A STUDY OF WATER-COOLED AIR CONDITIONERS

by

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ABSTRACT

Retrofitting a standard air-cooled air conditioner with a water-cooled condenser can increase the COP of the unit by 60% and increase the capacity of the unit by 40%. However, to fully realize these improvements, modification of the throttling device is necessary.

ACKNOWLEDGEMENTS

I would like to thank my parents, Richard L. Alexander and Delee Alexander for their support and patience in my work. It was through their generosity and trust that I was allowed to tinker with the air conditioner.

I would like to also thank my father, David J. Bergeron, Jr., for donating hardware used in this project.

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Thanks also go to Mr. Hyatt, "The Air Conditioning Man" who helped me install the system at my house and who donated most of the hardware used in this study.

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TABLE OF CONTENTS

LIST OF FIGURES AND GRAPHS

NOMENCLATURE

P_{E1} - pressure entering the evaporator, psig

P_{E2} - pressure leaving the evaporator, psig

P_{C1} - pressure entering the condenser, psig

P_{C2} - pressure leaving the condenser, psig

T_{E1} - evaporator temperature entrance, °F

T_{E2} - evaporator temperature exit, °F

T_{C1} - condenser temperature entrance, °F

T_{C2} - condenser temperature exit, °F

T_{W1} - water temperature entering condenser, °F

T_{W2} - water temperature leaving condenser, °F

V - voltage to compressor, volt

W - watts to compressor, watts

A - current through compressor, amps

$\dot{M}_W$ - flowrate of water in condenser, gal/min

h_{E1} - enthalpy entering the evaporator, BTU/lb

h_{E2} - enthalpy leaving the evaporator, BTU/lb

h_{C1} - enthalpy entering the condenser, BTU/lb

h_{C2} - enthalpy leaving the condenser, BTU/lb

COP - Coefficient of Performance

P.f. - power factor

$\dot{M}_R$ - refrigerant flowrate, lb_m/hr

Q_E - cooling capacity, BTU/hr

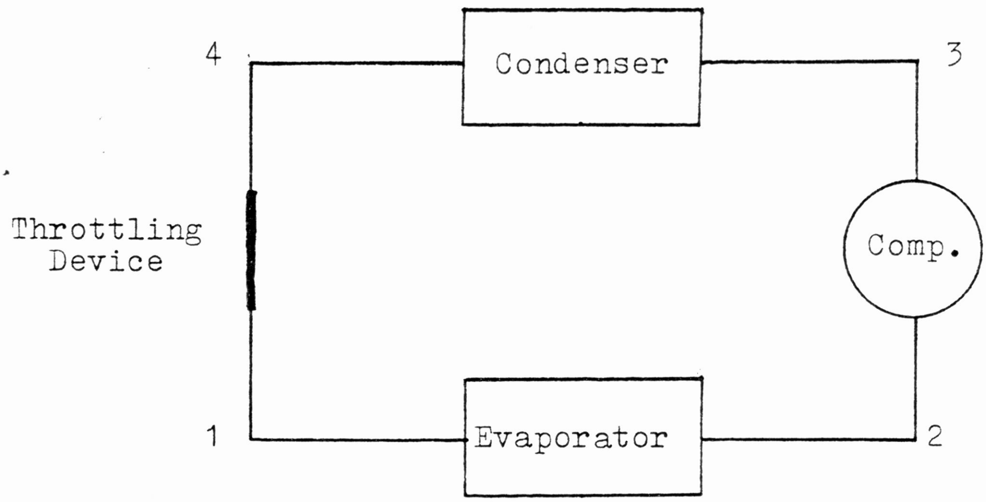
INTRODUCTION

Generally speaking, water-cooled air conditioners have superior performance characteristics compared to air-cooled units. Most residential units are air-cooled. If a supply of water is available, suitable for use in a water-cooled air conditioner, one may retrofit an existing air-cooled unit with a water cooled condenser. This should, according to theoretical calculations, improve the coefficient of performance by 60% and increase the cooling capacity of the unit by 40%. The student involved in this project did retrofit his 4-ton home A/C with a water-cooled condenser using pool water to condense the freon. This system realized only a 46% increase in COP and a 10% increase in capacity. The purpose of this project is to determine the cause of the difference in theoretical and actual performance, and correct the problem or problems.

Theory

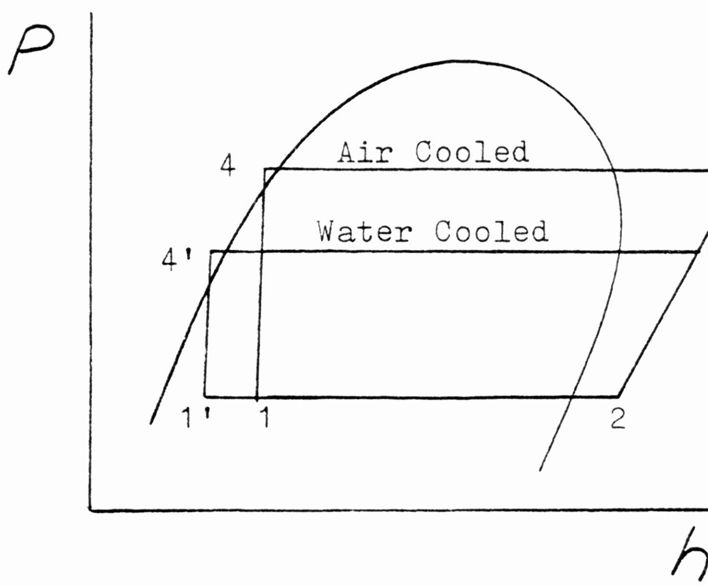
An air conditioning system is shown in Fig. 1. The thermodynamic cycle of a vapor compression air conditioning system is shown in Fig. 2.

At state 1, the working fluid (typically R-22) enters the evaporator where, at low pressure, the fluid evaporates to state 2. This low pressure vapor is then compressed by the compressor to state 3. The work required by the



A/C System

Fig. 1



Thermodynamic Cycle of
a Vapor Compression
Refrigeration System

Fig. 2

compressor is equal to the change in enthalpy of the vapor. Once at state 3, the high pressure enables the working fluid to condense to a high pressure liquid. This condensation occurs in the condenser. The pressure at which the vapor condenses depends upon the temperature of the condenser walls. Air-cooled units typically operate with a condenser temperature of 125°F. By water cooling the unit, this condensing temperature drops to 95°F. The work required by the compressor is significantly reduced. Once at state 4, the liquid is throttled to low pressure (state 1) using an expansion valve or capillary tube. Water-cooled units cool the liquid to a lower enthalpy state which results in a larger cooling effect in the evaporator per lb_m of working fluid. This, coupled with the fact that the mass flow rate increases in a water-cooled system, increases the cooling capacity of the unit dramatically. The mass flow rate should increase for two reasons. First, the reduced load on the compressor increases its speed (less rotating magnetic field slip). Secondly, volumetric efficiency losses decrease. A certain fraction of the vapor in the compressor is not pushed through on each stroke due to the fact that at top dead center there is some volume remaining in the cylinder. The amount of mass in this volume depends upon the density of the vapor. The density is less in a water-cooled unit since the pressure of this trapped vapor is only 170 psig compared to 270 psig for air-cooled units.

Therefore, according to theoretical calculations, water-cooling an A/C should have the following major effects on the unit's performance:

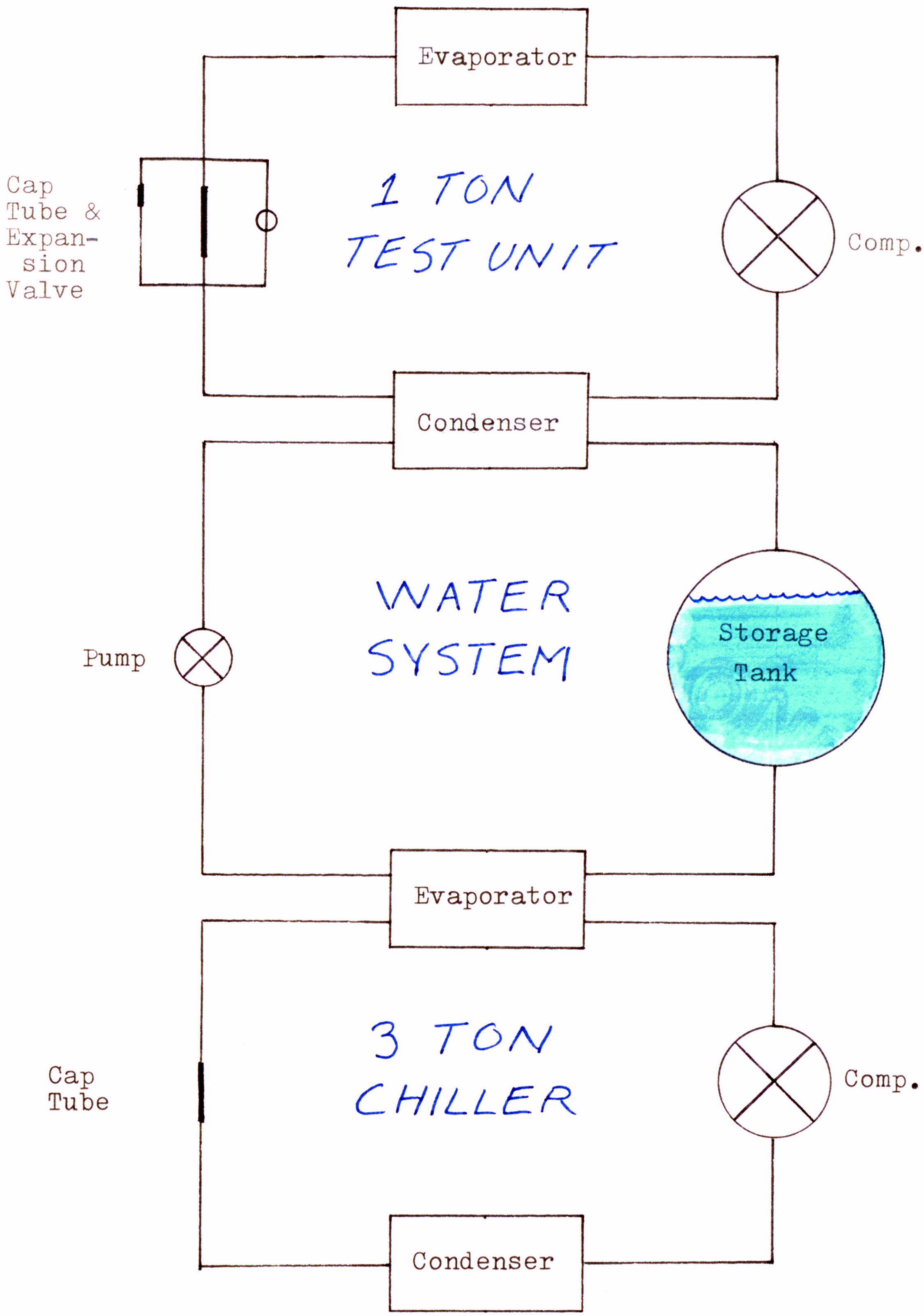
1. increased COP,
2. decreased condenser pressure,
3. increased unit capacity, and
4. increased mass flowrate.

APPARATUS

Testing was conducted in the Mechanical Engineering Shops, J. R. Thompson Building, on the Texas A&M University Campus. The test set is shown in Fig. 3. The one-ton system in this drawing was the testunit. The storage tank and 3-ton unit provided the necessary control over the condenser temperature of the one-ton unit. When the one-ton unit was operating alone, the water system would heat up at a rate of approximately 1.4°F per minute. When the 3-ton chiller operated alone, the water cooled at a rate of 2.5°F per minute. For testing, the chiller unit was allowed to cool the water to approximately 38°F. At this point, the test unit was turned on and allowed to reach equilibrium for five minutes. The chiller unit was then turned off and the system would begin heating. Data was taken as the condenser temperature rose. One data point was collected for every 10 psi rise in condenser pressure. The first data point was generally taken at a condenser pressure of 90 psig (T_C = 60°F) and the last taken at 310 psig (T_C = 135°F).

One-Ton Test Unit

A sketch of the one-ton system and the instrumentation is shown in Fig. 4. The test unit was equipped with three parallel throttling devices; two capillary tubes, and one



Test Set-up
Fig.3

Instrumented Test Unit

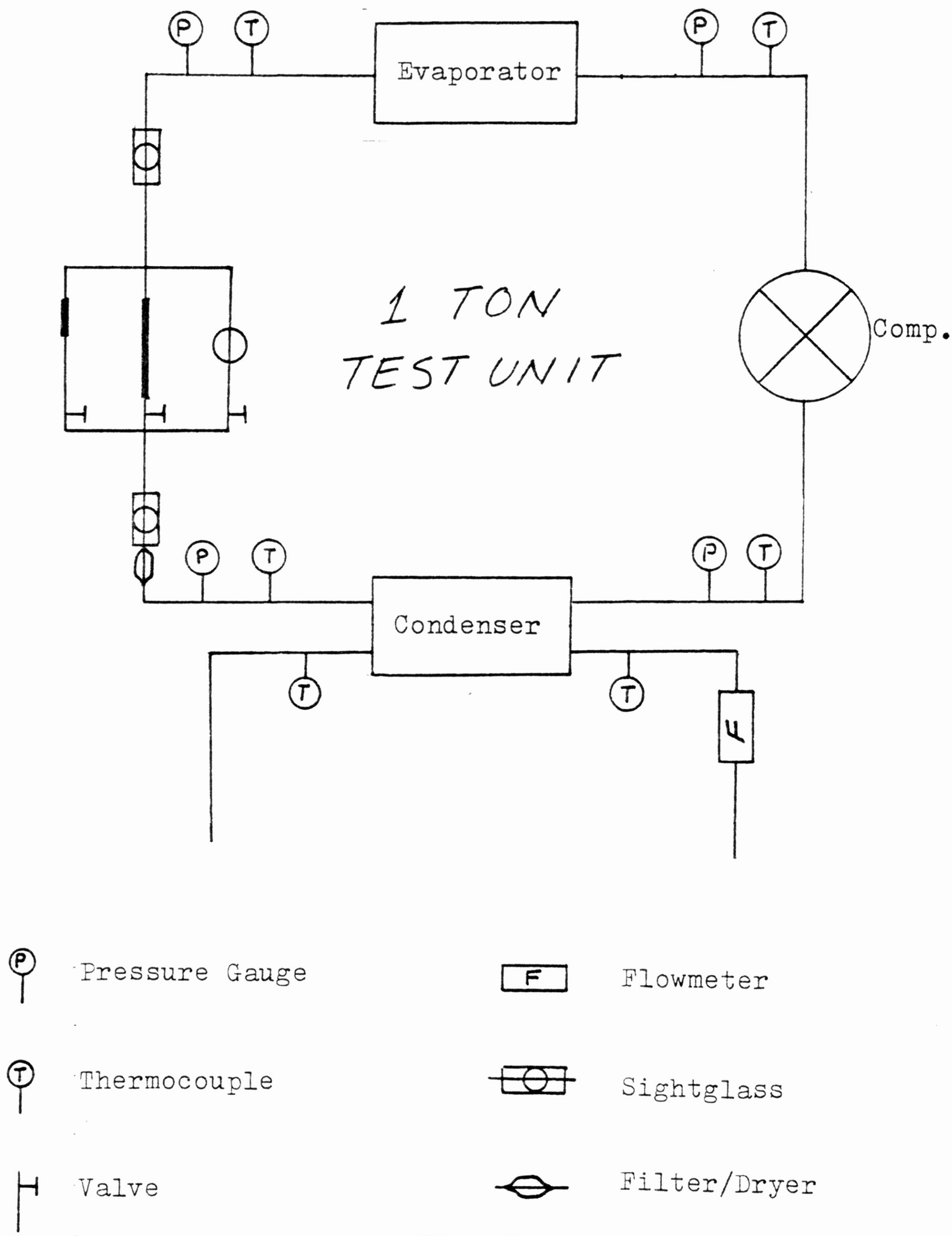


Fig. 4

externally equilized thermostatic expansion valve. Valving on the unit allowed any combination of the three throttling devices. There were four basic tests run on the unit.

1. The first test involved use of the normal capillary tube. Normal in the sense that it was sized for typical air-cooled operating conditions; [1] P_{COND} = 270 psig and P_{EVAP} = 70 psig.

2. The second test involved the use of the short capillary tube. It was sized for typical water-cooled operating pressures; [1]
P_C = 150 psig and P_E = 70 psig.

3. The third test used a normal E.E.T. expansion valve. Normal operating pressures for it were; 270 psig and 70 psig.

4. The last test used the same expansion valve but a superheat adjustment screw was opened to allow for normal operating pressures of 150 psig and 70 psig.

Tests

All four operating conditions were tested over a wide range of condenser temperatures (60 to 135°F). Below is a list of all the variables measured:

P_{E1} - Pressure gauge ± .5 psi

P_{E2} - Pressure gauge ± .5 psi

P_{C1} - Pressure gauge ± 2 psi

P_{C2} - Pressure gauge ± 2 psi

T_{E1}, T_{E2}, T_{C1}, T_{C2} — Thermocouples soldered to the refrigeration lines and insulated ± ½°C direct read out

T_{W1}, T_{W2} — Thermocouples placed in direct contact with the water-millivolt readout ±.02°F

V - voltmeter ± 1%

W - Wattmeter ± 1%

A - Ampmeter ± 1%

$\dot{M}_W$ — Turbine flowmeter millivolt readout ±.01 gal/min

The following variables were calculated from the data. These calculations are illustrated in Appendix A-1: COP, $\dot{M}_R$, Q_{EVAP}, P.f., and annual cost.

Mass Flowrate Measurements

The mass flowrate of the refrigerant (R-22) was measured indirectly. The enthalpy (energy/lb) of the R-22 was known at the entrance and exit of the condenser. The energy flow into the condenser water was known completely, therefore, using an energy balance, one could calculate the mass flowrate of the R-22. An indirect method was used because the instrumentation needed to measure the flowrate directly was too expensive.

RESULTS AND DISCUSSION

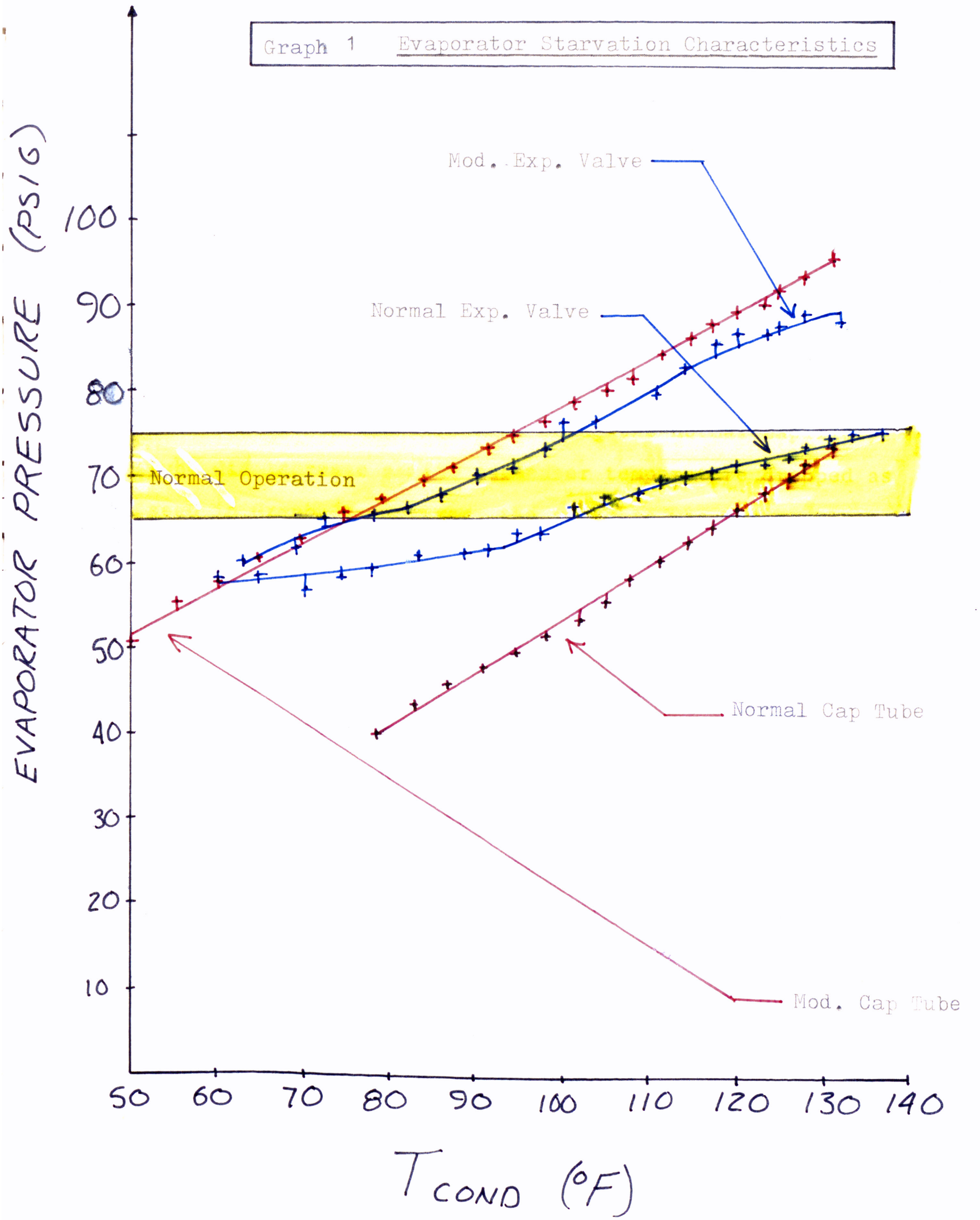
Starvation

It was apparent from the results that the biggest cause of off-design inefficiency was caused by evaporator starvation.

If one water-cools a standard air conditioner which uses capillary tubes for throttling, the new lower condenser pressure will not be able to force the design flow rate of a fluid through the capillary tubes. This will result in a drop in the evaporator pressure. This is undesirable since the work of the compressor is increased by a lowering of the evaporator pressure. A plot of the evaporator starvation is shown on Graph 1.

The normal expansion valve was much more suitable for the lower condenser pressures, as illustrated in Graph 1. At water-cooled condenser temperatures (85 to 95°F), the evaporator pressure was only slightly out of the normal range (65 to 75 psig).

The short capillary tube gave good evaporator pressures in the water-cooled range, but flooded the evaporator for normal use. A system which operates at an abnormally high evaporator pressure will not cool the air passing over it enough for proper dehumidification. If the pressure is much higher than normal (>85 psig), there is a chance that all the refrigerant will not evaporate, sending liquid to


Graph 1 Evaporator Starvation Characteristics
Mod. Exp. Valve
Normal Exp. Valve
Normal Operation
Normal Cap Tube
Mod. Cap Tube
EVAPORATOR PRESSURE (PSIG)
100
90
80
70
60
50
40
30
20
10
50
60
70
80
90
100
110
120
130
140
T COND (°F)

to the compressor.

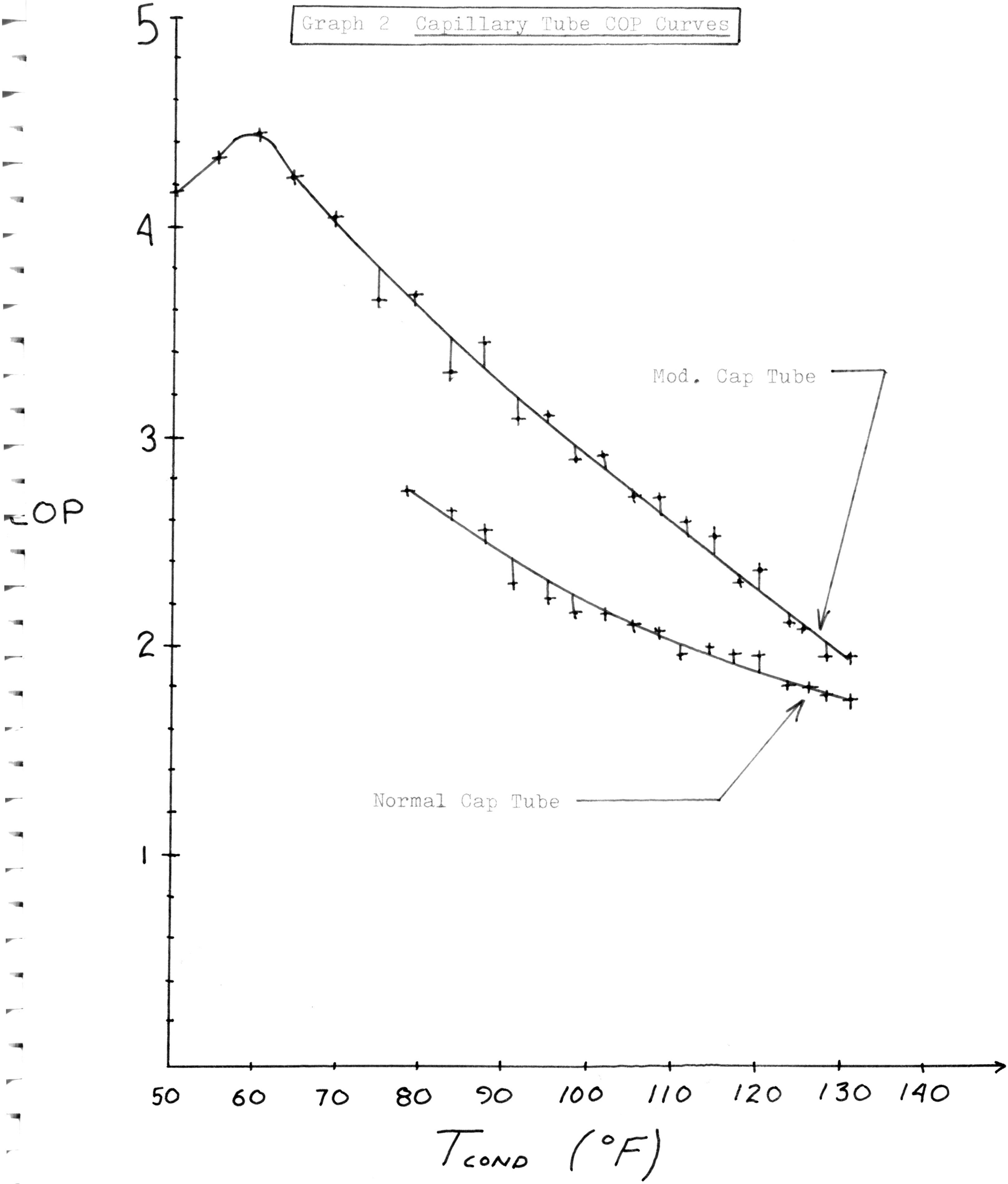
The opened expansion valve was able to maintain normal evaporator pressures when operating in the water-cooled condenser range.

COP

Generally speaking, the COP of any air condition cycle should increase if the condensing temperatures lowers or if the evaporating temperature rises.

The COP of the test unit using the normal capillary tube did increase as the condenser temperature dropped as seen in Graph 2. However, the increase was not nearly as good as one might first guess because evaporator starvation lowered the effective evaporator temperature.

The short capillary tube was able to realize a much greater increase in COP at water-cooled condenser temperatures. One may ask why the COP of the short capillary tube operating at normal air cooled condensing temperatures is higher than that of the normal capillary tube. The short tube has an abnormally high evaporator temperature when operating in this range which results in inadequate dehumidification and possible compressor damage. Points on the COP curve in the range of $T_C > 95°F$ should be considered illegal operating points for the short tube. This same argument applies to the opened expansion valve for $T_C > 95°F$.


Graph 2 Capillary Tube COP Curves
5
4
3
2
1
COP
Mod. Cap Tube
Normal Cap Tube
50
60
70
80
90
100
110
120
130
140
Tcond (°F)

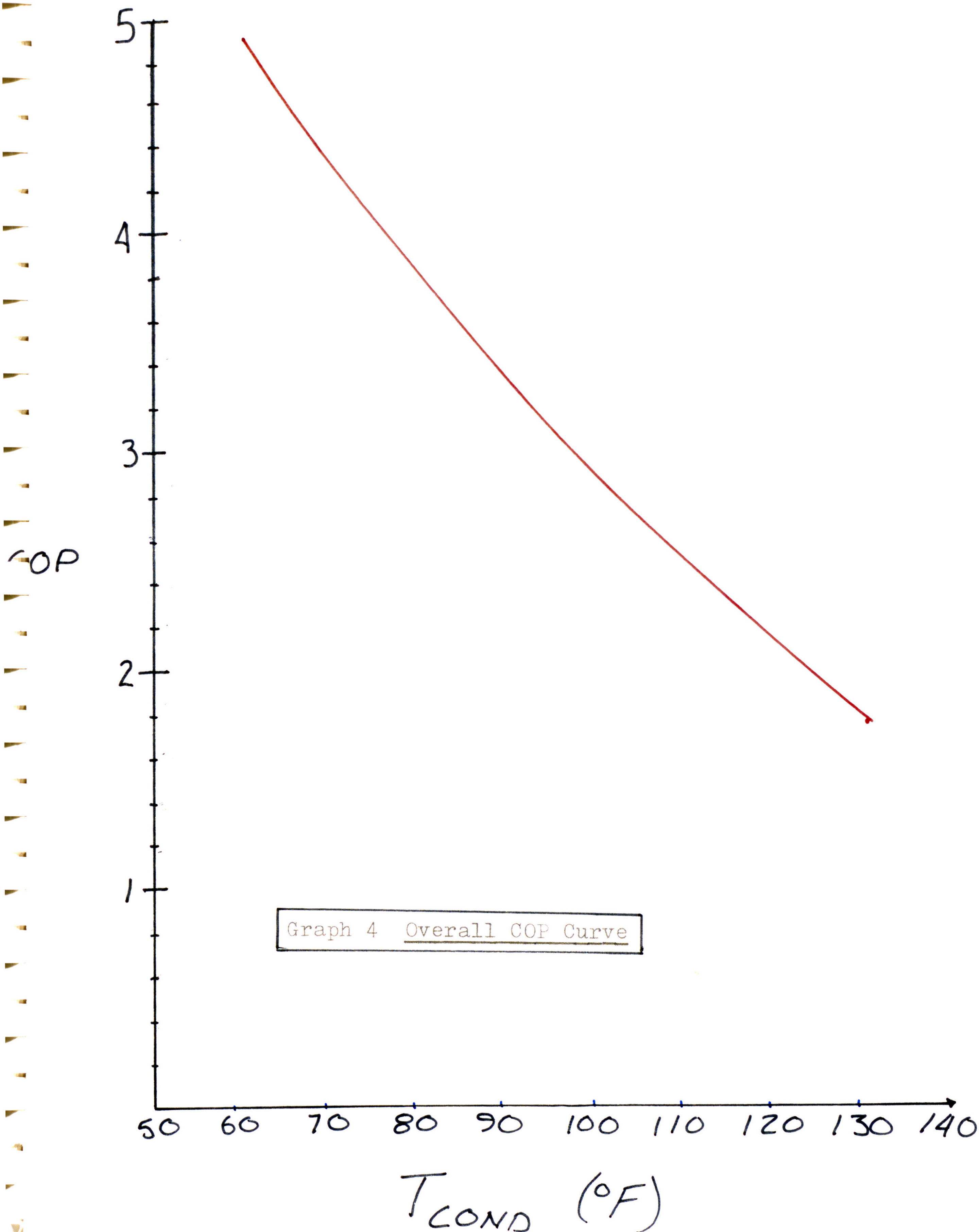
The COP of the test unit using the normal expansion valve increased excellently at water-cooled conditions, as seen in Graph 3. The opened expansion valve did realize a better COP due to the fact that starvation hurt the normal expansion valves' performance.

If some ideal expansion valves were available, its performance would follow the curve of Graph 4. This graph is a "legal" combination of the other four. This graph basically follows the performance of the normal throttling devices in the normal condensing temperature range and then follows the modified throttling device's performance in the water-cooled range.

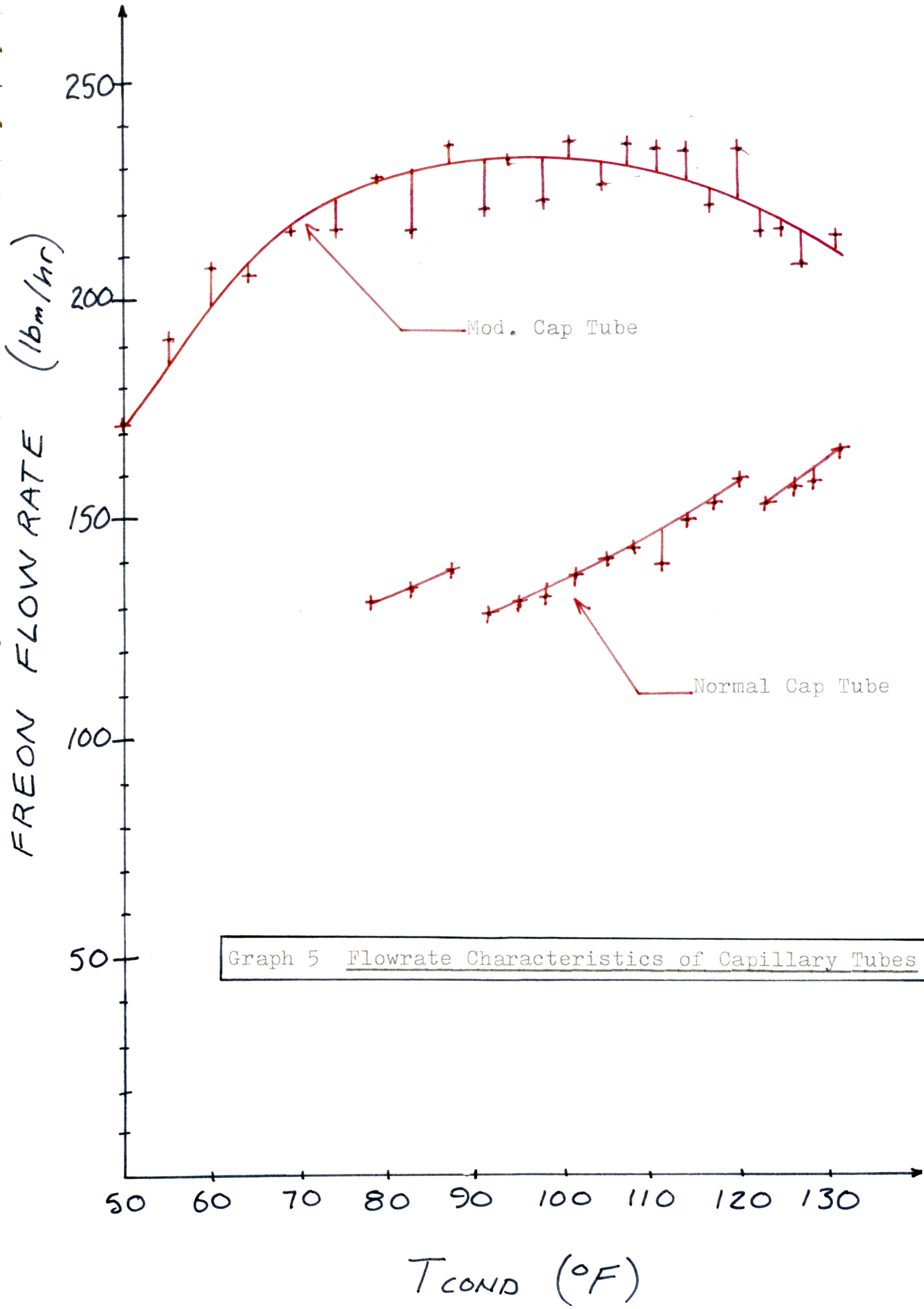
The annual cost of operating a 5-ton A/C system, based on the overall COP curve is shown in Graph 11.

Mass Flowrate

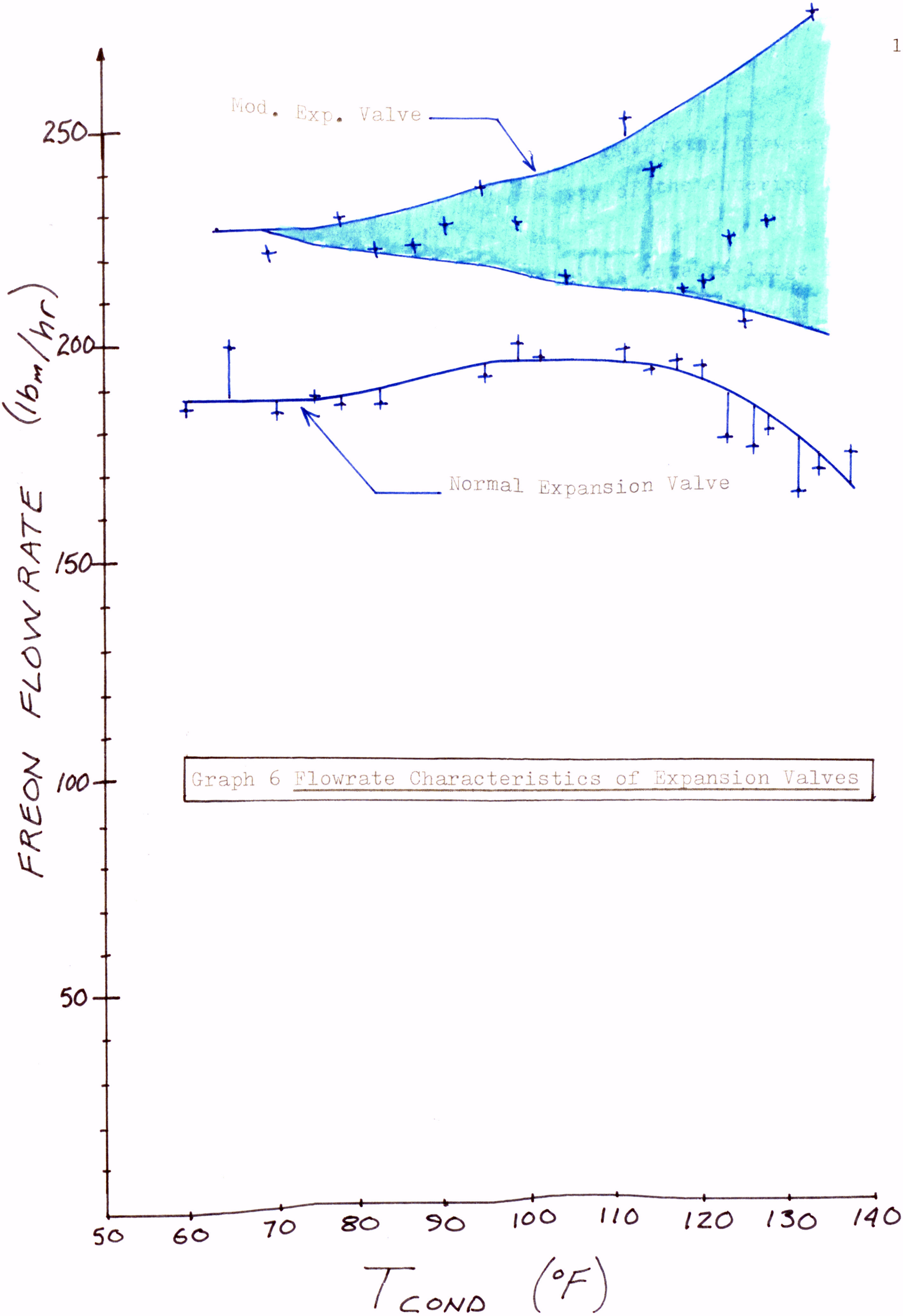
One may guess that the mass flowrate should increase as the condenser temperature drops since the compressor spins faster and the volumetric efficiency losses decrease. However, the starvation which occurs decreases the density of the fluid entering the compressor. The compressor should push through more volume per rate of time, but the decrease in density results in less mass flowrate. This is seen in Graphs 5 and 6. The decrease in mass flowrate that occured above T_C=115°F was caused by the fact that



Graph 4 Overall COP Curve



Graph 5 Flowrate Characteristics of Capillary Tubes



Graph 6 Flowrate Characteristics of Expansion Valves

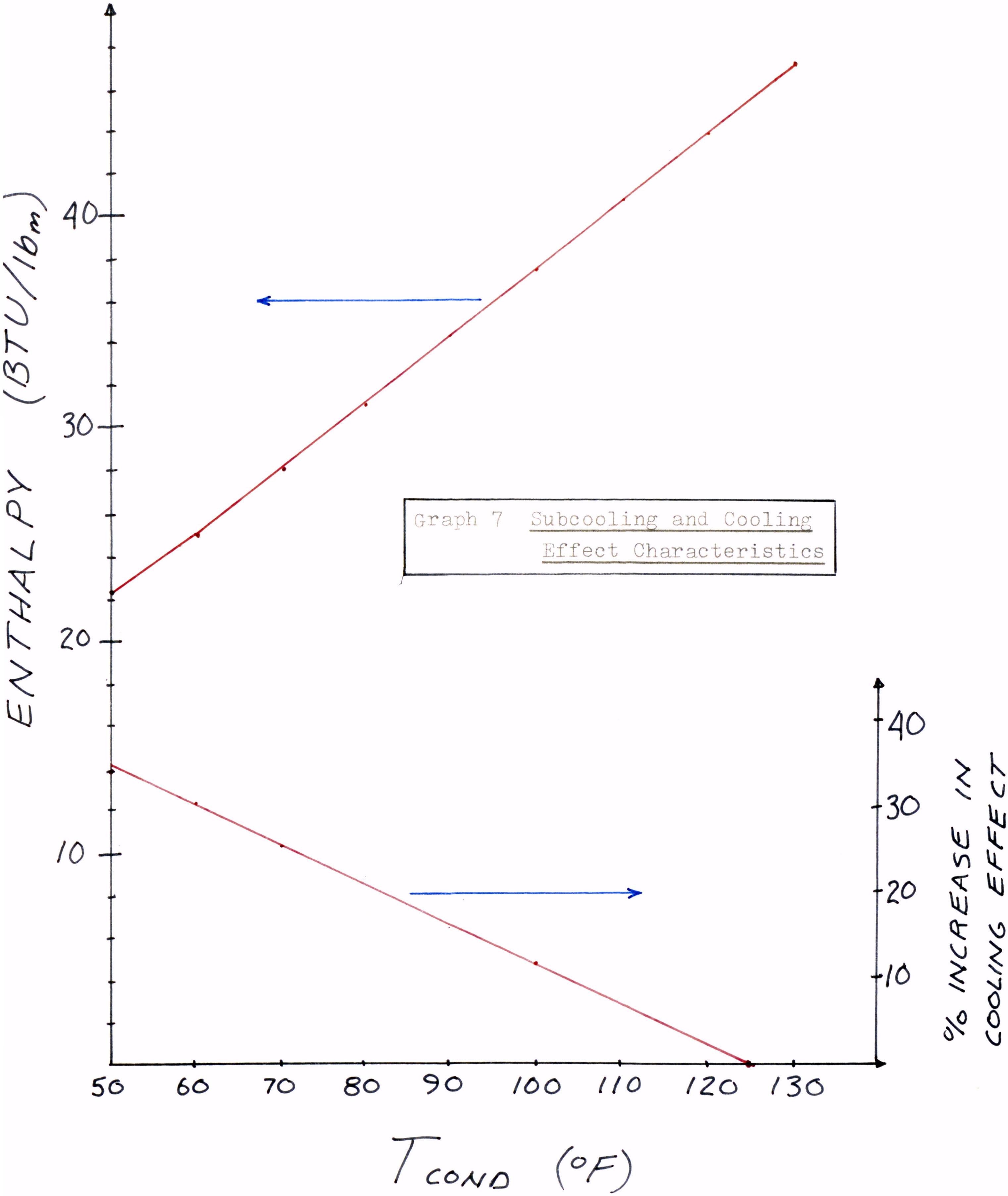
the compressor speed slowed since it is working between large pressure differences and the density of the entering fluid is high.

It is obvious that the skatter in data is large for the expansion valve tests. The expansion valve tends to "hunt" or oscillate between closed and opened during normal operation. The period of oscillation is approximately thirty seconds, but the pattern of change is not sinusoidal. To complicate matters, there is a phase shift between the patterns of each instrument reading. For this test, it was decided to gather each instrument's maximum reading during one oscillation period. As you can see by the data, this approach is limited. Discussions for improving the quality of the data are in the section entitled, "Suggestions for Future Tests."

Capacity

The capacity of any unit is related to the mass flowrate and the change in enthalpy of the refrigerant in the evaporator.

It has already been shown that for a normal capillary tube, the mass flowrate falls steadily. However, the enthalpy of the refrigerant entering the evaporator is lower (colder in a sense), therefore, more cooling effect is realized in evaporating the fluid. A plot of entering enthalpy vs. condensing temperature is shown on Graph 7.



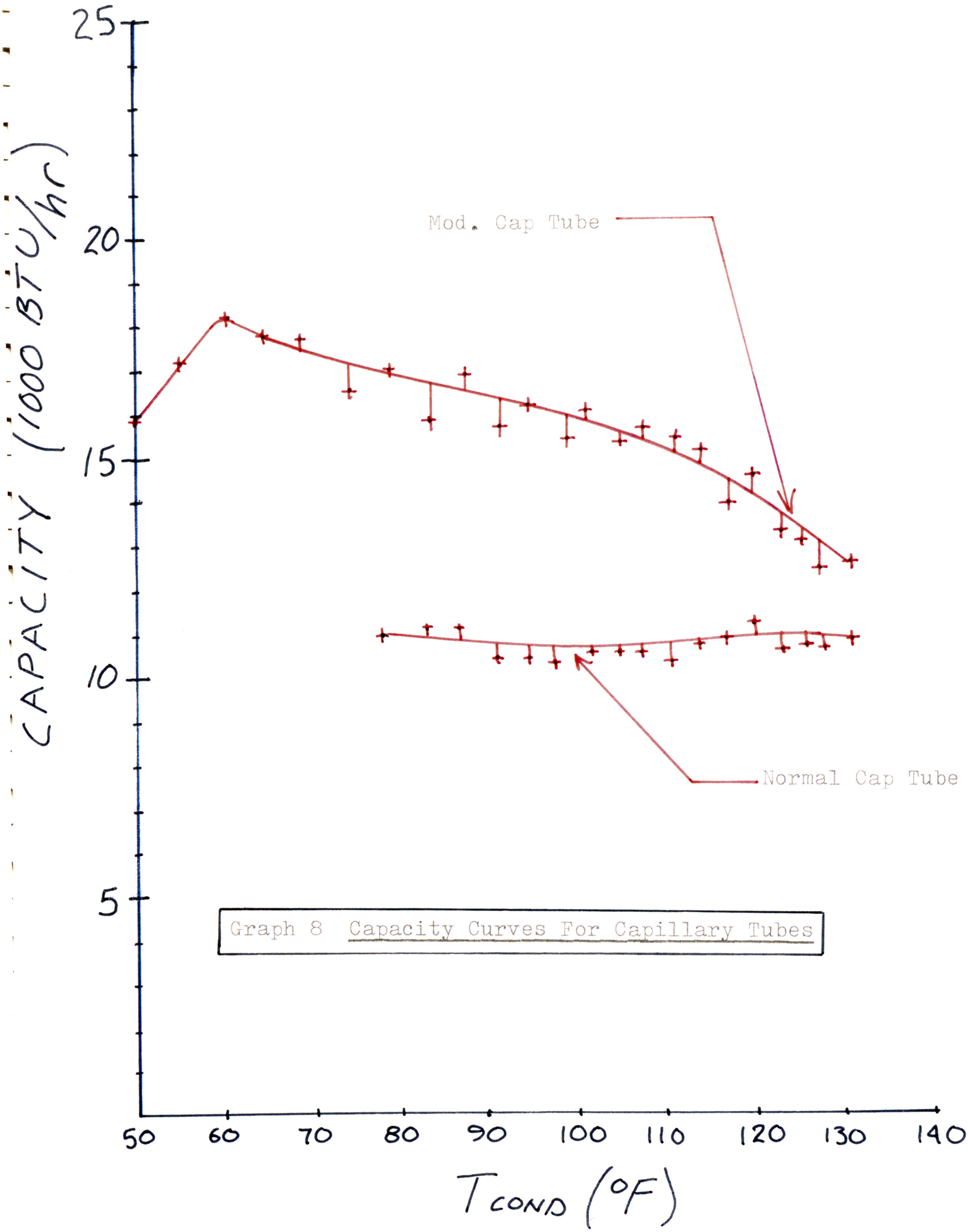
Graph 7 Subcooling and Cooling Effect Characteristics
ENTHALPY (BTU/lbm)
40
30
20
10
% INCREASE IN COOLING EFFECT
40
30
20
10
50
60
70
80
90
100
110
120
130
TCOND (°F)

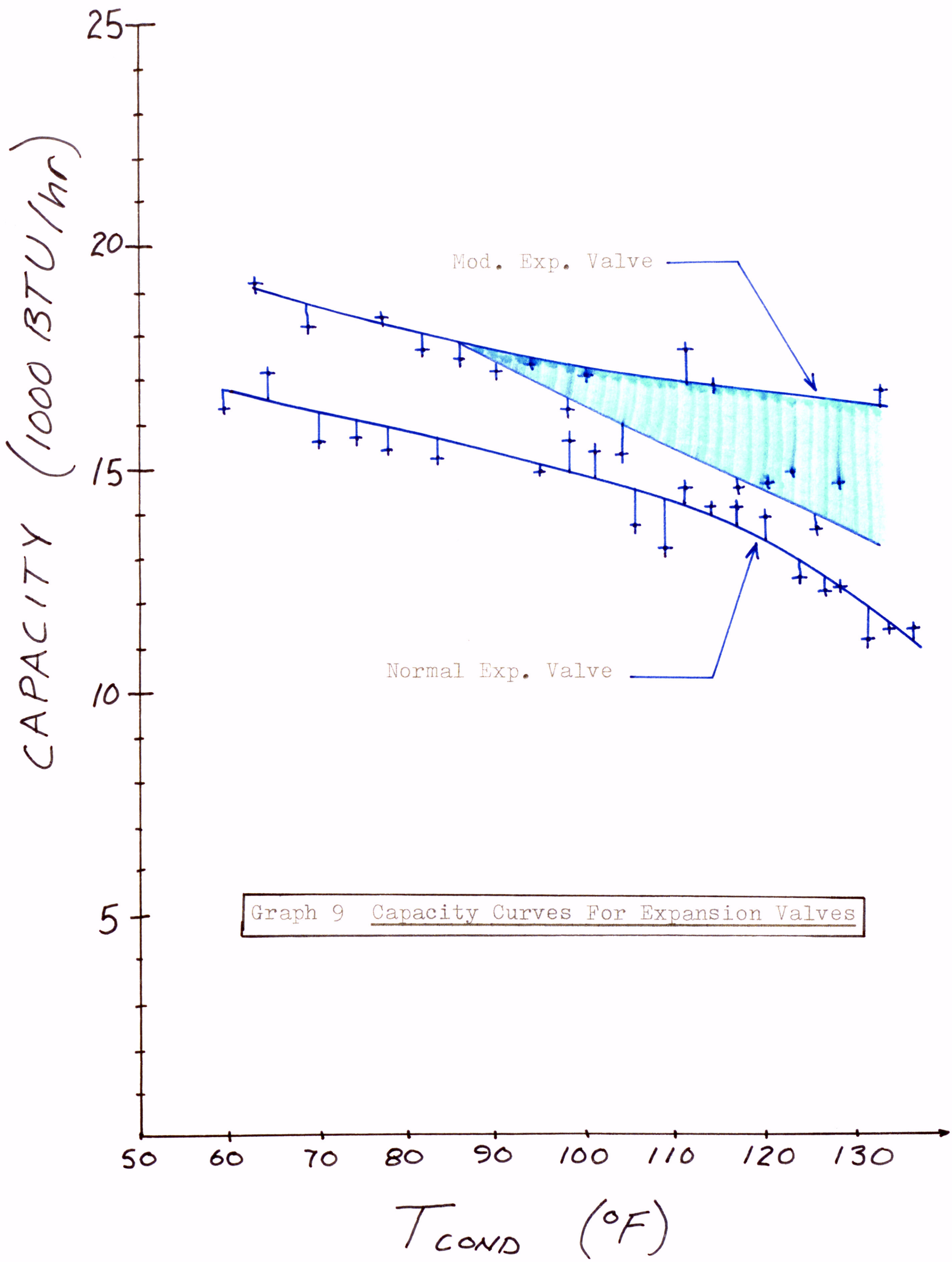
The bottom of this graph shows the percentage increase in cooling effect necessary to evaporate the fluid (per pound). The overall increase in cooling is equal to this times the flowrate. This graph applies to all four tests. For the normal capillary tube, the increase in cooling effect is offset by the decrease in mass flowrate, resulting in a flat capacity curve (Graph 8).

The modified capillary tube is able to show a significant increase in capacity due to the fact that the starvation is not occuring in the water-cooled range.

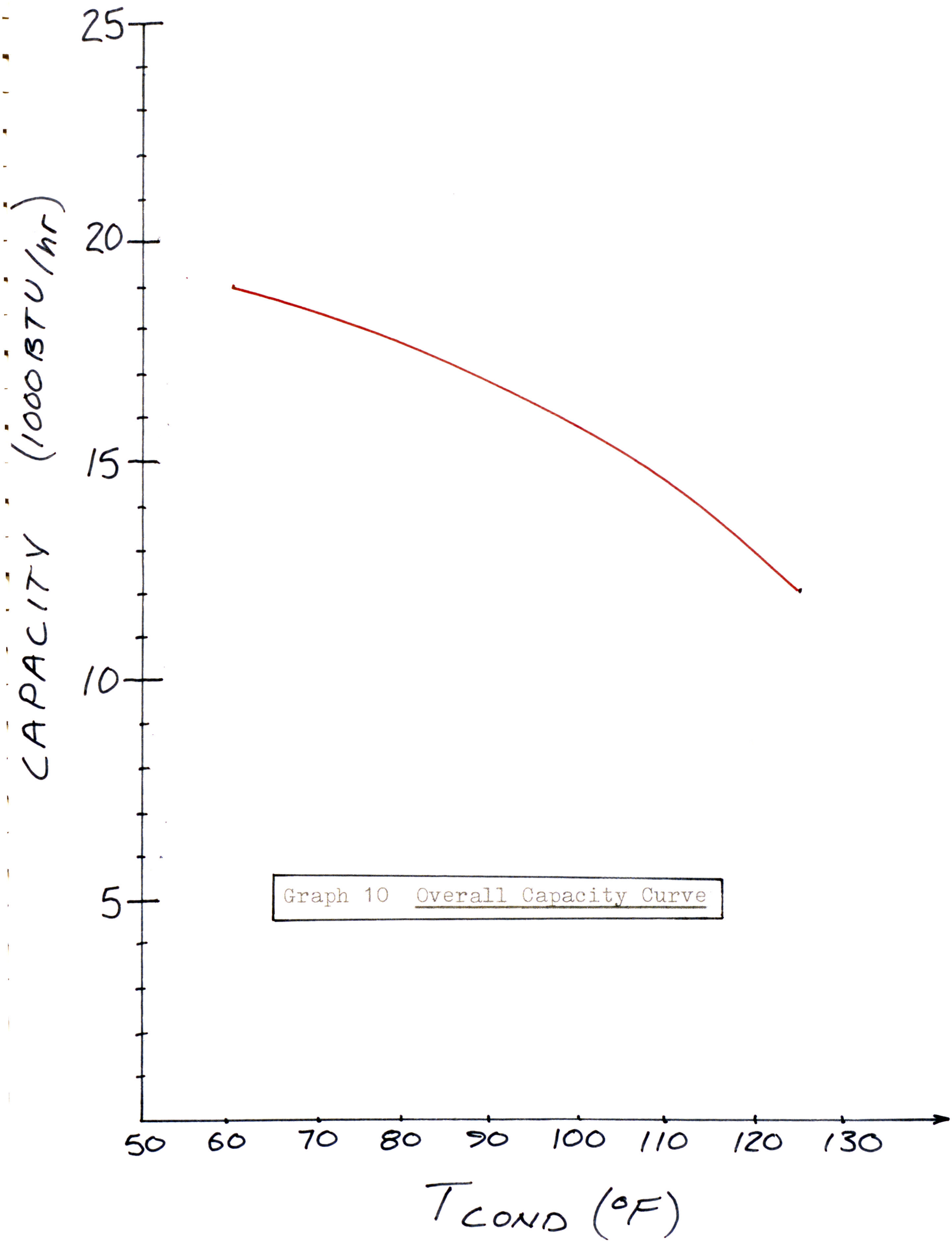
The normal expansion valve shows a good increase in capacity when water-cooled as seen in Graph 9. However, the modified expansion value performs better since no starvation occurs in the water-cooled range. Again, one may ask why does the modified capillary tube and expansion valve perform better than the normal equipment at air-cooled conditions. This is due to the fact that an abnormally high evaporator pressure allows an abnormally high flowrate, one should consider operating characteristics of the modified throttling equipment illegal above $T_C = 95°F$.

An ideal throttling device cooling capacity characteristic is shown in Graph 10. This graph shows the capacity that one may expect if the system is operating without starvation. This curve was derived from legal operating conditions of the four tests.

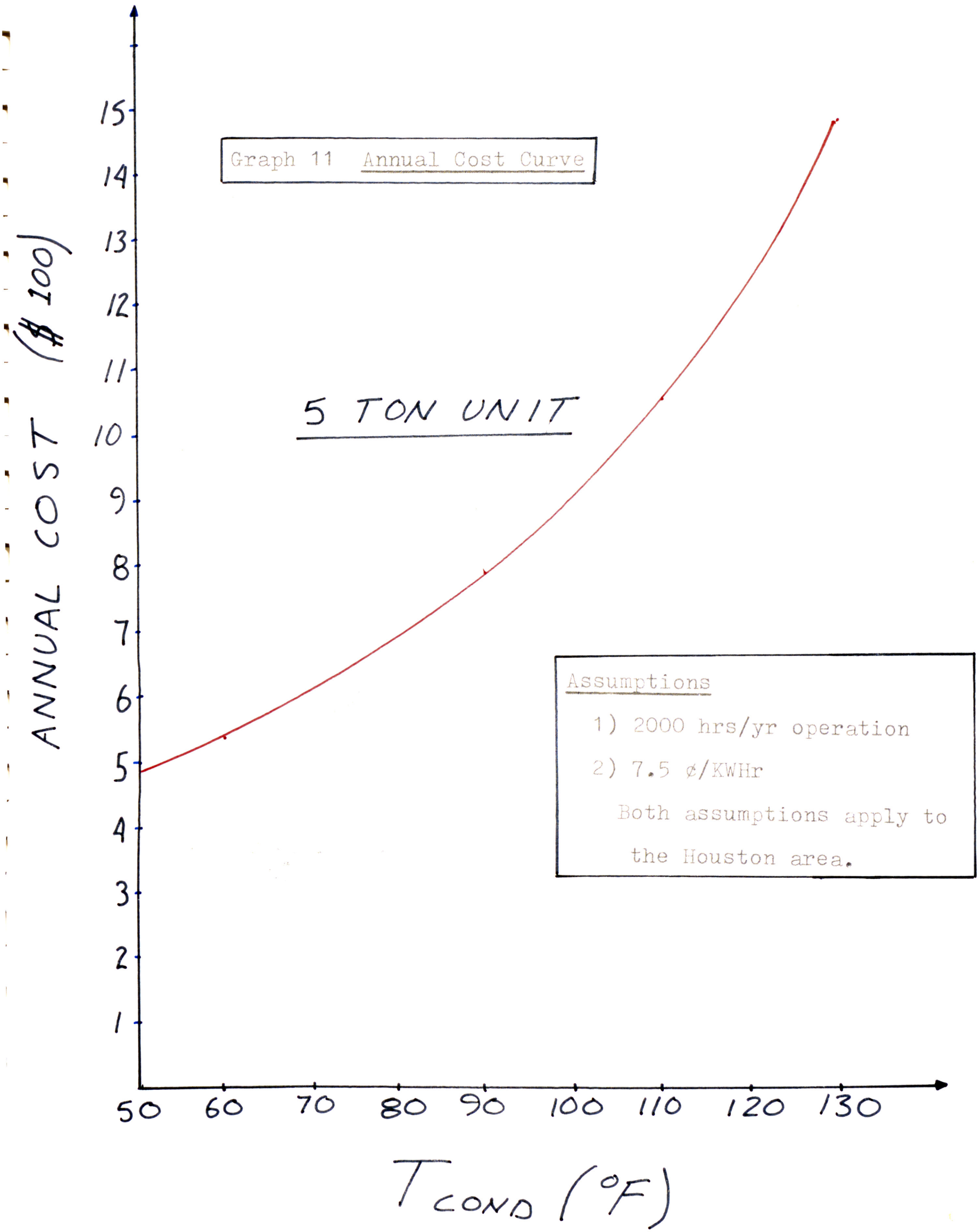

25
20
15
10
5
CAPACITY (1000 BTU/hr)
Mod. Cap Tube
Normal Cap Tube
Graph 8 Capacity Curves For Capillary Tubes
50
60
70
80
90
100
110
120
130
140
TCOND (°F)



Graph 9 Capacity Curves For Expansion Valves



Graph 10 Overall Capacity Curve



Graph 11 Annual Cost Curve
5 TON UNIT
ANNUAL COST ($ 100)
15
14
13
12
11
10
9
8
7
6
5
4
3
2
1
50 60 70 80 90 100 110 120 130
T_{COND} (°F)
Assumptions
1) 2000 hrs/yr operation
2) 7.5 ¢/KWHr
Both assumptions apply to the Houston area.

Power Factor

It was first thought that the reduced compressor load would cause the compressor motor P.f. to fall from unity, causing unnecessary current losses. However, the P.f. never fell below .98 for water-cooled operation.

SUMMARY OF RESULTS

1. Water-cooling a standard A/C system which uses a capillary tube results in an increase of COP of 29%, and no increase in capacity.

2. Modifying the throttling device by shortening the capillary tube or installing an expansion valve results in an increase in COP of 60% and an increase in system capacity of 36%.

3. Water-cooling a standard air-cooled A/C system which uses an expansion value increases the COP by 38% and increases the cooling capacity by 15%.

4. Modifying the expansion valve results in the performance listed in result number 2, above.

SUGGESTIONS FOR FUTURE TESTS

In testing a system such as an air conditioner where, when in stable operation, the variables measured oscillate, it is necessary to integrate the measured inputs over the entire cycle of oscillation. This would require, for this A/C system, the 14 pieces of data to be feed into a microcomputer using appropriate A/D converters. The computer would have to integrate the test data, search for enthalpies with the temperature and pressure information, and perform calculation of COP, $\dot{M}_R$, etc. This type of operation would cost approximately $15,000.00 and would be necessary if the scatter is to be eliminated.

A second important modification to this experiment would be to install a flowmeter to directly read the refrigerant flowrate. A third important modification needed to ensure system equilibrium is to increase the size of the storage drum to approximately 150 gallosn. This would result in a temperature rise rate of only .3°F per minute, ensuring an almost continuous equilibrium in the system.

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2. Wark, Thermodynamics, New York: McGraw-Hill, 1978.
3. Smith, Electric Devices and Systems, New York: McGraw-Hill, 1977.

APPENDIX

SAMPLE CALCULATIONS

RAW DATA

P_{E1} = 60.1 psig		h_{E1} = 26.3 BTU/lb_m	
P_{E2} = 43.8 psig		h_{E2} = 112.9 BTU/lb_m	
P_{C1} = 113 psig		h_{C1} = 127 BTU/lb_m	
P_{C2} = 110 psig		h_{C2} = 26.3 BTU/lb_m	
T_{E1} = 6°C		(Ref. A)	
T_{E2} = 15°C			
T_{C1} = 68°C			
T_{C2} = 14°C			
T_{W1} = .448 mV	(Ref. B)	52.3°F	
T_{W2} = .691 mV	(Ref. B)	63.3°F	
$\dot{M}_W$ = .0291 V	(Ref. C)	3.81 gal/min	
V = 214.4 V	x 1	214.4 Volts	
A = .587	x 10	5.87 Amps	
W = 61.8	x 20	1236 Watts	

References

Ref. A Thermodynamic Properties of Freon-22, E.I. duPont Company.

Ref. B Temperature Measurement Handbook, Omega Eng., Inc. Type-K Thermocouples.

Ref. C L. A. Hale, Calibration of Hersey Products Flowmeter with a D-to-A Converter, SW 3790.

Calculations

Energy Balance on Condenser:

$$\dot{M}_R(h_{C1}-h_{C2}) = \dot{M}_W C_P(T_{W2}-T_{W1}) \qquad \text{(Ref. 1)}$$

$$\dot{M}_R = \frac{\dot{M}_W C_P(T_{W2}-T_{W1})}{(h_{C1}-h_{C2})}$$

$$\dot{M}_R = \frac{(3.81\ \frac{gal}{min})(8.34\ \frac{lb_m}{gal})(\frac{60\ min}{1\ hr})(.9985\ \frac{BTU}{lbm°F})(63.3-52.3°F)}{(127\ -\ 26.3\ BTU/lb_m)}$$

$$\dot{M}_R = \underline{\underline{207.9}}\ lb_m/hr$$

$$Q_{EVAP} = \dot{M}_R(h_{E2}-h_{E1}) \qquad \text{(Ref. 1)}$$

$$Q_{EVAP} = (207.9\ bl_m/hr)(112.9-26.3\ BTU/lb)$$

$$Q_{EVAP} = \underline{\underline{18,004}}\ BTU/hr$$

$$COP = \underline{\underline{4.27}}$$

$$P.f. = \frac{WATT}{VI} = \frac{1236}{(214.4)(5.87)} = \underline{\underline{.982}} \qquad \text{(Ref. 2)}$$

Annual Cost

COP = 4.27 (EER 14.6 $\frac{BTU/hr}{WATT}$)

Running Time = 2000 hr/year (Ref. 2 ,Houston Area)

KWhr Cost = 7.5¢/KWhr (Houston, summer rates)

Assume a 5-ton system (60,000 BTU/hr):

$$\text{Cost} = \left(\frac{2000\ \text{hr}}{\text{yr}}\right)\left(\frac{\$.075}{\text{KWhr}}\right)\left(\frac{\text{hr Watt}}{14.6\ \text{BTU}}\right)\left(\frac{\text{KW}}{1000\ \text{WATTS}}\right)\left(\frac{60{,}000\ \text{BTU}}{\text{hr}}\right)$$

$$\text{Cost} = \underline{\underline{\$616.00}}$$