

VOLUMETRIC EFFICIENCY OF COMPRESSORS

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Introduction

In this paper an attempt has been made to collect all of the possible means of determining the volumetric efficiency of compressors. The ammonia compressor only is considered in this study, but the deductions and formulas as shown are applicable to all types of compressors with only very slight changes.

Very little of the information in this paper is original with the writer, it is a collection of observations and work of the leading authorities of this country on the subject of refrigeration. There has not been, to my knowledge, a very extended study of this particular phase of the subject of refrigeration by anyone person. For this reason, there is not available under one cover a complete discussion on volumetric efficiency. The purpose of this paper was to collect this material together.

I have tried to give the source of all borrowed information. Any omission in this regard is an oversight. I am especially indebted to a number of the leading refrigerating machine builders for the valuable information which they kindly supplied.

General Survey of the Field

Until a few years ago very little thought was given to the subject of the volumetric efficiency by refrigerating engineers. They knew what it was, and some of the causes, but very little significance was attached to the volumetric efficiency of compressors. Ice making and mechanical refrigeration is a comparatively recent development. Like all other things, the earliest refrigerating machines were very crude and little was known of their actual operation. As the science gradually progressed, refrigerating engineers became more interested in how to make the machines more reliable and efficient. When study was given to the compressor, a number of conditions were found which could be improved greatly. Professor J.E.Denton in his tests in a 75-ton refrigerating machine in the Knickerbocker Brewery in New York City was probably the first person to make any accurate or worthwhile tests on compressors. This test was carried on in about the year 1890. The ammonia tables and data were not accurate then, but Professor Denton metered the ammonia used and made careful measurements of temperatures and pressures. Since more accurate ammonia tables have been compiled Professor Denton's results have been very valuable in checking the theory and also in presenting the data for some valuable deductions. Since Professor Denton's work numerous other extensive tests have been conducted. Probably the most notable are those of the York Manufacturing Company and other manufacturers of refrigerating equipment.

In this discussion an attempt has been made to gather the existing data with regard to ammonia compressors and to choose what is considered to be the best available for the use of the operating engineer. In collecting this data the work of a number of the outstanding authorities on refrigeration were studied. Also, some data was collected by the writer on a 150-ton De La Vergue horizontal slow-speed double acting simple compressor and an 80-ton Ingersoll-Rand horizontal high speed double acting compound compressor.

General Theory

The term volumetric efficiency of ammonia compressors is a very much misunderstood expression in the refrigerating world to-day. The term volumetric efficiency without any qualifying statement does not mean much.

Apparent volumetric efficiency is defined as the ratio of that part of the compressor strokes during which ammonia is entering the cylinder to the total compressor stroke. Apparent volumetric efficiency shows the effect of clearance upon the operation of the compressor. A more extended discussion of this will be given later.

Superheating of the ammonia gas as it enters the compressor cylinder causes a loss of efficiency. This effect is shown by a term called volumetric efficiency due to superheating.

Real or true volumetric efficiency of a compressor is the ratio of the actual weight of ammonia handled by the compressor per stroke to the weight of the ammonia gas at the temperature and pressure just ahead of the suction valve that the cylinder would handle if filled.

Letting E_c = volumetric efficiency due to clearance

and E_s = volumetric efficiency due to superheating

$$E_t = E_c E_s, \text{ where } E_t = \text{real volumetric efficiency.} \text{-----}(1)$$

The volumetric efficiency of a refrigerating plant is often interputed to mean the volumetric efficiency of a compressor.

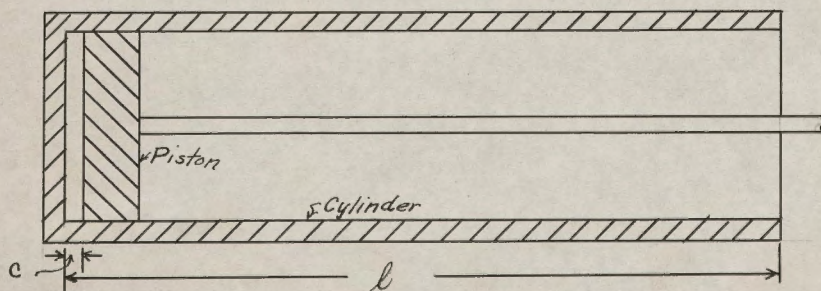
By using the piston displacement of the compressor, temperature at the compressor, pressures at the scale and oil separators, and the number of pounds of ammonia circulated in the system, it is possible to calculate the volumetric efficiency of the plant. It is evident that this is not the volumetric efficiency of the compressor alone because it includes pipe and valve losses which should not be charged to the compressor.

There are numerous items which tend to reduce the volumetric of a compressor. The principal items are :

1. Loss due to clearance in the compressor
2. Loss due to superheating of the gas as it enters the compressor.
3. Leakage of gas past discharge valves, suction valves and piston rings.
4. Reduction in pressure in cylinders below the pressure in the suction pipe so that the gas flow will flow through the valves.
5. Minor losses.

These will be discussed successively -

I Loss in volumetric efficiency due to clearance volume.



When a piston moves back and forth in a cylinder as shown in the diagram, the stroke of the piston cannot be quite as great as the length of the cylinder, because if it were, the piston would strike the end of the cylinder on every stroke and damage it. Therefore, there is always left a small space between the end of the cylinder and the extreme position of the piston. This space is known as the clearance. It is usually expressed as a percentage of the total cylinder volume. If the cylinder is designed as shown in the diagram

$$\% \text{ clearance} = \frac{\text{clearance volume}}{\text{total cyl. volume}} = \frac{\frac{\pi d^2}{4} c}{\frac{\pi d^2}{4} l} = \frac{c}{l} \text{ ----- (2)}$$

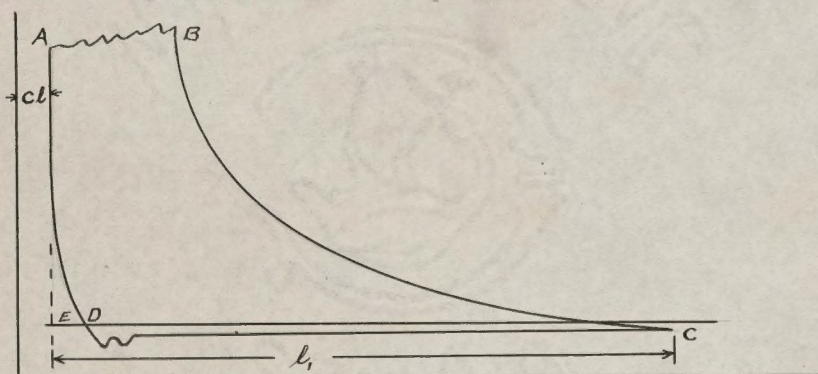
The ratio $\frac{c}{l}$ is known as the linear clearance.

Usually the cylinder is not designed as shown but has some valves and passages in the head and may be counterbored. In this case the clearance volume does not equal $\left(\frac{\pi d^2}{4} c\right)$ and the linear clearance is not the same as the volumetric clearance. In this case the clearance volume cannot be calculated except by a summation of minor volumes. It is then more commonly measured.

Measurement by displacement may be done very easily. Place the piston in the extreme head end position. Seal all the outlets and the piston circumference with grease so that water will not leak through or by them. Fill the clearance space with water, measuring the amount required to fill the space. The volume of water required to fill the clearance space is the clearance volume. The clearance volume divided by the total

volume of the cylinder will give the volumetric clearance.

Very often it is desired to determine the clearance of a machine which is in operation without stopping the machine and removing the cylinder head. This cannot be done absolutely accurately, but there are several approximate methods by which the clearance can be calculated with a fair degree of accuracy. These approximations make use of the indicator card taken from the compressor in question. The indicator card is a closed curve drawn with ordinates representing pressures and the abscissae representing volumes in the cylinder at all positions of the piston. The following is a typical indicator card from an ammonia compressor



Typical Ammonia Compressor Card

Fig. 2

At the point C the piston is at the beginning of its stroke and the cylinder is filled with ammonia gas. The gas is compressed along the line CB.

There are three possible states in which the gas may exist at the point C.

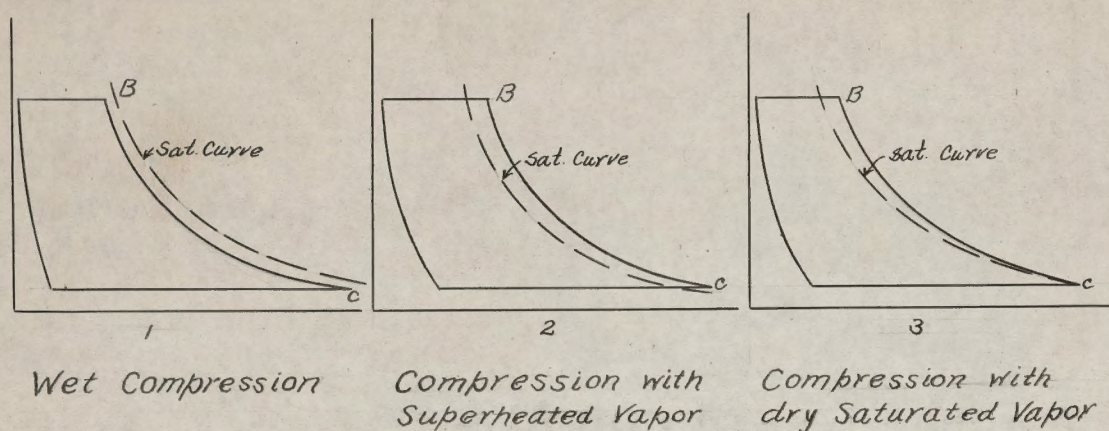


Fig. 3

(1) The gas at the point C might be wet, that is, contain particles of liquid. If the gas is wet enough it will still be wet at the point B, when it is discharged. In this case the compression curve BC follows the law

$$pv^{1.17} = p,v^{1.17} = c$$

(2) The gas may enter the compressor in a slightly superheated state. In this case the gas will have a higher degree of superheat when it reaches the point B. The compression curve will follow the law

$$p.v^{1.3} = p,v^{1.3} = c$$

(3) The gas may enter the compressor in a dry saturated state. In this case the gas immediately becomes superheated and follows the law for superheated gas

$$p.v^{1.3} = p,v^{1.3} = c$$

The first case is known as wet compression, and the last two are known as dry compression. Dry compression is most commonly used in this country.

At the point B the pressure in the cylinder is slightly greater than the pressure in the discharge pipe and condenser. Therefore, the discharge valve of the compressor opens and the ammonia gas in the compressor is discharged into the discharge line along the line BA. At the point A the discharge valve closes and the piston reaches the end of its stroke and the reverses its direction. The space cl is the linear clearance space which is filled with ammonia gas at the discharge pressure but it is not discharged. As the piston travels back the pressure in the cylinder is reduced and the gas trapped in the clearance space expands along the line AD. This is practically an adiabatic expansion and follows the same law as the compression curve. At the point D the pressure in the cylinder is the same as that in the suction pipe. When the pressure in the cylinder drops slightly below this, the suction valve opens and the ammonia gas is drawn into the cylinder along the line DC. It is readily seen that if there had been no clearance space to be filled with gas, ammonia would have been drawn into the cylinder during its whole backward stroke EC instead of only the distance DC. Therefore, because of the clearance space, the compressor cylinder is not able to handle as much ammonia as it would if there were no such clearance space.

Since volumetric efficiency is defined as the ratio of the amount of gas handled to the amount that should be handled by the compressor, the volumetric efficiency due to clearance, or apparent volumetric efficiency is

$$E_c = \frac{CD}{CE} \text{-----}(3)$$

since the volume of the cylinder is proportional to its linear length. Therefore, the apparent volumetric efficiency of a compressor may be determined approximately by measuring the lengths CD and CE on the indicator card. Great accuracy must be maintained in these measurements if the result is to be at all accurate.

The following is the derivation of the thermodynamic equation for apparent volumetric efficiency.

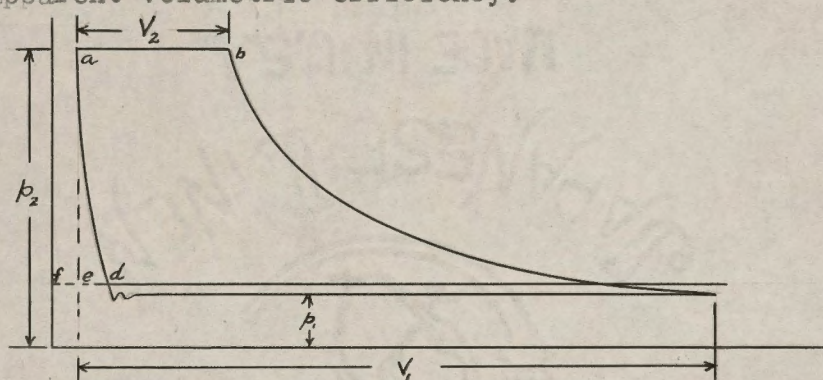


Fig. 4.

The compression and expansion curves follow the law

$$PV^n = P_1 V_1^n = P_2 V_2^n = C$$

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2} \right)^n$$

$$\frac{V_1}{V_2} = \left(\frac{P_2}{P_1} \right)^{1/n}$$

$$\frac{df}{ag} = \left(\frac{P_2}{P_1} \right)^{1/n}$$

$$df = de + ag$$

$$\frac{de + ag}{ag} = \left(\frac{P_2}{P_1} \right)^{1/n}$$

Solving for de

$$de = ag \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - ag = ag \left[\left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - 1 \right]$$

ag = clearance volume = Vc

$$de = Vc \left[\left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - 1 \right]$$

By definition and from equation (3)

$$E_c = \frac{dc}{ce} = \frac{ce - de}{ce}$$

ce = piston displacement = V,

$$E_c = \frac{V_1 - V_c \left[\left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - 1 \right]}{V_1}$$

$$E_c = 1 - \frac{V_c}{V_1} \left[\left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - 1 \right]$$

$$\frac{V_c}{V_1} = \% \text{ clearance} = c$$

$$E_c = 1 - c \left[\left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - 1 \right] \text{-----(4)}$$

The value of the exponent n in the above equation varies within certain limits but most authorities agree that the average value for n=1.3

$$E_c = 1 - c \left[\left(\frac{P_2}{P_1} \right)^{0.77} - 1 \right] \text{-----(5)}$$

In this equation which is theoretically correct it can be seen that the clearance of the machine must be known before the apparent volumetric efficiency can be calculated. Usually the machine is in operation and the clearance cannot be measured. The indicator card does not show the clearance but it does supply enough data to calculate the clearance. Two simple graphical methods will be shown.

In the Transactions of the A.S.M.E., vol.XXVI, the following method is described.

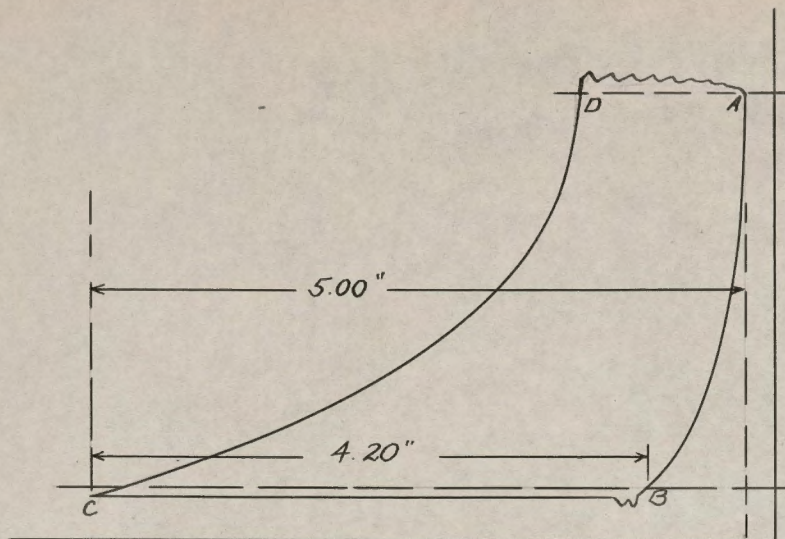


Fig. 5

The drawing shows an actual indicator diagram. With the use of a scale the following measurements were determined.

P = abs. pressure at point A = $98 \frac{\text{#}}{\text{sq.in.}}$

V = volume at point A = $X\%$ of total volume

P_1 = absolute pressure at point B = $20 \frac{\text{#}}{\text{sq.in.}}$

V_1 = volume at point B = $(X + \text{re-expansion})$

$$\text{Re-expansion} = \frac{5 - 4.20}{5} = .16 = 16\%$$

$$PV^{1.3} = P_1 V_1^{1.3}$$

Substituting numerical values

$$(98) \times (X)^{1.3} = (20) \times (X + 16)^{1.3}$$

dividing by $20 X^{1.3}$

$$\frac{98}{20} = \left(\frac{X + 16}{X} \right)^{1.3}$$

Raising to $\frac{1.33}{1.3}$ power

$$\left(\frac{98}{20} \right)^{.77} = \frac{X + 16}{X} = 3.40$$

$$X = 6.67\% \text{ clearance}$$

Mr. Garden T. Voorhees in his book entitled Mechanical Refrigeration gives the following method of determining clearance from an indicator card

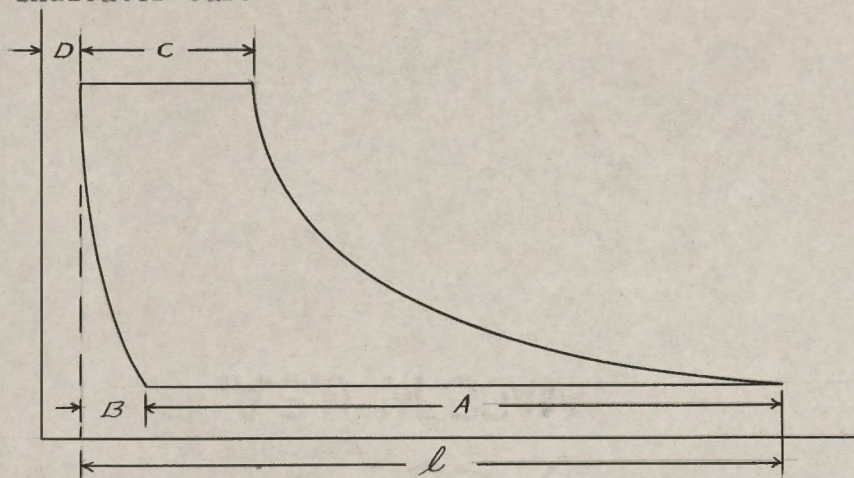


Fig. 6

The compression curve and the expansion curve^{are} assumed to follow the same law of variation, which is practically true.

With the above assumption the following proportion is true-

$$(l + D) : (C + D) = (B + D) : D$$

Multiplying the means and extremes

$$(l + D) D = (C + D) \times (B + D)$$

$$lD + D^2 = CB + CD + BD + D^2$$

Subtracting D^2 from each side of the equation

$$lD = CB + CD + BD$$

Transposing

$$lD - CD - BD = CB$$

$$D(1 - C - B) = CB$$

$$D = \frac{CB}{(1 - C - B)} = \frac{CB}{1 - (C + B)}$$

$$D\% = \frac{D}{l} = \frac{CB}{l[1 - (C + B)]} \quad \text{-----} (6)$$

From the above discussion it has been shown that the clearance of a compressor affects its volumetric efficiency. Since this has been found out the question of importance is, "Does it make any difference"?

First, consider the effect of clearance on the capacity of the compressor. It has been shown that if the compressor had no clearance at all, gas would be drawn into the cylinder as soon as the piston starts its suction stroke. It has also been shown that with clearance, the gas trapped must expand into the cylinder until its pressure is below that in the suction pipe before gas will be drawn into the cylinder. Sometime this gas trapped in the clearance space will occupy 10% to 15% of the total cylinder volume when expanded and, thereby, reduce the capacity of the compressor by that much. Reference to the accompanying curve will give some idea of the relation of clearance to the capacity of the compressor. The apparent volumetric efficiency may be as low as 60% to 80% for very small machines, 80% to 90% for medium sized machines, and 90% to 98% for large compressors. The efficiency is higher in large machines because the clearance space becomes a smaller part of the total cylinder volume.

The next consideration is the effect of clearance upon the power required to operate the compressor. Since the gas compressed in the clearance space expands each time it is compressed it should exert as much work on the piston each stroke as the piston does upon the gas. This would be true in actual practice if the gas did not lose any energy in the form of heat. There is a

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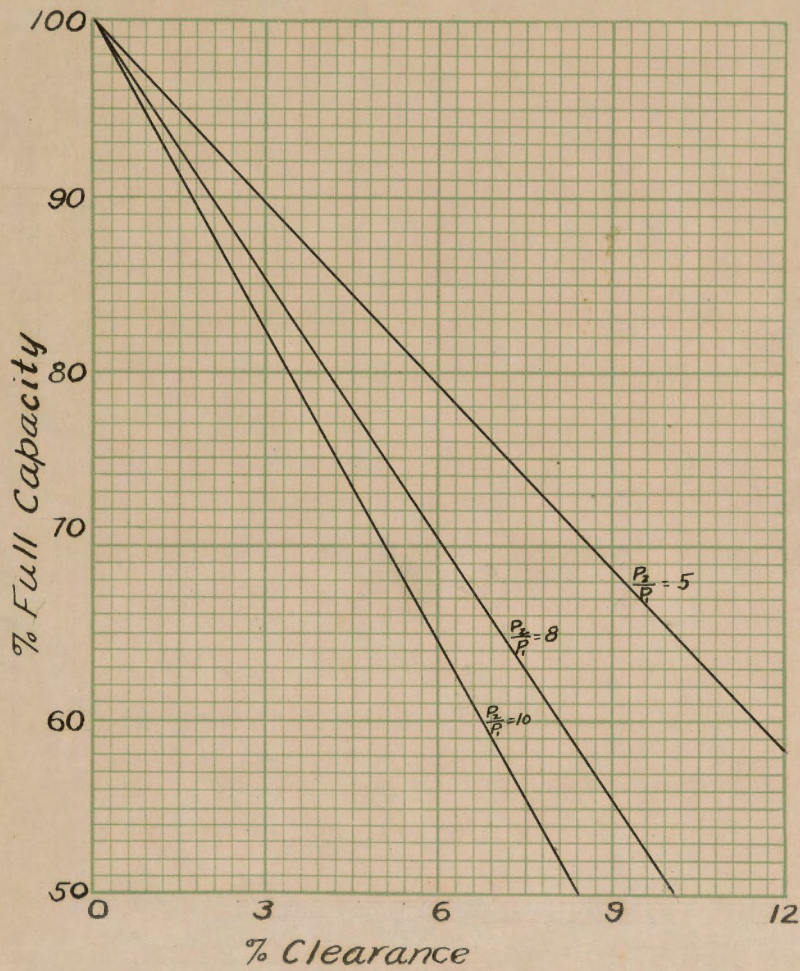


FIG. NO. 7

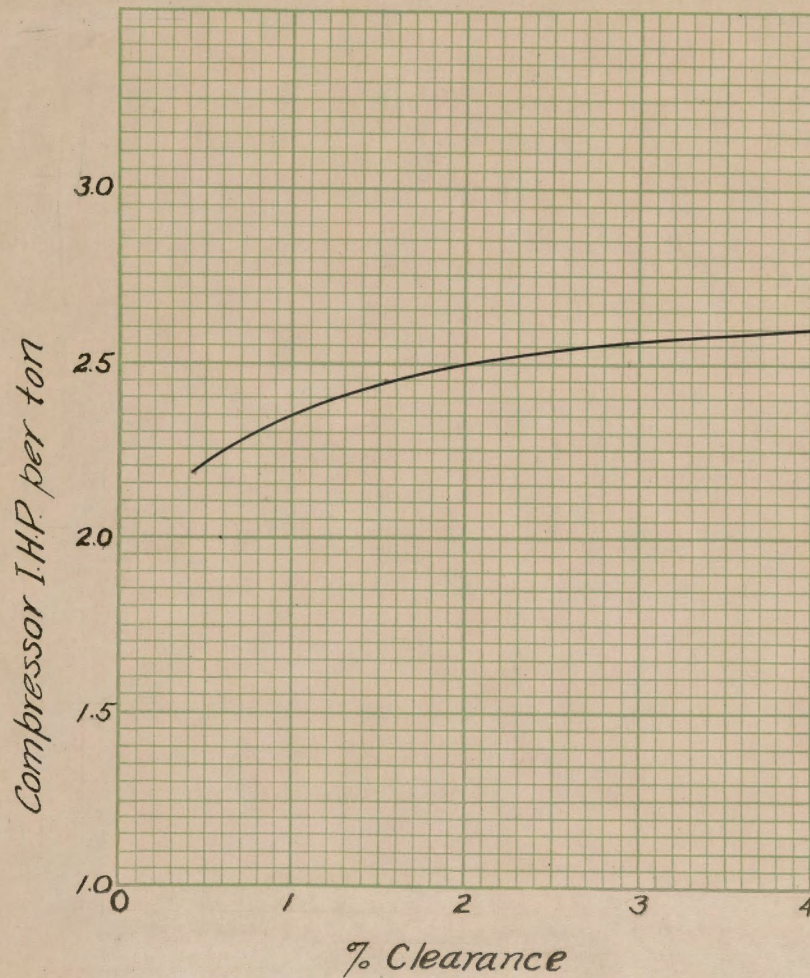
TITLE:-

Effect of Clearance on Capacity

INTERPRETATION OF CURVES

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FIG. NO. 8

TITLE:-

Effect of Clearance on Power

INTERPRETATION OF CURVES

little heat loss which causes a slight loss in power, but the loss is almost so small as to be negligible. In fact, many compressors are designed to have variable clearances so that the capacity of the machine may be varied. Curve number 2 shows the results of a test by Mr. Thomas Shipley on a $12\frac{1}{2}$ " x 18" double acting compressor with 185[#] gage condensing pressure and 5[#] gage suction pressure.

II Loss in Volumetric efficiency due to superheating.

Superheating of the ammonia gas as it enters the compressors is usually the greatest loss in the compressor operation. When the gas is compressed in the cylinder it becomes very hot because of the work done upon it. The hot gas heats the cylinder walls. As the new gas is drawn into the cylinder it comes in contact with the hot cylinder. The gas becomes heated and increases in volume. Since the gas increases in volume, a given mass of gas occupies a larger space and the cylinder will not hold as great a mass of gas. The wiredrawing effect produced as the gas passes through the suction valve also tends to raise its temperature. Also, because of the work necessary to operate the valves, there is a slight drop in pressure in the gas as it passes through the valves. This work done on the valves is accompanied by a rise in the temperature of the gas passing through the valves. If the gas is already in a saturated condition, it will be superheated slightly by this action. Likewise, the same effect will be produced as the gas passes through the discharge valve.

It is practically impossible to calculate mathematically by any theoretical equation the loss in efficiency due to superheating. Efficiencies due to superheating can only be determined accurately by an actual test on the particular compressor in question. It is what may be termed an operating characteristic of the compressor cylinder. Mr. Gardner T. Voorhees has derived an equation for volumetric efficiency due to superheating from valves obtained in actual tests. He gives the formula

$$E_s = 1 - \frac{t - t_o}{1324} \text{ ----- (7)}$$

E_s = volumetric efficiency due to superheating.

t_o = temperature of suction in degrees Fahrenheit.

t_1 = theoretical temperature after compression.

$$t_1 = (T - 460) T_o \left(\frac{P}{P_o} \right)^{0.24}$$

P_o and P_1 = pressures before and after compression.

In the above formula the constant 1324 was determined from the results of actual tests.

In the A.S.R.E. Journal, volume VII, the following method is described. It is more easily explained by using a specific case.

Assume a 10" x 16" compressor cylinder.

Suction pressure = 20[#] gage

Temperature corresponding to back pressure of 20[#] gage = 5.89 °F.

Specific volume for above pressure and
temperature = 7.991 cu.ft.

Superheat = 30 ° F

R.P.M. = 120

Displacement of compressor

$$\frac{3.1416 \times 10^2 \times 16 \times 120 \times 2}{4} = 301 \text{ cu.ft.}$$

Ammonia used per minute for 100 efficiency

$$\frac{301}{7.991} = 37.70 \text{ }^\#$$

For each degree increase in temperature

Back pres	30	27.5	25.1	20.11	19.97	15.6	1.3	8.2
Increase in vol.	.015	.0159	.0168	.0192	.0195	.022	.024	.0292

For each degree increase in temperature with back
pressure of 20[#] gage volume increase = .0195 cu.ft.
per pound.

With a superheat of 30 ° F.

$$.0195 \times 30 = .585 \text{ cu.ft. increase in volume.}$$

$$\text{New volume per lb.} = 7.991 + .585 = 8.576 \text{ cu.ft.}$$

Actual ammonia used per minute

$$\frac{301}{8.576} = 35.5 \text{ }^\#$$

Loss of ammonia per minute due to superheating

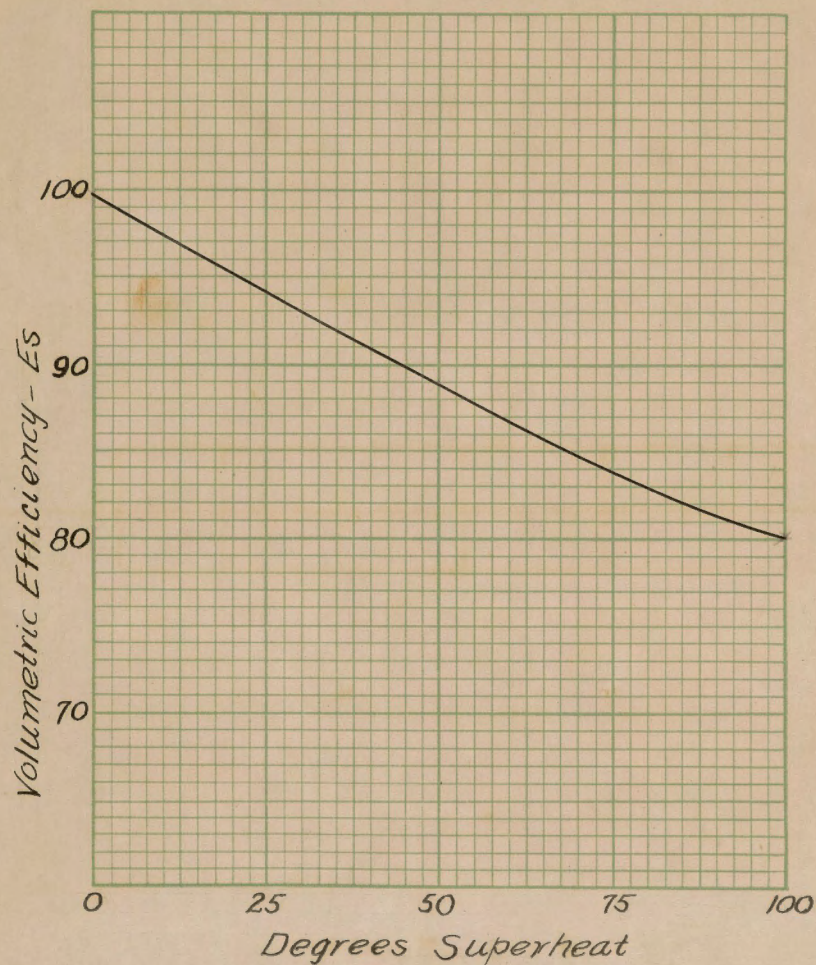
$$37.70 - 35.50 = 2.20 \text{ }^\#$$

Volumetric efficiency due to 30 F superheating

$$E_s = \frac{35.50}{37.70} = .93 = 93\%$$

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FIG. NO. 9

TITLE:-

Relation of Superheat to Volumetric Eff.

INTERPRETATION OF CURVES

*10" x 16" Compound Compressor
Suction pressure = 20[#] gage
R.P.M. = 120*

The accompanying figure 9 shows the effect of the different superheats on volumetric efficiency for a specific condition.

The loss in efficiency due to superheating is a total loss as just as much power is required to operate the compressor, although its capacity may be reduced as much as twenty percent. This loss due to superheating is the major loss in the compressor, so it is evident that its loss should be reduced as much as possible.

III Leakage of gas past discharge valves, suction valves, and piston rings.

The two major losses in a compressor have already been discussed. All of the other losses are so small as to be negligible in a properly operating machine. Leakage of the discharge valves, suction valves, and piston rings is a very common occurrence in machines which are not kept in first class condition. These troubles can be usually detected by the use of an indicator. On an indicator card taken from the compressor in question plot the adiabatic and isothermal compression curves, letting them coincide with the actual compression curve at the toe of the card.

Adiabatic compression is the compression of the gas in the cylinder in such a manner that no heat is lost or gained by the gas to its surroundings. The adiabatic compression curve for dry ammonia follows the law

$$p_1 v_1^{1.3} = p_2 v_2^{1.3}$$

$$p_2 = p_1 \left(\frac{v_1}{v_2} \right)^{1.3}, \quad \text{in which}$$

p_1 and v_1 are the pressure and volume of the gas at the beginning of the compression stroke. These two quantities are known. By use of the above equation the pressure corresponding to any other assumed volume can be calculated and plotted on the card. For anyone who is not familiar with solving equations similar to the above, a set of curves has been drawn which may be used without any calculation. First determine the clearance C of the compressor from the indicator card by one of the methods which has already been described.

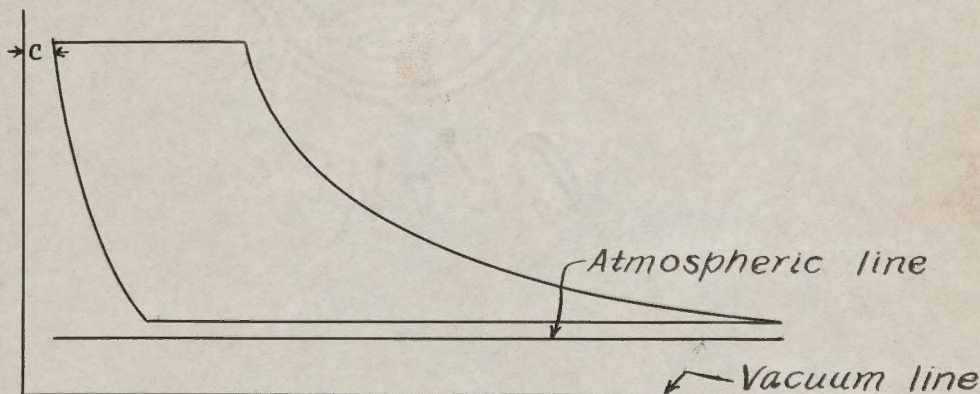
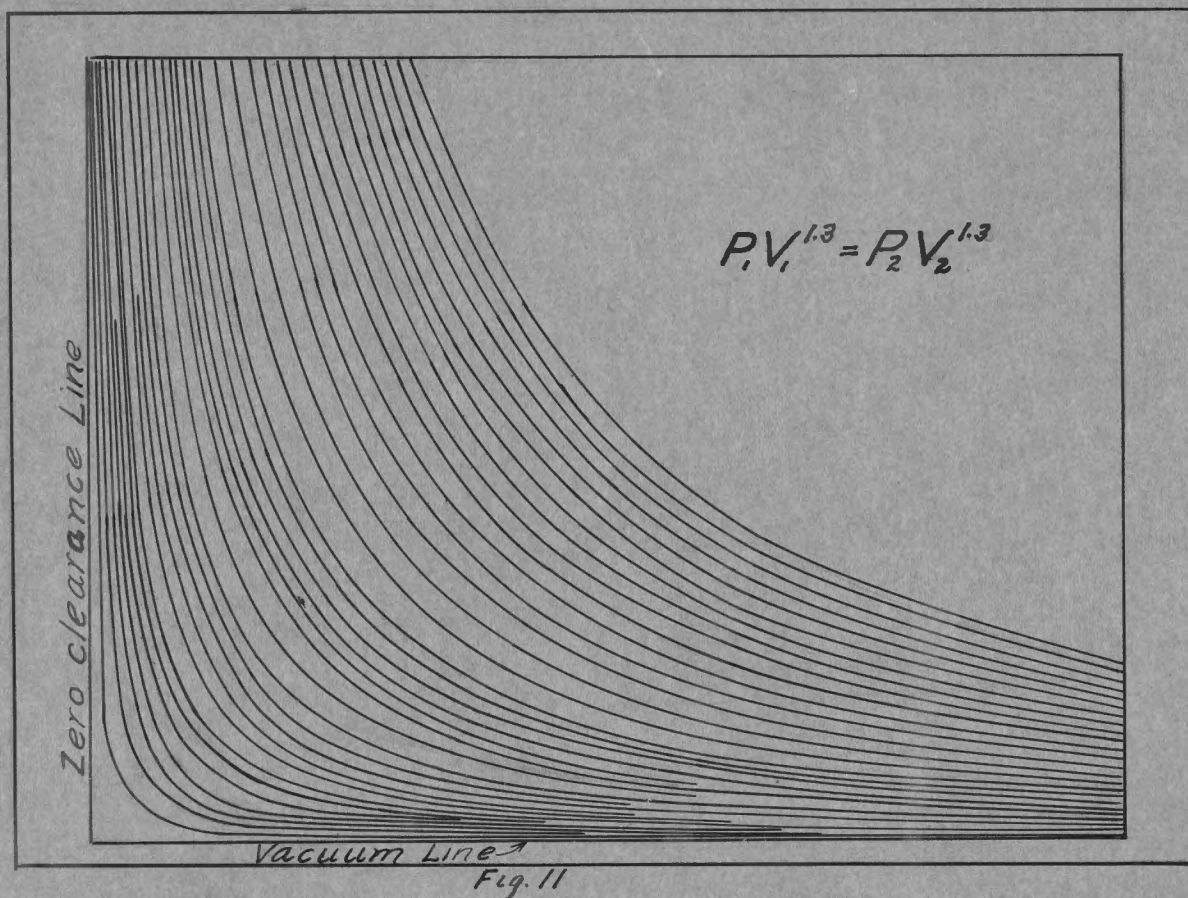


Fig. 10

Draw the vertical line on the card at a distance of C from the edge of the expansion curve as shown in the diagram. Next draw the vacuum line YZ on the card as has been described on



ADIABATIC CURVES

Draw zero clearance and zero pressure or vacuum line on the indicator card. Place card under curves so that zero clearance and vacuum lines coincide. Curve crossing the point on toe of card is the Adiabatic curve for the card.

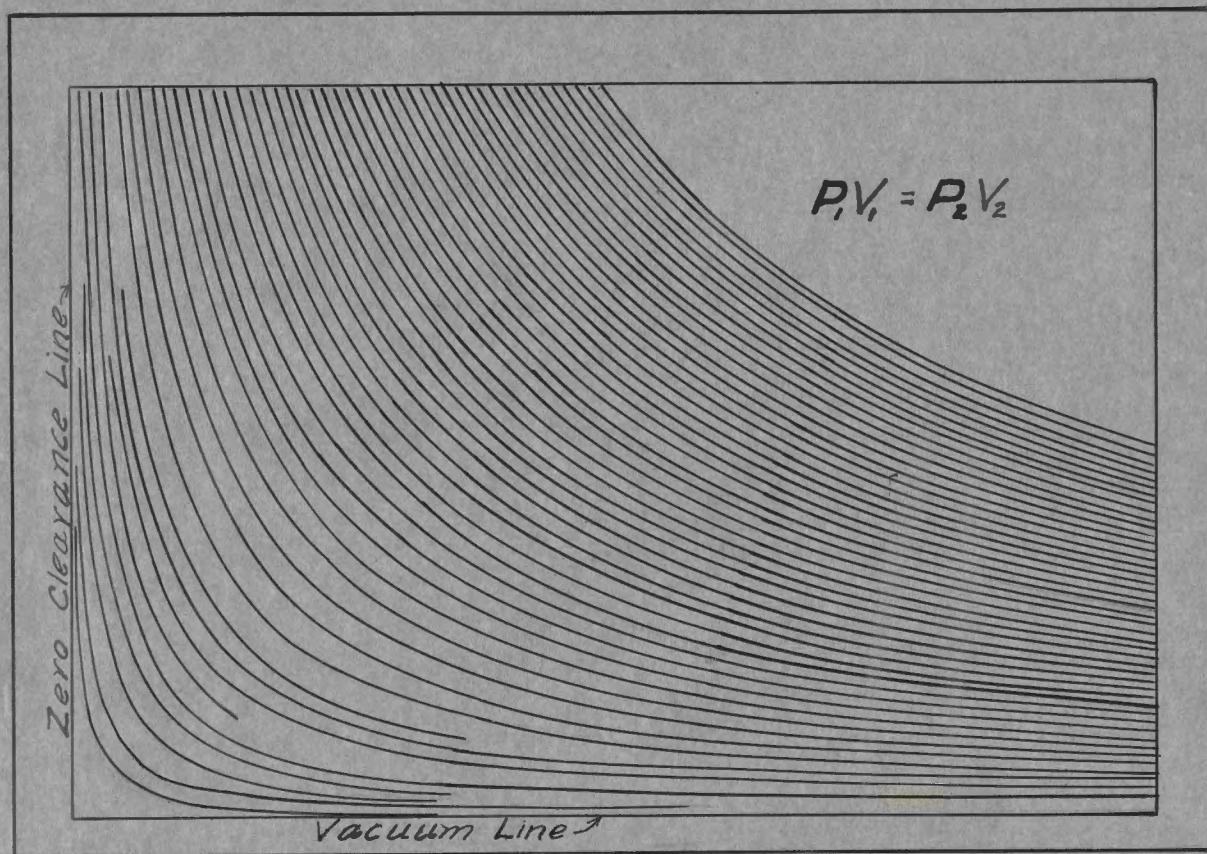


Fig. 12

ISOTHERMAL CURVES

Draw ZERO clearance and zero pressure or vacuum line on the indicator card. Place card under curves so that zero clearance and vacuum lines coincide. Curve crossing the point on toe of card is the Isothermal curve for the card.

a previous page. Place the curves, which are drawn on transparent paper over the card so that the border lines coincide with the lines XX and YZ. The curve which crosses the point A of the indicator card is the adiabatic compression curve for that indicator card.

Isothermal compression is compression in which the temperature of the gas compressed, remains constant throughout the operation. This can only happen with wet compression, that is, when the gas admitted to the cylinder contains particles of liquid ammonia. Isothermal compression follows the law

$$P_1 V_1 = P_2 V_2$$

$$P_2 = P_1 \left(\frac{V_1}{V_2} \right)$$

P_1 and V_1 are the absolute pressure and volume of the gas as it enters the cylinder. The pressure corresponding to any other assumed volume may be calculated from the equation and plotted on the indicator card. A set of isothermal curves has been included in the discussion which may be used instead of the calculation. Construct on the indicator card the zero clearance and zero pressure lines as when using the adiabatic curves as described above and use the curves in a like manner. These curves may be transferred to the card by pricking holes with a pin and joining the points with a curve.

The closer the actual compression curve lies to the adiabatic curve the better. If the actual curve lies above the adiabatic

curve it shows that the discharge valves are leaking and gas is coming in from the condenser. If the actual curve lies too far below the adiabatic curve, near the isothermal curve, it shows that the suction valves or the piston rings are leaking. These conditions can be probably more accurately determined by the use of a thermometer on the discharge line near the compressor. If the thermometer shows a higher temperature than it normally should, it is safe to say that the discharge valves are leaking, causing the discharge to superheat excessively. Likewise, if the thermometer records a lower temperature than it should, it shows that the suction valves or the piston rings are leaking.

Loss due to leakage is, of course, a total loss and should be remedied, by repairing the leaking parts.

IV Reduction in pressure in the cylinder necessary to operate the suction and discharge valves.

The loss of pressure of the gas in passing through the compressor valves is another minor loss. It is easily seen that the pressure in the cylinder must be slightly below that in the suction pipe before the gas will flow through the suction valve into the cylinder. Likewise, the pressure in the cylinder must be slightly greater than the pressure in the discharge pipe before the gas will flow from the compressor into the discharge line. This pressure drop is, of course, a loss to the compressor.

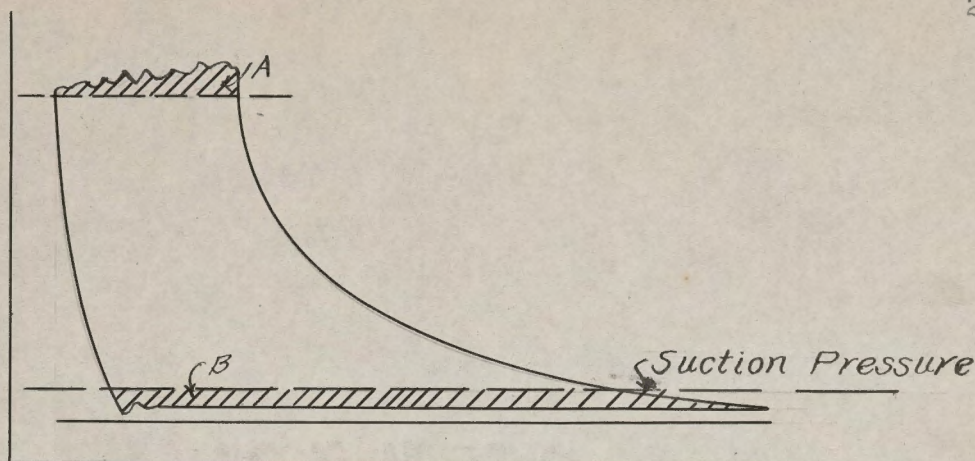


Fig. 13

On the above indicator card the shaded area A shows the work required to operate the suction valve. There has been a great deal of work done to eliminate the valve losses. This will be discussed later.

V Effect of speed on volumetric efficiency.

Compressors are divided into two classes with reference to speed. The slow speed compressor usually has a piston stroke of about 1.5 to 2.0 times the diameter of the cylinder. Gardiner's rule gives the maximum speed of slow speed compressors as :

$$\text{r.p.m.}_{ss} = \frac{376}{\sqrt{S}}$$

S = stroke of piston in inches.

High speed compressors have a piston stroke of from 1.0 to 1.5 times the diameter of the cylinder. Gardiner's rule for the maximum speed of a high speed compressor is.

$$\text{r.p.m.}_{hs} = \frac{850}{\sqrt{S}}$$

The average stroke for slow speed compressors is from 24 inches to 36 inches and the average stroke for high speed compressors is from 12 inches to 18 inches. As shown before, the clearance of a compressor is :-

$$\% \text{ clearance} = \frac{\text{clearance volume}}{\text{total cylinder volume}}$$

The clearance volume is kept as small as possible in all cases so it is practically the same in both high-speed and slow-speed compressors. At the same time the total cylinder volume of a high-speed cylinder is only about one-half as great as that of the slow-speed compressor. From the above equation it is evident that the per cent clearance for the high-speed compressor will be about twice that of the slow-speed compressor. It has already been shown that clearance is one of the factors affecting the volumetric efficiency of compressors.

The speed of a high-speed compressor is about three times that of the slow-speed type. This means that the valves on the high-speed compressor must operate about three times as often. The valves of the high-speed compressor would necessarily, then, become much more worn and have a greater leakage. The effect of leaky valves has already been discussed.

Since the valves of the high-speed compressors must operate rapidly as compared with the slow-speed compressors, they will have more inertia to overcome and stronger springs will be necessary to insure rapid closing. Because of this, more work will be required to operate the valves, which results in

a greater drop in pressure of the gas as it passes through the valves. It has already been shown that the drop in pressure results in a loss in volumetric efficiency.

There is one item in the high-speed compressor which in some degree counteracts the ill effects. The cylinder surface is much smaller than that of the slow-speed compressor. It has already been shown that the hot cylinder walls causes a superheating of the gas as it enters the cylinder. If the area of the exposed cylinder walls is decreased, the superheating effect will be reduced. Since superheating causes the largest loss in the compressor, this may be a large item of saving.

VI Other minor losses.

There are always a number of other minor losses which cannot be measured and make any calculation of volumetric efficiency slightly inaccurate. One minor item which may have a very slight effect on the operation of a compressor is variation in atmospheric pressure. Tests have been made which show that compressors operating in high altitudes where the atmospheric pressure is low, the volumetric efficiency of the compressor is slightly reduced. It has been found, also, that the speed at which a compressor operates affects its efficiency.

VIII Effect of external system on volumetric efficiency.

It was stated in the first part of this discussion that care

must be taken not to confuse volumetric efficiency of the compressor with the volumetric efficiency of the entire refrigerating system. Since the compressor losses have been discussed it might be well to show what some of the losses in the external system are.

Whenever a liquid or a gas flows in a pipe there is always a loss in pressure in the flowing medium. The more restricted the passage is, the greater will be the pressure drop. The restriction may be due to the small size of the pipe, the roughness of the pipe, or the fittings, such as; elbows, tees, valves, etc.

If the system from the compressor to the expansion valve, which includes the piping and the condenser, offers such resistance to the flow of the gas as to cause a large drop in pressure, it will have the same effect as if the valve on the discharge line of the compressor was partly closed. This causes the discharge pressure of the compressor to be increased greatly because more pressure is required to discharge the gas handled by the compressor. Likewise, if the piping system between the expansion valve and the compressor offer a large resistance to the flow of the gas, there will be a drop in pressure and the suction pressure of the compressor will be lower than it should be. Therefore, if the refrigerating system is restricted the discharge pressure of the compressor is increased and the suction pressure is lowered. Tests have shown that volumetric efficiency increases as

the difference between the discharge and suction pressure decreases. That is, an increase in the discharge pressure or a decrease in the suction pressure of a compressor lowers the volumetric efficiency of the compressor. This shows that if the external system causes a decrease in the volumetric efficiency of the compressor.

The various causes for loss in the efficiency of compressors have been discussed and methods have been shown how to calculate the amount of the losses. The question now is to find out how to determine the loss in the actual volumetric efficiency of the compressor as a whole.

The apparent volumetric efficiency of a compressor becomes the real volumetric efficiency when the temperature of the gas after it has entered the cylinder is equal to the temperature of the outside gas. If during admission to the cylinder the temperature of the gas is raised, the weight of gas admitted will be decreased in the ratio of the absolute temperature before and after admission. Therefore equation (4) for apparent volumetric efficiency

$$E_c = 1 - C \left[\left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - 1 \right], \text{ becomes}$$

$$E_R = \frac{T_1}{T} \frac{P}{P_1} \left(1 - C \left[\left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - 1 \right] \right) \text{ ----- (8)}$$

E_R = real volumetric efficiency.

T_1 = absolute temperature in suction pipe

T = absolute temperature when compression begins.

P_1 = absolute pressure in suction pipe.

P = absolute pressure at intake .

P_2 = discharge pressure.

This equation takes account of clearance loss and superheating loss but does not account for loss through leaky valves or piston rings, so it does not give an absolutely accurate result .

Referring to equation (1) for real volumetric efficiency

$$E_R = E_c \times E_s$$

From equation (4)

$$E_c = 1 - c \left[\left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - 1 \right]$$

$$E_c = 1 - c \frac{P - P_1}{P_1}$$

and from equation (7)

$$E_s = 1 - \left(\frac{t_1 - t_o}{1324} \right)$$

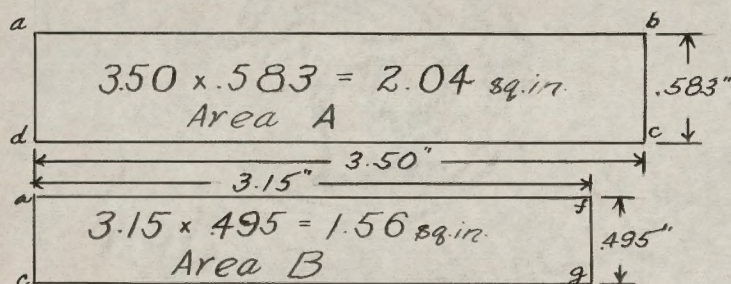
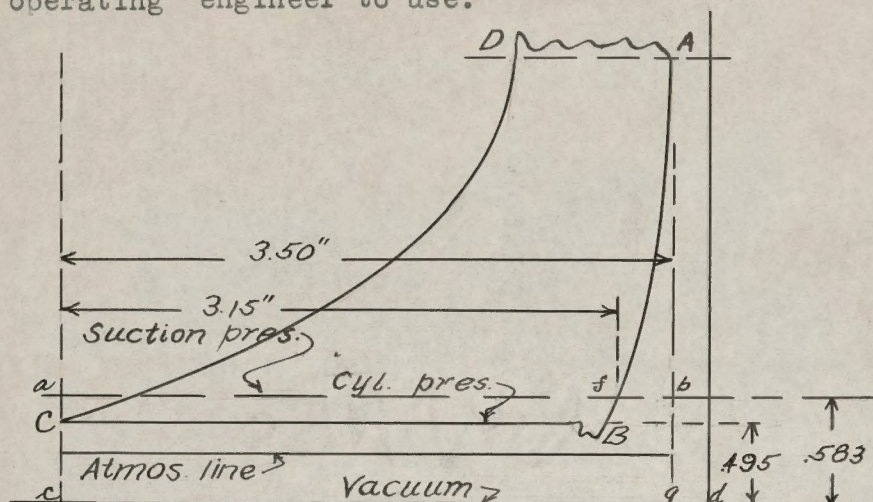
Substituting these in equation (1)

$$E_R = \left[1 - c \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - 1 \right\} \right] \left[1 - \frac{t_1 - t_o}{1324} \right] \text{-----} (9)$$

This expression for real volumetric efficiency also accounts for the major losses, clearance and superheating, but does not account for the minor losses.

The following is a simple approximate method for determining the volumetric efficiency of a compressor as described in the Transactions of A.S.M.E., volume XXVI, by F.E. Matthews. It

takes into account all of the major losses and is simple enough for any operating engineer to use.



$$\text{Vol. Eff.} = \frac{1.56}{2.04} = 76.5\%$$

Fig. 14

The theory is as follows :-

It is almost correct to assume that the specific volume of ammonia gas is proportional to its absolute pressure. The ordinates on the card represent pressures.

Ratio of cylinder back pressure to suction pipe back pressure equals

$$\frac{.495}{.583} = .85 = 85\%$$

Therefore, according to assumption, 85% weight of gas in the suction pipe enters the compressor cylinder. On the suction stroke, the gas does not enter the cylinder until the point B is reached. Therefore, from the measurements on the card the percentage of the stroke during which the gas enters the cylinder is

$$\frac{3.15}{3.50} = .90 = 90\%$$

The volumetric efficiency due to clearance is 85%

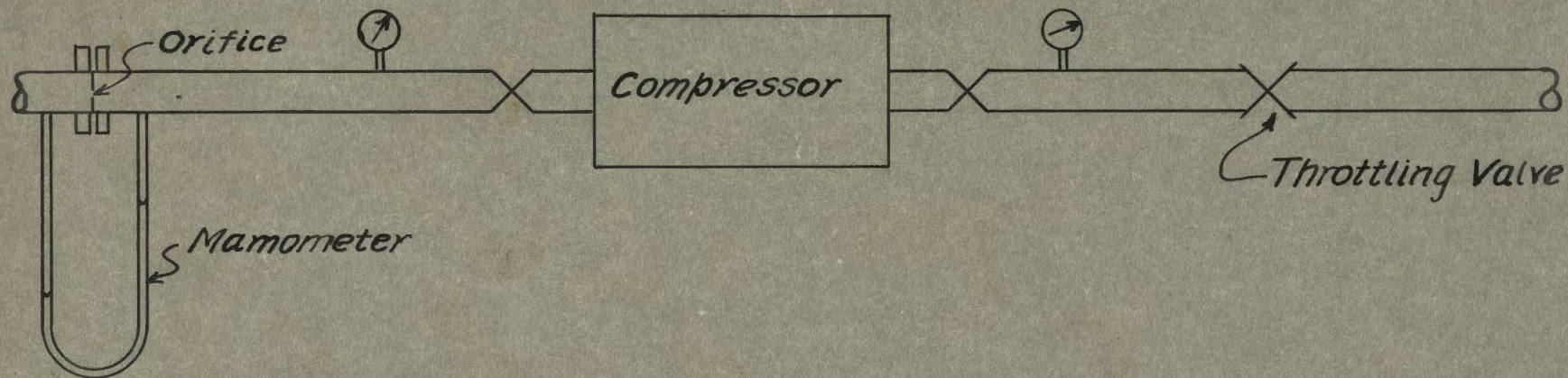
The volumetric efficiency due to superheating is 90%

Combining these two

$$E_e = .85 \times .90 = 76.5\%$$

In the graphical solution area A is proportional to the apparent or theoretical displacement of the compressor. The area B is proportional to the effective compressor displacement. The ratio $\frac{\text{area B}}{\text{area A}}$ gives the volumetric efficiency of the compressor.

All of the methods for determining the volumetric efficiency of a compressor that have been described so far have been approximate. There is only one way to determine volumetric efficiency absolutely accurately. That is to measure the actual amount of ammonia that has passed through a compressor and divide by the cylinder displacement for that given time. This method is used very often and is, of course, the best when when it is possible to use it. There have been designed



DIAGRAMATIC LAYOUT FOR
DETERMINING VOLUMETRIC EFFICIENCY

Fig. 15

some very elaborate weighing devices which can be used. There is included in this discussion a diagram which shows a very simple method for determining the amount of ammonia which is circulated in the system. The flow meter is a venture tube or an orifice with a manometer attached. Either of these devices can be calibrated to measure the flow fairly accurately. If absolute accurate results are desired an arrangement can be made to discharge the ammonia after it has passed through the compressor into a tank of water or other liquid which will dissolve ammonia rapidly. By weighing the tank before and after the test the weight of ammonia which has been used can be determined.

Effect of Volumetric Efficiency on the Operation of a Compressor

The study of volumetric efficiency in all its phases may be very interesting theoretically, but unless it has some direct bearing upon the actual performance of the refrigerating machine is not of any practical value and is not worth the time spent studying it. With the refrigerating machine the main interest is the capacity of the machine to produce the maximum refrigerating effect at the minimum cost. In the following discussion an attempt will be made to show just what effect volumetric efficiency has on the operation of the refrigerating plant.

It is largely due to what is the cause of the loss in volumetric efficiency as to what effect it has on the opera-

tion of a compressor. The first step to be considered is the capacity of the plant. Of course, the refrigerating capacity of a plant is directly proportional to the weight of the refrigerant circulated in the system. The maximum theoretical volume of refrigerant that can be pumped into a system is obviously equal to the compressor cylinder displacement. Anything that reduces the theoretical displacement of the compressor decreases the capacity of the plant. The definition of real volumetric efficiency of a compressor is the ratio of the actual weight of refrigerant handled to the theoretical amount that should be handled. Putting the definition in the form of an equation

$$\frac{\text{Effective displacement}}{\text{Theoretical displacement}} = E_r$$

$$\text{Effective displacement} = E_r \times \text{Theoretical displacement}$$

This equation shows that the capacity of the compressor is proportional to its volumetric efficiency. In other words, a reduction of 10% in the volumetric efficiency of the compressor will cause a reduction of 10% in the capacity of the machine. Since this is so, the lower the volumetric efficiency of the compressor, the larger the machine must be to produce a given amount of refrigeration. The larger a machine is the more it will cost, so it is of economic importance to keep the volumetric efficiency of the machine as high as possible. The capacity of a machine is usually expressed in tons of refrigeration. This term must not be confused with

tons of ice making capacity. One ton of refrigeration is defined as the removal of enough heat from the medium which is being cooled to melt one ton of ice at 32° F to water at 32° F. The latent heat of fusion of water, that is, the amount of heat required to melt one pound of ice at 32° F to water at the same temperature, is about 144 B.T.U. For one ton of ice the heat required to melt it would be

$$2000 \times 144 = 288,000 \text{ B.T.U.} = 1 \text{ ton of refrigeration}$$

$$\text{refrigeration per day} = \frac{288000}{24 \times 60} = 200 \text{ B.T.U.}$$

per minute.

One ton of refrigeration would be equivalent to one ton of ice-making if the water to be frozen was taken at 32° F and frozen into ice at 32° F. Usually the temperature of the water ranges from 80° F to 90° F and the temperature of the ice after freezing is about 5° F to 10° F. In actual practice, taking into account radiation and other losses, one ton of ice-making is equivalent to about 1.5 tons of refrigeration. The following is the way to determine the capacity of the machine in terms of tons of refrigeration. This done with the use of the indicator card.

First, determine the volumetric efficiency of the compressor from the indicator card by one of the methods described earlier in the discussion. Also, determine the suction pressure. From tables of properties of saturated ammonia find the specific volume of ammonia gas at the suction pressure. It is not necessary to enter a discussion of standard ton conditions

here, other than to state that .4079 pounds of ammonia is circulated per minute under standard ton conditions.

$$\begin{aligned} &.4079 \times \text{specific volumes at suction pressure} \\ &\quad \text{cylinder displacement per ton} \\ &= \text{cylinder displacement per ton per minute} \end{aligned}$$

$$\frac{\text{Cylinder displacement}}{\text{volumetric efficiency}} = \text{actual displacement per ton per minute.}$$

$$\begin{aligned} &\text{Actual volume} \times \text{weight of 1 cu.ft. at suction} \\ &\text{pressure} = \text{wt. flowing per stroke.} \end{aligned}$$

$$\begin{aligned} &\text{Strokes per min,} \times \text{wt. flowing per stroke} = \\ &= \text{wt. flowing per minute.} \end{aligned}$$

$$\begin{aligned} &\text{Wt. flowing per minute} \times \text{heating effect per} \\ &\text{lb. under compressor conditions} = \text{B.T.U. per minute.} \end{aligned}$$

$$\text{Tons refrigeration} = \frac{\text{B.T.U. per minute}}{200}$$

The heating effect is found from tables of properties of ammonia. It is the difference between the heat content at discharge conditions of pressure and temperature and the heat content at the suction conditions of temperature and pressure.

The next consideration is the effect of volumetric efficiency upon the power required to operate the machine. In the discussion on clearance, earlier in this paper, it was shown that theoretically clearance did not have any effect on the power

required to drive the compressor. Also, a curve is included which shows that on actual test there is a slight power loss as the clearance on a machine increases but that it is small. It has also been shown that loss of volumetric efficiency due to superheating was a complete loss. If low volumetric efficiency is due to excessive clearance there will not be very much power loss but if it is due to superheating or leakage the power loss will be considerable. The only way in which power loss due to low volumetric efficiency can be determined is by measurement on an actual test. The accompanying curves show the results of test on York machines which had a very small clearance so the volumetric efficiency was not affected very much by clearance.

The power is expressed in terms of indicated compressor horsepower per ton of refrigeration. The greater the power loss, the greater will be the I.H.P. per ton. Indicated horsepower of a compressor may be measured from the indicator card. The area of the card is proportional to the work done on the compressor. The following formula gives the horsepower of a compressor.

$$\text{H.P.} = \frac{P \times L \times A \times N}{33000}$$

P = mean effective pressure (M.E.P.)
in lbs. per sq. in.

L = Piston stroke in feet.

A = area of cylinder in sq. inches

N = number of strokes of cylinder
per minute.

The M.E.P. is gotten in the following manner : -

1. Measure the area of the indicator card in square inches by means of a planimeter.
2. Measure the length of the card in inches.
3. Divide the area by the length and multiply the result by the scale of the indicator spring used and the result will be the mean effective pressure.

If a planimeter is not available, divide the length of the card into ten equal divisions and draw a vertical line through each point found. Measure the length of each of these lines within the area inclosed by the diagram. Divide the sum of these ten lengths by ten and multiply by the scale of the indicator spring used in taking the card. The result is the mean effective pressure. If the compressor is double acting, take an indicator card for each ^{end} end of the compressor and find the power for each end similarly. The sum of the two will be the total compressor indicated horsepower. In determining the power for the crank end of the compressor, the area of the piston rod must be subtracted from the piston area for the value of A in the formula.

The indicated horsepower per ton is calculated by dividing the indicated horsepower of the compressor by the output of the compressor in tons of refrigeration. The method of de-

termining the capacity of a compressor has already been described.

The reduction in capacity of the compressor and the increase in the horsepower required per ton of refrigeration as discussed are the principal effects of low volumetric efficiency. An attempt was made to show these effects on a thermodynamic chart, commonly called a ³⁷Theta-Phi Diagram, but it was found that the diagram did not show anything of interest that could not be shown more easily on the indicator card. After analyzing the Theta-Phi Diagram the reason for this is readily seen. The $\theta\phi$ diagram is a combination of four diagrams showing the properties of ammonia with the following coordinates P,V; P,T; T,S; S,V.

P = pressure (lbs. per sq.inch)

V = volume (cubic feet)

T = temperature

S = entropy.

The indicator card is a diagram on the P,V plane. The indicator card is transferred to the P,V plane of the diagram. This diagram is in turn projected to the T,S plane of the $\theta\phi$ diagram. Any area in the P,V plane = $P \times V$ which is the work done on the compressor expressed in foot-pounds. Any area in the T,S plane = $T \times S$ which is the work done on the compressor expressed in different units.

From the definition

Let T be the temperature in $^{\circ}F$.

Let S be the entropy in Btu/R .

LABORATORY

DATE

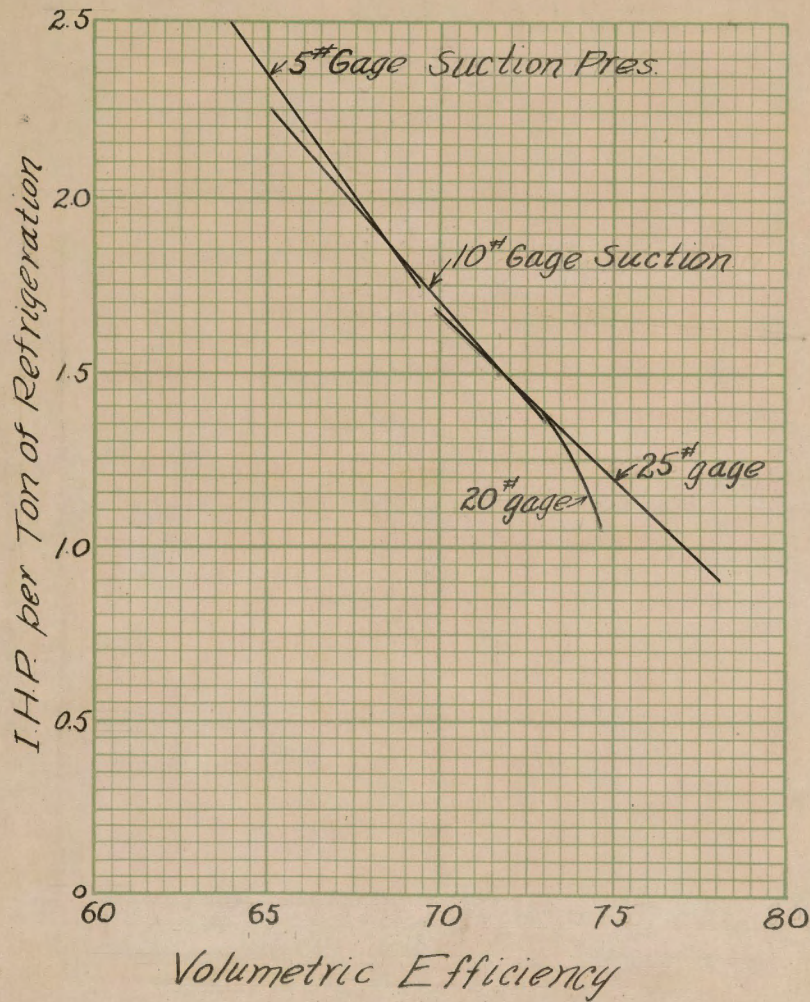


FIG. No. 16

TITLE:-

Relation of Volumetric Eff. to Power

INTERPRETATION OF CURVES

From definition

$$S = \frac{Q}{T}$$

Q = heat in B.T.U.

T = temperature in $^{\circ}F$

$$\text{Therefore, } T \times S = T \times \frac{Q}{T} = Q$$

The area in the T, S plane is, then, the work done on the compressor in heat units, B.T.U. Since the area in both planes represents the work done in the compressor, they both show the same thing, and neither, shows anything that the other cannot be made to show.

There are two general types of compressors, the horizontal double-acting machine and the vertical single-acting machine. The double-acting compressor makes two compression strokes per revolution, while the single-acting machine makes only one compression stroke per revolution. This being the case, it is evident that the single-acting machine must have a much larger cylinder volume than the double-acting compressor to have the same capacity. With larger cylinder volumes the cylinder walls exposed to the gas will be greater. Although the single-acting compressor has a larger cylinder surface exposed to the incoming gas, the superheating loss is less than in the double-acting machine, because, in the single-acting compressor the cylinder walls have a longer time to cool between strokes than the double-acting machine.

The percentage of clearance volume in the vertical single-acting compressor is very much smaller than in the horizontal double-acting compressor. The vertical single-acting machine

is usually made with a false or safety head. This false head, which is held in place by means of a spring, acts as the discharge valve and opens at the end of the compression stroke. With this arrangement the clearance volume can be so reduced as to be almost negligible in effect. The vertical single-acting compressor has a very high volumetric efficiency due to clearance.

In the horizontal double-acting compressor the false head cannot be used. Space must be left between the cylinder head and the piston when it is in the extreme position so that they will not strike. Sometimes this space represents several percent of the total cylinder volume. In the double-acting machine, clearance must be allowed at both ends of the cylinder. Because of the comparatively large clearance necessary both ends of the stroke, the horizontal double-acting compressor has a much lower volumetric efficiency due to clearance than the vertical single-acting machine.

In the vertical single-acting compressor the suction valves are in the piston. Since the valves open as the piston starts the down stroke, and close as the piston starts the upward stroke, the inertia of the valves help to operate them and only a very light spring is necessary. The pressure loss in gas in passing through these valves is very small. In the horizontal double-acting compressor the difference in pressure on each side of the valve is

all that operates it. Therefore, the pressure drop in the gas passing through this valve is larger than that in the vertical single-acting compressor.

The above statements show that the vertical single-acting compressor operates with a higher volumetric efficiency than the horizontal double-acting compressor. Actual tests on the different types of compressors show this to be true.

Efforts Which Have Been Made to Increase
Volumetric Efficiency

Since low volumetric efficiency has a harmful effect on the operation of a refrigerating machine, it is only natural to suppose that some effort would be made to make the volumetric efficiency as high as possible so as to reduce the loss as much as possible. The following statements will show that this is true.

As stated before the factors causing low volumetric efficiency are : -

1. Clearance
2. Superheating of suction gas.
3. Leakage past valves and piston.
4. Loss of pressure due to operation
of valves.
5. Operating pressures of the compressor.

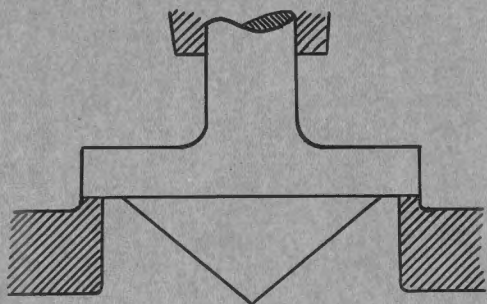
As the compression refrigerating machine has been developed, the compressors have been built with less and less clearance. Of course, it is not possible to reduce the clearance to zero because, otherwise, the piston would strike the end of the cylinder. Vertical single-acting compressors have less clearance space than the horizontal machines have. Some designs of the vertical type have a false head which moves back as the piston nears it. With such a scheme as this, the clearance may be very small, usually less than $\frac{1}{32}$ inch. The

newest designs of horizontal compressors have a clearance space of $\frac{1}{16}$ inch or less.

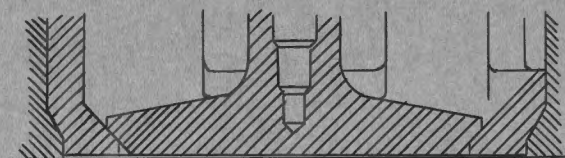
Superheating causes a great loss in compressor efficiency. The suction gas, as it enters the compressor, has a temperature of about 4 °F to 8 °F. The gas leaving the compressor has a temperature of around 175 °F to 200 °F. If the cold gas absorbs any of the heat radiated from the hot gas its volume increases and the volumetric efficiency is reduced. In the earlier designs of compressor the inlet and outlet pipes to the compressor were very close to each other, giving a chance for a large transfer of heat from the hot to the cold gas. The practice at the present time is to separate the two lines as much as possible. Another item which causes superheating is the hot cylinder walls. The compressed air which is very hot heats the cylinder walls. When the cold gas is drawn into the cylinder, the hot walls heat the gas and it expands. Most compressor cylinders are water-jacketed to keep them as cool as possible so as to cut down the loss. The greater the pressure to which the gas is compressed, the hotter will be the gas. It has been found that, because of this, the lower the discharge pressure of the gas, the higher will be the volumetric efficiency. This fact has given rise to the development of the two stage compressors. By the use of a high pressure cylinder and a low pressure cylinder, the pressure range of the cylinder is reduced and, likewise, the heating effect of the cylinder walls is greatly reduced.

Probably the one part of the compressor which has given the most trouble is the valves. If the valves are too heavy or the springs too strong an excessive amount of work is required to open the valves, which results in a loss. Likewise, if the springs are weak or the valves are not properly seated they will leak, causing loss. If the parts or passages are small there is considerable loss due to wiredrawing. The accompanying diagrams show some of the steps in the development of the valve. The earliest valves were the multiported rotary disk valve and the multiported rotary cylinder valve. Later the balanced piston valve and the semi-rotary or Corliss valves were used. All of these valves were mechanically operated. The poppet valve, which is a later development, is very widely used at the present time. A combination of the semi-rotary and the poppet valves was attempted and proved successful with the air compressor but it could not be used on the ammonia compressor. The plate valve and the feather type valve have attracted a great deal of attention during the past few years. It seems that the design of the feather valve has some desirable characteristics. The moving parts are very light, having practically no inertia. Also, it does not contain any spring. The diagrams are sufficient to give a general idea of the design and operation of the various types of valves.

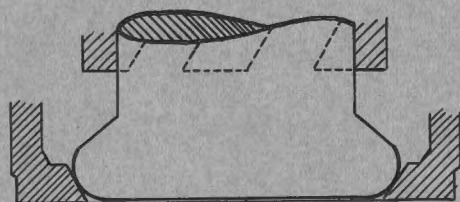
A point of interest with respect to valves is that it is possible, when using valves with very large openings, to have



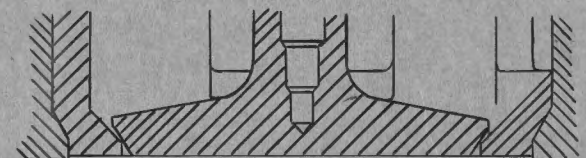
OLD TYPE POPPET VALVE



CUT DOWN POPPET VALVE

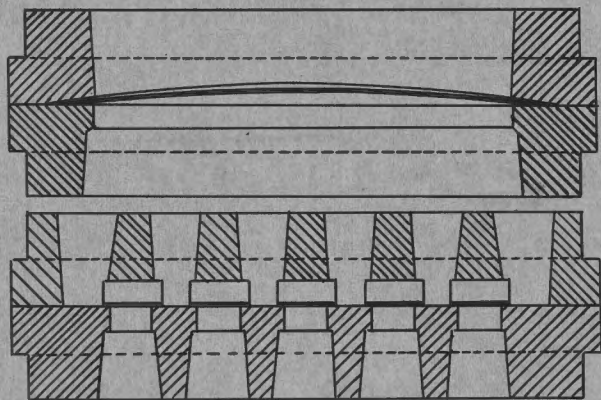


LARGE POPPET VALVE



POPPET VALVE WITH ANGLE CHANGED

VALVES *Fig 17*



LAILAW FEATHER VALVE

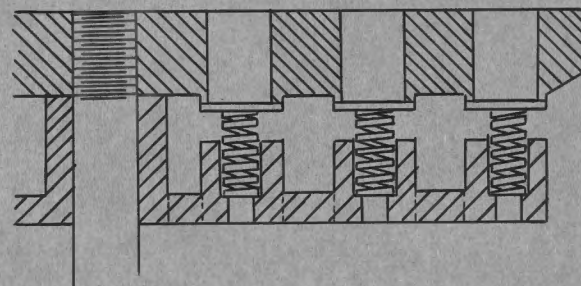


PLATE VALVE

VALVES

Fig. 18

the volumetric efficiency go above 100%. This does not mean , of course, that the cubic feet of gas handled by the compressor is greater than the piston displacement, but it does mean that the cubic feet of gas handled by the compressor is greater than that by the theoretical thermodynamic formulas, taking into account the re-expansion of the gas trapped in the clearance space and the superheating effect. This is true due to the fact that if the gas passages are not restricted in area the gas enters with a high velocity and has considerable inertia. This inertia causes more gas to flow into the cylinder than would ordinarily flow into it under normal conditions.

SUPPLEMENTARY TO THETA-PHI DIAGRAM DISCUSSION

Effect of Different Conditions Incident to
Volumetric Efficiency upon the Temperature Entropy
or ϕ - ϕ Diagram

The Theta-Phi Diagram is a thermo-dynamic chart which is a combination of four charts. Referring to the Theta-Phi Diagram at the end of this discussion, any point on the curve XY in quadrant I shows the relation between pressure and temperature for saturated ammonia gas. In Quadrant II there are two curves, $X_2'Y_2'$ and $X_2''Y_2''$. $X_2'Y_2'$ shows the relation between temperature and entropy for liquid ammonia. $X_2''Y_2''$ shows the temperature-entropy relationship for saturated ammonia gas. Entropy is defined as the ratio of the heat contained by a gas to its temperature.

$$S = \frac{Q}{T}$$

S = Entropy

Q = Heat (B.T.U.)

T = Temperature ($^{\circ}$ F.)

In Quadrant III are constant temperature lines. Their use will be shown later. The coordinates for Quadrant IV are pressure and volume. The curve X_4Y_4 shows the pressure-volume relationship for super heated gas, while any point above the curve shows the same for the gas containing particles of liquid ammonia.

Pressure and volume are the coordinates for the indicator card. Therefore the indicator card diagram may be constructed in Quadrant IV of the Theta-Phi Diagram. This figure can then be transferred to Quadrant II, having temperature-entropy coordinates. A point may be transferred from Quadrant IV to Quadrant II in the following manner. Draw a vertical and horizontal line through the point. From the point where the vertical line cuts the curve in Quadrant I draw a horizontal line. From the point when the horizontal line through the original point in question intersects the constant temperature line in Quadrant III which corresponds to the temperature of the given point, draw a vertical line. The point where this vertical line intersects the horizontal line in Quadrant II is the transferred position of the original point. The whole indicator card may be transferred to the temperature-entropy diagram by this means.

Area under a curve on the pressure-volume diagram in Quadrant IV represents work done on the compressor in foot-pound units. The area under a curve on the temperature-entropy diagram in Quadrant II represents the work done on the compressor in heat units (B.T.U.) These two areas, then, represent the same thing, expressed in different units. This different way of expressing the same thing helps to make some things about the indicator card more intelligible.

THE IDEAL THETA-PHI DIAGRAM FOR STANDARD TON CONDITIONS WITH CLEARANCE EFFECTS

Next, consider the actual case with clearance compressor. Area $A_4B_4G_4H_4$ represents an indicator card having clearance effects. Line C_4D_4 is the no clearance line. When transferred to the temperature-entropy diagram it becomes C_2D_2 . Likewise, G_4H_4 is the clearance line and becomes G_2H_2 when transferred. The indicator card with clearance when transferred to Quadrant II becomes $A_2B_2G_2H_2$. The area $G_2H_2D_2C_2$ is that lost due to clearance effects. It cannot be said that this lost area represents lost work because it has been shown before that clearance does not cause any loss of work, but it does cause a loss in the capacity of the compressor. The transferred area of the card $A_2B_2G_2H_2$ may be still considered as representing the effective work of the compressor. The area out to the no clearance line C_2D_2 is $A_2B_2C_2D_2$. This area represents the effective work which would have been done if there had been no clearance. The ratio of these two areas, then, is the volumetric efficiency due to clearance of the compressor.

The following deductions may be made for using the Theta-Phi Diagram to show the effect on volumetric efficiency of various conditions:

1. The area of the indicator card on the temperature-entropy plane represents the actual work done by the compressor.
2. The part of the area which is included between the lines $X'Y'$ and $X''Y''$ is the part of the work which is affected by the clearance. This area represents the heat involved in

the compression cycle for each pound of ammonia circulated not including the work dissipated by heat of the superheated area.

3. Any part of the area of the card which extends outside of the XY lines indicates a loss of volumetric efficiency by superheating, but may add to the heat actually carried away by each pound of condensing water since the temperature difference between cooling water and superheated gas is increased by this superheated condition.

4. Any area which is included between this horizontal lines representing the suction and discharge pressures when transferred to the TS plane and the lines X'Y' which is not included in the area representing the transferred indicator card represents a loss in volumetric efficiency.

To use the Theta-Phi Diagram for actual conditions, draw the actual indicator card as taken from the compressor in Quadrant IV and transfer it to the temperature-entropy diagram. It will be noticed, of course, that scale of pressure and volume are different on the Theta-Phi Diagram from that on the indicator card taken from the compressor. This fact will have to be taken into account when drawing the card on the diagram.

IDEAL THETA-PHI DIAGRAM FOR 100% VOLUMETRIC EFFICIENCY
WITH COMPRESSION MAINTAINING SATURATED GAS CONDITIONS

First, consider a case in which the ammonia gas is drawn into the cylinder in a saturated condition and that the gas remains saturated throughout compression. Also, assume that the compressor has no clearance. The indicator card for such a theoretical condition is shown on the diagram as $A_4B_4C_4D_4$. This card, transferred to the temperature-entropy plane is $A_2B_2C_2D_2$. The admission pressure line A_4D_4 and the discharge pressure line B_4C_4 when transferred become A_2D_2 and B_2C_2 . The area included by these two lines and the lines $X_2'Y_2'$ and $X_2''Y_2''$ represent the actual work done by the compressor on the gas. It has already been shown that

$$\frac{\text{Effective Work}}{\text{Actual Work}} = \text{Volumetric efficiency.}$$

The area of the indicator card included in the above area as shown when transferred on the temperature-entropy plan represents the effective work done by the compressor. Referring to the diagram, it is seen that for the case assumed, the useful work equals the actual work and the volumetric efficiency is 100%. That is, if gas enters the cylinder in a saturated condition and remains saturated throughout compression the volumetric efficiency would be 100%, provided the compressor had zero clearance. Of course, this is a hypothetical case and is not obtained in practice.

IDEAL THETA-PHI DIAGRAM FOR STANDARD TON CONDITIONS WITH NO CLEARANCE

Standard ton conditions have been described earlier in the paper. For standard ton conditions the gas enters the cylinder at a pressure of 19.59 pounds gage and at a temperature of $^{\circ}\text{F}$. The gas is discharged at a pressure of 154 pounds gage and a temperature of $^{\circ}\text{F}$. The figure $E_2F_2B_2C_2D_2$ is the transferred area. This represents the actual work done. But, as stated before, the effective work is represented by that part of the total transferred area which is enclosed by the lines $X_2'Y_2'$ and $X_2''Y_2''$. Therefore, in this case

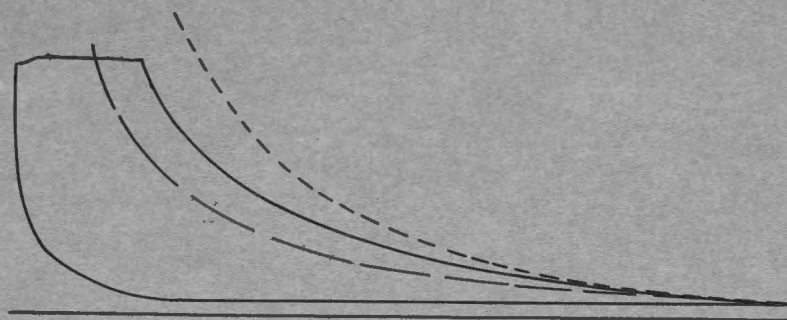
$$\text{Volumetric efficiency} = \frac{(\text{Area } B_2C_2D_2E_2) (K \times \text{area } E_2F_2B_2)}{\text{Area } B_2C_2D_2E_2}$$

In the above equation K is some factor which is less than one. The reason for the necessity of this factor is that, because of a higher temperature due to superheating the rate of heat transfer to the cooling water is greater than with a lower temperature and, therefore a large area as $E_2F_2B_2$ in the superheat region does not represent as much loss as a similar area of loss included between the $X_2'Y_2'$ and $X_2''Y_2''$ lines. The greater the degree of superheat with which the gas enters the compressor the farther apart A_2 and E_2 will become and increase the area of lost work, as all area outside of the XY lines represents lost work. It is, also, noticeable from the diagram that the nearer the actual compression curve falls to the saturation curve, the smaller will be the area of lost work, and, therefore, the higher is volumetric efficiency.

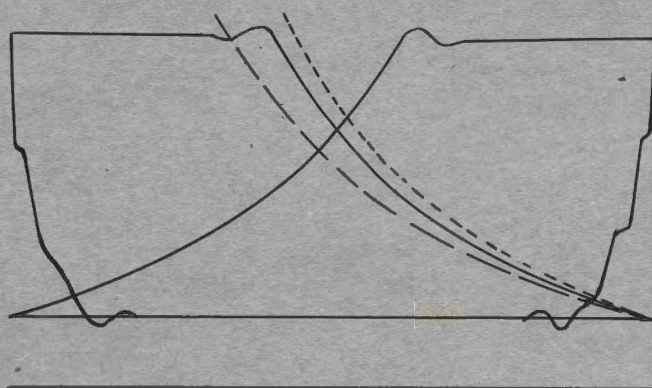
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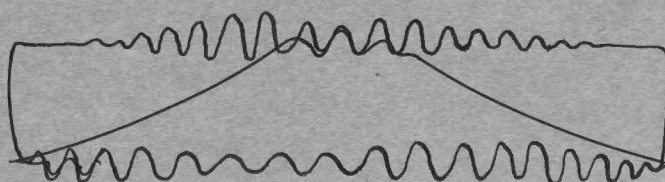
Appendix
Curves and Drawings



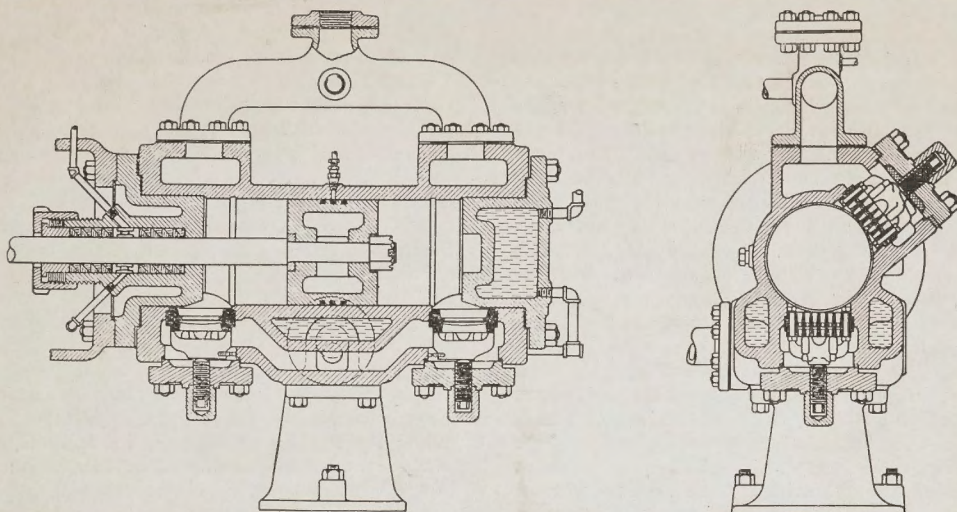
East Tennessee Packing Company
DeLavernge Machine
100# spring



Low Pressure 30# spring

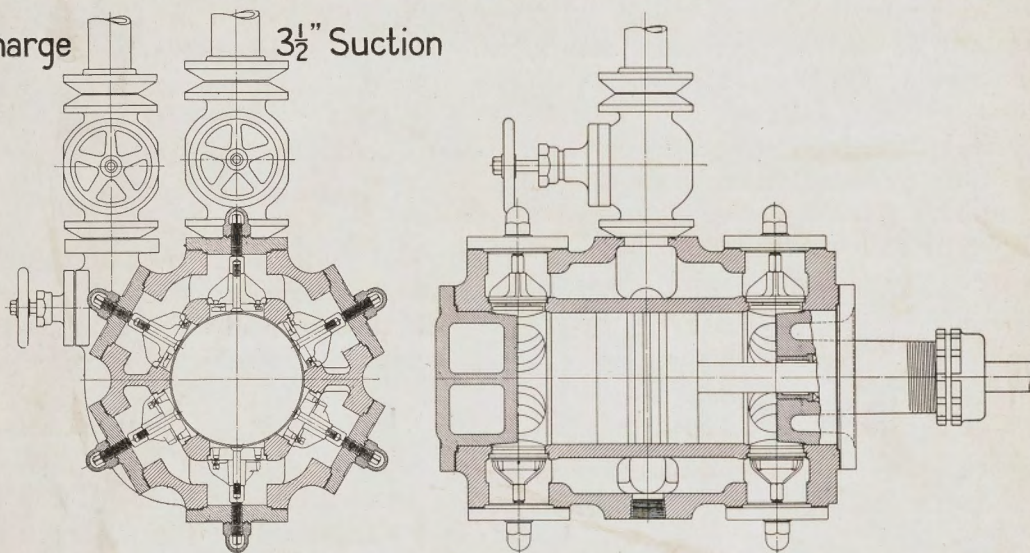


High Pressure 80# spring
Morgan Ice Company
Ingersoll-Rand Compound Compressor

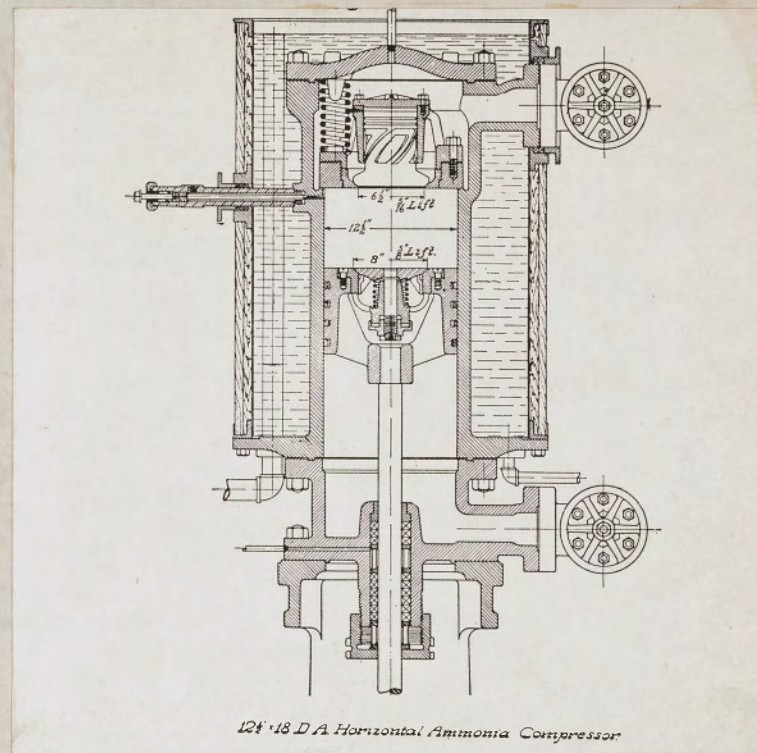
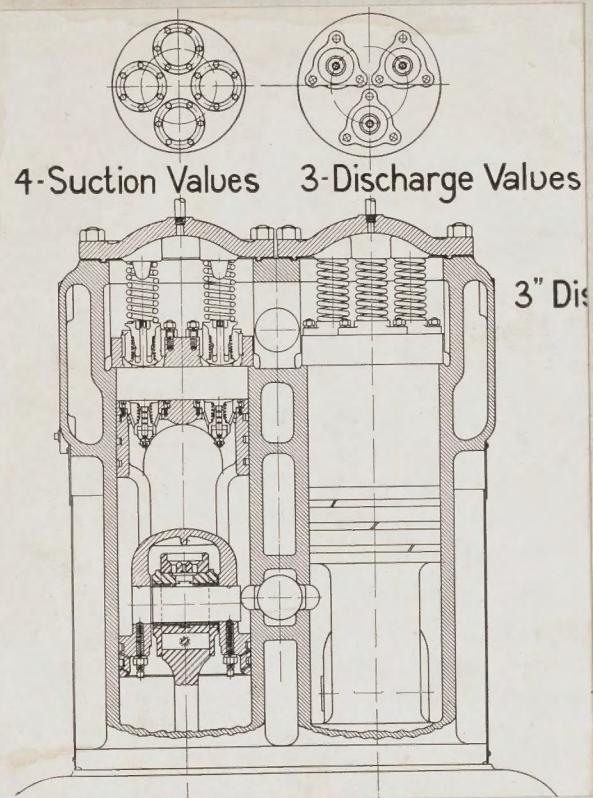


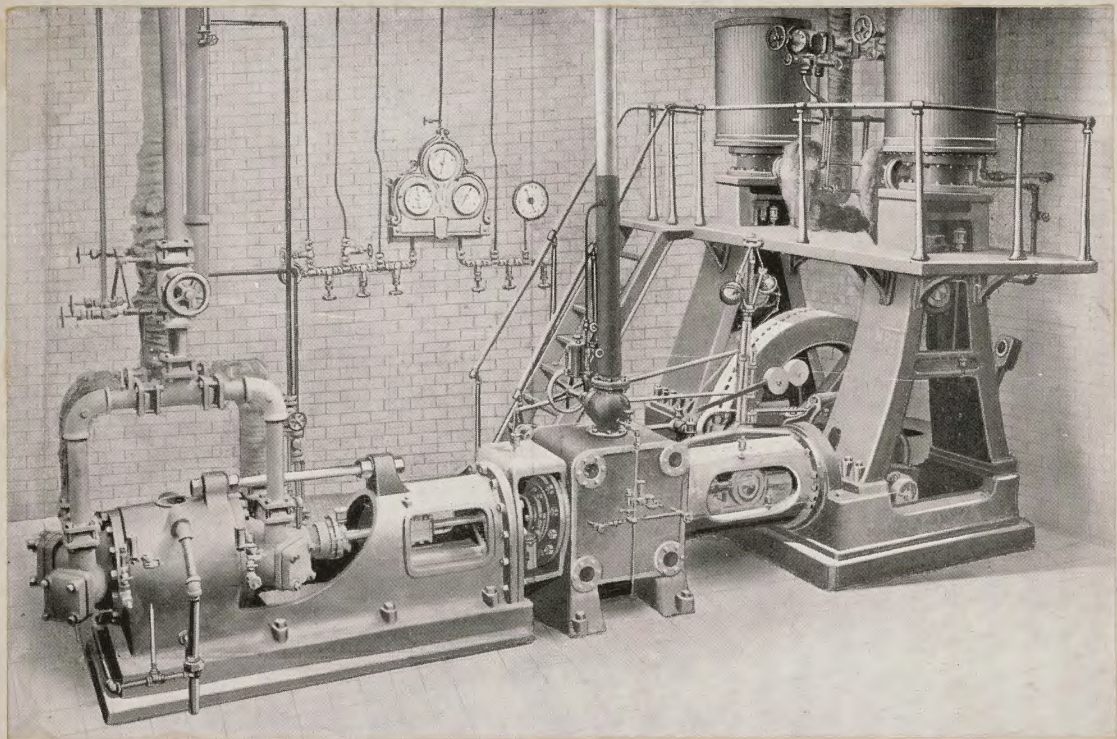
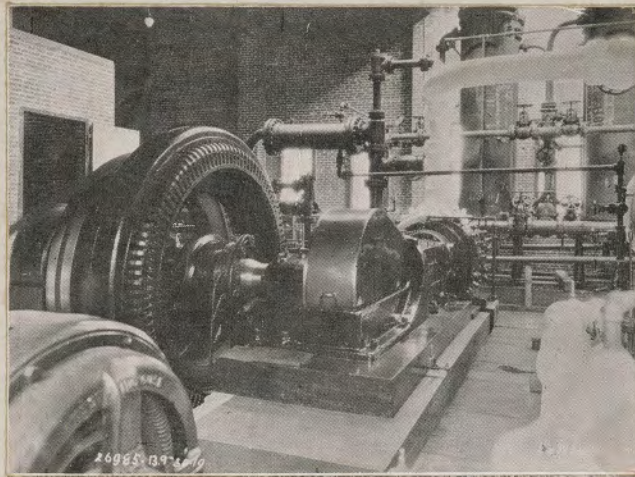
3" Discharge

3½" Suction



Horizontal Double-Acting Compressor Cylinders





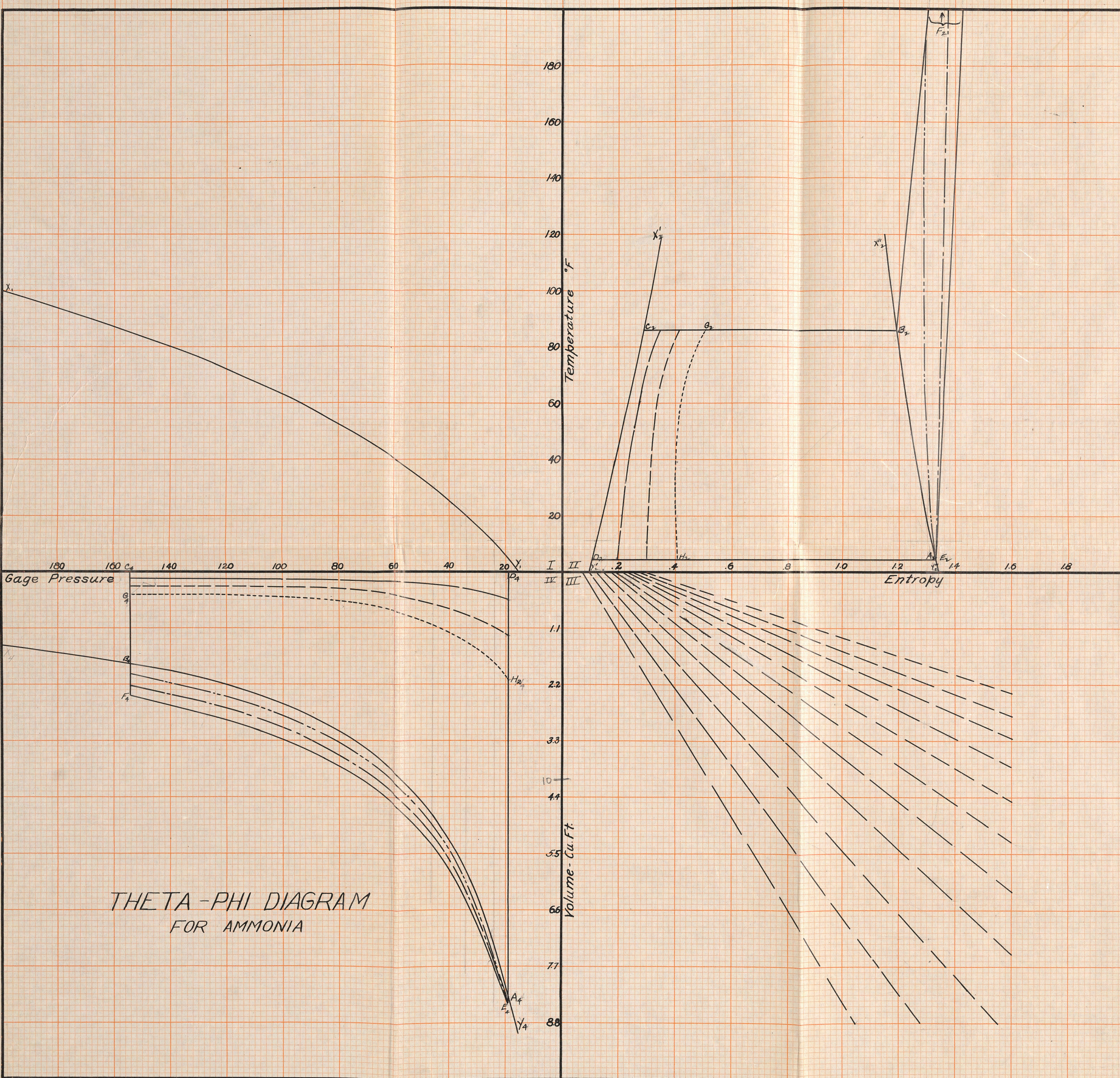
Ammonia Compressor Installations

Approval

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

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THETA-PHI DIAGRAM
FOR AMMONIA