SUBCOOLING LIQUID REFRIGERANT 12 IN A
VAPOUR COMPRESSION REFRIGERATION SYSTEM WITH
A LIQUID–VAPOR HEAT EXCHANGER

by

JOHN D. CHRISTIE

Submitted in Partial Fulfillment
of the Requirements for the
Degree of Bachelor of Science
at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

May, 1959

Signature of Author ............... John D. Christie ...........
Department of Mechanical Engineering, May 25, 1959

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Thesis Supervisor

Accepted by ............... B. C. Richtmyer ..............
Chairman, Departmental Committee on Theses

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Department of Mechanical Engineering
Massachusetts Institute of Technology
Cambridge 39, Massachusetts
May 25, 1959

Professor Alvin Sloane
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge 39, Massachusetts

Dear Sir:

Submitted herewith is a thesis entitled "Subcooling Liquid Refrigerant 12 in a Vapor Compression Refrigeration System with a Liquid-vapor Heat Exchanger" in partial fulfillment of the requirements for the degree of Bachelor of Science in Mechanical Engineering.

Respectfully submitted,

John D. Christie

Best Available Copy
ACKNOWLEDGMENT

I would like to express my appreciation and
gratitude to Professor A. L. Hesselschwerdt, Jr.
for his helpful guidance and to Mr. B. Dimond for
his assistance in the laboratory while I was
preparing this thesis.

Sincerely,

John D. Christie
# TABLE OF CONTENTS

Abstract

I. Introduction .............................................. 1

II. Theoretical Analysis ..................................... 3

III. Experimental Approach and Procedure .................. 7

IV. Results and Discussion of Results ...................... 13

V. Conclusions .............................................. 26

VI. Recommendations ......................................... 28

VII. Appendices

A. Sample Theoretical Calculation .......................... 30

B. Equipment Description and Specifications .............. 32

C. Tabulated Data ........................................... 34

D. Sample Experimental Calculations ...................... 36

E. Tabulated Results ......................................... 39

F. Nomenclature ............................................. 41

G. References and Bibliography ............................. 43
ABSTRACT

Subcooling liquid refrigerant in a vapor compression refrigeration cycle increases the system capacity. The purpose of this thesis is to relate theoretical and experimental performance characteristics to one method of subcooling Refrigerant 12, the use of a liquid-vapor heat exchanger.

Theoretical and experimental values of system coefficient of performance and horsepower per ton are found at various evaporating pressures for a system with and without a liquid-vapor heat exchanger. The theoretical performance of a system with an evaporating pressure of 25 psia and a condensing pressure of 110 psia is found for variations in heat exchanger effectiveness between 0 and 1.

Theoretically a liquid-vapor heat exchanger has a negligible effect on system coefficient of performance and horsepower per ton. Experiments on a system with a relatively short suction line showed that the use of a liquid-vapor heat exchanger does not increase the system performance enough to warrant its installation in a system for this reason. The system capacity is increased by installing a liquid-vapor heat exchanger.

Any future work done with liquid-vapor heat exchangers for the purpose of increasing performance in Refrigerant 12 systems should be limited to systems with long suction lines.
I. INTRODUCTION

Subcooling liquid refrigerant before the expansion valve in a vapor compression refrigeration cycle increases the evaporator capacity for a given evaporating pressure. If the vapor refrigerant leaving the evaporator is heated before it enters the compressor, the volume of the vapor is increased and more work is required to compress the vapor to a given pressure. When liquid refrigerant is subcooled in a liquid-vapor heat exchanger (heat exchanger between the liquid and vapor refrigerant), both of these effects occur. One purpose of this thesis is to theoretically predict the changes in system performance that occur when a liquid-vapor heat exchanger is used to subcool liquid Refrigerant 12. The theoretical performance of the system with a liquid-vapor heat exchanger is compared with the performance of the system without a heat exchanger. This comparison is made over a range of operating conditions.

In an actual system under standard operating conditions energy losses occur in the motor, compressor, evaporator, and piping. These losses decrease the system performance below the theoretical predictions. Under average conditions with the piping exposed to room air the vapor refrigerant in the suction line will gain heat from the atmosphere. This heat flow increases the specific volume of the refrigerant before compression and decreases the system performance. If an equivalent heat flow to the vapor refrigerant in the suction line can be taken from the liquid refrigerant leaving the condenser rather than from the ambient air, the
liquid refrigerant can be subcooled without any added loss in performance. The increase in performance due to subcooling the liquid in a liquid-vapor heat exchanger is dependent upon the magnitude of the heat flow to the suction line when a heat exchanger is not used. If the ambient air temperature is high or the heat transfer surface of the suction line is large, the increase in performance that can be obtained with a heat exchanger may warrant its installation.

Another purpose of this thesis is to compare the actual performance of a Refrigerant 12 system with a liquid-vapor heat exchanger and with no heat exchanger. This comparison is made over a range of operating conditions using Refrigerant 12 in the system. These experimental results are also compared with the theoretical calculations for the same conditions.
II. THEORETICAL ANALYSIS

Theoretical calculations were made for a Refrigerant 12 compression cycle with a liquid-vapor heat exchanger. A counterflow heat exchanger was assumed for these calculations because the maximum amount of heat transfer is possible with this type of heat exchanger. The average specific heat of Refrigerant 12 vapor is less than that of the Refrigerant 12 liquid. Therefore the heat exchanger effectiveness for this case is defined as:

\[
\epsilon = \frac{\frac{t_{vo} - t_{vl}}{t_{lo} - t_{li}}}
\]

when the effectiveness is 1, the temperature of the vapor out of the heat exchanger \( t_{vo} \) equals the temperature of the liquid entering the heat exchanger \( t_{li} \) and the maximum heat transfer occurs.

The coefficient of performance (COP) and horsepower per ton \( \frac{\text{HP}}{\text{ton}} \) were calculated for the cycle while the heat exchanger effectiveness \( \epsilon \) was varied between 0 and 1. Figure 1 is a plot of coefficient of performance versus temperature of liquid leaving the heat exchanger \( t_{lo} \). The plot shows that the maximum possible increase in COP is about 2 per cent. Figure 2 is a plot of \( \frac{\text{HP}}{\text{ton}} \) versus \( t_{lo} \). The maximum decrease in \( \frac{\text{HP}}{\text{ton}} \) is also about 2 per cent.

For all practical purposes the theoretical coefficient of performance and horsepower per ton for a cycle operating between these pressures is approximately constant. This is also shown to be true for other evaporating pressures. Further theoretical calculations were made using experimentally determined values for evaporating pressure, condensing pressure, and heat exchanger effectiveness. These results are shown in Figure 14, a plot of coefficient of performance versus evaporating pressure for cycles with and without a liquid-vapor heat exchanger. All the points in this
do not lie on two smooth curves because the condensing pressure varies between 109 psia and 121 psia. The important thing is that the theoretical COP is approximately the same for a cycle with a heat exchanger as it is for a cycle operating between the same pressures without a heat exchanger.

Theoretical calculations are shown in Appendix A with the corresponding assumptions for these calculations.
Figure 1:
Theoretical Coefficient of Performance vs. Temperature of Liquid Leaving Heat Exchanger for Refrigerant 12 Cycle Operating between 26 psia and 110 psia with Liquid-Vapor Heat Exchanger.
Figure 2

Theoretical Horsepower per Ton vs Temperature of Liquid Leaving Heat Exchanger for Refrigerant 12 Cycle Operating Between 26 psia and 110 psia with Liquid-Vapor Heat Exchanger

Temperature of Liquid Leaving Heat Exchanger - °F
III. EXPERIMENTAL APPROACH AND PROCEDURE

The purpose of making experimental runs was to obtain actual system characteristics to compare with theoretical results and to compare actual system characteristics with and without a liquid-vapor heat exchanger.

Two different sets of runs were made. The first 9 runs were made with the liquid-vapor heat exchanger in the system. Then 5 runs were made without the heat exchanger. The first set of runs was made with the heat exchanger so the pressure drop across the heat exchanger in the suction line could be measured. When the heat exchanger was removed, an equivalent length of piping was installed to provide an equal pressure drop between the same two points in the suction line. The system characteristics are then compared for approximately the same pressure drops in the suction line with and without the heat exchanger.

In all experimental runs a York Condensing Unit and a primary refrigerant calorimeter were used. The heat exchanger used in the first 9 runs was a single pass counter flow exchanger. Mineral wool blanket insulation was used to insulate the heat exchanger and all piping except the liquid line to the heat exchanger for the first 9 runs. This insulation was approximately one inch thick and was covered with aluminum foil to minimize heat transfer to and from the surroundings. The use of insulation improves the effectiveness of the heat exchanger and the accuracy of the heat exchanger effectiveness calculations.

A complete description of all equipment used is listed in Appendix B. A schematic diagram of the apparatus with the liquid-vapor heat exchanger is shown in Figure 3. The locations of all the data points are shown with their corresponding symbols. A photograph of the apparatus as it was set up for the first 9 runs is shown in Figure 4. Figure 5 is a photograph of the insulated heat exchanger showing the location of the pressure taps.
and the tubing leading to a mercury manometer.

The heat exchanger and all the insulation were removed for the last 5 runs. An equivalent length of 12.5 feet of 5/8 inch O.D. refrigeration tubing was installed between the two pressure taps. The piping was not insulated for these runs in an attempt to approach normal operating conditions. Other than the extra length of suction line and the lack of insulation, the apparatus was the same for the second set of runs.

All runs were made after the equipment had been operating for one or more hours to reach steady state conditions. Then the equipment reached steady state, data points were taken every five minutes for 45 minutes. The ten readings were then averaged before corrections and calculations were made. By taking an average of ten readings, errors due to small fluctuations in data and the taking of data were reduced.

All temperatures were measured with copper-constantan thermocouples inserted in glands which were located in the fluid flow. Temperature corrections were made using a standard calibration curve. All pressures except the barometric pressure and the pressure drop across the heat exchanger were measured with bourdon gages. All pressure gages were calibrated before any experimental runs were made and corrections were made during these calibrations. The pressure drop across the heat exchanger or equivalent length of piping was measured with a mercury manometer. The electrical input to the calorimeter and motor were measured with wattmeters and corrected with calibration curves.

The tabulated average data for all runs is shown in Table I, appendix C.

To compare the characteristics under different operating conditions, the actual COP, \( \frac{\text{W}}{\text{ton}} \), and evaporator capacity were calculated for each run. The heat exchanger effectiveness is calculated for each run were it was used. In calculating the COP, \( \frac{\text{W}}{\text{ton}} \), and evaporator capacity, the heat
Figure 3. Insulated liquid-vapor heat exchanger as installed in apparatus.
transfer to the evaporator from the surroundings is assumed to be negligible compared to the electrical input. This is a good assumption, especially when the evaporating temperature is relatively high.

After calculations were made for the actual performance, the experimental heat exchanger effectiveness was used to make theoretical calculations of COP and \( \frac{\eta}{\text{ton}} \) for a system operating between the same evaporating and condensing pressures.

For the second set of runs without the heat exchanger, the equivalent length of piping to be installed was calculated. From the equation for pressure drop in a pipe:

\[
p = \frac{0.0121 f L n^2 v}{d^5}
\]

the equivalent length of piping (L) was determined. The average L for a number of calculations was 12.5 feet.

Similar experimental and theoretical calculations were made for the second set of runs without the heat exchanger. The theoretical calculations with no heat exchanger are the same as those with a heat exchanger where \( \epsilon = 0 \).

Details of the calculations are shown in Appendix D.
IV. RESULTS AND DISCUSSION OF RESULTS

Figure 1 shows the theoretical increase in COP that is possible by using a heat exchanger in a Refrigerant 12 cycle operating between 26 psia and 110 psia. Figure 2 is a similar plot showing the effect of the liquid-vapor heat exchanger on $\frac{\Delta H}{\text{ton}}$. In both cases the maximum possible increase in performance is about 2 per cent. Similar results are shown for other evaporating pressures in Figure 1d. If operating conditions were approximately ideal and the COP and $\frac{\Delta H}{\text{ton}}$ were the most important characteristics of the system, the use of a liquid-vapor heat exchanger in the system would not be warranted. The increase in performance gained by subcooling the liquid Refrigerant 12 is not large enough to overcome the decrease in performance due to heating the vapor before the compressor and still cause an appreciable change on system COP.

Although the system COP and $\frac{\Delta H}{\text{ton}}$ do not change appreciably when a heat exchanger is used, the evaporator capacity will increase. If a given evaporator is in a system and the tonnage of the system has to be increased, it might be possible to do so by adding a liquid-vapor rather than installing a larger evaporator.

The experimental and theoretical COP for a system with a liquid-vapor heat exchanger are plotted versus evaporating pressure ($p_E$) in Figure 6. All the theoretical points do not lie on a smooth curve because the condensing pressure ($p_C$) varies between 110 psia and 121 psia. The experimental curve follows the same trend as the theoretical curve but the deviation between the two is greater at higher values of $p_E$. This is reasonable because the refrigerant flow is greater at higher values of $p_E$. Consequently the pressure drops in the system are larger, causing a greater decrease in COP.

Figure 7 is a plot of theoretical and experimental $\frac{\Delta H}{\text{ton}}$ versus $p_E$ for
the system with the liquid-vapor heat exchanger. Both curves have the same
trend with the experimental \( \frac{\nu}{\text{ton}} \) being at least twice as great as the cor-
responding theoretical value. The deviation between the two is greater at
lower values of \( P_E \). For lower values of \( P_E \) the specific volume of the re-
frigerant is greater, and changes in specific volume are greater for a given
\( P_E \). The specific volume of the refrigerant affects the actual work of
compression and the efficiency of the compressor. The effect of specific
volume is greater at lower values of \( P_E \); therefore, the effect on the \( \frac{\nu}{\text{ton}} \)
required is greater.

Heat exchanger effectiveness is plotted versus evaporating pressure
in Figure 8. Theoretically \( \epsilon \) is a function of heat exchanger geometry only.
The experimental plot of \( \epsilon \) versus \( P_E \) shows that \( \epsilon \) decreases as \( P_E \) increases.
As \( P_E \) increases the mean temperature difference between the two fluids in
the heat exchanger decreases. The mass flow rate of the refrigerant
through the heat exchanger increases as \( P_E \) increases. The actual heat
exchanger effectiveness \( \epsilon \) is dependent on the mean temperature difference
and/or the mass flow rate of the fluids. The heat loss from the heat ex-
changer to the ambient air decreases as the heat exchanger effectiveness
increases. If no heat losses occurred, the variation of \( \epsilon \) would be smaller
for the same variation of \( P_E \).

Two points on the experimental plot deviate considerably from the curve
in the figure. The high value of \( \epsilon = 0.519 \) was obtained when the heat loss
from the heat exchanger was unusually low. The value of \( \epsilon = 0.179 \), when
\( P_E = 59.22 \) psia, was obtained during a run when the system was not quite
at a steady state condition. The temperature of the vapor in the heat ex-
changer varied over \( 40^\circ \text{ F} \) during the run. These two experimental values are
known to be in error; therefore, they were not used to draw the curve.

Figure 9 is a plot of theoretical and experimental \( \text{COP} \) for the system.
with the extra suction line and no heat exchanger. The curves follow the same trends as the ones for the system with the liquid-vapor heat exchanger. The deviation is again greater at higher values of \( p_E \) because of increased pressure drops.

The \( \frac{\Delta p}{\text{ton}} \) of the system without the heat exchanger is plotted versus \( p_E \) in Figure 10. The trends are the same for the system without the heat exchanger as they are for the system with the heat exchanger.

The actual pressure drop across the heat exchanger and the extra suction line for the two different sets of runs is plotted versus evaporating pressure in Figure 11. The \( \Delta p_v \) across the extra suction line is larger than the \( \Delta p_v \) across the heat exchanger for the same evaporating pressure. This deviation shows that the wrong length of tubing was installed for the extra suction line. This error in installing the wrong length of pipe is due to unaccounted pressure losses in the fittings. An equivalent length of pipe was determined for each fitting but the results were not accurate. The maximum deviation between the two curves is approximately 0.2 psi. This deviation is not large enough to cause any considerable error in comparing the experimental results for the two sets of runs.

The actual COP for the two different sets of runs is plotted versus \( p_E \) in Figure 12. For any \( p_E \) the system COP is higher when the liquid-vapor heat exchanger is in the system. The absolute increase in performance is about the same for any \( p_E \), but the percentage increase in COP decreases as \( p_E \) increases.

A similar plot comparing \( \frac{\Delta p}{\text{ton}} \) for the two sets of runs is shown in Figure 13. The \( \frac{\Delta p}{\text{ton}} \) for the system with the heat exchanger is lower for any \( p_E \). The deviation between the curves increases at lower values of \( p_E \) because of the increasing effect of specific volume changes.

Figure 14 is a plot of theoretical COP versus \( p_E \) for the system runs
with and without the heat exchanger. The points do not lie on smooth
curves because of variations in condensing pressure. The plot shows that
the difference in theoretical performance for systems with and without the
liquid-vapor heat exchanger is negligible.

All tabulated results are shown in Table 2, Appendix E.
Figure 1
Experimental and Theoretical
Horsepower per Ton vs.
Evaporating Pressure for
Refrigerant 12 System with
York Condensing Unit and
Liquid-Vapor Heat Exchanger

\[
\begin{align*}
\text{Horsepower per Ton} & > 3 \\
\text{Evaporating Pressure - psia} & > 0 \quad 10 \quad 20 \quad 30 \quad 40 \quad 50 \quad 60
\end{align*}
\]

\[\Delta\text{ Experimental}\]
\[\bigcirc\text{ Theoretical}\]
Figure 4
Heat Exchanger Effectiveness vs. Evaporating Pressure for Liquid-Vapor Heat Exchanger in Refrigerant 12 System with York Condensing Unit
Figure 9

Experimental and Theoretical Coefficient of Performance vs. Evaporating Pressure for Refrigerant 13 System with York Condensing Unit and no Liquid-Vapor Heat Exchanger

- Experimental
- Theoretical
Figure 10
Experimental and Theoretical Horsepower per Ton vs. Evaporating Pressure for Refrigerant 12 System with York Condensing Unit and no Liquid-Vapor Heat Exchanger

- Experimental
- Theoretical
Figure 4

Theoretical Coefficient of Performance vs. Evaporating Pressure with and without Liquid-Vapor Heat Exchanger for Refrigerant 12 System (condensing pressures correspond to experiments).

Coefficient of Performance vs. Evaporating Pressure (bars)

- O - With Heat Exchanger
- □ - Without Heat Exchanger
V. CONCLUSIONS

From the calculations, and Figures 1, 2, and 1h, it is obvious that installing a liquid-vapor heat exchanger for the purpose of increasing the COP is not worthwhile from a theoretical standpoint. Theoretically the use of a liquid-vapor heat exchanger will increase the capacity of a given evaporator at a given evaporating pressure.

For the laboratory system which was tested the experimental values of system COP were between 20% and 45% of the theoretical values. This range of values holds for both sets of runs, with and without the liquid-vapor heat exchanger. Similarly the experimental values of system \( \frac{h}{\text{ton}} \) for the system tested were between 200% and 450% of the theoretical values. The maximum experimental system COP that can be obtained with this laboratory Refrigerant 12 system is approximately 50% of the theoretical value. The actual system COP deviates from the theoretical COP more at higher evaporating pressures while the actual system \( \frac{h}{\text{ton}} \) deviates from the theoretical \( \frac{h}{\text{ton}} \) more at lower evaporating pressures.

The actual liquid-vapor heat exchanger effectiveness decreases as the mean temperature difference of the two fluids decreases and the flow rate increases. Theoretically the heat exchanger effectiveness is a function of geometry only.

For an operating system containing Refrigerant 12 with a relatively short suction line, the increase in actual system COP achieved by using a liquid-vapor heat exchanger would not warrant its installation. The longer the suction line is in an operating system, the greater the heat transfer will be between the ambient air and the vapor, other conditions being constant. Therefore the advantage of a liquid-vapor heat exchanger to subcool the liquid refrigerant increases as the length of the suction line increases. The installation of a liquid-vapor heat exchanger to increase system COP
might be warranted in a Refrigerant 12 system with a long suction line.

If the evaporator capacity of a Refrigerant 12 system must be increased at a given evaporation pressure, the desired increase might be obtained with a liquid-vapor heat exchanger rather than a larger evaporator.
VI. RECOMMENDATIONS

Future work with liquid-vapor heat exchangers to subcool Refrigerant 12 should be limited to systems with long suction lines and to systems where it is desired to increase a given evaporator capacity. The increase in performance that can be obtained by installing a liquid-vapor heat exchanger in a system with a long suction line should be obtained.

The use of liquid-vapor heat exchangers and other methods of subcooling liquid refrigerant to increase system capacity could be compared. This comparison should be made on an economic basis as well as a mechanical performance basis.
VII. APPENDICES
APPENDIX A

SAMPLE THEORETICAL CALCULATION

Conditions and Assumptions

1. Vapor compression cycle with expansion valve
2. Refrigerant 12
3. No pressure drops in pipe lines, evaporator, or compressor manifolds and valves
4. Saturated vapor leaving evaporator
5. Saturated liquid leaving condenser
6. Isentropic compression
7. $P_E = 26 \text{ psia}$
8. $P_C = 110 \text{ psia}$

For a cycle with a liquid-vapor heat exchanger set the temperature of the liquid leaving the heat exchanger at $h_0^\circ F$.

$$P_E = 26 \text{ psia} \quad h_{v1} = 77.710 \text{ Btu/lb}$$
$$P_C = 110 \text{ psia} \quad h_{li} = h_{LCO} = 28.059 \text{ Btu/lb}$$
$$t_{lo} = h_0^\circ F \quad h_{lo} = 17.273 \text{ Btu/lb}$$

An energy balance around the heat exchanger fields:

$$h_{vo} = h_{v1} + (h_{li} - h_{lo}) = 77.710 + (28.059 - 17.273) = 88.496 \text{ Btu/lb}$$
$$h_{s} = h_{vo} = 88.496 \text{ Btu/lb}$$
$$P_{vo} = P_E = 26 \text{ psia} \quad s_{vo} = 0.19003 \text{ Btu/lb}^\circ F$$
$$s_d = s_s = s_{vo} = 0.19003 \text{ Btu/lb}^\circ F \quad P_d = 110 \text{ psia}$$
$$h_d = 101.551 \text{ Btu/lb}$$
$$h_{LEX} = h_{lo} = 17.273 \text{ Btu/lb}$$
$$\text{COP} = \frac{h_{LE} - h_{LEX}}{h_d - h_s} = \frac{77.710 - 17.273}{101.551 - 88.496} = 4.629h$$
\[
\frac{\text{hp}}{\text{ton}} = \frac{w(h_d - h_s)}{h_2 h_2} \frac{200}{w(h_{\text{in}} - h_{\text{EX}})} = \frac{1.71 h}{\text{COP}}
\]

\[
\frac{\text{hp}}{\text{ton}} = \frac{1.71 h}{4.5294} = 1.0162 \text{ hp/ton}
\]

For a cycle without a heat exchanger \( t_{10} = t_{10} = t_{\text{LCO}} \) and for a cycle with an infinite counterflow heat exchanger \( t_{\text{V0}} = t_{11} \).

The results used for Figures 1 and 2 are:

<table>
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<th>( t_{10} )</th>
<th>COP</th>
<th>( \frac{\text{hp}}{\text{ton}} )</th>
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<tr>
<td>87.23</td>
<td>1.5455</td>
<td>1.0370 (no heat exchanger)</td>
</tr>
<tr>
<td>80</td>
<td>1.5563</td>
<td>1.0316</td>
</tr>
<tr>
<td>70</td>
<td>1.5693</td>
<td>1.0281</td>
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<tr>
<td>60</td>
<td>1.5851</td>
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<td>50</td>
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<tr>
<td>40</td>
<td>1.6294</td>
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<td>33.24</td>
<td>1.6167</td>
<td>1.0164 (Infinite heat exchanger)</td>
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APPENDIX B

EQUIPMENT DESCRIPTION AND SPECIFICATIONS

York Condensing Unit

Maker: York Corporation, York, Pennsylvania

Type: 422 FW

Speed: 375 rpm

Rating at 100 F Condensing Temperature:

a. 4890 Btu per hour at 20 F evaporator temperature
b. 7900 Btu per hour at 40 F evaporator temperature

Maximum condensing pressure: 138 psig

Motor: 5/4 hp

Bore: 2 5/8 in.

Stroke: 1 1/4 in.

Number of cylinders: 2

Piston displacement per revolution: 0.0075 cu. ft.

Refrigerant: Freon-12

Normal Refrigerant charge: 7 lbs.

Receiver refrigerant capacity: 40 lbs.

Crank case oil charge: 5 pints

Control equipment:

Condenser water: Penn Electric—Type XLI

Pressure safety: Minneapolis-Honeywell—Type L-h13-1
Evaporator:
Type—Primary refrigerant calorimeter (flooded evaporator)
Insulation—Wood-covered cork, heat transfer factor 1.315 Btu/hr—°F

Expansion Valve:
Maker: Detroit Lubricator
Type: Thermostatic
Number: 783
Setting: ±10°F superheat

Heat Exchanger:
Model Number: 75-x

Wattmeters
Motor: 0-2000 watts
Evaporator heater: 0-3000 watts

Copper constantan thermocouple
Potentiometer—Leads and Northrop

Bourdon gages
Evaporator exit: 0-100 psig
Vapor into heat exchanger: 0-200 psig 0-30 in. Hg vacuum
Vapor out of heat exchanger: 0-200 psig 0-30 in. Hg vacuum
Suction: 0-150 psig 0-30 in. Hg vacuum
Discharge: 0-150 psig 0-30 in. Hg vacuum

Mercury Manometer
### APPENDIX C

#### TABLE I

**TABULATED EXPERIMENTAL DATA**

<table>
<thead>
<tr>
<th>Date</th>
<th>4/28/59</th>
<th>4/30/59</th>
<th>4/30/59</th>
<th>5/2/59</th>
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<td>45</td>
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APPENDIX D

SAMPLE EXPERIMENTAL CALCULATION

Run number 2 is used for an example.

Assumptions for calculations:

1. \( P_{DE} = P_E \)
2. \( P_d = P_c \)
3. Heat losses from the calorimeter are negligible.

\( p_b = 29.829 \) in. \( h_g = 29.829 (0.4912) \) psia = 14.67 psia

\( p_e = p_{DE} = 23.03 \) psig = 37.70 psia

\( t_e = 22.79 ^\circ F \)

\( p_c = p_d = 100.75 \) psig = 115.42 psia

\( t_c = 90.64 ^\circ F \)

\( \text{(COP)}_A = \frac{W_H}{W_H} = \frac{1905}{726} = 2.63 \)

\( \text{(COP)}_A = \frac{14.71 h}{\text{(COP)}_A} = \frac{14.71 h}{2.63} = 1.80 \text{ hp/ton} \)

\( q_E = 3.413 \text{ W}_H = 3.413 (1905) = 6500 \text{ Btu/hr} \)

\( \epsilon = \frac{t_{vo} - t_{vi}}{t_{li} - t_{vi}} = \frac{li.3 - 31.4}{70.7 - 31.4} = 0.2h2 \)

\( P_{vi} = 22.18 \text{ psig} = 36.85 \text{ psia} \quad t_{vi} = 31.4 ^\circ F \quad h_{vi} = 81.003 \text{ Btu/lb} \)

\( P_{vo} = P_{vi} - \Delta p_v = 36.85 - .21 = 36.64 \text{ psia} \quad t_{vo} = li.3 ^\circ F \quad h_{vo} = 82.263 \text{ Btu/lb} \)

\( t_{li} = 8\text{4.7} \quad h_{li} = 27.46h6 \text{ Btu/lb} \)

\( t_{lo} = 70.65 \quad h_{lo} = 2li.200 \text{ Btu/lb} \)

\( (\Delta h)_l - (\Delta h)_v \text{ he} = (h_{li} - h_{lo}) - (h_{vo} - h_{vi}) = 1.33h \text{ Btu/lb} \)
For \((\text{CO}_2)_T\) and \((\text{H}_2)_{\text{ton}} T\)

\[
P_{\text{EE}} = 37.70 \text{ psia} \quad t_{\text{EE}} = 22.79 \text{ F} \quad h_{\text{EE}} = 79.575 \text{ Btu/lb}
\]

\[
t_{\text{LOCO}} = t_c = 90.637 \text{ F} \quad h_{\text{LOCO}} = 28.86 \text{ Btu/lb}
\]

From the definition of \(\epsilon = \frac{t_v - t_{vi}}{t_{li} - t_{vi}}\)

\[
t_v = t_{vi} + \epsilon (t_{li} - t_{vi})
\]

\[
t_v = 22.79 + 0.21(90.64 - 22.79) = 39.19 \text{ F}
\]

\[
P_v = P_{\text{EE}} = 37.70 \quad h_v = 82.17 \text{ Btu/} \text{lb} \quad s_v = 0.17205 \text{ Btu/} \text{lb}^\circ \text{R}
\]

\[
s_d = s_s = s_v = 0.17205 \text{ Btu/} \text{lb}^\circ \text{R} \quad p_d = 115.42 \text{ psia}
\]

\[
h_d = 117.70 \text{ Btu/} \text{lb}
\]

From energy balance around heat exchanger

\[
h_{li} = h_{li} - (h_v - h_{vi}) = 28.86 - (82.153 - 79.675) = 26.331 \text{ Btu/} \text{lb}
\]

\[
h_{\text{LEX}} = h_{li} = 26.331 \text{ Btu/} \text{lb}
\]

\[
(\text{CO}_2)_T = \frac{h_{\text{EE}} - h_{\text{LEX}}}{h_d - h_s} = \frac{79.575 - 26.331}{91.040 - 82.153} = 6.00
\]

\[
\left(\frac{\text{hp}}{\text{ton}}\right) = \frac{\text{h}_{\text{li}}}{6.00} = 0.787 \text{ hp/ton}
\]

To find equivalent length of pipe for \(p\) across the heat exchanger

\[
h_{\text{EE}} = 80.768 \text{ Btu/} \text{lb}
\]

\[
h_{\text{LEX}} = 21.189 \text{ Btu/} \text{lb}
\]

\[
w = \frac{3.613 \text{ ft/min}}{(h_{\text{EE}} - h_{\text{LEX}}) 60} = 3.613 \frac{(1905)}{56.579 \text{ (50)}} = 1.92 \text{ lb/min}
\]

\[
v_{\text{avg}} = v \text{ at 36.7 psia and 37.9 F}
\]

From equation for \(p = 0.0121 \frac{\text{flw}^2v}{d^2}\)

\[
L = \frac{\text{flw}^2v_{\text{avg}}}{0.0121d^2v_{\text{avg}}}
\]

From Figure 6.6, Severns and Fellows (L) \(f = 0.02\)

From Figure 10.2, Severns and Fellows (L) \(d = 0.545 \text{ in. for 5/8 in. refrigeration tubing.}\)
Solving for \( L \) yields \( L = 11.5 \text{ ft} \).

The calculations for runs without a heat exchanger are the same as this sample calculation except that \( \epsilon = 0 \). For runs without the heat exchanger no equivalent pipe length \( L \) was calculated.
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<th>3</th>
<th>4</th>
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<td>2.32</td>
<td>1.05</td>
</tr>
<tr>
<td>(COP)$_T$</td>
<td>3.10</td>
<td>9.27</td>
<td>4.50</td>
<td>6.22</td>
<td>6.71</td>
<td>5.22</td>
<td>10.22</td>
</tr>
<tr>
<td>($\frac{\eta}{\text{ton}}$)$_T$—hp/ton</td>
<td>0.733</td>
<td>0.509</td>
<td>1.23</td>
<td>0.705</td>
<td>0.770</td>
<td>1.571</td>
<td>0.462</td>
</tr>
<tr>
<td>$Q_L$—Btu/hr</td>
<td>9520</td>
<td>10810</td>
<td>21830</td>
<td>4623</td>
<td>3530</td>
<td>2050</td>
<td>11330</td>
</tr>
<tr>
<td>$h_v$—Btu/lb</td>
<td>0.115</td>
<td>0.121</td>
<td>——</td>
<td>——</td>
<td>——</td>
<td>——</td>
<td>——</td>
</tr>
<tr>
<td>$h_v$—Btu/lb</td>
<td>23.006</td>
<td>22.853</td>
<td>——</td>
<td>——</td>
<td>——</td>
<td>——</td>
<td>——</td>
</tr>
<tr>
<td>$h_v$—Btu/lb</td>
<td>23.730</td>
<td>24.479</td>
<td>——</td>
<td>——</td>
<td>——</td>
<td>——</td>
<td>——</td>
</tr>
<tr>
<td>$h_l$—Btu/lb</td>
<td>2.312</td>
<td>28.312</td>
<td>——</td>
<td>——</td>
<td>——</td>
<td>——</td>
<td>——</td>
</tr>
<tr>
<td>$h_l$—Btu/lb</td>
<td>26.809</td>
<td>27.065</td>
<td>——</td>
<td>——</td>
<td>——</td>
<td>——</td>
<td>——</td>
</tr>
<tr>
<td>($\Delta h_l - \Delta h_v$)$_{he}$—Btu/lb</td>
<td>0.779</td>
<td>0.621</td>
<td>——</td>
<td>——</td>
<td>——</td>
<td>——</td>
<td>——</td>
</tr>
</tbody>
</table>
APPENDIX F

NUMERICAL

*d* diameter

*f* friction factor

*h* enthalpy

*L* equivalent length of pipe

*p* pressure

*Q_E* evaporator capacity

*s* entropy

*t* temperature in degrees Fahrenheit

*v* specific volume

*w* flow rate

*W* Watts

*COP* coefficient of performance

*hp* horsepower per ton

*Δ* finite difference

*ε* heat exchanger effectiveness

SUBSCRIPTS

*A* actual or experimental

*b* barometer

*c* conditions in condenser

*d* discharge from compressor

*E* conditions in evaporator

*EE* evaporator exit
H  calorimeter heater
he  heat exchanger
LX  liquid refrigerant before expansion valve
LO  liquid out of condenser
li  liquid into heat exchanger
lo  liquid out of heat exchanger
m  compressor motor
s  suction at compressor
T  theoretical
vi  vapor into heat exchanger
vo  vapor out of heat exchanger
REFERENCES AND BIBLIOGRAPHY


