13.4</td>
</tr>
<tr>
<td>6</td>
<td>10.2</td>
</tr>
<tr>
<td>4</td>
<td>7.9</td>
</tr>
</tbody>
</table>

The above value of $T_m$ from (4) and (5) have been plotted in Fig. 5 and show the difference in the value for the two methods of calculation.

4. The values of $S$ per pound of condensate were calculated within the range of temperatures shown in Table I using a value of $U = 300$. These values are shown in Table V and are plotted in Fig. 6.

<table>
<thead>
<tr>
<th>$T_m$</th>
<th>$T_3 - T_4$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>6</td>
</tr>
<tr>
<td></td>
<td>Vacuum</td>
</tr>
<tr>
<td>25&quot;</td>
<td>28&quot;</td>
</tr>
<tr>
<td>10</td>
<td>341</td>
</tr>
<tr>
<td>15</td>
<td>227</td>
</tr>
<tr>
<td>20</td>
<td>170</td>
</tr>
<tr>
<td>25</td>
<td></td>
</tr>
<tr>
<td>30</td>
<td></td>
</tr>
</tbody>
</table>
Figure 6
Square Feet of Surface
for Surface Condensers

- - - - 25 inch Vacuum
--- - - - 28 inch Vacuum

T_m

T_3 - T_2 = 15
T_3 - T_2 = 6
T_5 - T_2 = 15
T_5 - T_2 = 6

15 per Pound of Condensation
5. Considering a 1000 k. w., unit using 20,000 pounds of steam per hour. If \((T_2 - T_1) = 15\) degrees and \((T_m - T_2) = 17\) degrees we find from Fig. 5 that \(T_m\) is 23 and from Fig. 6, \(S\) is 0.15. This gives 20,000 x 0.15 equals 3000 square feet of surface. A ratio of from 2 1/2 to 4 has been assumed by certain manufacturers for ascertaining the value desired, approximately.

The square feet of surface is determined by multiplying the k. w. capacity by the above factor. Thus assuming 3 as an average we have 3 x 1000 equals 3000 square feet.
CHAPTER V

DISCUSSION OF PUMPS

1. Condenser pumps may be divided into two classes: (a) wet air pumps, and (b) dry air pumps. The former handles both water and air, and the later air alone. Ordinary jet condenser wet air pumps handle simultaneously the circulating water, condensed steam, entrained air, and vapors. The surface condenser wet air pump handles the condensed steam, air, and vapors only. Wet air pumps may be driven by the main engine or independently and may be direct acting or fly-wheel driven. The fly-wheel type may be steam, electric, or belt driven. Dry air pumps are virtually air compressors, compressing air from the condenser pressure to that of the atmosphere. They are generally of the fly-wheel type.

2. In determining the amount of air, it is generally assumed that the percentage of air entering is from 2 to 5 per cent of the volume entering the condenser. The volume of the vapor, condensate, and cooling water can readily be determined thus giving the theoretical capacity of the pump. It is essential to make certain allowances for losses and slippage and the manufacturers have done this by determining various factors of design.

3. For a 28 inch vacuum and 70 degree cooling water one manufacturer considers the wet air pump for a surface condenser should be 30 times the volume of the condensate; for a 27 inch vacuum a ratio of 25 to 1 is used; and for a 26 inch vacuum the ratio is considered as 15 to 1. Other manufacturers exceed this ratio depending upon the design of the apparatus and make of pump. Where a dry air pump is used as in the Alberger surface condenser, the hot well pump displacement is made 2 or 3 times the volume of the condensate. The dry air pump is given a displacement of 1 cubic foot for each pound of condensate with a 28 inch vacuum.

4. In a jet condenser the displacement of the pump is made about 54 times the volume of the condensate for a 26 inch vacuum and 70 degree cooling water. In an installation where a higher degree of vacuum is desired a dry air
pump is used in conjunction with the wet air pump. Thus for a 28 inch vacuum and 70 degree cooling water the volume of the wet air pump is made about 54 times the volume of the condensate, and the dry air pump displacement is made 90 times the volume of the condensate. It is customary to calculate all volumes and displacements in gallons per minute for the sizes of pumps can be determined on this rating.

In the jet condensers the wet air pump acts as a circulating pump if a force pump is not used. In the surface condensers a pump must be installed to circulate the water through the tubes. This pump may be of the following types, (a) direct acting steam, (b) fly-wheel or rotative, (c) centrifugal, (d) rotary, (e) power driven. The volume of circulating water necessary for a given condition may be found from Fig. 1, 2, 3, or 4.

5. In calculating the horse power of the engine necessary to drive the pump an average head of 15 feet may be taken although the head will depend to a large extent upon the design of the installation, including the pipe sizes and turns.
CHAPTER VI

THEORETICAL DISCUSSION OF PUMP CYCLE

1. The following discussion of the air cycle shows up several points of more theoretical interest than of practical value in the design of pumps.

In the above cycle (4 - 1) is the suction line taking in the air at the condenser pressure. The line (1 - 2) is compression which is assumed adiabatic; (2 - 3) is the exhaust line in which air is forced out of cylinder. The distance (3 - 5) represents the amount of clearance in the cylinder. The line (3 - 4) represents the adiabatic expansion of the residual air in the clearance volume. In the following analysis it is assumed that the media is air and that the expansion and compression are adiabatic.

2. The work of the cycle is obtained by adding the several areas and subtracting the negative work; thus

\[ W = W_{12} + W_{25} - W_{61} - W_{3564} \]

\[ = W_{12561} - W_{3564} \]

where \( W_{12} \) = work of adiabatic compression
\[ W_{25} = \text{work of exhausting air} \]
\[ W_{61} = \text{work of suction} \]
\[ W_{3564} = \text{net work of expansion} \]
\[ W_{12561} = \text{net work of compression} \]
\[ W_{12} = \frac{p_1 v_1 - p_2 v_2}{n - 1} \]
\[ W_{25} = -p_2 v_2 \]
\[ W_{61} = p_1 v_1 \]
\[ W_{12561} = \frac{p_1 v_1 - p_2 v_2}{n - 1} + p_1 v_1 - p_2 v_2 = \frac{n}{n - 1} (p_1 v_1 - p_2 v_2) \]
\[ W_{3564} = -\frac{n}{n - 1} (p_4 v_4 - p_3 v_3) \]
\[ W = W_{12561} - W_{3564} \]
\[ = \frac{n}{n - 1} (p_1 v_1 - p_2 v_2) - \frac{n}{n - 1} (p_4 v_4 - p_3 v_3) \]
\[ = \frac{n}{n - 1} p_1 v_1 \left[ 1 - \left(\frac{p_2}{p_1}\right)^\frac{n - 1}{n} \right] - \frac{n}{n - 1} p_4 v_4 \left[ 1 - \left(\frac{p_3}{p_4}\right)^\frac{n - 1}{n} \right] \]

Since \( p_3 = p_2 \) and \( p_4 = p_1 \)
\[ W = \frac{n}{n - 1} \left[ 1 - \left(\frac{p_2}{p_1}\right)^\frac{n - 1}{n} \right] (p_1 v_1 - p_1 v_4) \]
\[ v_4 = v \left(\frac{p_3}{p_4}\right)^\frac{1}{n} = k v \left(\frac{p_2}{p_1}\right)^\frac{1}{n} \]

where \( k \) is the ratio of \( v_3 \) to \( v_1 \)
\[ W = \frac{n}{n - 1} \left[ 1 - \left(\frac{p_2}{p_1}\right)^\frac{n - 1}{n} \right] \left[ p_1 v_1 - k p_1 v_1 \left(\frac{p_2}{p_1}\right)^\frac{1}{n} \right] \]
\[ W = \frac{n p_1 v_1}{n - 1} \left[ 1 - \left(\frac{p_2}{p_1}\right)^\frac{n - 1}{n} \right] \left[ 1 - k \left(\frac{p_2}{p_1}\right)^\frac{1}{n} \right] \]
\[
= \frac{n v_1}{n-1} \left[ 1 - \left( \frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - k \left( \frac{p_2}{p_1} \right)^{\frac{1}{n}} \right] + k p_2
\]

\[
\frac{dW}{dp_1} = \frac{n v_1}{n-1} \left[ 1 - \frac{1}{n} \left( \frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - \frac{1-n}{n} k \left( \frac{p_2}{p_1} \right)^{\frac{1}{n}} \right] = 0 \quad \text{(8)}
\]

\[
\frac{d^2W}{dp_1^2} = \frac{n v_1}{n-1} \left[ -\frac{1}{n} \left( \frac{1-n}{n} \frac{p_2}{p_1}^\frac{n-1}{n} + \frac{1-2n}{n} \right) k \left( \frac{p_2}{p_1} \right)^{\frac{1}{n}} \right] \quad \text{(10)}
\]

When \( n = 1.3 \), (9) becomes

\[
1 - 0.77 R^{0.231} = 0.231 k R \quad \text{(9a)}
\]

And when \( n = 1.4 \) (9) becomes

\[
1 - 0.715 R^{0.285} = 0.285 k R \quad \text{(9b)}
\]

Solving for \( k \)

\[
k = \frac{1 - 0.77 R^{0.285}}{0.231 R^{0.77}} \quad \text{from (9a)}
\]

\[
k = \frac{1 - 0.715 R^{0.285}}{0.285 R^{0.715}} \quad \text{from (9b)}
\]

3. From the theoretical discussion the factor \( k \) which represents the clearance volume enters into and it is possible to solve (9a) and (9b) for various degrees of vacuum which gives the clearance necessary to obtain maximum work. These values have been plotted in Fig. 7 and 8. Fig. 7 gives the relation between \( R \) and \( k \). \( R \) represents the ratio of atmospheric pressure (30 inches of mercury) to the absolute pressure in the condenser. Fig. 8 shows the relation between \( k \) and the absolute condenser pressure. The values are also shown in
Table V, VI, and VII. From this discussion the value of clearance for any vacuum can be found although the values of $k$ are negative for certain values of $p_1$. These results are of little practical value.

**TABLE VI**

$n = 1.3$

<table>
<thead>
<tr>
<th>Vacuum In. of Hg.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>10</th>
<th>15</th>
<th>30</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R$</td>
<td>30</td>
<td>15</td>
<td>10</td>
<td>7.5</td>
<td>6</td>
<td>5</td>
<td>3</td>
<td>2</td>
<td>1</td>
</tr>
<tr>
<td>$k$</td>
<td>-0.217</td>
<td>-0.236</td>
<td>-0.23</td>
<td>-0.206</td>
<td>-0.181</td>
<td>-0.148</td>
<td>0.0185</td>
<td>0.241</td>
<td>1.0</td>
</tr>
</tbody>
</table>

**TABLE VII**

$n = 1.4$

<table>
<thead>
<tr>
<th>Vacuum In. of Hg.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>10</th>
<th>15</th>
<th>30</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R$</td>
<td>30</td>
<td>15</td>
<td>10</td>
<td>7.5</td>
<td>6</td>
<td>5</td>
<td>3</td>
<td>2</td>
<td>1</td>
</tr>
<tr>
<td>$k$</td>
<td>-0.268</td>
<td>-0.278</td>
<td>-0.258</td>
<td>-0.221</td>
<td>-0.186</td>
<td>-0.145</td>
<td>0.024</td>
<td>0.278</td>
<td>1.0</td>
</tr>
</tbody>
</table>

4. Using a clearance of 3 per cent, which is common practice, the value of the work $W$ was calculated for various degrees of vacuum. These results are shown in Tables VIII and IX and in Fig. 9 and 10.

**TABLE VIII**

$n = 1.3$

<table>
<thead>
<tr>
<th>Vacuum In. of Hg.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>10</th>
<th>15</th>
<th>30</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R$</td>
<td>30</td>
<td>15</td>
<td>10</td>
<td>7.5</td>
<td>6</td>
<td>5</td>
<td>3</td>
<td>2</td>
<td>1</td>
</tr>
<tr>
<td>$W$</td>
<td>0.302</td>
<td>0.572</td>
<td>0.755</td>
<td>0.877</td>
<td>0.982</td>
<td>1.049</td>
<td>1.190</td>
<td>1.078</td>
<td>0</td>
</tr>
</tbody>
</table>
TABLE IX

\[ n = 1.4 \]

<table>
<thead>
<tr>
<th>Vacuum In. of Hg.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>10</th>
<th>15</th>
<th>30</th>
</tr>
</thead>
<tbody>
<tr>
<td>R</td>
<td>30</td>
<td>15</td>
<td>10</td>
<td>7.5</td>
<td>6</td>
<td>5</td>
<td>3</td>
<td>2</td>
<td>1</td>
</tr>
<tr>
<td>W</td>
<td>0.377</td>
<td>0.646</td>
<td>0.825</td>
<td>0.952</td>
<td>1.041</td>
<td>1.109</td>
<td>1.23</td>
<td>1.093</td>
<td>0</td>
</tr>
</tbody>
</table>

The maximum amount of work is obtained from 19 to 20 inches vacuum. It is to be observed, however, that the degree of vacuum most desirable for the pump is not that which gives the greatest efficiency for the unit which is the deciding factor.
Figure 7

Ratio of $P$ to $K$

$R = \frac{P_2}{P_1} = \frac{14.7}{P_i} = \frac{20''}{P_i}$

$K = \text{CLEARANCE RATIO}$
FIGURE 8
RATIO OF P, AND K

P, ABSOLUTE - INCHES MERCURY
VACUUM - INCHES MERCURY
Figure 10
Ratio of $p_r$ and $W$

$\eta = 1.4$
$\eta = 1.3$

$0 \quad 10 \quad 20 \quad 30$
$p$, Absolute Inches Mercury

0 10 20 30
Vacuum - Inches Mercury