AN ANALYTICAL AND GRAPHICAL SOLUTION FOR THE MOST ECONOMICAL SIZE OF PIPING TO BE USED IN THE SUCTION, DISCHARGE AND LIQUID LINES OF A REFRIGERATION SYSTEM

THESIS FOR THE DEGREE OF M. S.
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JOSEPH E. RIZZUTO

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AN ABSTRACT

Submitted to the School of Graduate Studies of Michigan State University of Agriculture and Applied Science in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

Department of Mechanical Engineering

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Approved Donald J. Renwick Major Professor

AN ABSTRACT

The selection of the proper size of piping is of the utmost importance in the design of an efficient refrigeration system. In most cases the only basis for the selection of the refrigeration piping are rules of thumb or arbitrarily choosing pipe or tubing to match the connecting fittings of the various pieces of equipment. Such a basis of selection, however, can be very costly in the long run and penalize the economy of a system for its entire lifetime.

In designing a refrigeration system, what we are primarily interested in is a system that will accomplish its task with the least possible total cost. The two factors of concern to be included in this total cost are the initial cost of pipe and the operating cost for power to drive the compressor. The two factors tend to act against each other, that is the larger the size of pipes used the greater will be their cost but the lower will be the cost of power required to drive the compressor. On the other hand the smaller the size of pipes used the lower will be their cost but the higher will be the cost for power required to drive the compressor. Hence it was found that there will be some optimum point at which the sum total of these two costs will be a minimum. It is the purpose of this paper to present an analytical solution for the selection of pipes which will result in the lowest total cost for the system.

The paper treats the suction, discharge and liquid lines for use with Freon-12 as the refrigerant. The equations and form of the solution would be exactly the same regardless of the choice of refrigerant but if a refrigerant other than Freon-12 is used, extreme care must be taken to

substitute the properties and values for the desired refrigerant into the given equations.

The paper also presents a nomograph to facilitate the solving of the equations, and is designed for use with Freon-12 only.

It was found that the liquid line posed a slightly different consideration than the other two lines. The main consideration in the case of the liquid line is that the refrigerant enter the expansion device 100% in the liquid state. Most expansion devices depend upon this for proper operation. It is seen that the available pressure drop for the liquid line will depend upon the condensing temperature and the degree of subcooling. The most economical size of pipe for this line will, therefore, be the smallest pipe that will allow only the available pressure drop and still insure 100% liquid at the expansion device.

In view of the large number of the possible variables treated, the solutions presented here will for the majority of cases replace an individual economic study. In other cases it will check the designer from resorting to extremes that would destine the system for uneconomical operation.

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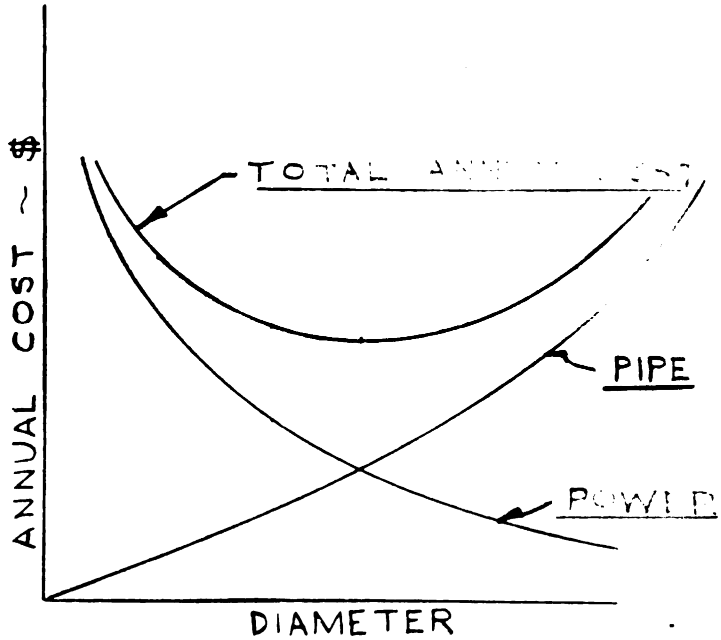
INTRODUCTION

The economic aspects of an engineering problem, although often of prime importance, are sometimes completely overlooked. Most engineering problems involve either a consideration or a comparison of costs. In attacking any engineering problem such as the design or selection of equipment to perform a certain task there may be an infinite number of selections or combinations that could do the job. However, of these alternatives, there can be found one which will do the job at the least total cost. It is this solution that the engineer must seek.

Before coming to a conclusion in an economic study, it is necessary to examine all the factors or variables which would influence our decision. It should be emphasized that the omission of any one factor may lead to an erroneous conclusion.

In the selection of piping for a refrigeration system, there are two primary factors which influence our choice. The first factor to be considered is the initial cost of pipe. The second factor is the operating cost for power required to drive the compressor. The magnitude of these two costs tend to oppose one another, that is, the larger the size of pipes used the greater will be their cost but the lower will be the cost of power required to run the compressor. On the other hand, the smaller the size of pipes used, the lower will be their cost but the higher will be the cost of power to drive the compressor. On the following page is shown a typical plot of the costs of power and pipe vs. the diameter of pipe used. From it, it can be seen that there will be some optimum point



at which the total cost for both piping and power will be a minimum. This paper presents an analytical solution for the optimum size of piping to be used in a refrigeration system. This paper treats the suction, discharge and liquid lines, although the considerations in the case of the liquid lines are found to be somewhat different. In the liquid line the most economical size of pipe is the smallest pipe that will still allow the refrigerant to be 100% in the liquid state at the entrance to the expansion valve.

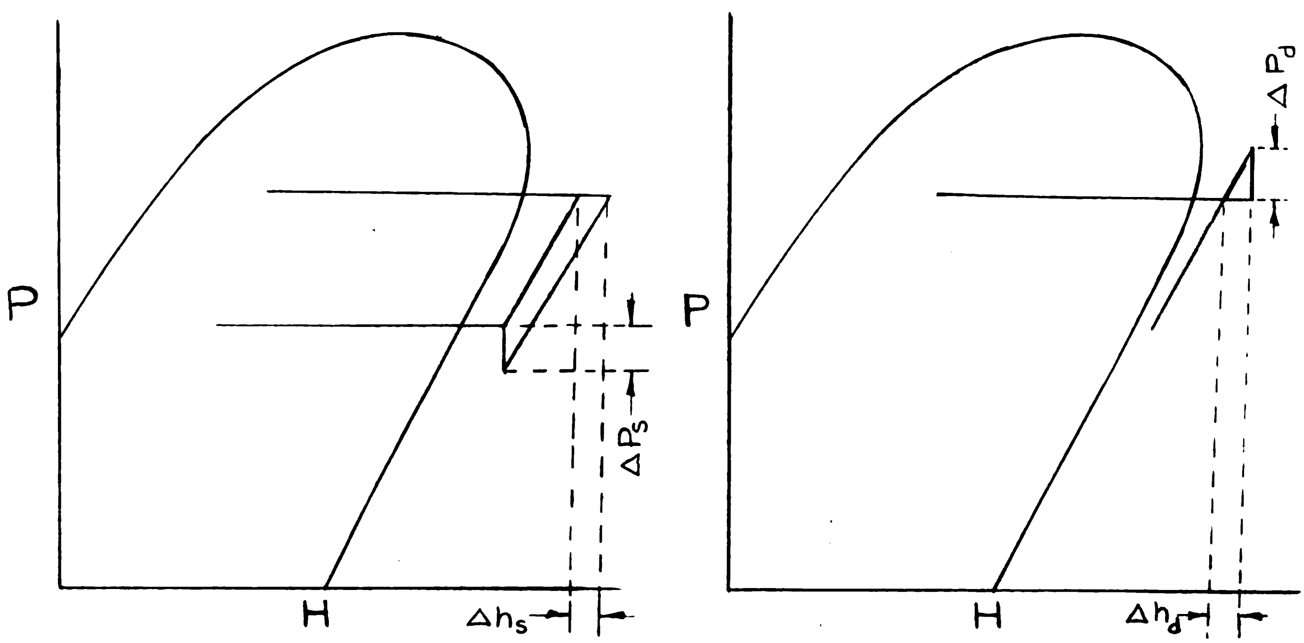
The approach to this problem is perfectly general, although the graphs and calculations made to facilitate selection may be used only with Freon-12. This is also the case for the nomographical solution of the equation for optimum pipe size which is presented to make pipe size selection an easy task.

The question of refrigeration pipe size selection has been investigated by many authors using several different approaches. In all cases, however, it was found that these authors rely only on experience and rules of thumb. In no instance could there be found an economic approach to the problem. This paper gives an analytic and accurate solution for the optimum size of pipe which will result in a minimum of total cost.

THE SUCTION AND DISCHARGE LINES

As previously stated the two prime factors influencing the selection of piping for the suction and discharge lines are the cost of piping and the cost of power to drive the compressor. These two factors are related to one another through the pressure drop. It is the fact that a change in pressure drop will alter these two costs in opposite directions that makes possible an optimum solution. Actually we could solve for an optimum pressure drop for a refrigeration pipe line but a pipe size is of much more significance to the designer.

Below are P-H diagrams showing the effect of pressure drop in both the suction and discharge lines.



Pressure drop in the suction line. Pressure drop in the discharge line.

Figure 1

ΔP_s and ΔP_d are taken to be the pressure drops in the suction and discharge lines respectively. It may be seen from these diagrams that the additional increments of enthalpy that the compressor must add because of the pressure drops are Δh_s and Δh_d.

We will let m_s and m_d represent the theoretical B.t.u. per pound per p.s.i. of pressure drop that must be added by the compressor. Hence,

$$m_s = \frac{\Delta h_s}{\Delta P_s} \qquad ; \qquad m_d = \frac{\Delta h_d}{\Delta P_d}$$

<u>Determination of m_s and m_d for the suction and discharge lines.</u>

It should be noted that m represents the slope of an isentropic line on a P-H diagram.

From the Second Law of Thermodynamics we may write

$$dh = Tds + vdp$$

Along an isentropic line ds = 0 so that:

$$\left(\frac{\partial h}{\partial P}\right)_S = v$$

From the above relation it can be seen that the slope of an isentropic line on a P-H diagram is equal to the specific volume.

But

$$m = \left(\frac{\partial h}{\partial P}\right)_s$$

$$m = v\,\frac{FT^3}{\#_m} = \frac{144v}{J}\;\frac{\frac{BTU}{\#_m}}{\frac{\#_f}{IN^2}}$$

The specific volume of the refrigerant, and hence m, was found to vary with the pressure and degree of superheat. The values of m are plotted for the suction and discharge lines on pages 6 and 7, respectively.

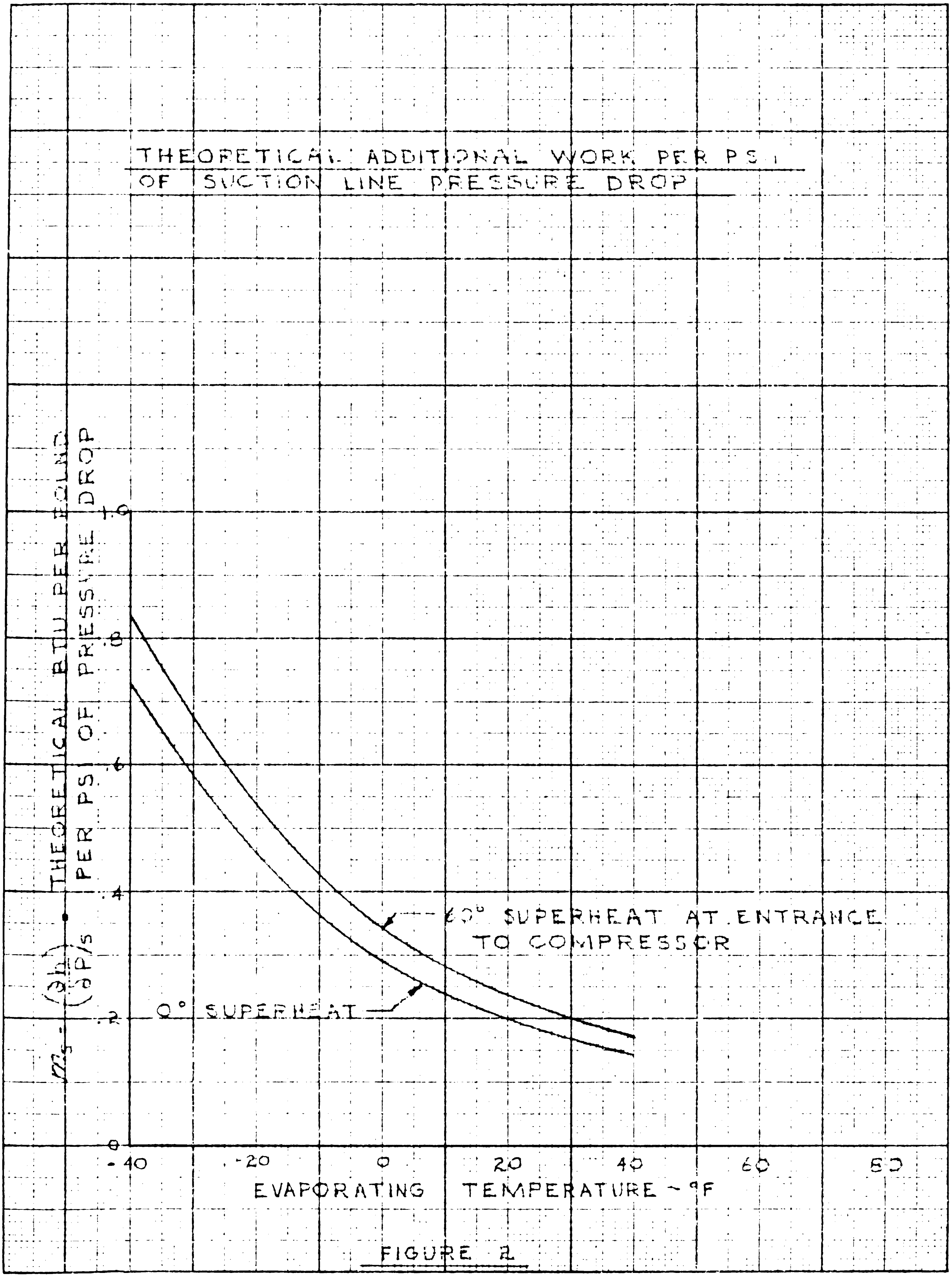

THEORETICAL ADDITIONAL WORK PER PSI
OF SUCTION LINE PRESSURE DROP
THEORETICAL BTU PER POUND
PER PSI OF PRESSURE DROP
1.0
.8
.6
.4
.2
0
-40
-20
0
20
40
60
80
EVAPORATING TEMPERATURE ~ °F
60° SUPERHEAT AT ENTRANCE TO COMPRESSOR
0° SUPERHEAT

FIGURE 2

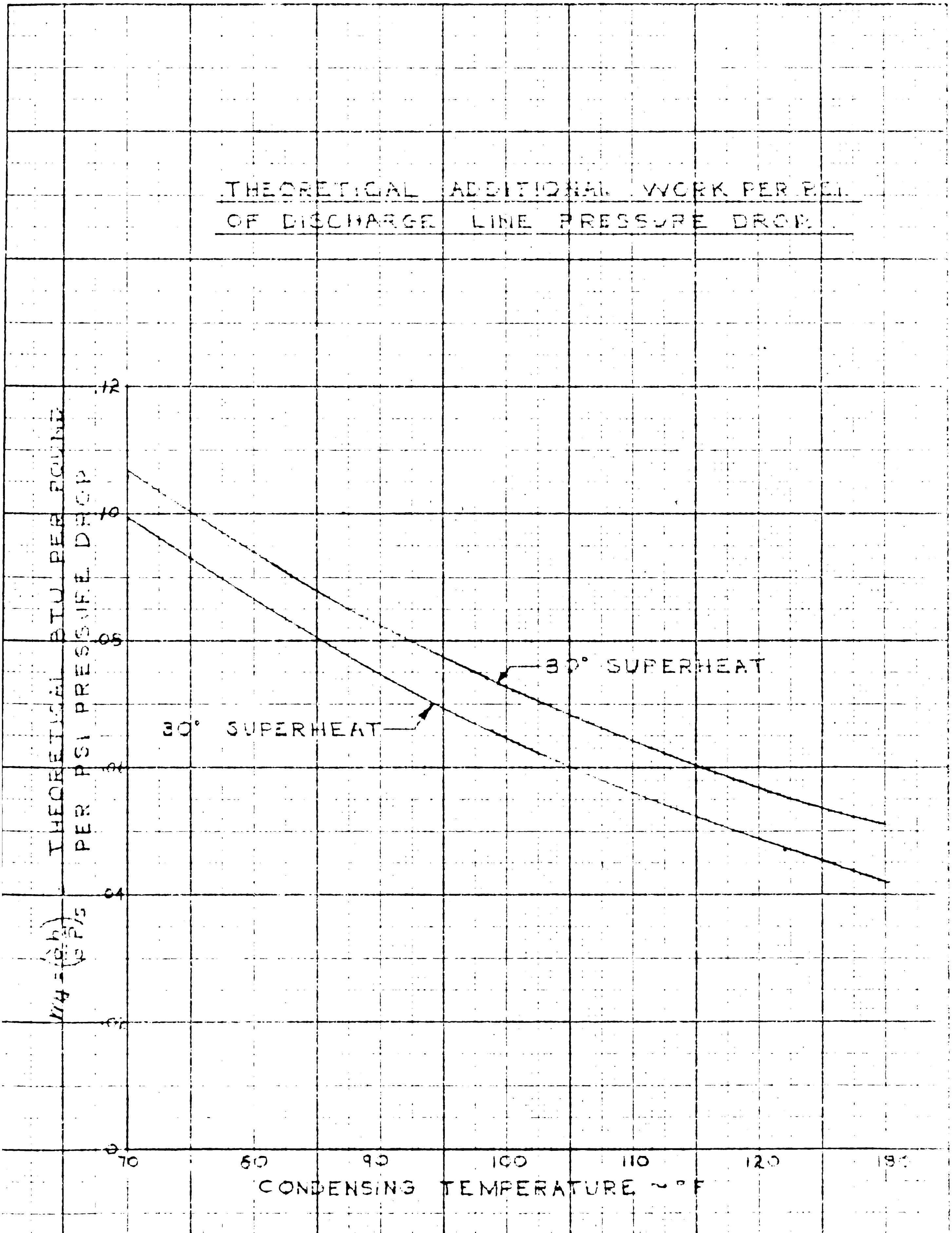

THEORETICAL ADDITIONAL WORK PER PSI
OF DISCHARGE LINE PRESSURE DROP
THEORETICAL BTU PER POUND
PER PSI PRESSURE DROP
.12
.10
.08
.06
.04
.02
0
30° SUPERHEAT
30° SUPERHEAT
70
80
90
100
110
120
130
CONDENSING TEMPERATURE ~ °F

FIGURE 3.

We will now determine the annual operating cost, Y.

Let:

b = annual cost to compress refrigerant from evaporator pressure to condenser pressure

C = increment in annual operating cost due to frictional pressure drop

$$Y = b + C$$

h = hours of operation per year

R = cost of electricity - cents per k.w.h.

η_{is} = isentropic compression efficiency - %

η_M = mechanical efficiency - %

η_{mot} = motor efficiency - %

w = refrigerant flow rate - lbs./sec.

The theoretical B.t.u. per second of power that must be added due to frictional pressure drop is H_T.

$$H_T = wm \Delta P$$

The actual B.t.u. per second of power will therefore be H_A.

$$H_A = \frac{wm \Delta P}{\eta_{is}\eta_M} = \frac{778}{550} \frac{wm \Delta P}{\eta_{is}\eta_M} \text{ H.P.}$$

The Kilowatts of power required will be:

$$Q = \frac{778}{550}(0.746) \frac{wm \Delta P}{\eta_{is}\eta_M\eta_{mot}}$$

$$C = \frac{QhR}{100} = hR \frac{778}{550} \left(\frac{.746}{100}\right) \frac{wm \Delta P}{\eta_{is}\eta_M\eta_{mot}}$$

$$C = .01058 \frac{hRwm \Delta P}{\eta_{is}\eta_M\eta_{mot}}$$

$$Y = b + .01058 \frac{hRwm \Delta P}{\eta_{is}\eta_M\eta_{mot}} \tag{1}$$

We will let:

$$U = \frac{.01058hRm}{\eta_{is}\eta_M\eta_{mot}}$$

Then $$Y = b + Uw\Delta P \tag{2}$$

The next step in the solution is to determine an expression for the cost of piping. In order to treat this cost together with the operating cost, it is advisable that it also be put on the annual cost basis. Recognizing the time value of money this may be done by multiplying the first cost of the piping by a capital recovery factor. This capital recovery factor will depend upon the current rate of interest and the expected life of the system. A table of capital recovery factors is given in the Appendix. The life of the system may be taken as either the life of the piping itself or that of the system as a unit if that is shorter. After its useful life, the piping may have some salvage value as scrap.

The annual cost of capital recovery may now be computed as follows. Let:

A = Annual cost of capital recovery - \$

L = Length of pipe - feet

B = Cost per foot per inch of diameter (installed) - \$

S = Salvage value per inch of diameter after n years - \$

n = Life of system - years

i = Interest rate - percent

D = Diameter of pipe - inches

K = Capital recovery factor $= \dfrac{i}{(i + 1)^n - 1} + i$

Then:

$$A = LD \left[(B - S) \left\{ \frac{i}{(i + 1)^n - 1} + i \right\} + Si \right]$$

$$A = LD \left[K (B - S) + iS \right]$$

Let us assume that the salvage value can be expressed as a percentage X of B.

$$S = XB$$

$$A = LD \, [KB - KXB + iXB]$$

$$A = LDB \, [K - KX + iX]$$

Let:

$$Z = [K - KX - iX]$$

$$A = LDBZ \qquad (3)$$

The total annual cost T will now be the sum of the operating cost plus the annual cost of capital recovery.

$$T = Y + A \qquad (4)$$

$$T = b + Uw \Delta P + LDBZ \qquad (5)$$

At this point it would be desirable to derive an expression for the frictional pressure drop in a pipe in terms of the variables involved. We may start with the Fanning equation which states:(1)

$$\Delta P = \frac{4fL\rho V^2}{2gD} \qquad (6)$$

From the continuity equation it is seen that:

$$V = \frac{w}{A\rho} = \frac{4w}{\rho \pi D^2}$$

$$V^2 = \frac{16w^2}{\rho^2 \pi^2 D^4}$$

For smooth pipes or tubing such as copper, brass, glass, etc., the following empirical relation has been proposed by Stoever.[1]

$$f = \frac{0.0653}{(Re)^{0.228}} \qquad \text{at Re from 4,000 to 1,000,000} \qquad (7)$$

It should be noted that this is the range of Reynolds numbers which will be encountered in refrigeration piping.

$$Re = \frac{DV\rho}{\mu}$$

$$f = \frac{0.0653\,\mu^{.228}}{D^{.228}V^{.228}\rho^{.228}}$$

Combining these relations we arrive at our final expression for the frictional pressure drop.

$$\Delta P = \frac{0.2}{g}\,\frac{Lw^{1.77}\mu^{.228}}{\rho\, D^{4.77}}$$

Putting ΔP in lbs/in^2 and D in inches, we get:

$$\Delta P = 6.02\,\frac{Lw^{1.77}\mu^{.228}}{\rho\, D^{4.77}} \qquad (8)$$

Substituting this in Equation 5, we get:

$$T = b + 6.02\,\frac{ULw^{2.77}\mu^{.228}}{\rho\, D^{4.77}} + LZBD \qquad (9)$$

It should be noted that in the expression for Y and T, the increment in annual cost C is included for only one line. The suction and discharge lines must be solved for separately, and in solving for one, the incremental cost in the other will be treated as a constant. It, together with b, will drop out when we differentiate the total cost with respect to D.

For minimum total cost:

$$\frac{dT}{dD} = 0$$

$$\frac{dT}{dD} = 0 = -4.77 \times 6.02\ UL\ \frac{\mu^{.228}}{\rho}\ \frac{w^{2.77}}{D^{5.77}} + LZB$$

$$D^{5.77} = 28.7\ \frac{U}{Z}\ \frac{\mu^{.228}}{\rho}\ \frac{w^{2.77}}{B}$$

Let $e = 28.7\ \frac{U}{Z}\ \frac{\mu^{.228}}{\rho} = \frac{.304hRm}{[K - KX - iX][\eta_{is}\eta_{M}\eta_{mot}]} \left(\frac{\mu^{.228}}{\rho}\right)$

We may now write our expression for the most economical pipe size diameter.

$$D^{5.77} = \frac{ew^{2.77}}{B} \tag{10}$$

THE NOMOGRAPHIC SOLUTION TO THE EQUATION FOR THE MOST ECONOMICAL PIPE SIZE DIAMETER

$$D^{5.77} = \frac{ew^{2.77}}{B} \qquad (10)$$

$$e \;\frac{.304hRm}{[K - KX - iX][\eta_{is} \times \eta_{M} \times \eta_{mot}]} \left(\frac{\mu^{.228}}{\rho}\right)$$

Let B_1 = Cost per foot of pipe.

This is more convenient for our final graph since the cost data is given by the manufacturers in this form.

Equation (1) will then be:

$$D^{4.77} = \frac{ew^{2.77}}{B_1} \qquad (11)$$

This equation may be written:

$$4.77 \log D + \log B_1 = \log e + 2.77 \log w \qquad (12)$$

Let $\log K = 4.77 \log D + \log B_1$

Then $\log K = 2.77 \log w + \log e$

Or $\log e + 2.77 \log w = \log K$

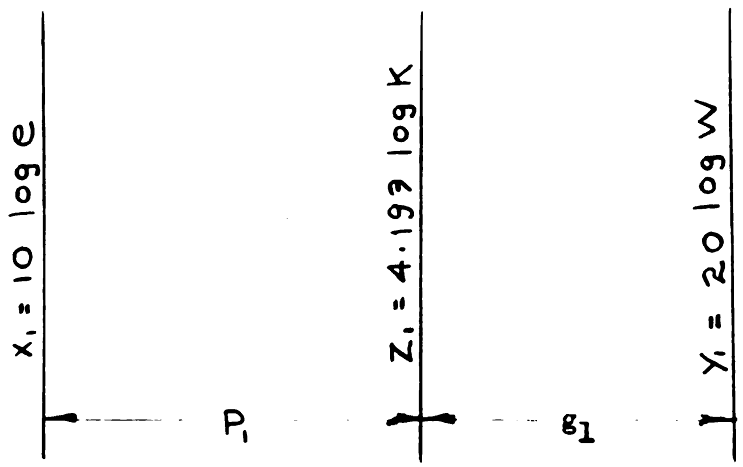
$x_1 = m_1 \log e \quad ; \quad Y_1 = m_2(2.77 \log w) \quad ; \quad Z_1 = m_3 \log K$

Let: $m_1 = 10$

$$m_2 = \frac{20}{2.77} = 7.220$$

$$m_3 = \frac{m_1 m_2}{m_1 + m_2} = \frac{7.220 \times 10}{17.220} = 4.193$$

$x_1 = 10 \log e \quad ; \quad y_1 = 20 \log w \quad ; \quad Z_1 = 4.193 \log K$



$$\frac{P_1}{g_1} = \frac{m_1}{m_2} = \frac{10}{7.22}$$

For $P_1 + g_1 = 10''$

$$P_1 = \frac{10}{17.22} \times 10 = 5.807''$$

$$g_1 = \frac{7.22}{17.22} \times 10 = 4.193''$$

$$4.77 \log D + \log B_1 = \log K$$

$x_2 = m_4(4.77 \log D)$; $Y_2 = m_5 \log B_1$; $Z_2 = Z_1 = 4.193 \log K$

Scale K will be used as the common line or factor in the two equations $Z_2 = Z_1$. This follows from the original equation in which we fixed K:

$$4.77 \log D + \log B_1 = \log K = \log e + 2.77 \log w$$

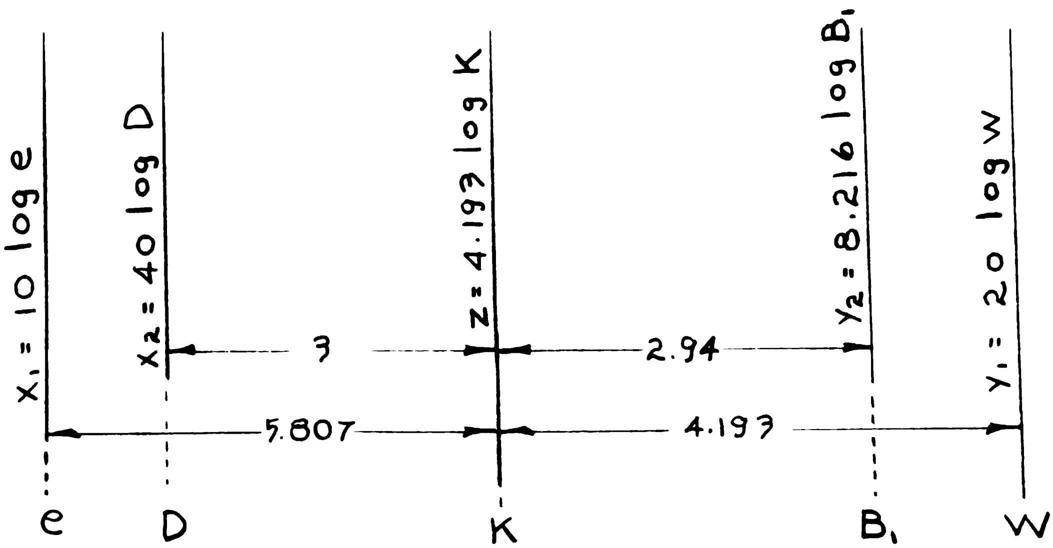
Let $m_4 = \dfrac{40}{4.77} = 8.386$

$$m_3 = \frac{m_4 m_5}{m_4 + m_5} \quad ; \quad 4.193 = \frac{8.386\, m_5}{8.386 + m_5} \quad ; \quad m_5 = 8.216$$

$x_2 = 40 \log D$ $\quad Y_2 = 8.216 \log B_1$ $\quad Z_2 = 4.193 \log K$

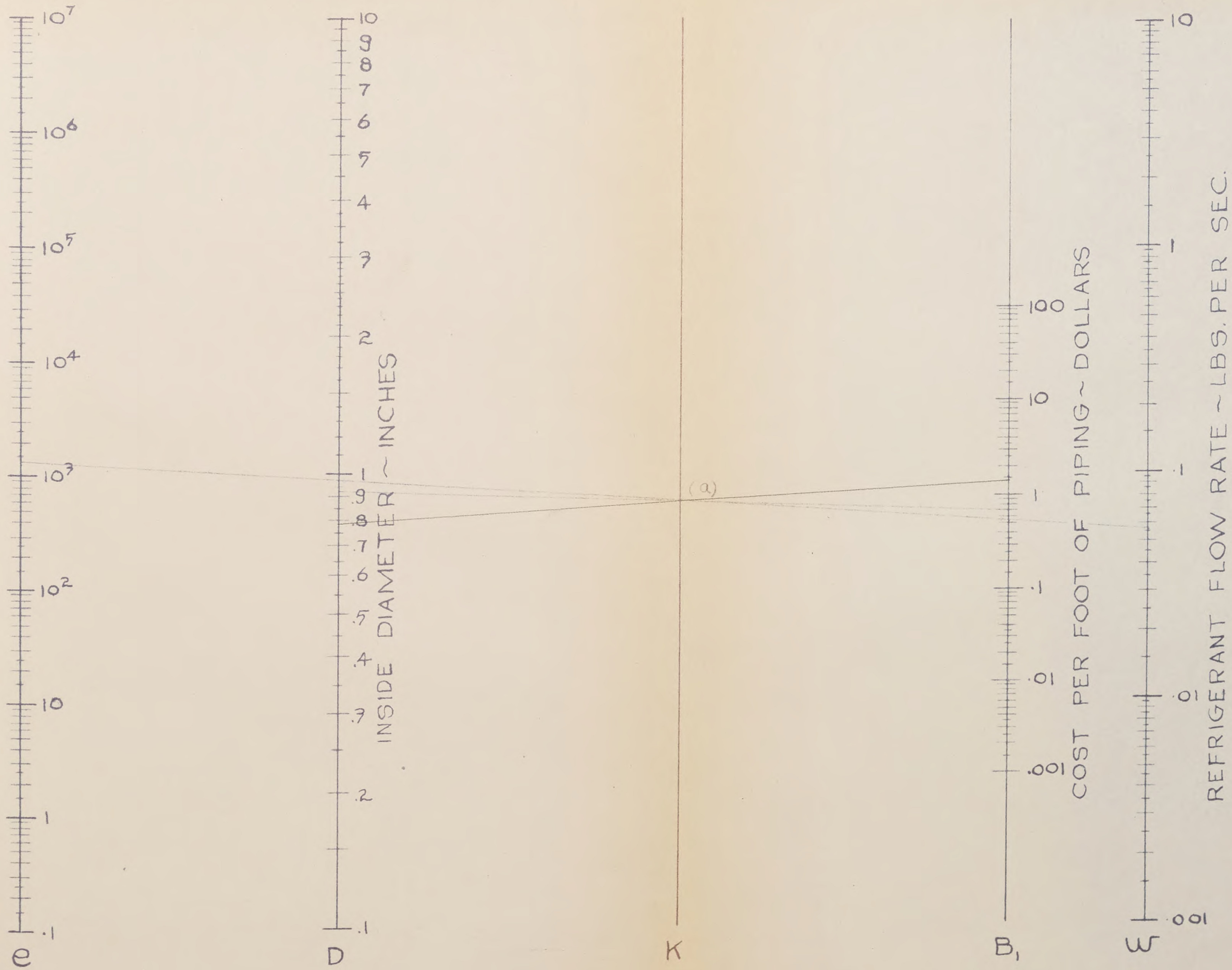
$$\frac{P_2}{g_2} = \frac{m_4}{m_5} = \frac{8.386}{8.216} = \frac{3}{2.94}$$

$$P_2 = 3" \quad ; \quad g_2 = 2.94"$$



The actual lengths of one cycle on the various scales will be in direct proportion to the coefficients of the log terms. No scale need be put on K since it is used only as a pivot line. The range and vertical positioning of three of the four scales is completely arbitrary, but once this is done the fourth must be done by a numerical computation of a point on the particular scale.

$$D^{4.77} = \frac{e\,w^{2.77}}{B_1}$$



MOST ECONOMICAL PIPE SIZE FOR SUCTION AND DISCHARGE LINES

FIGURE 4

The use of the equations and nomograph will now be demonstrated by the solution of a typical problem for sizing a suction line.

Assume

h = 2,000 hours of operation per year

R = 2 cents per k.w.h. for electricity

n = 20 years of life for system

i = .05 interest rate

X = .02 salvage value at end of n years

η_{is} = .75 isentropic compression efficiency

η_M = .80 mechanical efficiency

η_{mot} = .82 motor efficiency

Evaporation temperature = 10°F and superheat = 60°F at entrance to the compressor.

Determine

The most economical size of pipe to use for the suction line on a 1-ton system (Freon-12).

From Table of Viscosities $\mu = 768 \times 10^{-8} \frac{\#}{\text{ft-sec}}$

From Freon-12 Mollier Diagram $\rho = \frac{1}{v} = \frac{1}{1.55} = .645 \frac{\#}{\text{ft}^3}$

From Graph $m_s = .4$

From Table of Capital Recovery Factors $K = 0.08024$

$$e = \frac{.304hRm}{[K - KX - iX][\eta_{is}\eta_M\eta_{mot}]}\left(\frac{\mu^{.228}}{\rho}\right) =$$

$$\frac{.304\,(2{,}000)(2)(.4)}{[.08024 - .08024(.02) - (.05)(.02)][.75 \times .80 \times .82]}\ \frac{6.80}{100 \times .645}$$

$$e = 1.34 \times 10^3$$

Assuming a refrigeration effect of 60 $\frac{\text{B.t.u.}}{\#}$ the mass flow for 1-ton refrigeration will be:

$$w = \frac{200 \frac{\text{B.t.u.}}{\text{Min}}}{60 \frac{\text{B.t.u.}}{\#}} \times \frac{1}{60} \frac{\text{Min}}{\text{Sec}} = .0556 \frac{\#}{\text{Sec}}$$

By connecting the calculated values of w and e we obtain point (a) on the K scale. Using this as our pivot point and our price data as a guide we draw a line through (a) so that the diameter is as close as possible to its corresponding price per foot.

For this case we find the most economical diameter to be 7/8 inch.

NOTE: To be theoretically correct the price per foot should include the exact cost of fittings and valves since their price will vary with the diameter. The equivalent length of the line should then be calculated and the cost per equivalent foot of pipe found. This should be done using the above solution as a first approximation. The final diameter will vary slightly from this owing mostly to the approximation made for the average price including fittings as given in the footnote on page 19.

TABLE I

SAMPLE PIPE COST DATA

O.D. Size Inches	Wall Thickness Inches	Price Per Foot $	Average Price[a]* Per Foot Including Fittings $	Installation[b] Cost Per Foot $	Total Cost per Foot $
3/8	.030	.218	.349	.090	.439
1/2	.035	.319	.510	.105	.615
5/8	.040	.448	.717	.180	.897
3/4	.042	.547	.875	.300	1.175
7/8	.045	.659	1.054	.420	1.474
1 1/8	.050	.909	1.454	.510	1.964
1 3/8	.055	1.227	1.963	.780	2.743
1 5/8	.060	1.548	2.477	.900	3.377
2 1/8	.070	2.377	3.803	1.010	4.813
2 5/8	.080	3.368	5.390	1.350	6.740
3 1/8	.090	4.320	6.912	1.560	8.472
3 5/8	.010	5.696	9.114	1.800	10.914
4 1/8	.110	7.143	11.429	1.910	13.339

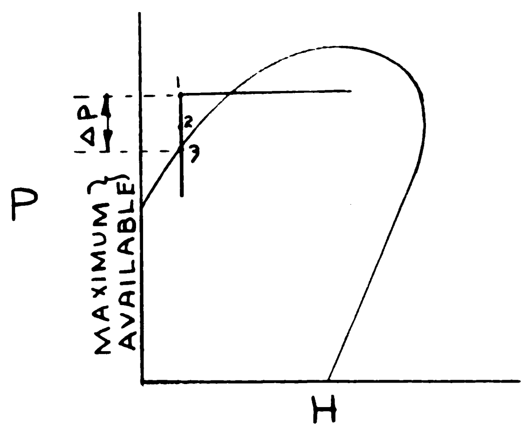
* a This assumes that the fittings will amount to 60% of the pipe cost.

b This is based on labor at $3 per hour and hours per foot as recommended by reference, "How to Make your Job Estimates Detailed and Accurate", 1955-56 REFRIGERATION AND AIR CONDITIONING Catalog of Catalogs by Commercial Refrigeration and Air Conditioning Magazine, pp. 154-155.

THE LIQUID LINE

The factors influencing the selection of a liquid line size are somewhat different from those for a suction or a discharge line. The prime consideration in selecting a liquid line is that the refrigerant enters the expansion device 100% in the liquid state. Most expansion devices depend upon this for proper performance. If an expansion device is supplied with a liquid vapor mixture, the capacity of the device is greatly reduced. This would disrupt the operation of the entire system. To prevent this, the pressure directly ahead of the expansion device must exceed the saturation pressure corresponding to its temperature. It is seen therefore that the available pressure drop for the liquid line will depend directly upon the condensing temperature and the degree of subcooling.

PRESSURE DROP IN THE LIQUID LINE



1. Refrigerant enters liquid line.
2. Refrigerant enters expansion device.
3. Refrigerant begins to change to vapor phase.

Figure 5

The available pressure drop will consist of the following parts:

$$\Delta P_{\text{Available}} = \Delta P_{\text{Friction}} + \Delta P_{\text{Diff. in Elevation}} + \Delta P_{\text{Safety Factor}} \qquad (13)$$

The ΔP for friction and safety factor will be positive, but the change in pressure due to the difference in elevation may be either positive or negative depending upon the relative positions of the condenser and the expansion device. If the expansion device is located above the condensing unit the ΔP will be positive, and if the expansion device is located below the condensing unit the ΔP will be considered negative. This ΔP due to the difference in elevation is often neglected and may be the cause of a serious error in design. This is especially true if there is a very large vertical lift preceding the expansion device. The Freons as a group are heavy refrigerants with Freon-12 having a pressure drop of approximately .57 p.s.i. per foot of static liquid head. Values for other refrigerants are given in the Appendix.

Equation (13) may be written in the form:

$$\underset{\text{(or Net)}}{\Delta P_{\text{Friction}}} = (\Delta P_{\text{Available}} - \Delta P_{\text{S.F.}}) - \Delta P_{\text{Diff. in Elev.}} \qquad (14)$$

It is recommended that the term ($\Delta P_{\text{Avail.}} - \Delta P_{\text{S.F.}}$) be taken as $.9\,\Delta P_{\text{Avail.}}$, and should at all times be at least 3 p.s.i. less than the available pressure drop. Values for the available pressure drop for Freon-12 are shown for various refrigerant conditions by the graph on page 24. Subtracting this from the first term, we arrive at the ΔP which may be allowed for the frictional drop.

$$\Delta P_{\text{Net}} = 6.02\,\frac{Lw^{1.77}}{D^{4.77}}\;\frac{\mu^{.228}}{\rho} \cdot (\text{Equ. } 8)$$

It is shown in the Appendix that the plots of density and viscosity vs. temperature are very nearly linear. The term $\frac{\mu^{.228}}{\rho}$ may, therefore, be written in the form:

$$\frac{\mu^{.228}}{\rho} = \frac{(.000672\,[.580 - .0006T])^{.228}}{(149.15 - .1261T)} \qquad (15)$$

This term was found to be very nearly a constant at common liquid line temperatures.

$$\frac{\mu^{.228}}{\rho} = 1.70 \times 10^{-3}$$

This brings our equation for $\Delta P_{Friction}$ into its final form:

$$\Delta P \frac{\#}{in.^2 - 100'} = 1.025 \frac{W^{1.77}}{D^{4.77}} \qquad (16)$$

A graph of this equation is shown on page 26.

In this analysis it is assumed that the state of the liquid refrigerant is such that its pressure is high enough above the saturation pressure to allow for the difference in elevation and some frictional pressure drop. This however may not always be the case. As was previously emphasized, a large vertical lift may by itself exceed the allowable pressure drop. In this case two alternatives are possible, both of which increase the available pressure drop. The first is to raise the discharge pressure of the condenser. The second method is to use a heat exchanger to further subcool the liquid refrigerant. This second method seems by far the better of the two alternatives and is becoming common in modern installations. A liquid suction heat exchanger will not only help to subcool the liquid refrigerant but will further superheat the

the suction line vapor which is desirable from safety standpoint. On the other hand, increasing the discharge pressure of the compressor will increase the operating costs and is to be avoided if possible.

AVAILABLE PRESSURE DROP VS CONDENSING TEMPERATURE
FOR LIQUID LINE USING FREON-12

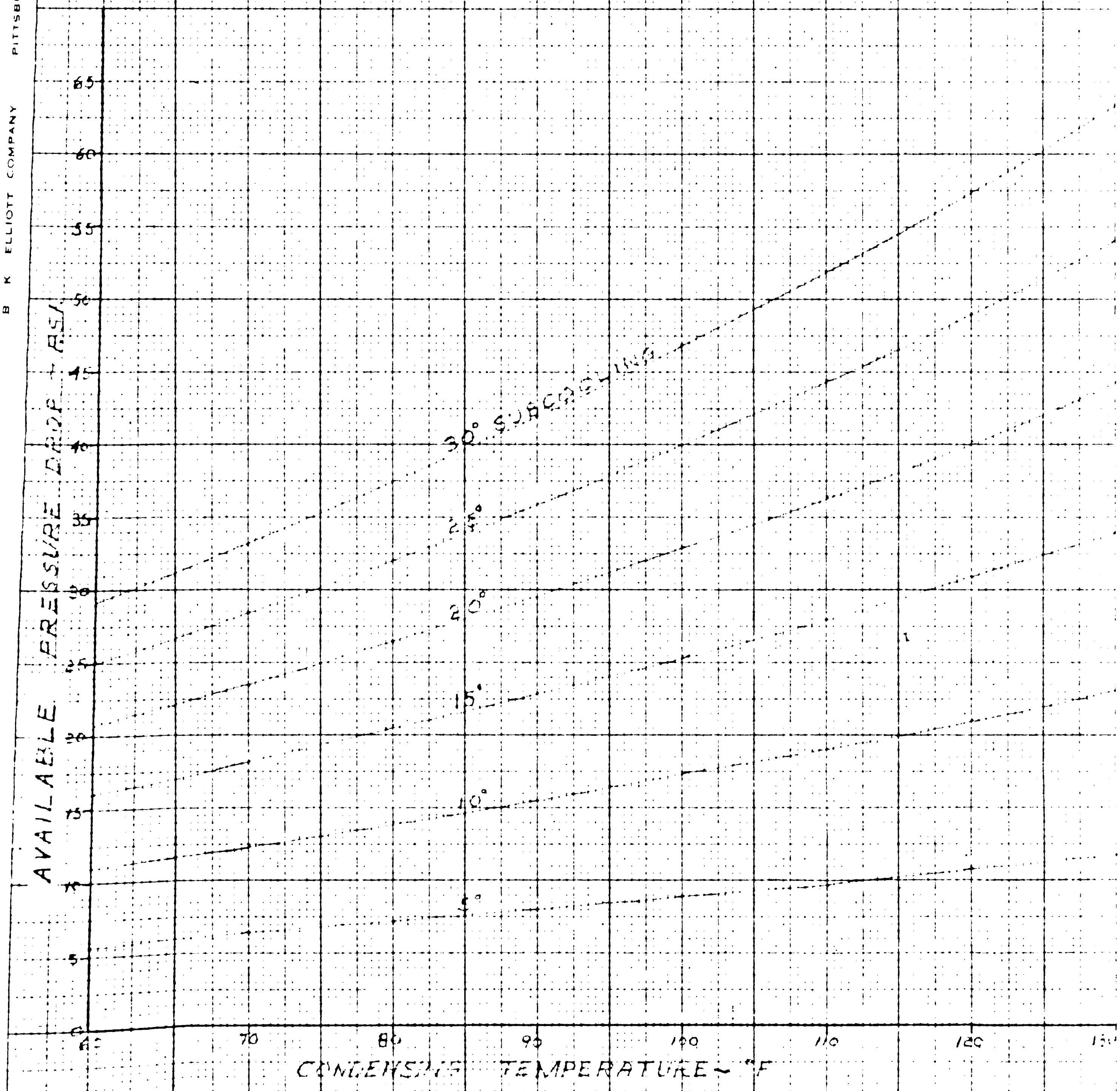


FIGURE 6

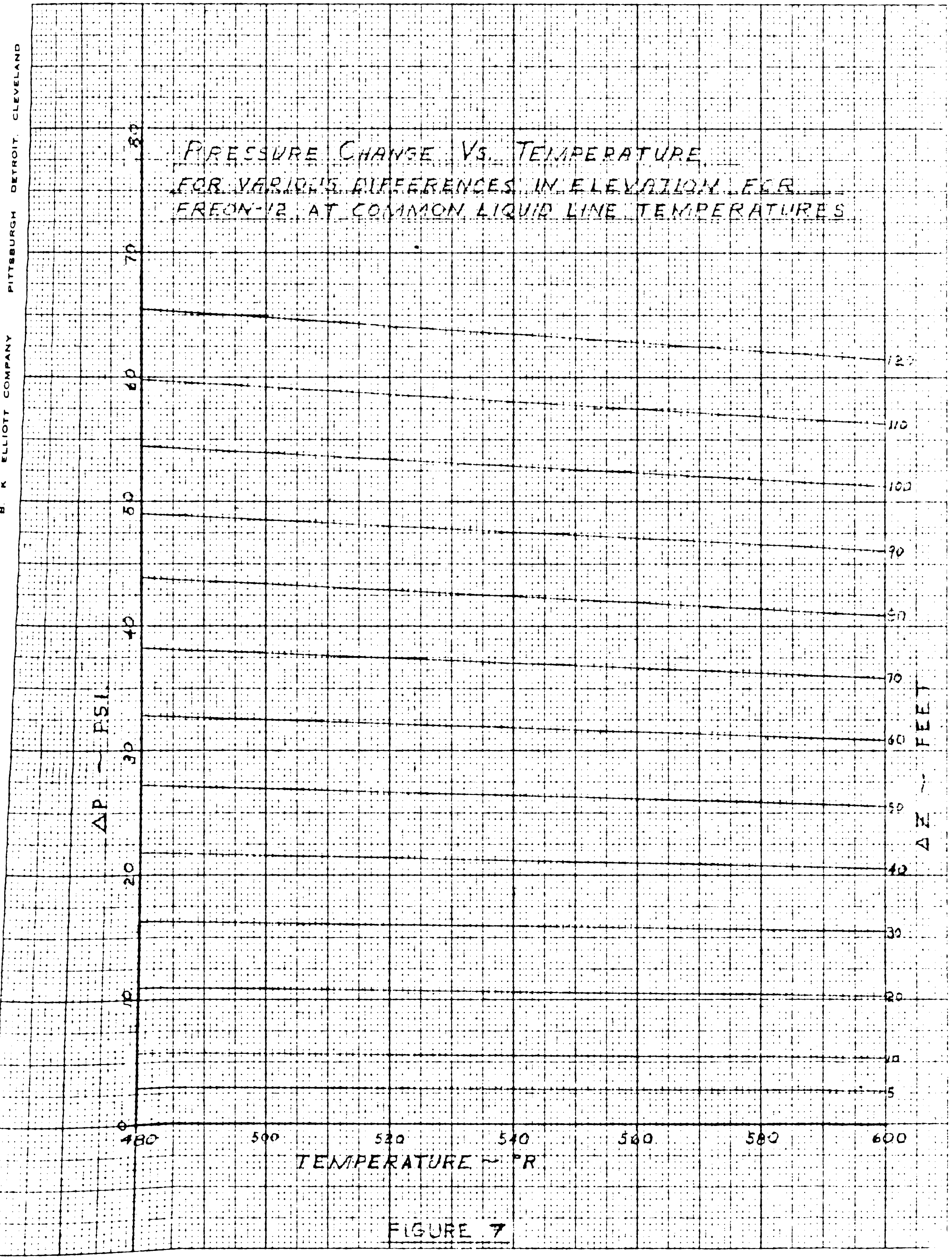
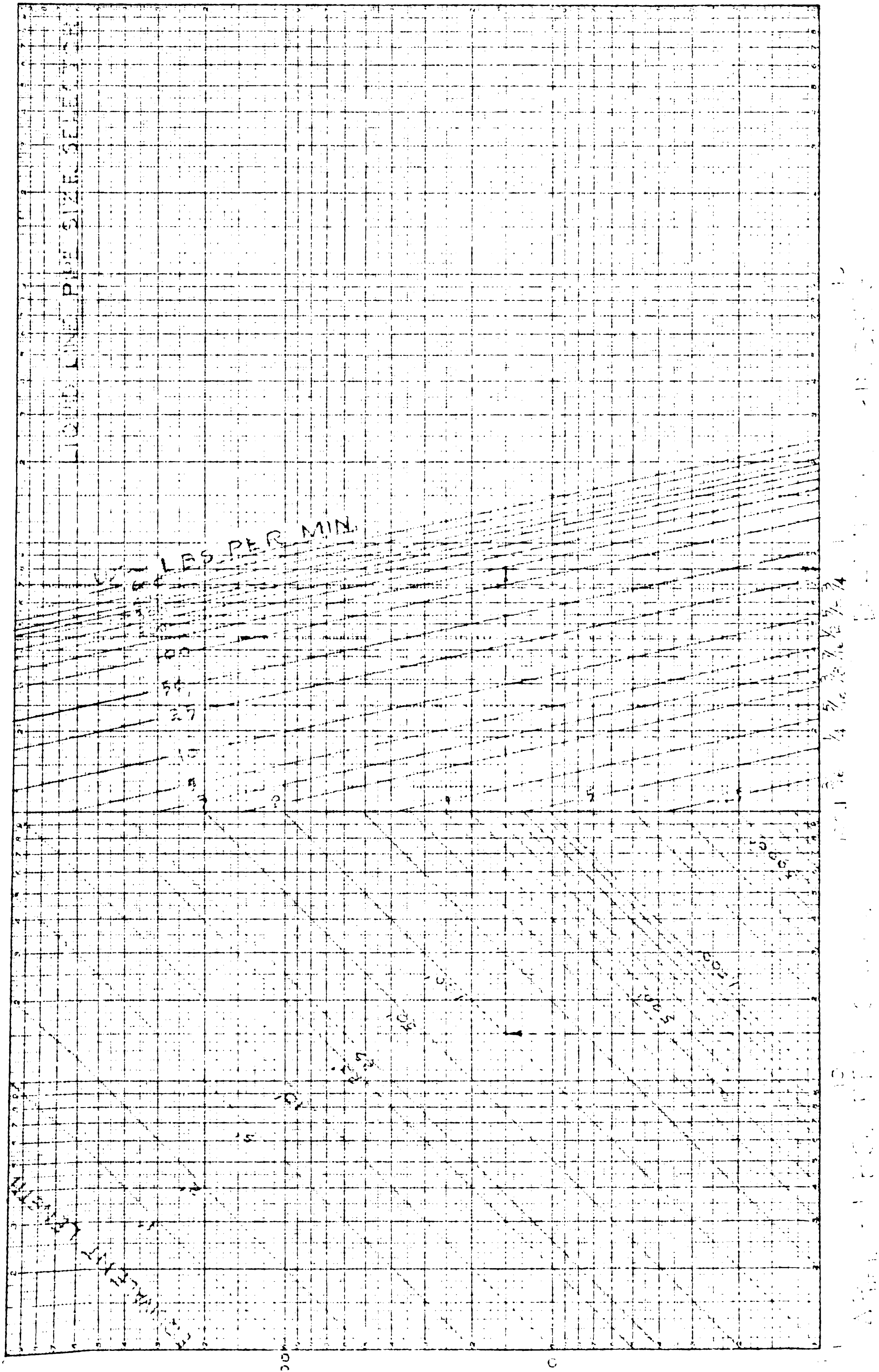

PRESSURE CHANGE VS. TEMPERATURE
FOR VARIOUS DIFFERENCES IN ELEVATION FOR FREON-12 AT COMMON LIQUID LINE TEMPERATURES
ΔP ~ PSI
ΔZ ~ FEET
TEMPERATURE ~ °R
0
10
20
30
40
50
60
70
80
480
500
520
540
560
580
600
5
10
20
30
40
50
60
70
80
90
100
110
120

FIGURE 7


LBS. PER MIN.

DISCUSSION AND CONCLUSION

We have derived in this paper an expression for the most economical size of pipe to be used in a refrigeration line in terms of the variables of our system. This equation was found to be:

$$D^{4.77} = \frac{ew^{2.77}}{B_1} \qquad (1)$$

where $e = \dfrac{.304hRm}{[K - KX - iX][\eta_{is}\eta_M\eta_{mot}]} \quad \dfrac{\mu^{.228}}{\rho}$

The accuracy to which we are able to select the most economical size of pipe will depend naturally upon how closely we are able to approximate each of the factors in the equation. It is of interest to see how the misjudgment of them might affect our choice of a diameter.

First of all it would be necessary to calculate the required mass rate of flow w required for the desired tonnage at the desired evaporator temperature. This might be done very accurately by sketching the cycle on a P-H chart and determining the refrigeration effect, R.E., in $\frac{\text{B.t.u.}}{\#}$.

$$w\frac{\#}{\text{sec}} = \text{Tonnage} \times \frac{\frac{200}{60}}{\text{RE}}$$

An approximation which is often made is using a refrigeration effect of 60 $\frac{\text{B.t.u.}}{\#}$ for Freon -12 and 80 $\frac{\text{B.t.u.}}{\#}$ for Freon -22.(3)

Let us assume that w was approximated to be 5% too large. This would result in an error of $(1.05)^{\frac{2.77}{4.77}} - 1 = .031$ or 3.1%. Similar calculations for the other variables in Equation (1) are given in a table below.

Var.	% Error of Given Variable	Reflected Error in Diameter
w	5%	3.1%
e	10%	2%
B_1	10%	2%
All of Above		7.5%*

*Assuming none are compensating.

The cost per foot of pipe may be determined from manufacturers price lists. It should include the cost of installation and all fittings and valves because these will vary with the pipe size. This would, therefore, have to be done after first getting a rough approximation with the use of the nomograph.

From the nomograph it may be seen that the range of mass flow rates is from .001 to 10 lbs./sec. This corresponds approximately from .02 to 180 tons. The range of diameters covered is from .1 to 10 inches. It should be emphasized in high tonnage systems requiring large diameter pipe, extreme care is necessary. The price of fittings and valves used in these systems is extremely high. In such cases it would probably be advisable to make a complete economic study.

In the solution for the liquid line it was stated that the most economical size of pipe is the smallest size that would still insure 100% liquid at the entrance to the expansion device. It might be argued, however, that the available pressure drop might be increased by raising the condensing temperature. This however we are assuming to be fixed at the optimum condensing temperature and this has been shown by several authors to be a function of the cost of water for cooling the condenser and the cost of electricity to run the compressor. A combination of studies of the liquid line size and optimum condensing temperature might be possible but is beyond the scope of this thesis.

It is admitted that in the past many authors have treated the problem of sizing refrigeration lines. Under typical conditions their solutions would probably prove to be quite adequate. This is so because in all cases they are based on arbitrary rules formed from past experience. Blindly following these rules, however, may lead the designer into serious trouble.

It is of interest to note that the diameter found in the sample solution for the suction line is the same as would be obtained using the methods of reference 1 and 3. However, over the past twenty years, the price of electricity has remained about constant whereas pipe costs have more than doubled. This indicates that using these methods in the past would have resulted in undersized pipe. The reader may conclude that, if we are to continue to select pipe by these arbitrary methods, the time will soon be here when we must revise our rules of thumb. We will no longer be able to use rules formulated under an entirely different set of economic conditions.

The solution for the most economical size of pipe to be used in the three refrigeration lines gives adequate treatment to all the factors which influence the selection. It will therefore give reliable results over a wide range. In conjunction with the nomograph its simplicity is also comparable with other methods.

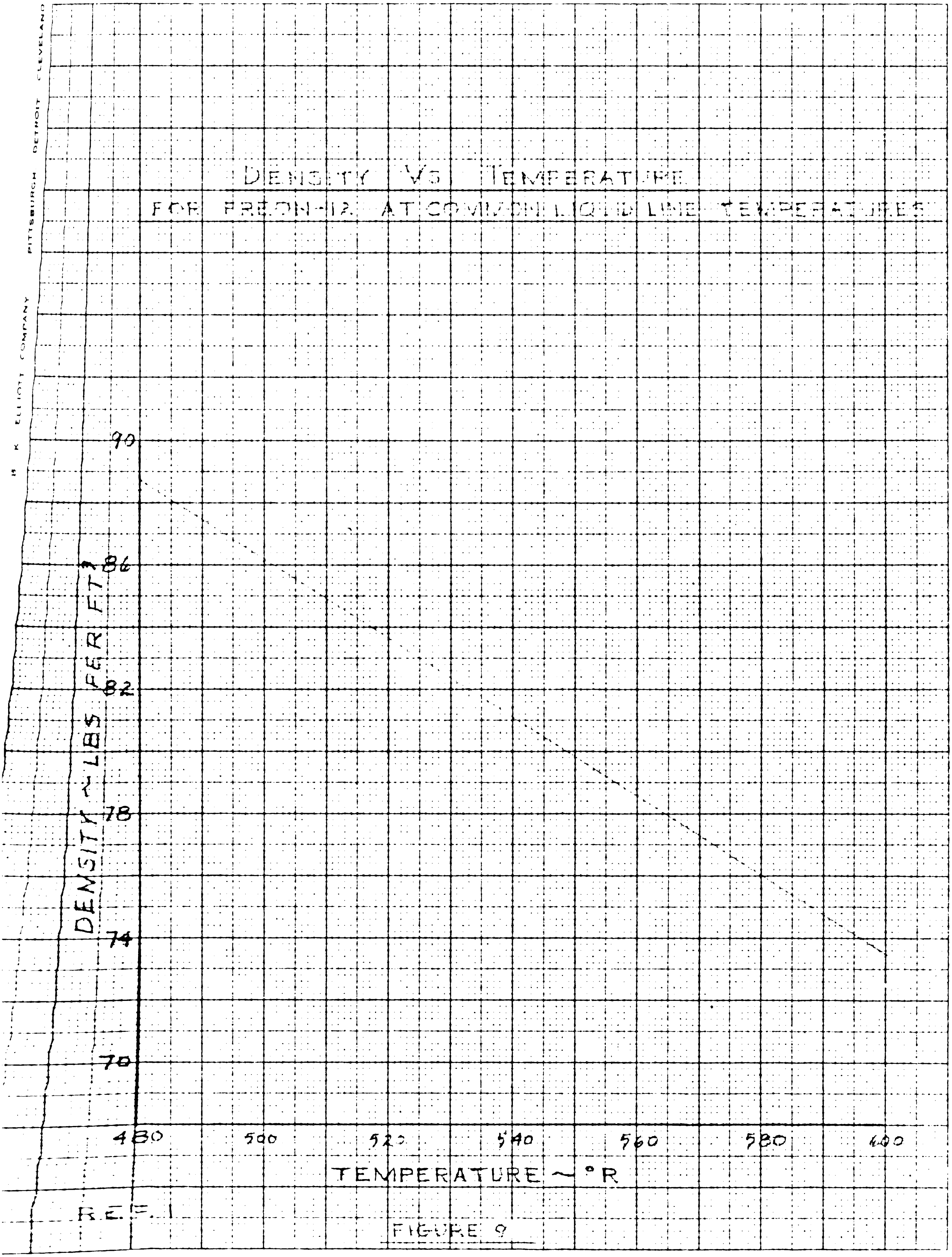
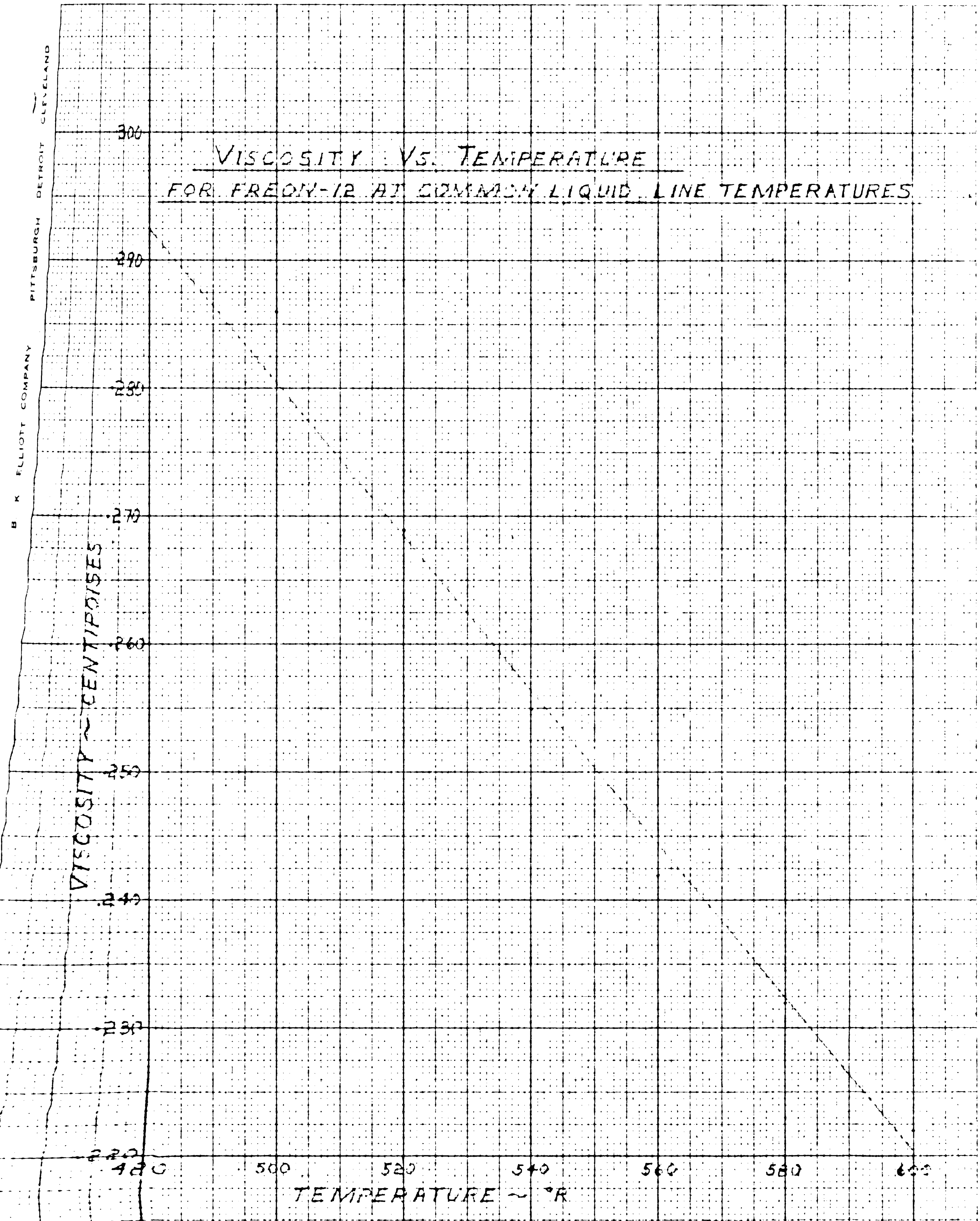

DENSITY VS TEMPERATURE
FOR FREON-12 AT COMMON LIQUID LINE TEMPERATURES
DENSITY ~ LBS PER FT³
90
86
82
78
74
70
480
500
520
540
560
580
600
TEMPERATURE ~ °R
REF. 1

FIGURE 9



REF. 1

FIGURE 10

DETERMINATION OF $\frac{\mu^{.228}}{\rho}$ FOR THE LIQUID LINE*

Density vs. Temperature

$$\rho = C_1 - m_1 T$$

$$\rho = 86.1 - \frac{86.1 - 76.02}{80} (T - 500)$$

$$\rho = 86.1 - .1261 (T + 500)$$

$$\rho = 86.1 - .1261T + 63.05$$

$$\rho = 149.15 - .1261T$$

Viscosity vs. Temperature

$$\mu = C_2 - m_2 T$$

$$\mu = .280 - \frac{.280 - .220}{100} (T - 500)$$

$$\mu = (.580 - .006T) \text{ Centipioses}$$

$$\mu = .000672 (.580 - .006T) \frac{\#}{\text{Ft-Sec}}$$

$$\frac{\mu^{.228}}{\rho} \approx 1.70 \times 10^{-3}$$

*The equations for the density and viscosity are put in the slope intercept form and the values of the slope and intercept for the equations for density and viscosity are obtained from Figures 9 and 10 respectively.

TABLE II (1)

PRESSURE DROP PER FOOT OF STATIC LIQUID HEAD
FOR VARIOUS REFRIGERANTS AT COMMON
LIQUID LINE TEMPERATURES*

Freon-11	0.64 p.s.i. per foot
Freon-12	0.57
Freon-21	0.60
Freon-22	0.51
Ammonia	0.26
Carbon Dioxide	0.33
Sulphur Dioxide	0.60
Methyl Chloride	0.40

*(Approximately 533°R)

TABLE III

CAPITAL RECOVERY FACTORS[9]

n \ i	0 %	2 %	4 %	6 %	8 %	10 %	12 %	15 %	20 %
1.	1.000	1.020	1.040	1.060	1.080	1.100	1.120	1.150	1.200
2.	0.500	0.515	0.530	0.545	0.561	0.576	0.592	0.615	0.655
3.	0.333	0.347	0.360	0.374	0.388	0.402	0.416	0.438	0.475
4.	0.250	0.262	0.276	0.289	0.302	0.316	0.329	0.350	0.387
5.	0.200	0.212	0.225	0.237	0.250	0.264	0.277	0.298	0.334
6.	0.167	0.179	0.191	0.203	0.216	0.230	0.243	0.264	0.301
7.	0.143	0.155	0.167	0.179	0.192	0.205	0.219	0.240	0.277
8.	0.125	0.137	0.149	0.161	0.174	0.187	0.201	0.223	0.261
9.	0.111	0.123	0.134	0.147	0.160	0.174	0.188	0.210	0.248
10.	0.100	0.111	0.123	0.136	0.149	0.163	0.177	0.199	0.239
11.	0.091	0.102	0.114	0.127	0.140	0.154	0.168	0.191	0.231
12.	0.083	0.095	0.107	0.119	0.133	0.147	0.161	0.184	0.225
13.	0.077	0.088	0.100	0.113	0.127	0.141	0.156	0.179	0.221
14.	0.071	0.083	0.095	0.108	0.121	0.136	0.151	0.175	0.217
15.	0.067	0.078	0.090	0.103	0.117	0.131	0.147	0.171	0.214
16.	0.063	0.074	0.086	0.099	0.113	0.128	0.144	0.168	0.211
17.	0.059	0.070	0.082	0.095	0.109	0.124	0.140	0.165	0.209
18.	0.056	0.067	0.079	0.092	0.107	0.122	0.138	0.163	0.208
19.	0.053	0.064	0.076	0.090	0.104	0.120	0.136	0.161	0.206
20.	0.050	0.061	0.074	0.087	0.102	0.117	0.133	0.160	0.205
25.	0.040	0.051	0.064	0.078	0.094	0.110	0.128	0.154	0.202
30.	0.033	0.044	0.058	0.073	0.089	0.107	0.124	0.152	0.201
40.	0.025	0.037	0.051	0.066	0.084	0.102	0.121	0.151	0.200
50.	0.020	0.032	0.047	0.063	0.082	0.101	0.120	0.150	0.200
100.	0.010	0.023	0.041	0.060	0.080	0.100	0.120	0.150	0.200
∞		0.020	0.040	0.060	0.080	0.100	0.120	0.150	0.200

TABLE IV

VISCOSITY OF FREON-12 VAPOR* (8)

t°F	$\frac{\text{lbs.}}{\text{ft.-sec.}}$	.228
-40	712 $\times 10^{-8}$	6.70 $\times 10^{-2}$
-20	732	6.73
0	758	6.79
20	778	6.81
40	799	6.84
60	826	6.90
80	846	6.95
100	866	6.99
120	887	7.02
140	907	7.07
160	927	7.10
180	940	7.13
200	960	7.16
220	980	7.19
240	1000 $\times 10^{-8}$	7.22 $\times 10^{-2}$

*Viscosities are of saturated vapor at one atmosphere. Changes in pressure of less than one atmosphere have a negligible effect upon the viscosity of the vapor.

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