AN ANALYTICAL AND NOMOGRAPHICAL SOLUTION FOR THE OPTIMUM OPERATION OF THE WATER-COOLED REFRIGERATION CONDENSER

Thesis for the Degree of M. S.

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Rafael Bawabé

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ESIS

This is to certify that the

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An Analytical and Nomographical Solution for the Optimum Operation of the Water-Cooled Refrigeration Condenser

presented by

Rafael Bawabe'

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Donald J. Renwick
Major professor

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AN ANALYTICAL AND NOMOGRAPHICAL SOLUTION FOR THE OPTIMUM OPERATION OF THE WATER-COOLED REFRIGERATION CONDENSER

BY

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A THESIS

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Department of Mechanical Engineering

1954

In the water-cooled vapor-compression refrigeration unit the main operational costs, aside from labor, are for power and cooling water. The relationship between these two is such that, for any given conditions, the amount of power needed would be inversely proportional to the amount of cooling-water used. The more water used, the lower would be the condensing temperature and consequently less power for compression would be required, if water is expensive and the amount of water used is small the condensing temperature would be high and obviously more power would be needed for the compression of the refrigerant. If one thinks in terms of the cost of purchasing the water and power, the above relationship would immediately suggest that there must be some optimum condition at which the total cost of operation is a minimum. That is, given the cost of water and the cost of power, a water-cooled refrigeration unit operating at some suction temperature would have to operate at a certain condensing temperature which is the most economical for these conditions. The amount of water and the amount of power to be consumed by this unit must balance each other so that the resulting total cost is a minimum.

The paper proposesa completely analytical solution for the optimum condensing temperature taking into consideration all the variables involved. In the derivation of the equations two main assumptions were made: 1) compression is isentropic, 2) the heat to be removed by the condenser is the refrigeration-effect plus the theoretical energy added to the refrigerant by compression. An important part in the derivations is the proof that, for any suction temperature, the relationship between the condensing temperature and the compression required is linear. The solution proposed is in terms of temperatures rather than pressures and so it is applicable to more than a single refrigerant. It is shown that the refrigerant used has very little effect on the solution for the optimum condensing temperature. A comparison of the solution proposed with different methods and solutions for the same problem by various authors shows its advantages and simplicity.

In addition the paper also presents a nomographic solution for the optimum condensing temperature. The nomograph is simple to use and can be used for any water-cooled compression refrigeration unit using any of the common commercial refrigerants. This nomograph is particularly convenient for use by operators of refrigeration equipment who do not have a technical education.

References:

Macintire, H.S.; Refrigerating Engineering, 1st rev. ed. John Wiley & Sons, Inc., New York, 1940, pp. 181-182

Buehler, L.; Economical Use of Condenser Water for Ammonia compression Refrigeration System, Ice and Refrigeration, Vol. 112, No. 1 (January 1947) pp.17

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I. INTRODUCTION

The cost of operation of any industrial piece of equipment is of prime importance to the engineer, and the problem of achieving optimum operating conditions would usually require very careful analysis of engineering economics. Often, the actual design of equipment must be directly tied to the initial cost and to the cost of operation during the useful life of the equipment. The cost of operation itself is usually a function of several variables which have to be adjusted so as to result in minimum expenses.

In the water-cooled refrigeration units the main operational costs, aside from labor, are for power and cooling water. The relationship between these two is such that, for any given conditions, the amount of power needed would be inversely proportional to the amount of cooling water used. The more water used, the lower would be the condensing temperature and consequently less power for compression would be required; if water is expensive and the amount of cooling water used is small the condensing temperature would be high and obviously more power would be needed for the compression of the refrigerant. If one thinks in terms of the cost of purchasing the water and power, the above relationship would immediately suggest that there must be some optimum condition at which the total cost of operation is a minimum. That is, given the cost of water and the cost of power, a water-cooled refrigeration unit

operating at some suction temperature would have to operate at a certain condensing temperature which is the most economical for these conditions. The amount of water and the amount of power to be consumed by this unit must balance each other so that the resulting total cost is a minimum.

The main purpose of this paper is to give a solution for the optimum condensing temperature taking into consideration all the variables involved. The literature on the subject is quite limited. The problem has been attacked by a few different methods (see reference 1, 2, 4, 6 and 8 in the Bibliography), however, these references do not offer a generalized, complete and accurate solution. The solution proposed in this paper is completely analytical and generalized so that it can be applied to any water-cooled refrigeration unit using any of the most commonly used refrigerants. In addition, a nomograph for the solution of the derived equation is given, so that an operator of refrigeration machinery without technical training can easily get a solution and adjust for the proper condensing temperature.

II. DERIVATION OF THE EQUATIONS

The actual vapor-compression refrigeration cycle differs from the theoretical cycle mostly by the amount of the superheat of the refrigerant in the evaporator and in the lines before entering the compressor, and by the degrees of subcooling in the condenser. The calculations on the cycle here were based on the following assumptions:

1) $10^{o}F$ superheating of refrigerant before leaving evaporator.

2) Additional $10^{o}F$ superheat before entering compressor.

3) $10^{o}F$ subcooling of the refrigerant before leaving condenser.

4) The compression is isentropic.

The first three assumptions are not essential for the derivation, but the result obtained could be of more practical value to the operating engineer. It will be shown later that the deviations in the actual refrigeration cycle from the theoretical one would have little influence on the results obtained, however, the assumption that the vapor compression is isentropic is quite essential.

As it has been stated in the Introduction the principal operational costs in a refrigeration plant are those for energy and water and hence the two quantities to be considered are the compression to be done by the compressor and the amount of heat to be removed by the condenser. It is clear that if the energy added to the refrigerant by compression is known, then the total heat to be removed in the condenser is the refrigeration-effect plus this added energy in the compressor. Some additional heat might be added by the friction in the cylinders, however, this is usually compensated by the cooling in the head of the compressor which is done by waste water from the condenser. The amount of heat that might be added by friction, even if no cooling of the head is provided for, is usually quite small when compared with the total heat to be removed by the condenser and it can be neglected here. If one wishes to include this quantity, which is usually unknown and at best can only be roughly estimated, it can be done without any difficulty. The quantity can be added as percentage of the total heat to be removed and its inclusion bears no consequence on the equations derived below.

Based on the above assumptions, calculations of the theoretical compression required per ton-hour were made for different refrigerants at various operating conditions and the results were plotted as shown in Figure 1 and Figures 1A-5A in the Appendix.

The plots show that, for any given suction temperature within the range of normal operating conditions, the rise in the energy required for compression is linear with the condensing temperature. These results are quite important for the following derivations since they show the increase in the energy required per one degree rise in the condensing temperature is independent of the condensing temperature. The calculations and the plots were based on suction and condensing

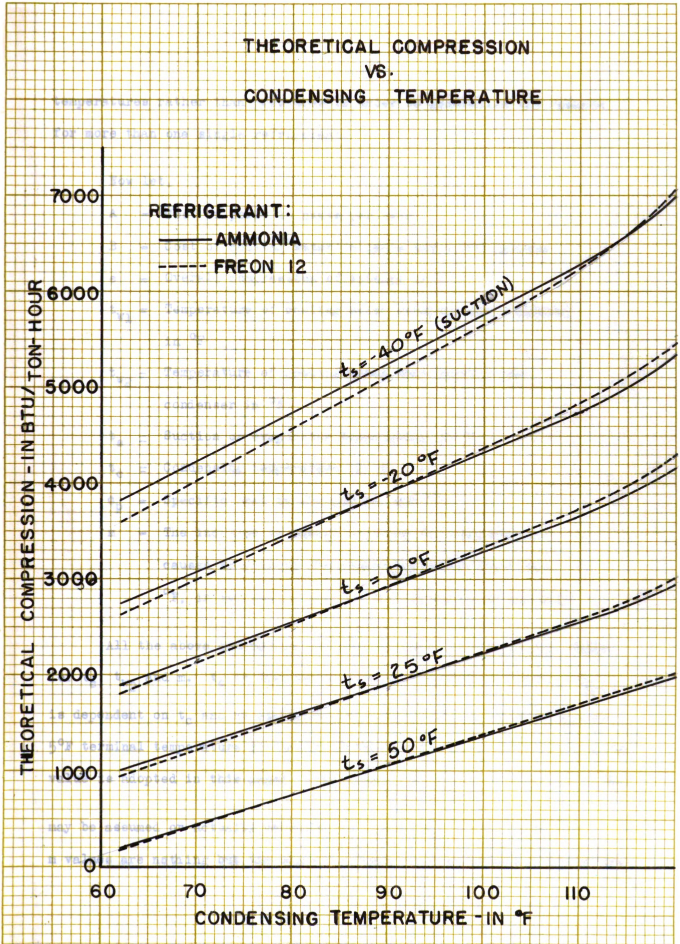
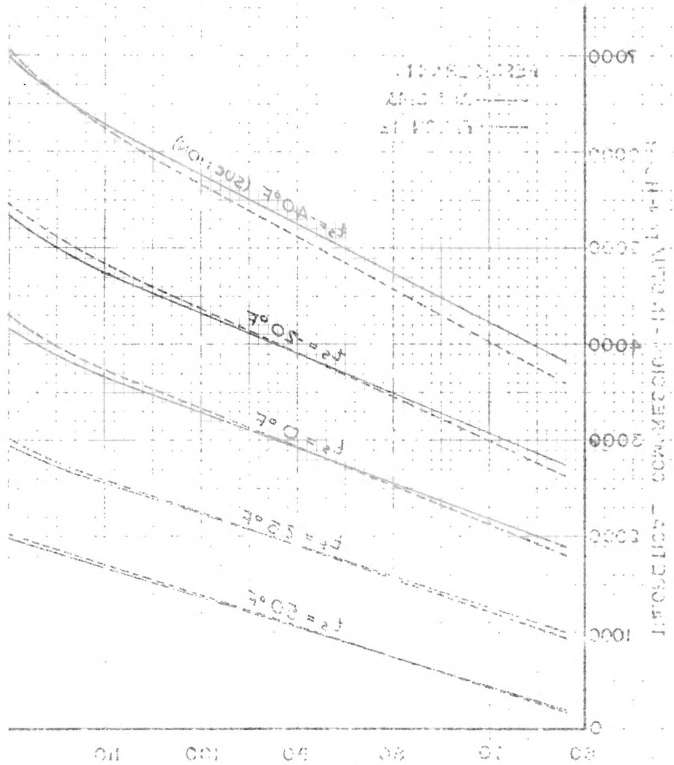

THEORETICAL COMPRESSION
VS.
CONDENSING TEMPERATURE
REFRIGERANT:
AMMONIA
FREON 12
t_s = -40°F (SUCTION)
t_s = -20°F
t_s = 0°F
t_s = 25°F
t_s = 50°F
THEORETICAL COMPRESSION - IN BTU/TON-HOUR
0
1000
2000
3000
4000
5000
6000
7000
60
70
80
90
100
110
CONDENSING TEMPERATURE - IN °F

FIG.1

THEORETICAL COMPRESSION

VS.

CONDENSING TEMPERATURE



CONDENSING TEMPERATURE - IN °F

FIG. 1

temperatures rather than pressures in order to generalize the results for more than one single refrigerant.

Now let,

A = Cost of power in cents per kw-h

B = Cost of cooling water in cents per 1,000 gallons

e = Overall compression efficiency in %

t_{w_1} = Temperature of cooling water entering the condenser in ^oF

t_{w_2} = Temperature of cooling water when leaving the condenser in ^oF

t_s = Suction or evaporator temperature in ^oF

t_c = Condensing temperature in ^oF

c_p = Specific heat in Btu per lb. per ^oF

m = The increase in energy required for compression caused by raising the condensing temperature one ^oF, in Btu per ^oF per ton of refrigeration per hour

All the above quantities are usually known or measurable except for t_c, t_{w_2} and m. t_c is the variable whose solution is sought. t_{w_2} is dependent on t_c and an assumption is needed here. An assumption of 5^oF terminal temperature difference would be quite satisfactory and this value is adopted in this paper, from this $\underline{t_{w_2} = t_c - 5}$. Any other value may be assumed or actually measured and then used in the equation. m values are nothing but the slopes of the lines in Figures 1A-5A in the

Appendix. (Tabulated values for m, for different refrigerants and different suction temperature, may be found in Table 1A in the Appendix.)

If the condensing temperature is to be t_c, then the theoretical compression work necessary per one ton of refrigeration per hour is: $m(t_c - t_s)$ Btu, and with the overall efficiency being e, the energy input is:

$$\frac{m(t_c - t_s)}{3412(e/100)} \text{ Kw-h per ton per hour}$$

The total heat to be removed by the condenser is $12,000 + m(t_c - t_s)$ Btu per ton per hour; from this the amount of cooling water required is:

$$\frac{12,000 + m(t_c - t_s)}{c_p(t_{w_2} - t_{w_1})} \text{ lbs. of water per ton per hour}$$

Introducing the costs of electricity and water and substituting for t_{w_2} one gets the total cost of operation for one ton of refrigeration per hour:

$$(1) \quad C = \frac{Am(t_c - t_s)}{3412\ (e/100)} + \frac{\left[12,000 + m(t_c - t_s)\right] B}{(t_c - 5 - t_{w_1})\ 8,330} \text{ cent per ton per hour}$$

c_p was assumed to be equal to unity.

From thermodynamic considerations in order to minimize the overall cost, C, the equation should be differentiated with respect to t_c and equated to zero.

$$(2) \quad \frac{dC}{dt_c} = \frac{Am}{3412(e/100)} + \frac{mB(t_c - 5 - t_{w_1}) - \left[12,000 + m(t_c - t_s)\right]B}{8,330\left[t_c - (t_{w_1} + 5)\right]^2} = 0$$

solving from this equation for t_c:

$$(3) \quad t_c = (t_{w_1} + 5) \pm 0.064\sqrt{\frac{B}{A}e\ \left(t_{w_1} + 5 + \frac{12,000}{m} - t_s\right)}$$

The negative sign should be discarded since the condensing temperature cannot be lower than $t_{w_1} + 5$. It will be equal to $t_{w_1} + 5$ if water cost $B = 0$.

The final equation then, is:

$$(4) \quad t_c = (t_{w_1} + 5) + 0.064\sqrt{\frac{B}{A}\, e\, (t_{w_1} + 5 + \frac{12,000}{m} - t_s)}$$

which is the solution sought for the optimum condensing temperature.

III. THE NOMOGRAPHIC SOLUTION

The construction of a nomograph with seven variables is not a simple problem, especially when some of the variables have functional relationships such that they are not easily separable as in the above equation (Eq. 4). The details and proof of the construction of the nomograph presented in this paper (Figure 2), will not be given since this is not the primary concern of the paper. However, a brief outline of the method might be of interest and that is given here.

The equation

$$t_c = (t_{w_1} + 5) + 0.064\sqrt{\frac{B}{A} e \left(t_{w_1} + 5 + \frac{12,000}{m} - t_s\right)}$$

is separated in the following manner:

let $K = 0.064\sqrt{\frac{B}{A} e}$ then,

$$t_c = (t_{w_1} + 5) + K\sqrt{t_{w_1} + 5 + \frac{12,000}{m} - t_s}$$

where K now is a variable coefficient of a certain function and it would be one axis in the nomograph to be constructed from this reduced equation for t_c. To get this axis write $\frac{K}{\sqrt{e}} = 0.064\sqrt{\frac{B}{A}}$ and this type of relationship can be easily represented nomographically by the "double" Z-chart method. In Figure 2, K is the central axis and it is not graduated since its value is not required and the K axis serves only as a pivot-line for the continuation of the nomograph.

Now, if constant values are assigned to m and t_s the above reduced equation becomes a polynomial in t_{w_1} of the form:

$$f_1(x) + K\, f_2(x) - L = 0$$

where,

$x = t_{w_1}$

$f_1(x) = t_{w_1} + 5$

$f_2(x) = \sqrt{t_{w_1} + 5 + \frac{12,000}{m} - t_s}$

$L = t_c$

m and t_s are fixed constants

Polynomials of this form have nomographic representation, however, $f_2(x)$ here has two "variable constants" in it and the most that one can incorporate within $f_2(x)$ for a "net-chart" is one "variable constant". In other words, in the construction of a nomograph for the variables x, K and L one can assign an additional variable and for each given value of this added variable a different curve for x results since $f_2(x)$ changes with different values of this constant that is incorporated with it. This additional variable is usually called "variable constant" and the resulting nomograph would have a "network" between x and this "variable constant".

In the above reduced equation there are two such "variable constants", namely, m and t_s and the equation must be reduced further in order to enable polynomial nomographic representation. Fortunately, it is possible in this case to get a good approximate solution when the above equation is reduced further to include only one "variable constant". Looking at the values for m in Table 1A. in the Appendix it is seen that the variations in m for different refrigerants at any given suction temperature, t_s, are quite small and an average value would be quite satisfactory. It is natural then to assume that by assigning a value to t_s the value of m is fixed too, and hence m and t_s are considered as one "variable constant". Now, the construction of the "net-chart" nomograph is possible, since $f_2(x)$ has been shown to have only one "variable constant".

The last assumption or "reduction" in the equation has the advantage of making the nomograph completely independent from any primary calculations or finding a value for m, and if one considers temperatures only, the refrigerant circulated in the unit has no influence on the results.

The procedure followed in the construction of the nomograph for the reduced equation is quite lengthy and it is not in the scope of this paper to describe this procedure or prove the construction. The literature on nomography is quite abundant and many authors deal with representation of polynomials by different methods.

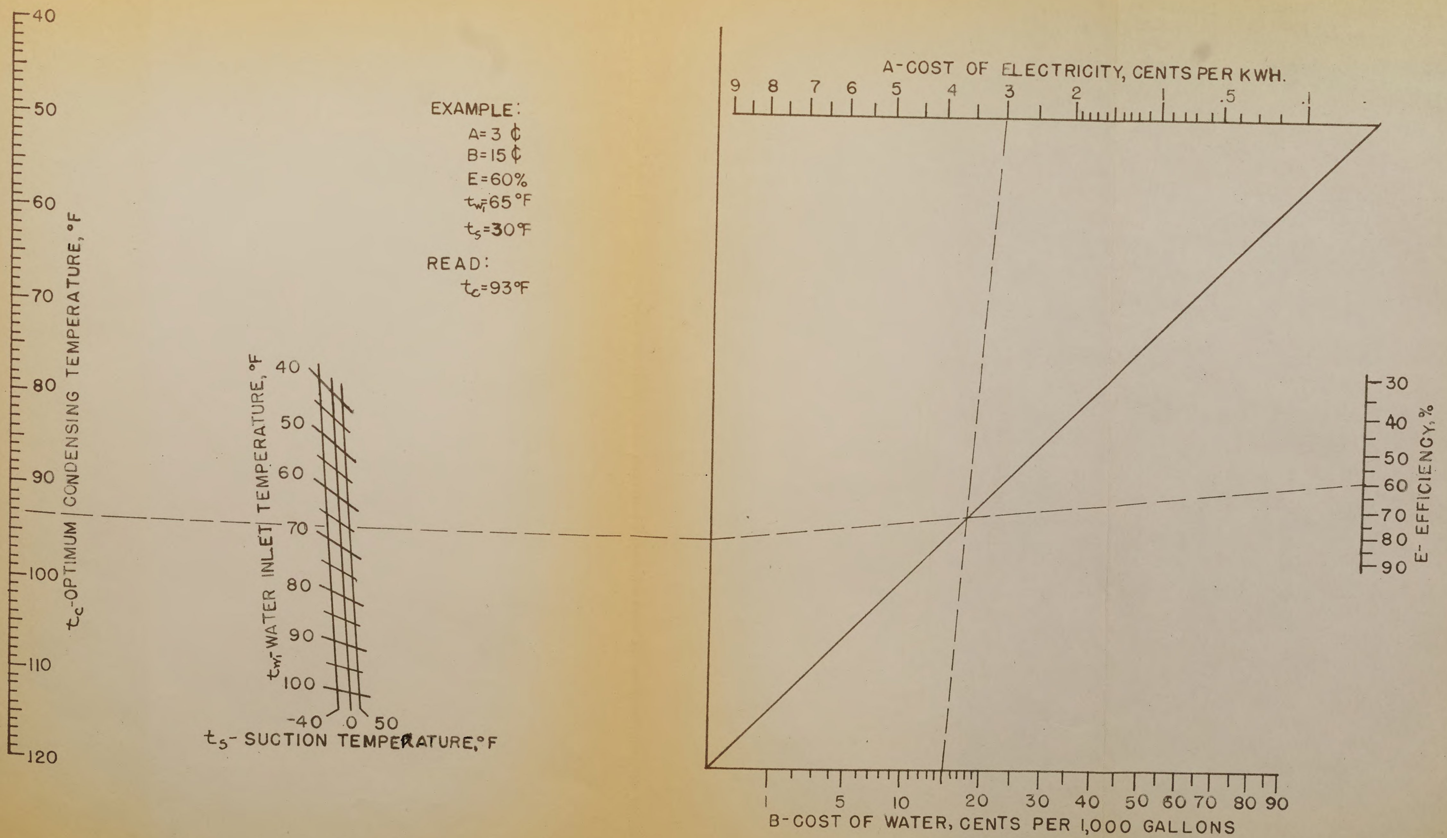


FIG. 2. OPTIMUM CONDENSING TEMPERATURES FOR COMPRESSION REFRIGERATION SYSTEMS

IV. DISCUSSION AND CONCLUSIONS

A comparison of the equations derived in this paper with other equations or solutions, for the same problem, proposed by different authors would show best the advantages and the simplicity of the equations proposed here.

A solution similar, to a certain extent only, to the solution given above was proposed by H. J. Macintire (see references 1 and 2). His equation solves for the economical temperature rise of the cooling water rather than for the optimum condensing temperature or pressure. The total cost of operating a refrigeration unit according to this equation is:

$$C = \frac{Amt_d}{e} + \frac{0.0072\ HB}{t_d}$$

The notation is the same as used in this paper except for:

t_d = economical temperature rise of the cooling water in ^oF.

H = the heat to be removed by the condenser in Btu per ton per minute.

There are two mistakes in this equation and one of them is an appreciable one: 1) The difference between the suction temperature and the condensing temperature (t_d) has the same numerical value as the temperature rise in the cooling water (t_d). This situation is practically impossible. Only if the cooling water entered at the same temperature as the evaporator suction temperature, and also if there were no terminal temperature difference between the condensing temperature and the water outlet temperature, could this be mathematically correct. From both a practical and physical standpoint this would be an impossible operating condition for a refrigeration system. If the t_d terms, in the above equation, are intended to be equal and have the defined value above (of economical water temperature rise), then a fairly large part of the electrical cost for operating the compressor has been left out of this total cost equation. If the above equation were to represent the total cost the first term in the equation should be:

$$\frac{Am(t_d + t_{w_1} + 5 - t_s)}{e}$$

2) The second error is in H. The heat to be removed by the condenser is assumed to be known and constant or a trial and error solution would be necessary since H is dependent on the varying condensing temperature. However, the error in here is not too serious, especially since one can find approximate values for H in the literature.

While this equation for total cost of operation is considerably in error as noted above in No. 1, the solution for the economical water temperature rise yields fairly good values, since most of the error disappears in the process of differentiating the equation with respect to t_d when trying to solve for minimum cost. It is of interest to note how often this erroneous equation has appeared in refrigeration textbooks. It first appeared in this exact form in Macintire's first edition and in the revised edition of his book and also in the new book by Macintire and Hutchinson (reference 2). The equation was also picked by Jordan and Priester in their textbook (see reference 3).

A completely different approach to the problem has been proposed by Boehmer (references 6 and 7), but his solution is limited to "Freon 12" only and for units from 10 to 60 hp. His equation applies only to relatively high suction temperatures as are encountered in air-conditioning units and to some extent he relies on manufacturers' data without allowing for different efficiencies for different units. In addition, no direct solution for the optimum condensing pressure or temperature is possible. The graph proposed for finding the optimum condensing pressure has a limited range of values.

Another earnest attempt to solve the problem for ammonia condensers only, was made by L. Buehler (see references 4 and 5) and an elaborate set of equations and graphs has been proposed by him. The method is limited to ammonia as a refrigerant and for a relatively small range of suction temperatures. Different suction temperatures require different equations and no generalization is possible. In the derivation of his equation he used manufacturers' data to a great extent and the equations he arrives at are actually empirical equations rather than analytical solutions.

Returning now to the solution proposed in this paper. Of all the variables involved in the solution for t_c, the temperature of the entering cooling water, t_{w_1}, has the largest effect. All other variables change only as the square-root and their effects on t_c vary accordingly. The suction temperature, t_s, has very little influence on t_c; this can be best seen in the nomographic solution (Figure 2) where the curves for t_s are crowded together. This fact might seem a little surprising, but noticing the lines in Figure 1, (and Figures 1A. to 5A.) it is seen that the different lines for different t_s are almost parallel and one might expect just a linear relationship between t_s and the total cost of operation.

The dependence of t_c on m and the refrigerant used is not too significant either. m is actually a measure to how "economical" the refrigerant is. The variations here are not too big. Table 1A. shows that ammonia and Freon 11 are probably the most "economical" refrigerants, since they require less work for compression with rising t_c and this only for low evaporator temperatures.

From the above considerations one may safely assume that deviations in the actual refrigeration cycle from the cycle considered here (see page 3) will have little effect on the solution for the optimum condensing temperature. One may say that the factors that appreciably affect the solution for optimum operating conditions are not inherent within the refrigeration unit itself, but are external to it such as the costs of water and electricity and in particular the temperature of the available cooling water.

In the same plant, variations in the temperature of the cooling water may be quite large during the different seasons of the year and this immediately brings the problem of automatic control for the system. This phase of the problem requires special investigation by itself and no solution is being proposed here. However, an indication as to the way by which this control might be accomplished is worthwhile mentioning. Equation 4, for the condensing temperature yields at the same time the economical temperature rise in the cooling water, for, $t_c = t_{w_2} + 5$

$$\therefore t_{w_2} - t_{w_1} = .064\sqrt{\frac{B}{A}\, e\left[(t_{w_1} + 5) + \frac{12,000}{m} - t_s\right]}$$

Since this economical temperature difference depends on t_{w_1} the problem of control, based on this temperature difference, is not a simple one, but it can be simplified for an approximate solution. The radical in the equation is not too sensitive to changes in t_{w_1}. With a typical value for m, say $m = 40$ and $t_s = 10^{\circ}F$ the term $\frac{12,000}{m}$ is obviously dominant and variations in t_{w_1} of 10 to 20°F will not change the value of the radical appreciably and a seasonal average for t_{w_1} will be quite sufficient. The control system then will have to measure the inlet temperature of the water, t_{w_1}, but control the outlet temperature, t_{w_2}. The economical temperature rise then, is considered as constant.

One of the most important factors in the derivations given in this paper was the proof that m is independent of the condensing temperature. This must be modified, since actually it holds true only up to a certain condensing temperature. This can be best seen in the curves used to determine the values for m in Figures 1A.-5A. in the Appendix.

All the curves are straight lines up to about $t_c = 115^oF$ and above this temperature the lines curve upward. The temperature at which the lines start to curve is different for different refrigerants as can be seen in the different plots. For ammonia, for instance, the lines are straight up to $t_c = 120^oF$. The reason for these deviations at high condensing temperatures could be due to the fact that the refrigerant is at high superheat conditions, high pressure and temperature, and the amount of work required to compress it at that region is more than the work needed at lower temperatures and pressures. The constant entropy lines on a p-h diagram would indicate this.

Fortunately, it is rarely that in a water-cooled condenser it would be economical to operate at condensing temperature higher than 120^oF. Economically speaking, high condensing pressures are advisable only when electricity rates are extremely cheap or water costs very high. For most practical purposes the independence of m on t_c holds and the equations derived are definitely applicable for most industrial refrigeration machinery.

The above extensive discussion and derivations might give some distorted view as to the importance of operating at the optimum condensing temperature. To be sure, it is quite desirable to operate at the appropriate conditions, but the optimum solution is not very critical and one should not exaggerate its necessity. An example will best show the effect of deviations from the optimum condensing temperature.

Assume:

Refrigerant: Freon 12

$t_s = 30^\circ F$ $\quad t_{w_1} = 65^\circ F$

Electric rate, A = 3¢ per kw-h

Cost of water, B = 15¢ per 1,000 gallons

Overall efficiency, e = 60%

Interpolating in Table 1A, m = 33. Substituting in eq. 4, or from the nomograph (Figure 2), one finds $t_c = 93^\circ F$. Equation 1. can be used now to find the minimum total cost of operation:

$$C = \frac{3(33)(93-30)}{3412}\,\frac{(100)}{(60)} + \frac{[12,000+(33)(93-30)]\,15}{(93-5-65)(8,330)} =$$

$$= 3.07 + 1.12 = 4.19\text{¢ per hour per ton.}$$

Now, if the unit operates at $t_c = 100^\circ F$ the total cost of operation is found to be 4.25¢ per hour per ton, and if the unit operates at $t_c = 85^\circ F$ the total cost would be 4.32¢ per hour per ton.

These differences are not too large to warrant strict adherence to the optimum t_c, but the differences grow quite fast as the deviation from the optimum point gets larger. With large refrigeration units these differences may amount to large sums in the long run. The problem should not be taken very lightly. Many of the refrigeration plants today operate without the slightest attention to the problem; either because the operators are not aware of it or because of lack of information on how to achieve optimum conditions. This is especially true in plants where there is not constant engineering supervision. (There

is some severe criticism in the literature for the neglect of this problem.) With this in mind the nomograph presented in this paper was constructed to facilitate the achievement of a solution by operators without technical training.

In conclusion it will be added that, since solutions for the optimum condensing temperature are available, it is the duty of the operating engineer to try and achieve these optimum conditions. The minimum total cost of operation of a refrigeration plant can be determined without too much difficulty and it deserves the proper attention from the refrigerating engineer.

A P P E N D I X

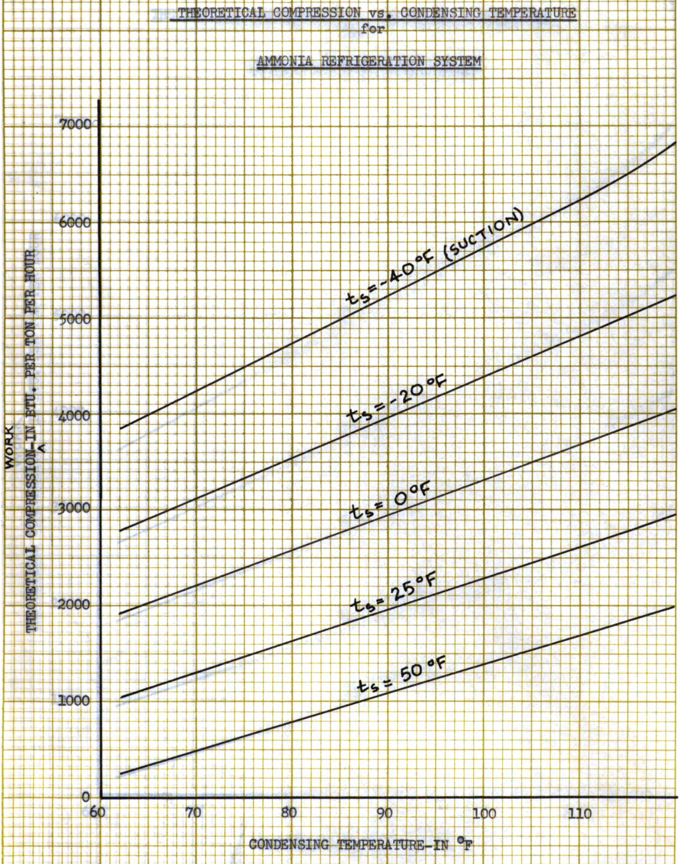

THEORETICAL COMPRESSION vs. CONDENSING TEMPERATURE
for
AMMONIA REFRIGERATION SYSTEM
THEORETICAL COMPRESSION WORK IN BTU. PER TON PER HOUR
7000
6000
5000
4000
3000
2000
1000
0
60
70
80
90
100
110
CONDENSING TEMPERATURE-IN °F
ts=-40°F (SUCTION)
ts=-20°F
ts= 0°F
ts= 25°F
ts= 50°F

FIG. 1A

REFRIGERATION SYSTEM

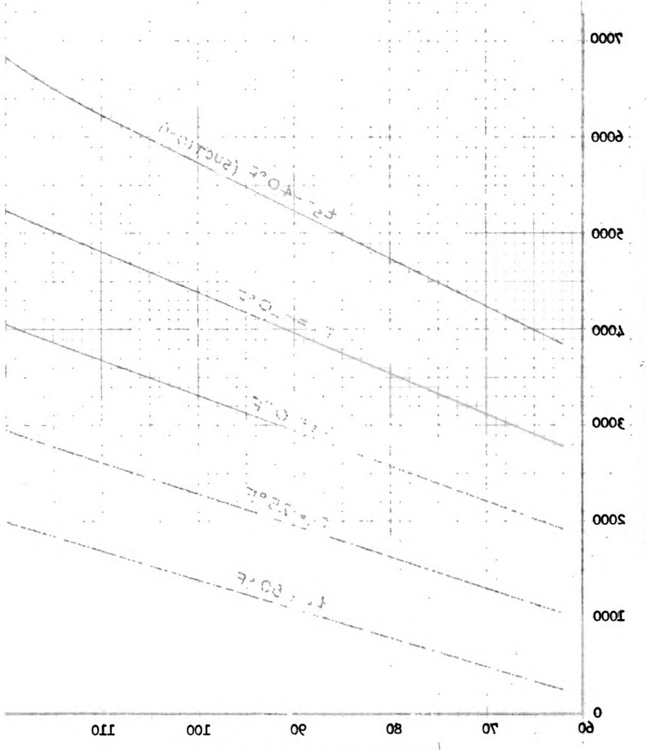


FIG. 1A

THEORETICAL COMPRESSION vs. CONDENSING TEMPERATURE
for
FREON-12 REFRIGERATION SYSTEM

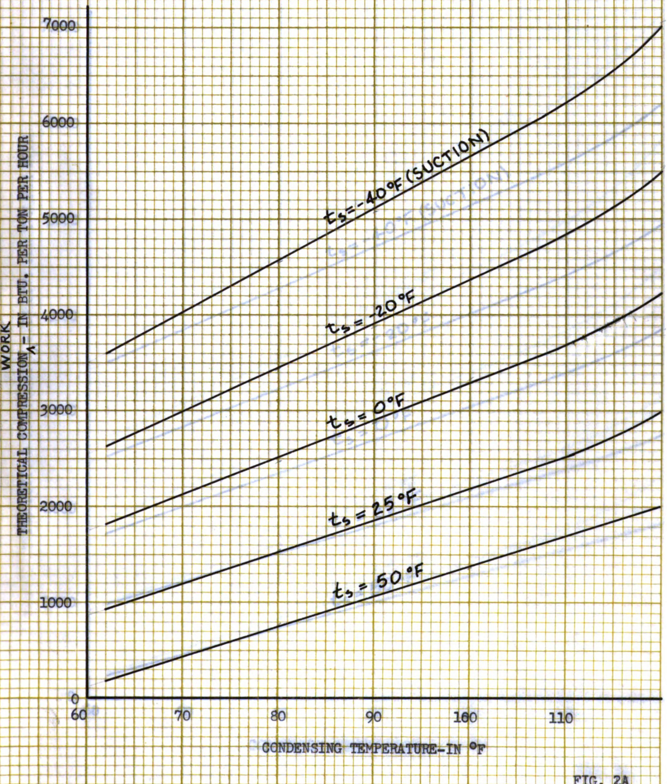


FIG. 2A

THEORETICAL COMPRESSION vs. CONDENSING TEMPERATURE
for

FREON-12 REFRIGARETION SYSTEM

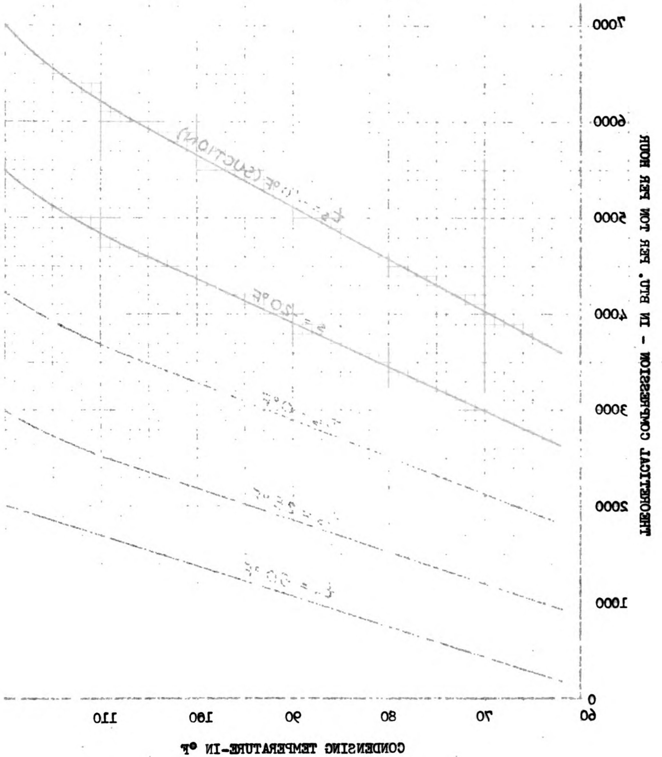


FIG. 2A

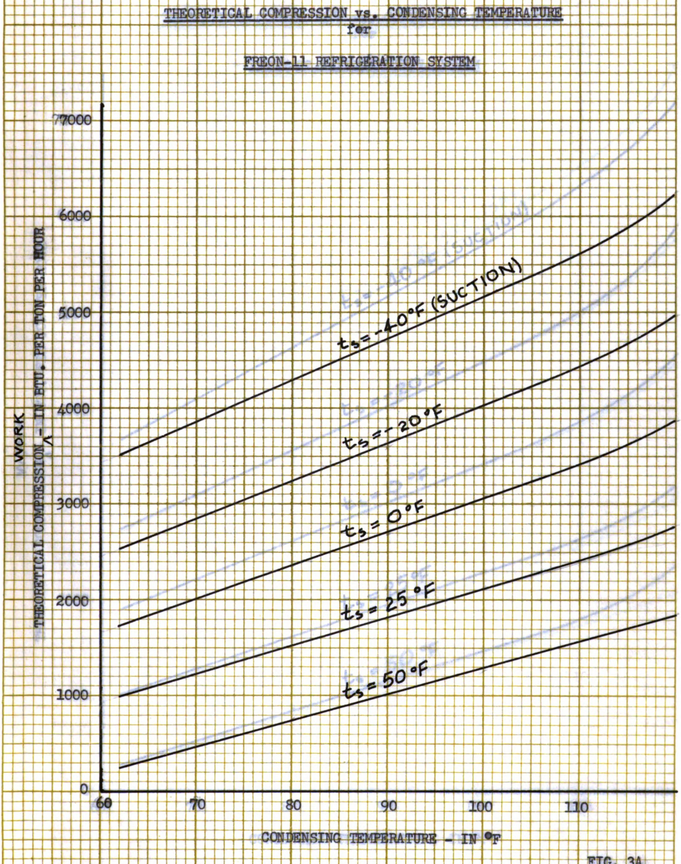

THEORETICAL COMPRESSION vs. CONDENSING TEMPERATURE
for
FREON-11 REFRIGERATION SYSTEM
WORK
THEORETICAL COMPRESSION - IN BTU. PER TON PER HOUR
7000
6000
5000
4000
3000
2000
1000
0
60
70
80
90
100
110
CONDENSING TEMPERATURE - IN °F
t_s = -40°F (SUCTION)
t_s = -20°F
t_s = 0°F
t_s = 25°F
t_s = 50°F

FIG. 3A

THEORETICAL COMPRESSION vs. CONDENSING TEMPERATURE
for

FREON-11 REFRIGERATION SYSTEM

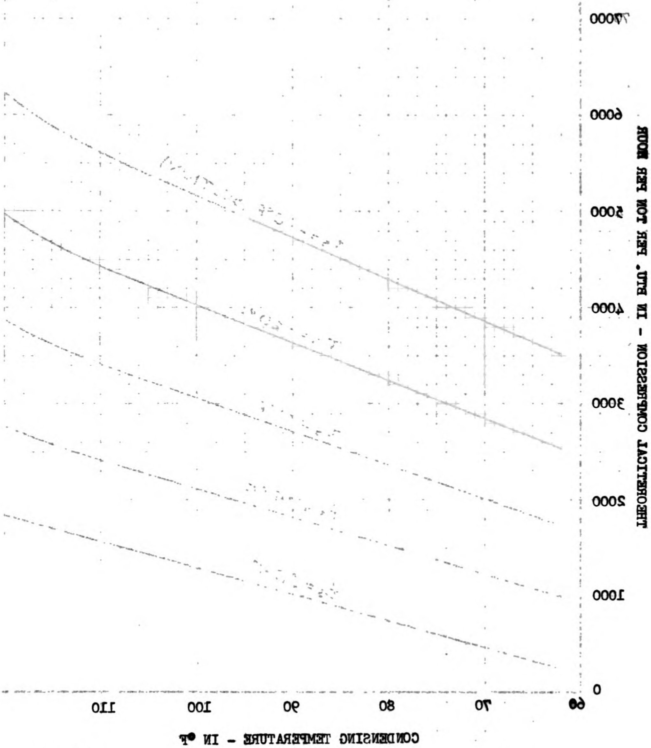


FIG. 3A

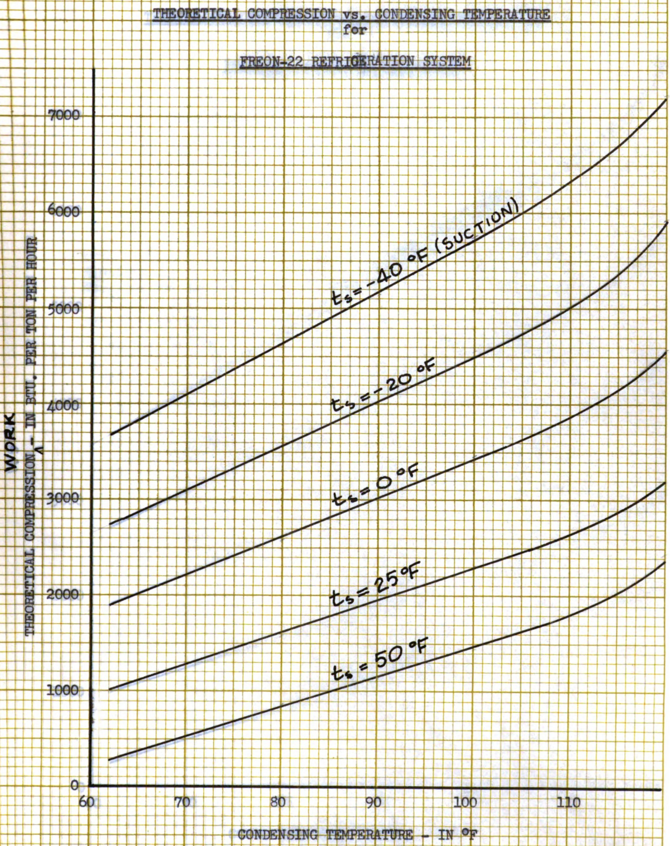

THEORETICAL COMPRESSION vs. CONDENSING TEMPERATURE
for
FREON-22 REFRIGERATION SYSTEM
WORK
THEORETICAL COMPRESSION - IN BTU. PER TON PER HOUR
7000
6000
5000
4000
3000
2000
1000
0
60
70
80
90
100
110
CONDENSING TEMPERATURE - IN °F
ts = -40 °F (SUCTION)
ts = -20 °F
ts = 0 °F
ts = 25°F
ts = 50 °F

FIG. 4A

THEORETICAL COMPRESSION vs. CONDENSING TEMPERATURE
for
FREON-22 REFRIGERATION SYSTEM

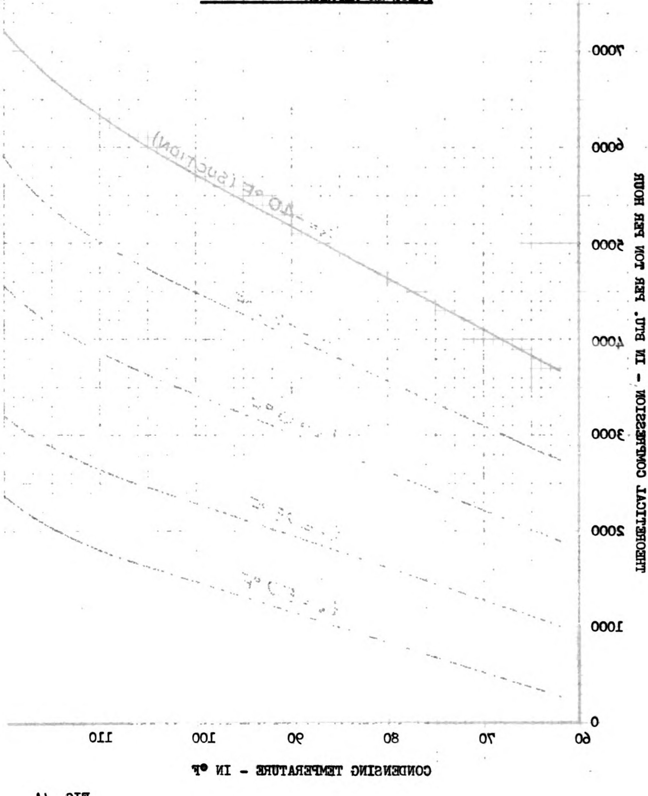


FIG. 4A

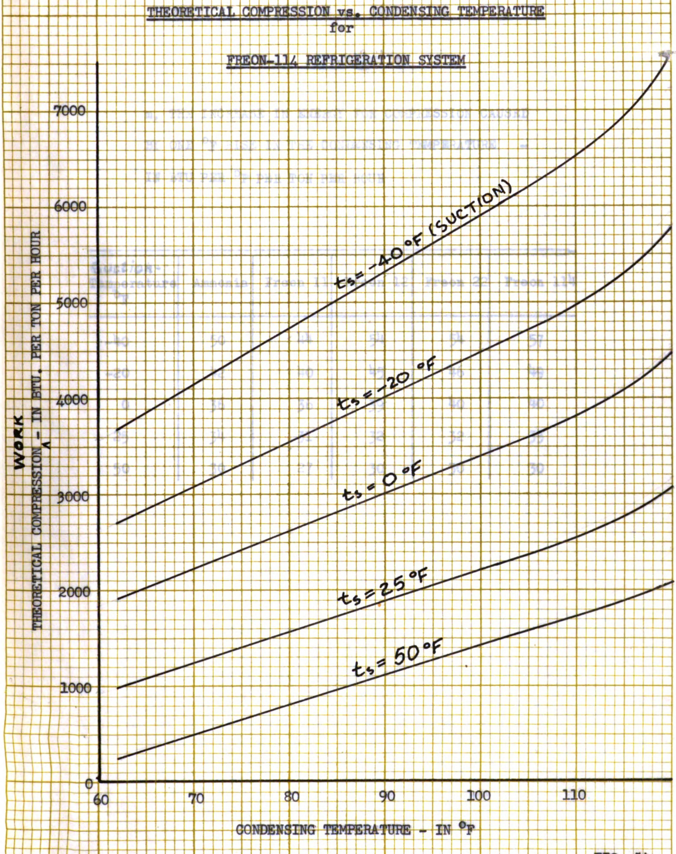

THEORETICAL COMPRESSION vs. CONDENSING TEMPERATURE
for
FREON-114 REFRIGERATION SYSTEM
7000
6000
5000
4000
3000
2000
1000
0
WORK
THEORETICAL COMPRESSION - IN BTU. PER TON PER HOUR
60
70
80
90
100
110
CONDENSING TEMPERATURE - IN °F
ts= -40 °F (SUCTION)
ts= -20 °F
ts= 0 °F
ts= 25 °F
ts= 50 °F

FIG. 5A

THEORETICAL COMPRESSION vs. CONDENSING TEMPERATURE
for
FREON-114 REFRIGERATION SYSTEM

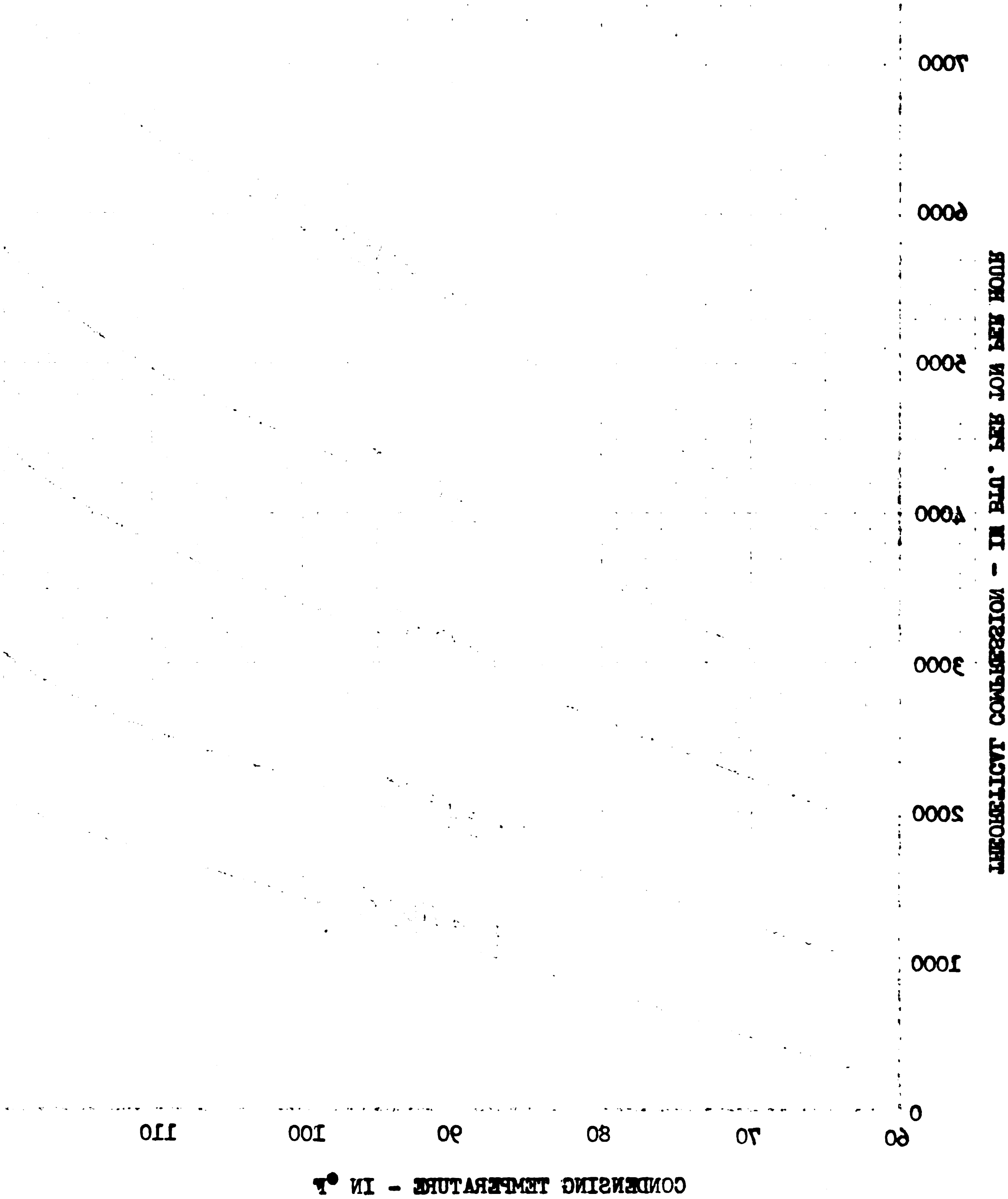


FIG. 5A

TABLE 1A

m, THE INCREASE IN ENERGY FOR COMPRESSION CAUSED BY ONE °F RISE IN THE CONDENSING TEMPERATURE - IN BTU PER °F PER TON PER HOUR

Suction Temperature °F	Ammonia	Freon 11	Freon 12	Freon 22	Freon 114
-40	50	44	54	54	57
-20	42	40	45	46	49
0	36	36	39	40	40
25	34	31	32	32	33
50	30	27	30	30	30

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