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Heat Pump Research Programme at University College Dublin (1976)

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HEAT PUMP RESEARCH PROGRAM AT UNIVERSITY COLLEGE DUBLIN

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contd.

1 INTRODUCTION

It is felt that a considerable disservice to the technology of heat pumps has been done by exaggerated performance claims which have appeared in the media from time to time. Even in the literature a wide range of C.O.P. values are quoted. The result is a general suspicion of the whole idea of heat pumps. Moreover, there is a distinct absence of data which would allow heat pump systems to be compared with conventional types. The probable net result is that heat pumps are not being given due consideration in situations where they could be of benefit.

On the subject of heat pumps, opinions seem to be either black or white while grey is unacceptable to either side. Those in favour of heat pumps point to boiler efficiencies of perhaps 50%. Those against heat pumps compare highly developed mass produced systems with heat pumps which have not been carefully designed for the particular circumstances of use.

It is felt that any proposed heat pump in an optimised form should only be compared with its likely competitors, i.e. existing or new systems which have been given the benefit of similar development.

The heat pump research being carried out at U.C.D. is only part of a program designed to improve the utilization of energy. The other main areas in this program are the improvement of burner efficiency for industrial boilers and the recovery of useful work from process steam.

2 OBJECT OF THE HEAT PUMP RESEARCH WORK BEING CARRIED OUT AT U.C.D.

In this project it is intended to evaluate heat pumps for the Irish climate, with particular emphasis on domestic and small scale commercial applications. It is hoped that the results of this project will provide a sound basis on which the economics of a heat pump in any domestic or commercial situation can be evaluated in the Irish context. An attempt will also be made to find the optimum heat pump system for Irish conditions.

The project work includes the following steps:-

1. A study of the literature is being undertaken to determine the present state of the art. This also includes collection of manufacturers' data on current heat pump equipment.
2. A Computer Program has been drawn up to simulate heat pump operation. This is being used to determine the sources of inefficiency in existing heat pump designs and will serve as the basis for component selection and optimisation. An engineering approach has been adopted in constructing the program; each component is represented by theoretical equations, using simplifying assumptions where possible in order to determine the main performance characteristics; the program will be verified by comparison with the measured performance of actual heat pump plants, discrepancies will be investigated and modifications made as required.
3. Tests are being carried out on an air to air heat pump plant to determine the actual performance under various weather conditions. These results will be used to verify the Computer Program and will also provide a record of the performance obtainable from a standard-type heat pump which has not been built specifically for Irish climatic conditions.
4. An investigation will be carried out to determine which refrigerant is most suitable for domestic and commercial applications in Ireland.

5. Using the simulation program, the components will be optimised and a Heat Pump system will be designed specifically for Irish conditions.
6. Having optimised the Heat Pump, weather data for a complete Irish heating season will be fed into the simulation program to determine the seasonal c.o.p. which can be expected.
7. An economic evaluation will be made to determine whether Heat Pumps can compete with other forms of domestic heating in Ireland. Use will be made of the results obtained from the simulation program and all relevant factors will be taken into account.

3 DESCRIPTION OF THE HEAT PUMP PLANT

The Heat Pump unit at U.C.D. is situated on the roof of the mechanical engineering building in Upper Merrion Street. It supplies heat to a large drawing office which occupies approximately half of the flat roof and is exposed on all sides.

The unit was assembled in Sweden by Klimatkyla AB. It is an air-to-air type with a rated power input of 3 kW and is designed to operate either as a heat pump or as an air conditioner. The changeover is accomplished by means of a four way reversing valve.

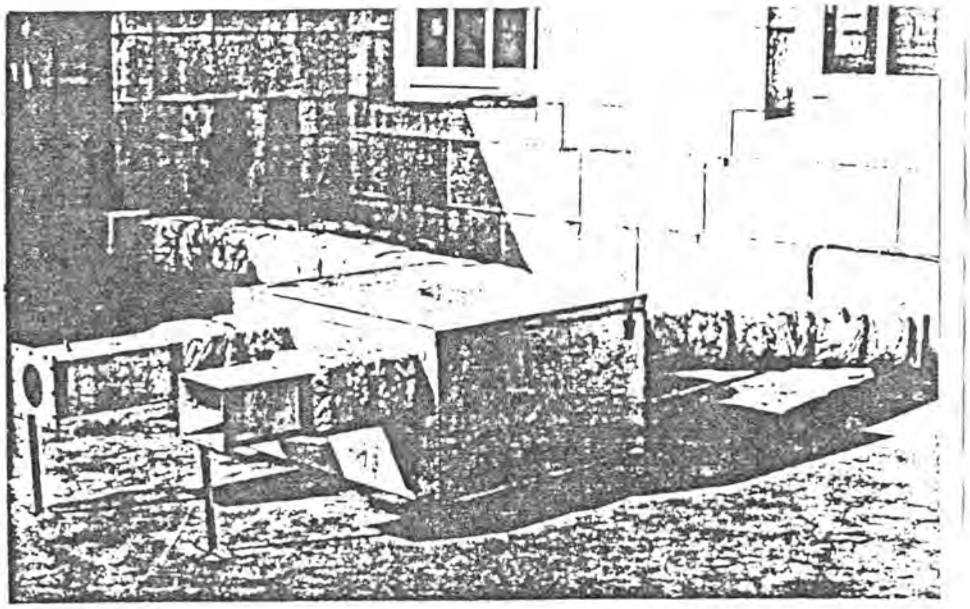


fig. 3-1 Heat Pump plant situated on the open roof outside the drawing office.

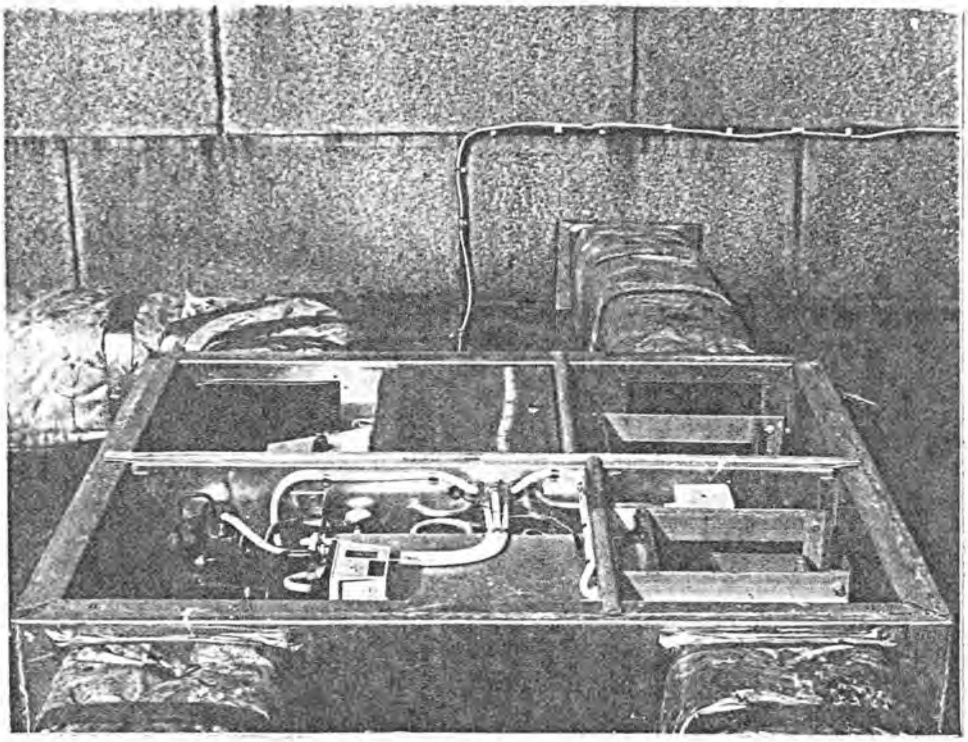


fig. 3-2 View of the plant with the cover removed.

3-1 Compressor:

This is a three phase "Tecumseh" hermetic compressor model AH5520E, designed for air conditioning and heat pump applications using refrigerant 22. It contains internal line-break motor thermal protection. An external motor protector is also fitted and a "Ranco" high pressure safety control cuts off the current to the compressor relay if the discharge pressure exceeds a preset limit. A wrap-around crankcase heater is fitted to prevent accumulation of liquid refrigerant while the compressor is not running.

The compressor is situated in the outside air compartment of the unit just in front of the evaporator.

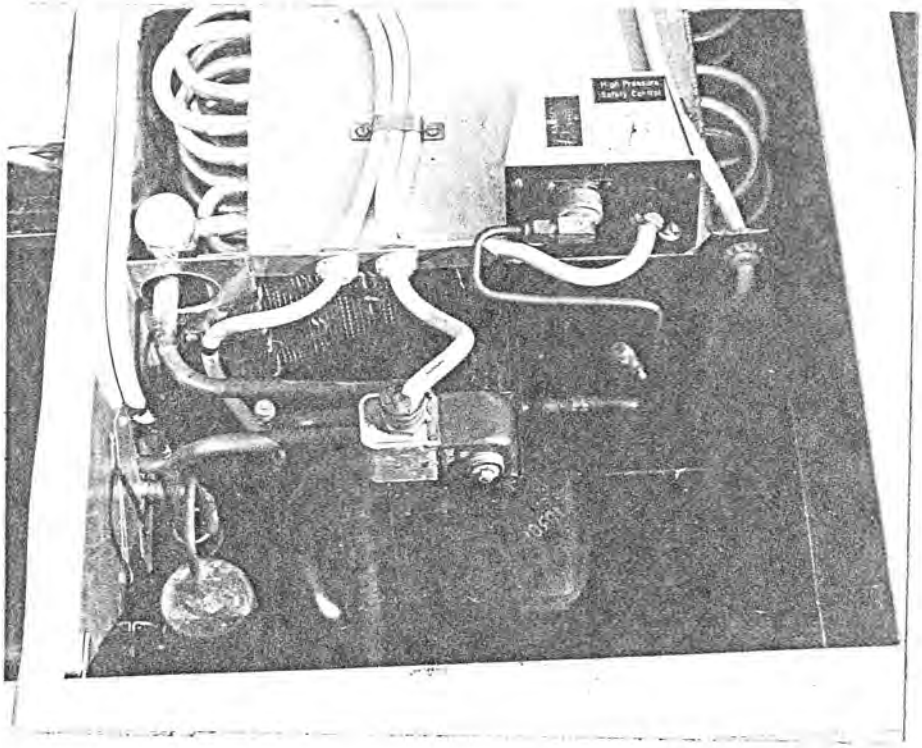


fig. 3-3 Outside air compartment.

On the lefthand side of the picture the insulated partition between the warm and cool sides of the unit is seen. The outside air intake duct is just visible on the right hand side. In the foreground is the compressor with the four way valve on top. In the background the high pressure safety control is positioned on top of the evaporator. In the bottom left hand corner part of the capillary can be seen behind the liquid receiver.

3-2 Evaporator and Condenser:

These are both three circuit units with horizontal $\frac{1}{2}$ inch O.D. tubes and vertical corrugated fins (approx. 6.3 fins/inch). The face dimensions of both are 12 inches wide by $13\frac{1}{2}$ inches high. The evaporator has 63 tubes and the condenser (being shorter) has 54. Beneath the evaporator is a tray containing the defrost heater. The room air passes through a filter before entering the condenser.

Expansion takes place through a capillary situated in the outside air compartment. A liquid receiver is joined in a T-connection to the refrigerant line from the condenser and this is also situated in the outside air compartment.

3-3 Electric Circuit:

The compressor and fans are three phase and there is a relay for each. These are situated in the control box on top of the condenser along with the compressor current protector and the defrost timer.

The plant was originally controlled by a thermostat in order that the compressor and fans would cut in or out in order to maintain a set room temperature. However, the heat requirement of the drawing office is much larger than the output of the plant. Thus the heat pump was unable to alter the room air temperature significantly and the compressor generally either ran continuously or remained stopped, depending on the thermostat setting. Moreover, heat and power measurements are complicated if the compressor cuts in and out at random intervals. For these reasons it was decided to eliminate the thermostat so that the compressor could be switched on or off manually. This also facilitated a change which was made to the defrost cycle (see below).

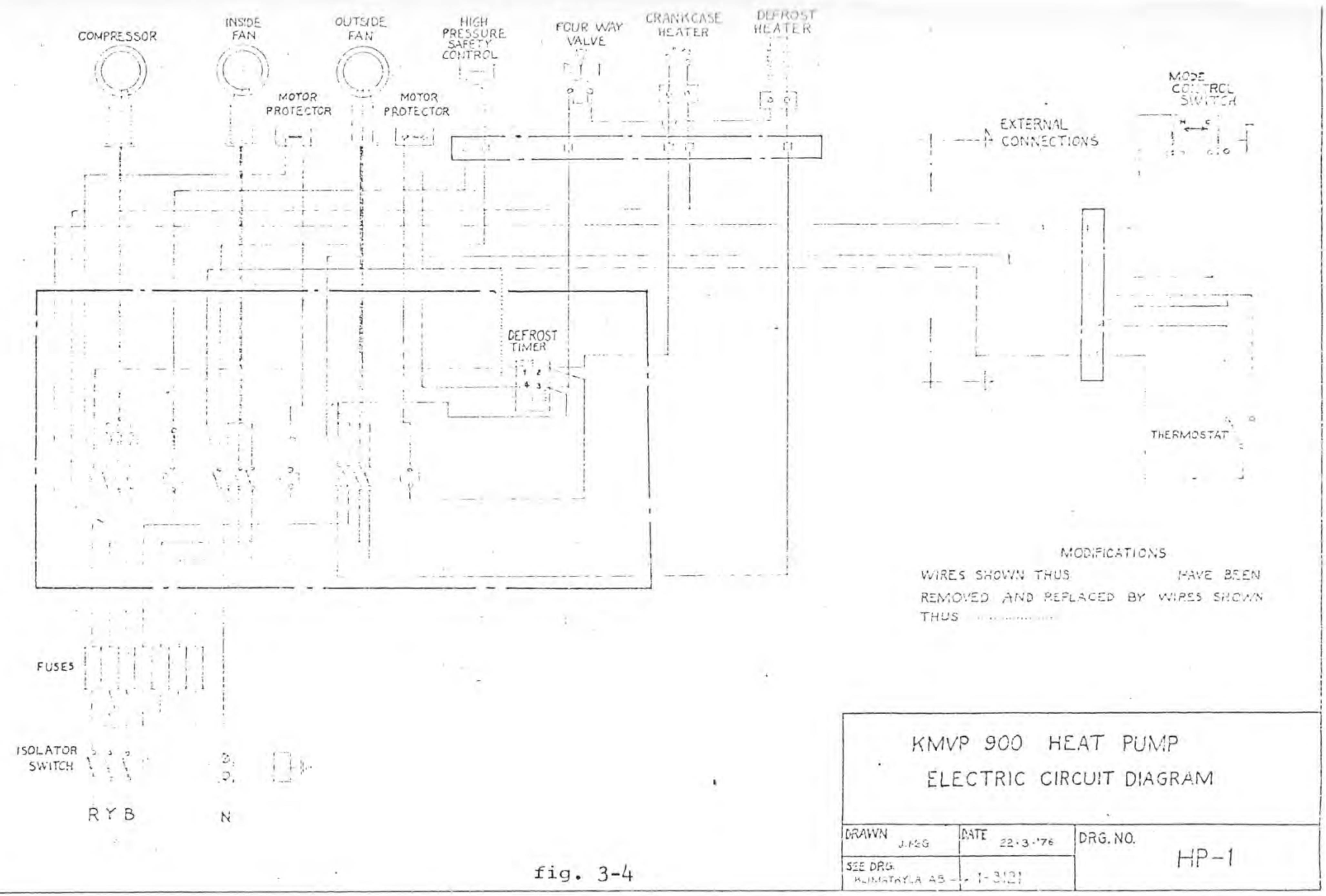

COMPRESSOR
INSIDE FAN
OUTSIDE FAN
HIGH PRESSURE SAFETY CONTROL
FOUR WAY VALVE
CRANKCASE HEATER
DEFROST HEATER
MOTOR PROTECTOR
MOTOR PROTECTOR
MODE CONTROL SWITCH
EXTERNAL CONNECTIONS
DEFROST TIMER
THERMOSTAT
MODIFICATIONS
WIRES SHOWN THUS HAVE BEEN REMOVED AND REPLACED BY WIRES SHOWN THUS
FUSES
ISOLATOR SWITCH
R Y B
N
KMVP 900 HEAT PUMP
ELECTRIC CIRCUIT DIAGRAM
DRAWN J.M.G
DATE 22·3·'76
DRG. NO.
HP-1
SEE DRG.
KLIMATKYLA AB - 1-3121

fig. 3-4

3-4 Defrost System:

Defrost is accomplished by means of a resistance heater beneath the evaporator and operates on a two hourly cycle. After each ninety minutes of running the defrost relay is energised and if the temperature sensing bulb in the evaporator reads a temperature below $0^{o}C$ the defrost cycle is brought into operation. The defrost cycle ends after thirty minutes or when the temperature rises above $0^{o}C$, whichever is the sooner.

The original mode of operation of the defrost system was to bring on the defrost heater and stop the outside fan while leaving the compressor and inside fan running. The objective seems to have been to continue to supply some heat to the room air during the defrost cycle However, this system did not succeed in defrosting the evaporator during the thirty minutes available. At the end of a day's running the evaporator was found to be fully iced up, the flow rate of the outside air was zero and the exit temperature of the air leaving the condenser was only slightly higher than the room temperature. Subsequently it was found that even on the first defrost cycle the evaporating temperature fell to $-30^{o}C$. This system of defrosting was clearly useless.

The changes which have been made are shown on the electric circuit diagram for the heat pump. Terminal 1 has been disconnected from the compressor relay and has been connected instead to terminal 3. Also, the compressor relay is now energized from terminal 4 instead of from the thermostat.

The results of the changes are:-

1. The unit can only operate as a heat pump.
2. The thermostat has been eliminated and the mode control switch acts as an ON/OFF switch.
3. When the control switch is in the ON position the compressor and fans operate continuously. If, during a thirty minute interval in every two hours, defrost is required, the defrost heater comes on and the compressor and fans stop.

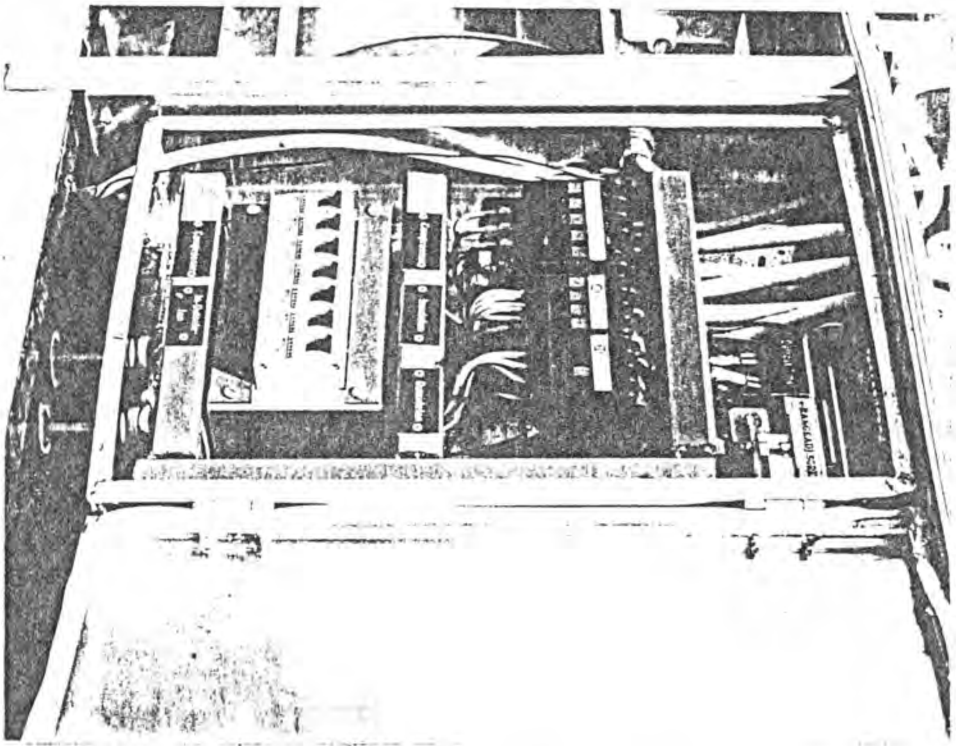


fig. 3-5 Control box.

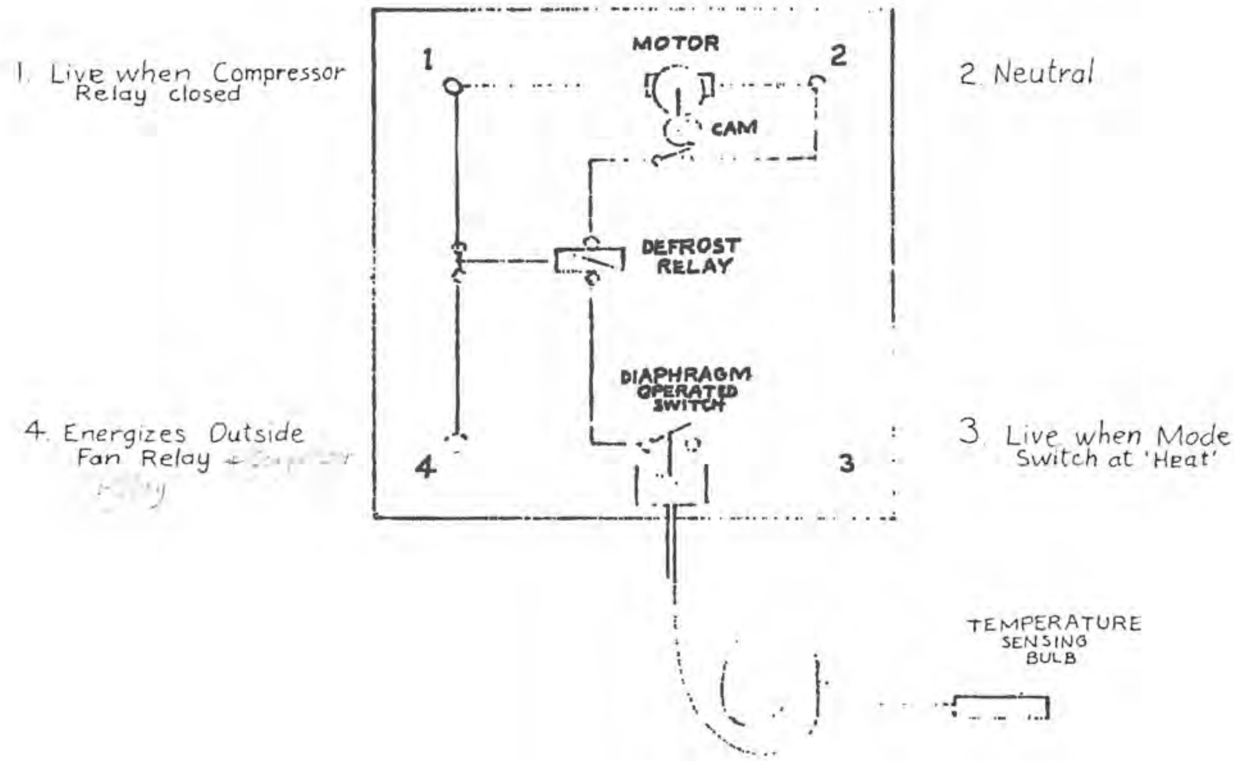


fig. 3-6 Schematic diagram of the defrost timer.

3-5 **Fans:**

There is one fan for the room air circuit and one for the outside air circuit. They are both backward bladed centrifugal and are immediately downstream of the condenser and evaporator respectively.

3-6 **Ducting:**

The ducting is 9 in. by 8 in. rectangular, fabricated in aluminium. It is insulated with 3 in. fibreglass wrapped with light polythene sheet.

By varying the position of a hinged damper, the condenser air can be drawn entirely from the room or from outside or can be a mixture of both.

4 INSTRUMENTATION

4-1 Electric Power Measurement:

A kW hr meter is fitted to the heat pump power supply cable. This is used to measure the total integrated power consumed by the unit in a given time interval. The smallest division on the scale is 0.01 kW hr.

A terminal board is situated inside the drawing office from which the compressor power can be measured. A multirange wattmeter is used for all instantaneous power measurements. It can be used for both single phase a.c. and for three-wire three phase a.c. with balanced loading.

4-2 Temperature Measurement:

A number of tests were carried out using four resistance thermometers and a multipoint temperature recorder. However, the particular instruments used were found to be inaccurate and very difficult to calibrate. Therefore, it was decided to use mercury-in-glass thermometers.

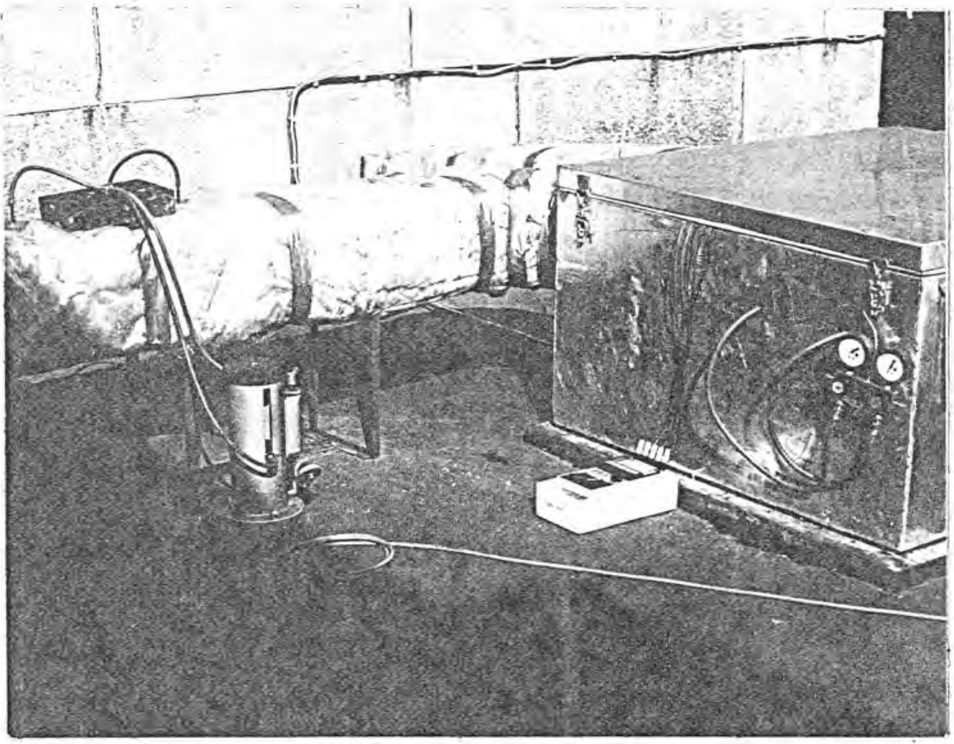
Both wet and dry bulb temperatures are taken for the outside air as it enters the heat pump and as it leaves. These four thermometers have a range from $-35^{\circ}C$ to $+50^{\circ}C$ and are graduated to $0.5^{\circ}C$.

For the room air only dry bulb temperatures are measured at entry and exit. These two thermometers have a range from $-10^{\circ}C$ to $110^{\circ}C$ and are graduated to $1.0^{\circ}C$.

The temperature of the refrigerant vapour leaving the compressor is measured by a thermocouple with the cold junctions in melting ice. The thermocouple is taped to the discharge line just as it leaves the compressor casing. With the thermocouple in place the discharge line is insulated with fibreglass.

4-3 Refrigerant Pressure Measurement:

Pressure tappings fitted with Schraeder valves are provided on the suction and discharge lines near the compressor. Flexible leads from a System Analyser are connected to these and the pressures are read on its two pressure gauges. These give the pressure in kg/sq.cm and also give the corresponding saturation temperatures for refrigerant 22 in deg. C.



Some of the measuring instruments.

The micromanometer below the duct is connected to the pressure tappings of the room air orifice. The rubber tube on the ground comes from the outside air orifice. The wattmeter and system analyser are also connected up to the heat pump unit.

4-4 Measurement of Air Flow Rates:

The flow rate of the room air is measured by a 6 inch ~~plate~~ orifice with D and D/2 taps, situated in the intake duct coming from the room. The flow rate of the outside air is measured by an open 6 inch orifice with a single D/2 tap. This is situated on the end of the intake duct.

The pressure differential is measured by a micromanometer reading to .0005 inches water gauge. A dial barometer is used to determine the atmospheric pressure.

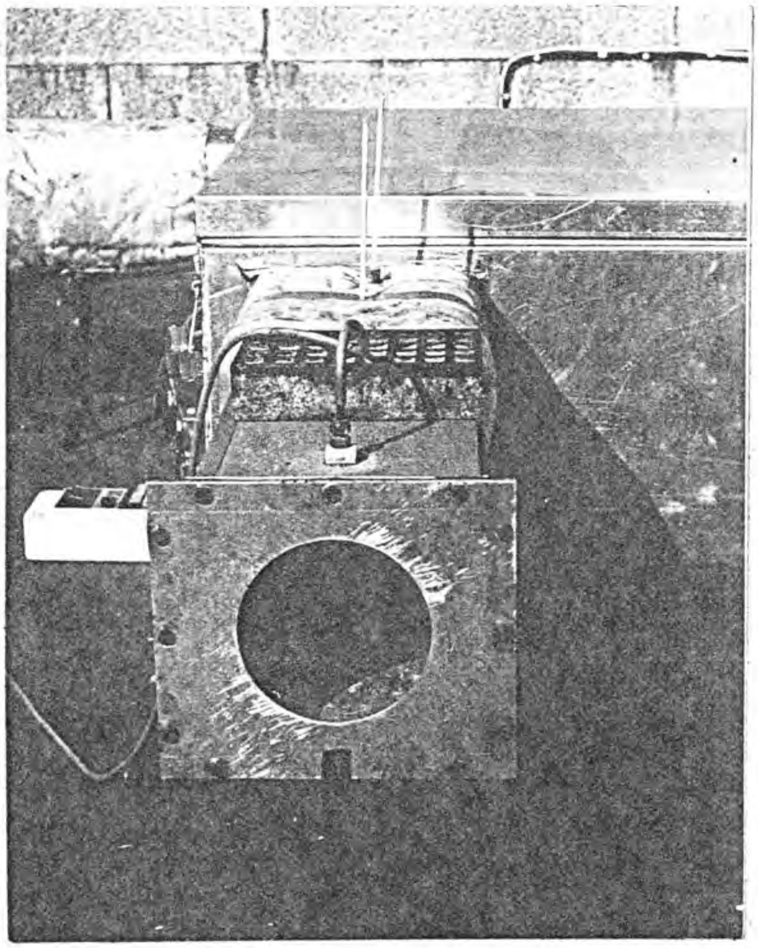


fig. 4-2 Outside air orifice with its pressure tapping.

The wet and dry bulb thermometers are also shown in position.

5 MEASURED PERFORMANCE DATA

Summary of Experimental Results for Tests carried out on 24th March, 1976.

1.	Test Number	1	2	3	4	5
2.	Starting Time	11.00	11.15	11.45	13.35	14.00
3.	Finishing Time	11.10	11.25	11.55	13.45	14.10
4.	Room Air Temperature °C	13.7	13.9	14.3	14.8	13.5
5.	Outside Air DBT °C	11.5	11.4	11.0	13.0	13.2
6.	Outside Air WBT °C	9.2	9.3	9.1	10.4	10.5
7.	Heat Output, kW	3.09	3.12	3.13	3.14	3.11
8.	Total Power Input, kW	1.83	1.80	1.80	1.96	1.89
9.	Coefficient of Performance	1.69	1.73	1.74	1.61	1.65
10.	Heat Removed from Outside Air, kW	2.34	2.27	2.22	2.41	1.87
11.	Latent/Total Heat Removed from Outside Air	.362	.376	.384	.443	.292
12.	Compressor Efficiency	.563	.625	.681	.509	.861
13.	Power Consumption of Fans, kW	.435	.371	.399	.446	.381
14.	Power Consumption of Defrost Heater, kW = .733					
15.	Measured Duration of Defrost Cycle, minutes	$8\frac{1}{2}$	$8\frac{1}{2}$	$8\frac{1}{2}$		
16.	Calculated Duration of Defrost Cycle, minutes	12.5	13.2	13.2		
17.	Evaporating Temperature °C	-6.6	-6.6	-6.6	-5.5	-6.0
18.	Condensing Temperature °C	37.5	37.5	38.5	40.0	38.0
19.	Refrigerant Temp. at Exit from Compressor °C	80.5	81.5	83.5	83.0	83.0
20.	Refrigerant Temp. at Exit from Evaporator °C	12.4	12.4	12.4	13.5	13.0

21. Difference between Condensing and Evaporating Saturation Temperatures °C	68.1	69.1	71.1	69.5	70.0
22. Difference between Room Air and Outside Air Temperatures °C	2.2	2.5	3.3	1.8	0.3
23. Evaporator Heat Balance heat to refrigerant/heat from air	1.26	1.29	1.32	1.22	1.61
24. Condenser Heat Balance heat from refrigerant/heat to air	1.15	1.14	1.11	1.18	1.21
25. Evaporator Heat Transfer Coefficient kW/°C	.154	.149	.149	.157	.116
26. Condenser Heat Transfer Coefficient kW/°C	.202	.209	.202	.191	.196

5-2 Notes on Experimental Results:

The heat pump is switched on and after it has been running for some time readings are taken during specified time intervals.

After the 90 minute cycle during which tests 1, 2 and 3 were made, a defrost cycle lasting $8\frac{1}{2}$ minutes occurred. After the 90 minute cycle during which tests 4 and 5 were made no defrost cycle was required. Therefore, in calculating the heat removed from the outside air the latent heat of fusion has been included for the first three tests. It has been assumed that for these three tests all of the moisture removed formed ice on the evaporator surfaces.

The figure given for compressor efficiency is the rate of heat transfer to the condenser air minus the rate of heat transfer from the evaporator air all divided by the power input to the compressor.

The calculated duration of the defrost cycle is the time it would take the defrost heater (.733 kW) to supply the latent heat of fusion of the moisture removed from the air if the measured test conditions had been sustained for 90 minutes.

The refrigerant temperatures at exit from the evaporator have been estimated from a previous test as a thermocouple was not fitted at this point for this particular set of tests.

For test 5 the "room air" was drawn from outside by changing the damper position on the intake duct.

Using the manufacturer's flow characteristic for the compressor and the measured refrigerant conditions the heat transfers to and from the refrigerant in the evaporator and condenser respectively have been calculated. Isenthalpic expansion has been assumed and, in the absence of an experimental reading, the refrigerant is assumed to be in the saturated liquid condition at entry to the capillary.

The heat transferred from the refrigerant does not balance with that transferred to the air in the condenser. Similarly there is an imbalance in the evaporator heat transfers. These figures suggest that the refrigerant flow rate is somewhat less than that predicted from the manufacturer's data. The greater heat imbalance in the evaporator may be due to the heat transfer from the compressor to the air which has been ignored.

The evaporator and condenser heat transfer coefficients are based on the heat transfer rates from and to the outside air and room air respectively.

6 DESCRIPTION OF THE HEAT PUMP SIMULATION PROGRAM

The basis of the heat pump simulation program is a theoretical refrigerant cycle for a particular refrigerant such as that illustrated on a T - s diagram and on a P-h diagram in fig. 6-1.

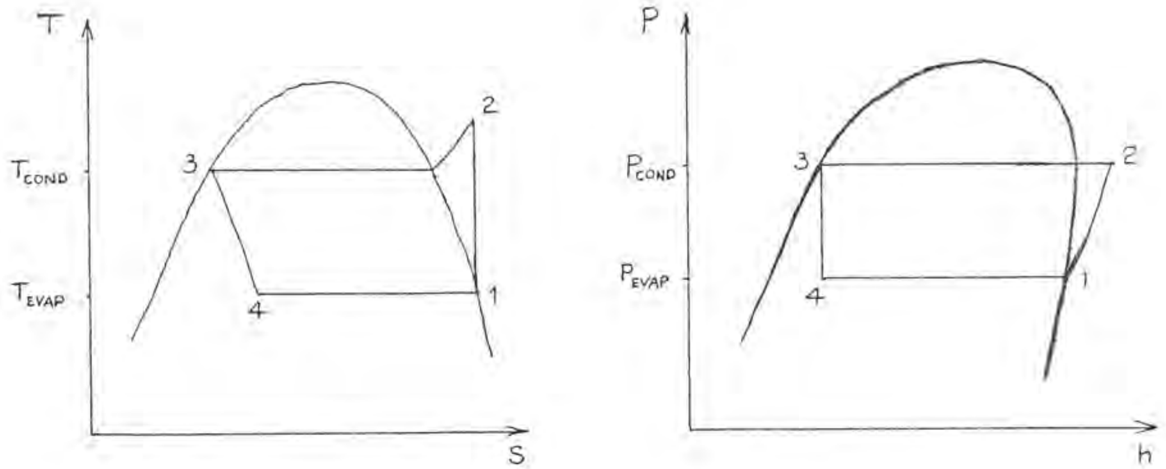


fig. 6-1 Refrigerant Cycle.

The cycle shown can be described as follows:-

1 ⟶ 2 Isentropic compression of the saturated vapour.

2 ⟶ 3 Cooling of the vapour followed by condensation - all at constant pressure.

3 ⟶ 4 Isenthalpic expansion of the saturated liquid.

4 ⟶ 1 Evaporation at constant pressure.

Empirical equations are available for the thermodynamic properties of a large number of refrigerants (ref. 1) and these are ideal for computer use. In the heat pump simulation a package of subprograms for the properties of refrigerants 12, 22 and 502, published by Kartsounes and Erth (ref. 2), has been used. Using these subprograms it is possible to simulate any refrigeration cycle which can be represented on a T - s diagram.

Besides the refrigerant cycle, the most important factors in determining the performance characteristics of the heat pump are the evaporator and condenser ratings and the flow characteristic of the compressor. In order to simulate a given heat pump, manufacturer's data or experimental data must be obtained for these components. In writing the simulation program the Klimatkyla air-to-air heat pump unit at U.C.D. has been the subject, but the resulting program is readily adaptable to any size or type of heat pump.

6-1 Evaporator and Condenser Simulation:

These are modelled as heat transfer surfaces between the refrigerant and the air. The mass flow rate and specific heat of the air are assumed to remain constant. For a given evaporator or condenser unit, the heat transfer equation can be written as

$$H = A\,U\,\Delta t$$

where H = kW transmitted

U = transmittance coefficient, $kW/m^2\,^oC$ this is the reciprocal of the sum of the thermal resistances of (i) the refrigerant film, (ii) the metal thickness and (iii) the air film.

Δt = mean temperature difference between the air and the refrigerant, ^oC.

A = effective surface area, m^2

For a given evaporator or condenser, U is assumed to remain constant over the working range and Δt is assigned the following value

$$\Delta t = T_{cond} - \left(\frac{T_{air_C} + T_{air_D}}{2}\right)$$

for the condenser

$$\text{or} \quad \Delta t = \left(\frac{T_{air_A} + T_{air_B}}{2}\right) - T_{evap}$$

for the evaporator

where the subscripts A, B, C and D refer to the positions in the air circuits shown in fig. 6-2

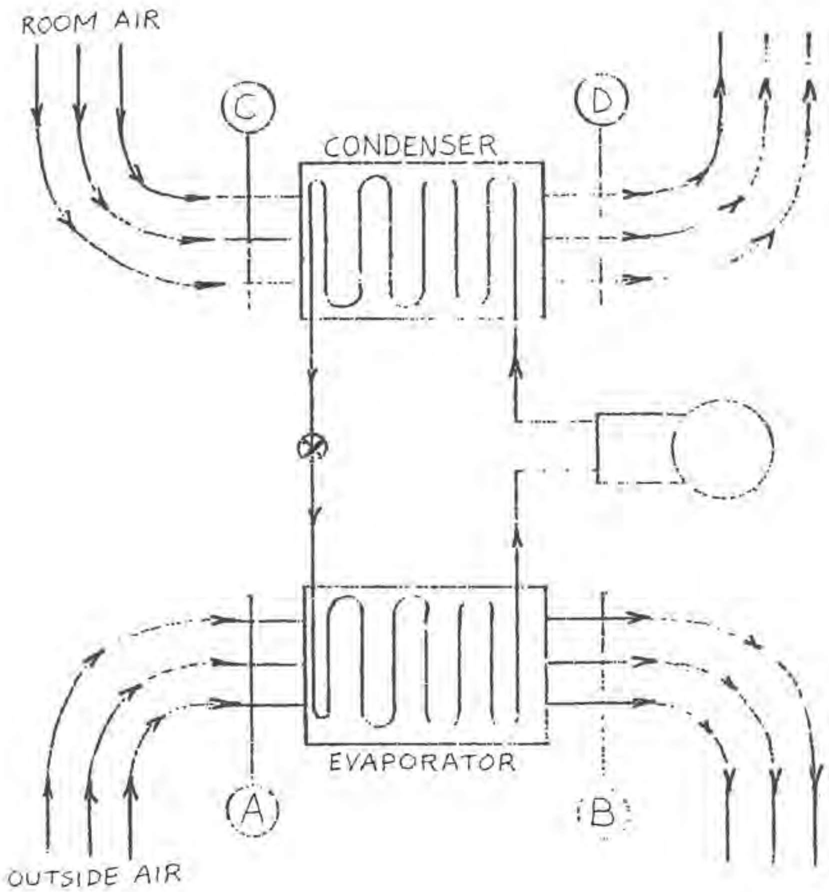


fig. 6-2

Thus, for a given evaporator or condenser the product A U is a constant. This is termed the heat transfer coefficient and has the units kW/^{o}C.

6-2 Compressor Flow Characteristic Simulation:

The mass flow rate of refrigerant through the compressor depends on:-

swept volume
speed
clearance
specific volume of vapour at inlet condition
specific volume of vapour at outlet condition

For a given compressor the swept volume and clearance are fixed, while the speed varies only slightly with load. The pressures at entry and exit are the evaporating and condensing pressures respectively, while the temperatures are the saturation temperatures plus the amount of superheat in each case. From refrigerant tables it is found that the variation in specific volume with amount of superheat is comparatively small.

Therefore, the refrigerant flow characteristic can be approximated by either a function of the saturation temperatures or of the saturation pressures.

The performance data supplied by Tecumseh for the compressor presents flow rate as a function of the saturation temperatures in graphical form. This was replotted as a function of the saturation pressures and the resulting curves were found to be more nearly linear. These are shown in fig. 6-3.

Straight line functions of P_{EVAP} were fitted to each of the four curves shown by the least squares technique. The slopes and intercepts were then expressed as a function of P_{COND} using the same technique.

$$\text{REFRIGERANT FLOW} = m\, P_{EVAP} + C$$

$$\text{where} \quad m = f\left(P_{COND}\right) \quad \text{linear}$$

$$\text{and} \quad c = f\left(P_{COND}\right) \quad \text{linear}$$

The resulting equation for the refrigerant mass flow rate is

$$REFFLO = COM_1 + COM_2 \times P_{EVAP} + COM_3 \times P_{COND} + COM_4 \times P_{EVAP} \times P_{COND}$$

where REFFLO is the refrigerant mass flow rate

COM_1 to COM_4 are the constants derived from the manufacturer's data

P_{EVAP} + P_{COND} are the saturation pressures in the evaporator and condenser respectively.

This equation is of the same form as that used by C.P. Crall (ref. 3) in the simulation of a reciprocating compressor for an air conditioning plant.

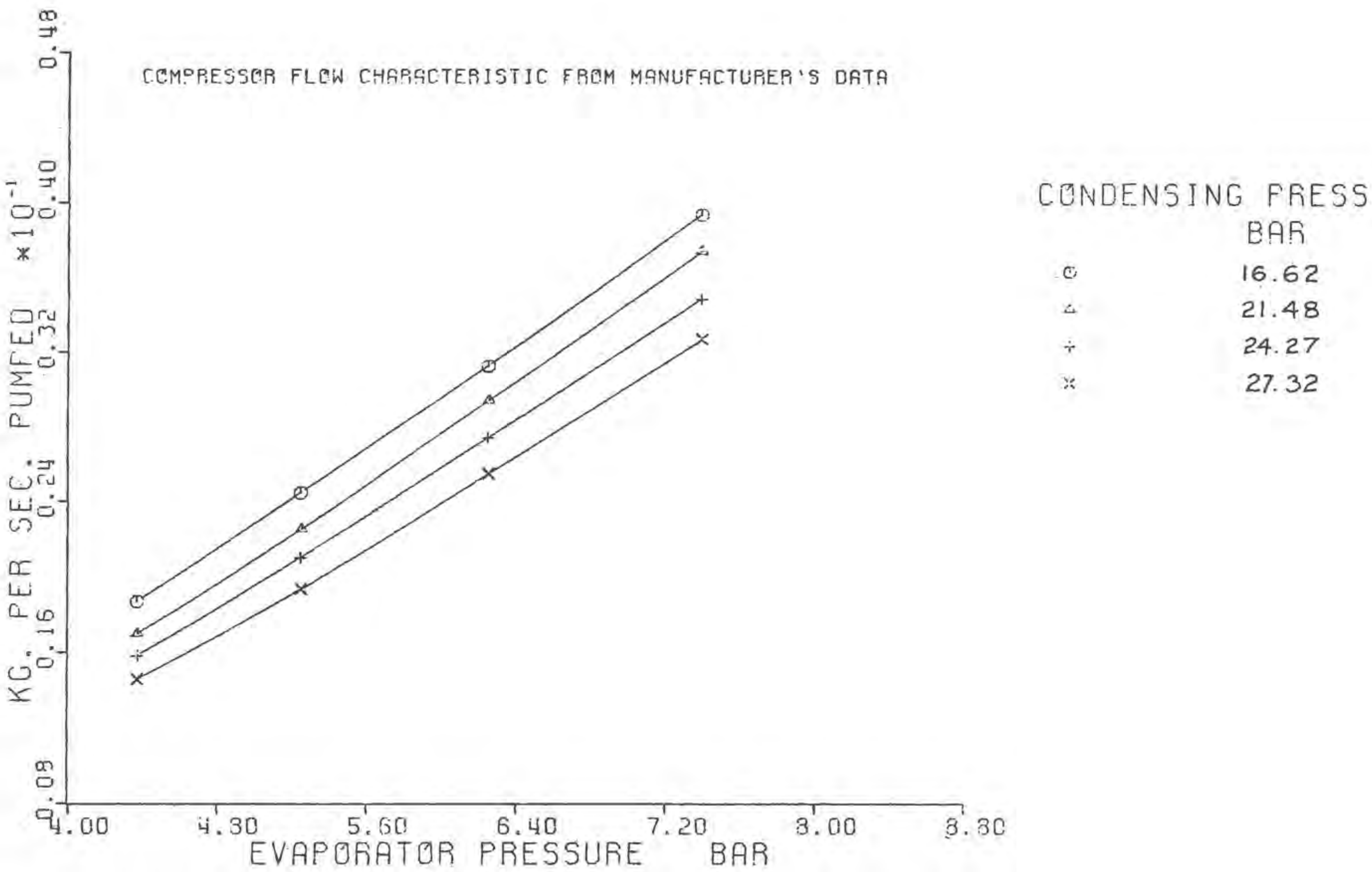


fig. 6-3 Mass flow rate of refrigerant 22 through Tecumseh compressor AH5520E as a function of saturation pressures at entry and exit.

6-3 Compressor Efficiency:

From the theoretical refrigerant cycle and the refrigerant mass flow rate, the indicated compressor power based on isentropic compression is found. The actual electric input to the compressor is calculated by dividing by the compressor efficiency.

$$\text{compressor efficiency} = \frac{\text{theoretical indicated power for isentropic compression}}{\text{measured electric power input}}$$

6-4 Power Consumption of Auxiliary Equipment:

The power consumptions of the inside and outside fan are assumed to remain constant. The only other item in the heat pump plant at U.C.D. which consumes power, while the unit is actually running in the heating mode, is a small crankcase heater which always remains on.

6-5 How the Simulation Program Operates:

1) Constants are read in for refrigerants 12, 22 and 502 which completely define their thermodynamic properties by means of the empirical equations in the property subprograms.
2) Data is read in to define the particular heat pump plant.
3) The operating conditions are read in. These consist of the outside air DBT and the room air DBT. The program has been written to give tabular results for a range of room and outside air temperatures.
4) The evaporating temperature is assumed by giving it a value below the outside air temperature.
   The condensing temperature is assumed by giving it a value higher than the room air temperature.
5) The resulting refrigerant flow rate is calculated.
6) The heat transfers in the evaporator and in the condenser are calculated.
7) For both the evaporator and the condenser two estimates of the exit air temperature are calculated. The first estimate is based on the required mean temperature difference between the refrigerant and the air. The

second estimate is based on the required temperature difference between the air at entry and at exit. If both estimates of the air exit temperature for both the evaporator and the condenser agree within a defined tolerance then the correct values of evaporating temperature (T_{EVAP}) and condensing temperature (T_{COND}) have been found for the current estimated refrigerant flow. If the tolerance on the air exit temperature is not met for both the evaporator and the condenser then new estimates are calculated for T_{EVAP} and T_{COND} and the iteration process is repeated from step 6).

8) The refrigerant flow rate is calculated for the current values of T_{EVAP} and T_{COND}.

9) If the difference between the calculated refrigerant flow rate and the estimated value is greater than a defined tolerance a new estimate of the refrigerant flow rate is calculated and the iteration process is repeated from step 6).
   If the tolerance is met then the correct values of T_{EVAP} and T_{COND} have been found.

10) The simulation results are now readily found from the parameters calculated during the iteration process.

6-6 Input Data:

1) Refrigerant Data -
   This is contained in a block data subprogram and so does not have to be read in each time the program is run.

2) Plant Data -
   Evaporator & Condenser
   Mass flow rate of outside air thru' evaporator, kg/s
   Mass flow rate of room air thru' condenser, kg/s
   Evaporator heat transfer coefficient, kW/^{o}C
   Condenser heat transfer coefficient, kW/^{o}C

Specific heat of air, kJ/kg ^{o}C
Inside fan power consumption kW
Outside fan power consumption kW

Compressor
Constants for flow characteristic
Compressor efficiency
Crankcase heater power consumption, kW

Refrigerant being used (12, 22 or 502)

3) Operating Conditions -
Outside air DBT ^{o}C
Room air DBT ^{o}C

6-7 Simulation Results:

1) Coefficient of performance: This is the heat output divided by the total electric power input.
2) Indicated C.O.P.: This is the heat output divided by the indicated compressor power.
3) Mass Flow Rate of Refrigerant, kg/s.
4) Temperature of Air Leaving Evaporator, ^{o}C.
5) Temperature of Air Leaving Condenser, ^{o}C.
6) Heat Transferred to Room Air, kW.
7) Heat Removed from Outside Air, kW.
8) Evaporating Temperature ^{o}C.
9) Condensing Temperature ^{o}C.
10) Mean Temperature Difference between Air and Refrigerant in the Evaporator, ^{o}C.
11) Mean Temperature Difference between Refrigerant and Air in Condenser, ^{o}C.

6-8 Performance of the Simulation Program to Date:

The simulation program described above is only a rough approximation to an actual heat pump plant. No provision has yet been made for superheating of the vapour entering the compressor or subcooling of the liquid leaving the condenser. The most serious omission is probably the fact that frosting of the evaporator and the latent heat involved are ignored. Nevertheless, the program is capable of illustrating the nature of the performance characteristics of actual heat pump plant. The process of gradually modifying the program to approximate the real system more closely will be continued until it is able to predict the measured performance of the real system.

The C.O.P. values being obtained with this program are already close to those obtained with the actual plant. It is clear that the main reason for the very large drop from the ideal Carnot C.O.P. between the outside air the room air temperatures and the C.O.P. of an actual plant is the additional temperature gradient required to transfer heat to and from the refrigerant.

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HEAT PUMP SIMULATION  -  IDEAL REFRIGERANT CYCLE COMBINED WITH
EVAPORATOR AND CONDENSER HEAT TRANSFER COEFFICIENTS, A COMPRESSOR
FLOW CHARACTERISTIC, FIXED POWER INPUTS FOR FANS ETC. AND A
CONSTANT COMPRESSOR EFFICIENCY.


REFRIGERANT CYCLE

1 - 2 ISENTROPIC COMPRESSION
2 - 3 COOLING OF SUPERHEATED VAPOUR AND CONDENSATION AT CONSTANT PRESSURE
3 - 4 ISENTHALPIC EXPANSION
4 - 1 EVAPORATION AT CONSTANT PRESSURE
POINTS 1 AND 3 ON SATURATION LINE


EVAPORATOR AND CONDENSER DATA

MASS FLOW RATE OF ROOM AIR THROUGH CONDENSER, KG/S          = 0.2880
MASS FLOW RATE OF OUTSIDE AIR THRU' EVAPORATOR, KG/S        = 0.4240
EVAPORATOR HEAT TRANSFER COEFFICIENT, KW/DEG. C             = 0.1908
CONDENSER HEAT TRANSFER COEFFICIENT, KW/DEG. C              = 0.2060
SPECIFIC HEAT OF AIR, KJ/(KG DEG. C)                        = 1.0120
INSIDE FAN POWER CONSUMPTION, KW                            = 0.2440
OUTSIDE FAN POWER CONSUMPTION, KW                           = 0.1910


COMPRESSOR DATA

CONSTANTS FOR COMPRESSOR FLOW CHARACTERISTIC
COM1 =-0.11247E-01    COM2 = 0.83191E-02
COM3 =-0.37107E-04    COM4 =-0.81069E-04

COMPRESSOR EFFICIENCY                                       = 0.4940

CRANKCASE HEATER POWER CONSUMPTION, KW.                     = 0.0685

REFRIGERANT  22
```

fig. 6-4 Data input to the simulation program.

COEFFICIENT OF PERFORMANCE

ROOM TEMP. DEG. C	OUTSIDE TEMP. DEG. C					
	-10.0	-5.0	0.0	5.0	10.0	15.0
10.0	3.2	3.2	3.2	3.3	3.3	3.3
15.0	2.8	2.9	2.9	2.9	3.0	3.0
20.0	2.6	2.6	2.7	2.7	2.7	2.7
25.0	2.4	2.4	2.4	2.5	2.5	2.5

INDICATED C.O.P. (BASED ON INDICATED COMPRESSOR POWER)

ROOM TEMP. DEG. C	OUTSIDE TEMP. DEG. C					
	-10.0	-5.0	0.0	5.0	10.0	15.0
10.0	6.5	6.5	6.6	6.7	6.7	6.8
15.0	5.8	5.9	6.0	6.0	6.1	6.1
20.0	5.3	5.3	5.4	5.5	5.5	5.6
25.0	4.8	4.9	4.9	5.0	5.1	5.1

HEAT TRANSFERED TO AIR IN CONDENSER KW

ROOM TEMP. DEG. C	OUTSIDE TEMP. DEG. C					
	-10.0	-5.0	0.0	5.0	10.0	15.0
10.0	1.61	2.00	2.40	2.81	3.21	3.62
15.0	1.58	1.97	2.36	2.76	3.16	3.57
20.0	1.54	1.93	2.32	2.71	3.11	3.51
25.0	1.50	1.88	2.26	2.65	3.04	3.44

EVAPORATING TEMPERATURE DEG. C

ROOM TEMP. DEG. C	OUTSIDE TEMP. DEG. C					
	-10.0	-5.0	0.0	5.0	10.0	15.0
10.0	-18.7	-15.9	-13.0	-10.3	-7.5	-4.8
15.0	-18.4	-15.4	-12.5	-9.8	-6.9	-4.1
20.0	-18.0	-15.0	-12.1	-9.2	-6.3	-3.4
25.0	-17.6	-14.6	-11.5	-8.6	-5.6	-2.7

CONDENSING TEMPERATURE DEG. C

ROOM TEMP. DEG. C	OUTSIDE TEMP. DEG. C					
	-10.0	-5.0	0.0	5.0	10.0	15.0
10.0	20.6	23.2	25.8	28.4	31.1	33.8
15.0	25.4	27.9	30.5	33.1	35.8	38.4
20.0	30.1	32.7	35.2	37.8	40.4	43.0
25.0	34.8	37.3	39.9	42.4	45.0	47.6

fig. 6-5 Some results of the simulation program.

By definition of the input parameters to the simulation program, defining the system, the performance predictions must correspond very closely to the measured performance at the operating conditions at which the input parameters were determined experimentally. The value of the program will lie in its ability to predict the performance at any other set of operating conditions and/or with system components of different ratings. This ability will depend on the validity of the assumptions made in constructing the simulation model and it is this which will determine the final model complexity.

The results shown in fig. 6-5 show surprisingly little variation in C.O.P. over a full range of operating conditions. This is due to aregulating effect of the heat transfer units and the compressor flow characteristic. If the evaporating and condensing temperatures are far apart the refrigerant flow rate is low and the rates of heat transfer through the evaporator and condenser are correspondingly small. Thus, the temperature gradients for heat transfer are small and the C.O.P. is relatively high. If, on the other hand, the evaporating and condensing temperatures are close together, the rates of heat transfer are large and the large gradients required tend to reduce the C.O.P.

The heat output characteristic of an air source heat pump is well illustrated by the results in fig. 6-5, the output being small when the outside air temperature is low.

7 THEORETICAL COMPARISON OF REFRIGERANTS 12, 22 and 502

The subprograms given in ref. 2 have been used to calculate refrigerant thermodynamic properties in order to compare the theoretical performance characteristics of the three refrigerants for a number of refrigeration cycles. The results have been output in both tabular and graphical form.

This comparison of refrigerants 12, 22 and 502 is still in progress and, as yet, no attempt has been made to determine which is the most suitable for any particular application. In this section the graphical results obtained to date are presented with little interpretation. It should be remembered that any comparison of refrigerants cannot be based on theoretical performance alone; more practical considerations such as oil solubility, cost, electrical properties and toxicity are often the deciding factors.

fig. 7-1: The Carnot coefficient of performance is the value obtained with a reversible cycle pumping heat from a source at a given low temperature to a sink at a given high temperature. No real cycle is reversible and so the C.O.P. is always less than the Carnot value.

figs. 7-2, 7-3, 7-4: For each refrigerant C.O.P. values are plotted for the reversed Rankine type cycle against evaporating and condensing temperatures. The effectiveness of these cycles can be judged by comparison with the curves plotted for the reversed Carnot cycle.

In this instance the condensing temperature is the lowest temperature at which heat is removed from the refrigerant and the evaporating temperature is the highest temperature at which heat is added to the refrigerant. In general, where the vapour is superheated before compression and the liquid is subcooled before expansion this will not be the case. Therefore, the evaporating and condensing temperatures of an actual refrigerant cycle are not strictly comparable with the low and high temperatures of the reversed Carnot cycle.

At an evaporating temperature of $-10^{\circ}C$ and a condensing temperature of $40^{\circ}C$ a C.O.P. of approximately 5.2 is obtained with refrigerant 12, while the Carnot C.O.P. between the same temperatures is approximately 6.3.

Refrigerants 12 and 22 give practically the same C.O.P. values. Those for refrigerant 502 are lower especially at high condensing temperatures.

<u>figs. 7-5, 7-6, 7-7</u>: Exit temperatures after isentropic compression from the saturated vapour condition are plotted for the three refrigerants. While a high exit temperature may have useful applications, such as the provision of hot water by means of a desuperheating heat exchanger, it has an adverse effect on compressor life (ref.5).

<u>figs. 7-8, 7-9, 7-10</u>: The compression ratio is the specific volume of the refrigerant vapour before compression divided by the specific volume after compression. This is an important factor in compressor selection and design.

<u>figs. 7-11, 7-12, 7-13</u>: C.O.P. values are plotted for a reversed Rankine type cycle with the liquid refrigerant subcooled by $10^{\circ}C$ before expansion. The increased C.O.P. can be seen by comparison with figs. 7-3 to 7-5.

<u>figs. 7-14, 7-15, 7-16</u>: C.O.P. values are plotted for a reversed Rankine type cycle with the refrigerant vapour superheated by $10^{\circ}C$ before compression. By comparison with figs. 7-3 to 7-5 the change in C.O.P. is seen to be insignificant except for refrigerant 502 with high condensing temperatures where the C.O.P. is slightly increased.

<u>figs. 7-17, 7-18, 7-19</u>: Exit temperatures after isentropic compression are plotted for refrigerant vapour superheated by $10^{\circ}C$. The effect of superheat at entry is to increase the exit temperature by a corresponding or slightly lesser amount.

fig. 7-20: Saturation pressure is plotted against temperature for each of the three refrigerants. It is seen that a system operating with refrigerant 12 operates at considerably lower pressure than one using refrigerants 22 or 502 between the same saturation temperatures.

Note:
An important parameter which has not yet been plotted is the specific volume of the vapour at entry to the compressor. This determines the swept volume of the compressor necessary to give the required heat output.

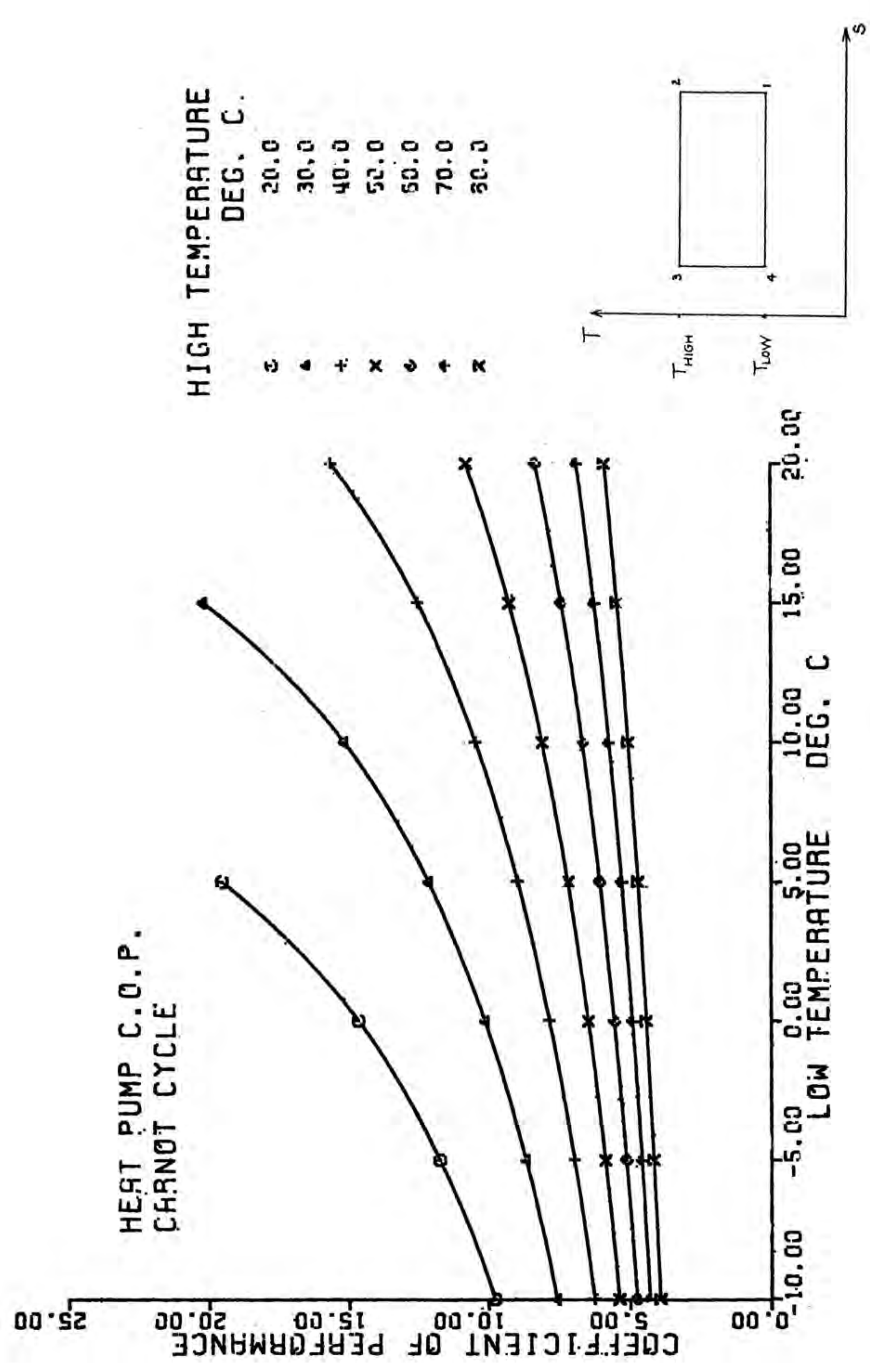

HEAT PUMP C.O.P.
CARNOT CYCLE
HIGH TEMPERATURE
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0
COEFFICIENT OF PERFORMANCE
0.00
5.00
10.00
15.00
20.00
25.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
LOW TEMPERATURE DEG. C
T
T_{HIGH}
T_{LOW}
S
1
2
3
4

fig. 7-1

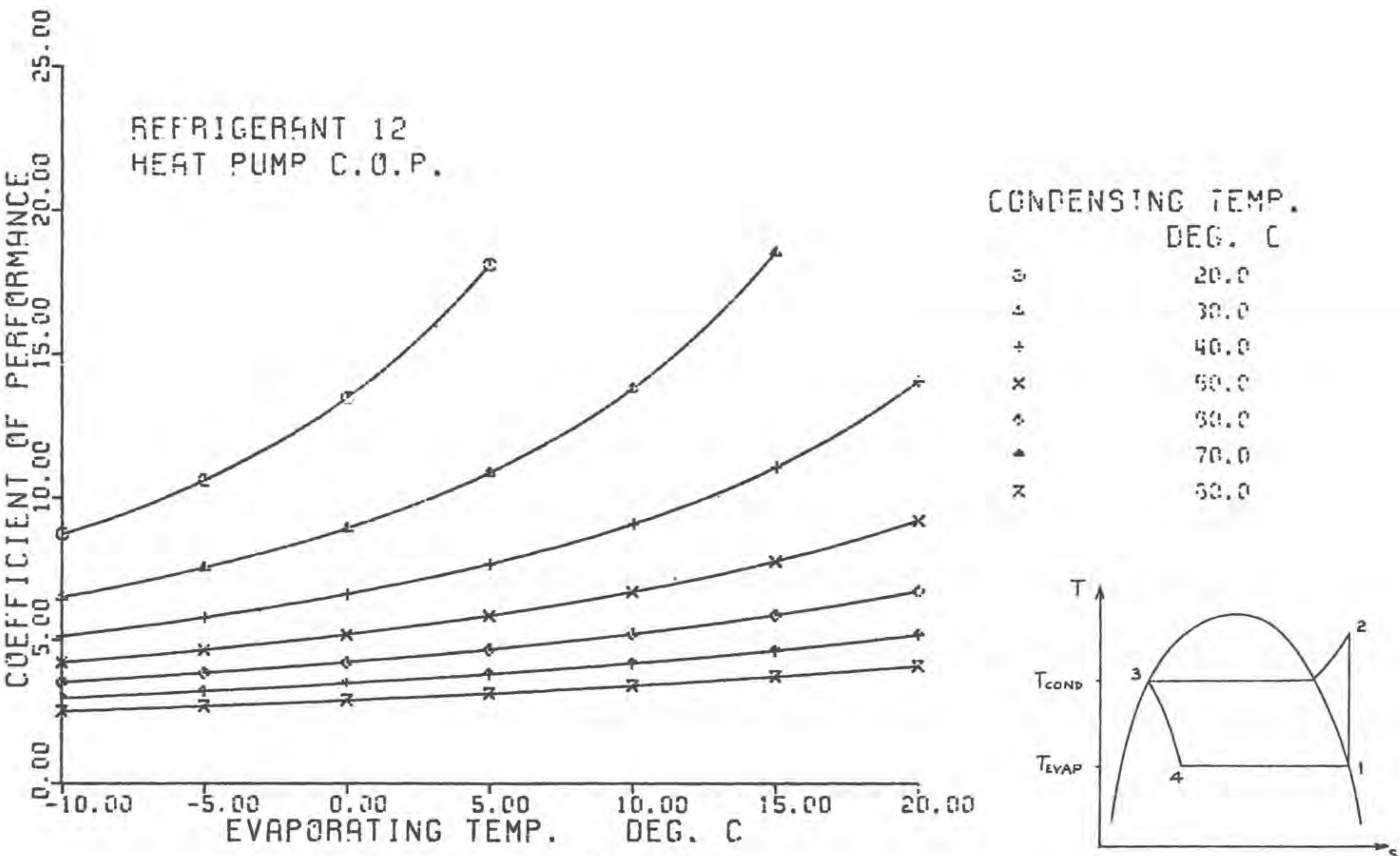

REFRIGERANT 12
HEAT PUMP C.O.P.
COEFFICIENT OF PERFORMANCE
0.00
5.00
10.00
15.00
20.00
25.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C
CONDENSING TEMP.
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0
T
S
T_{COND}
T_{EVAP}
1
2
3
4

fig. 7-2

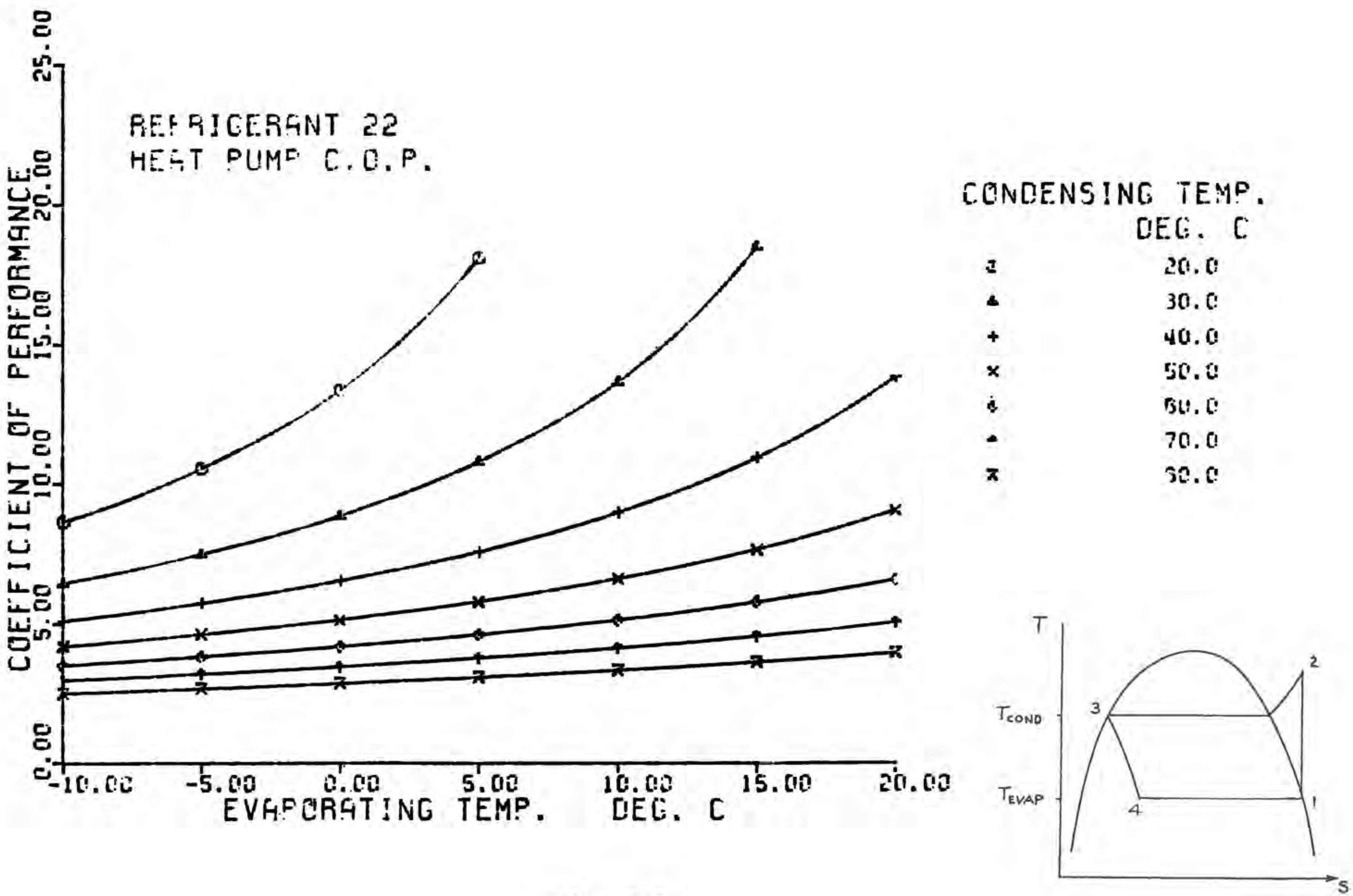

REFRIGERANT 22
HEAT PUMP C.O.P.
COEFFICIENT OF PERFORMANCE
0.00
5.00
10.00
15.00
20.00
25.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C
CONDENSING TEMP.
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0
T
T_{COND}
T_{EVAP}
1
2
3
4
S

fig. 7-3

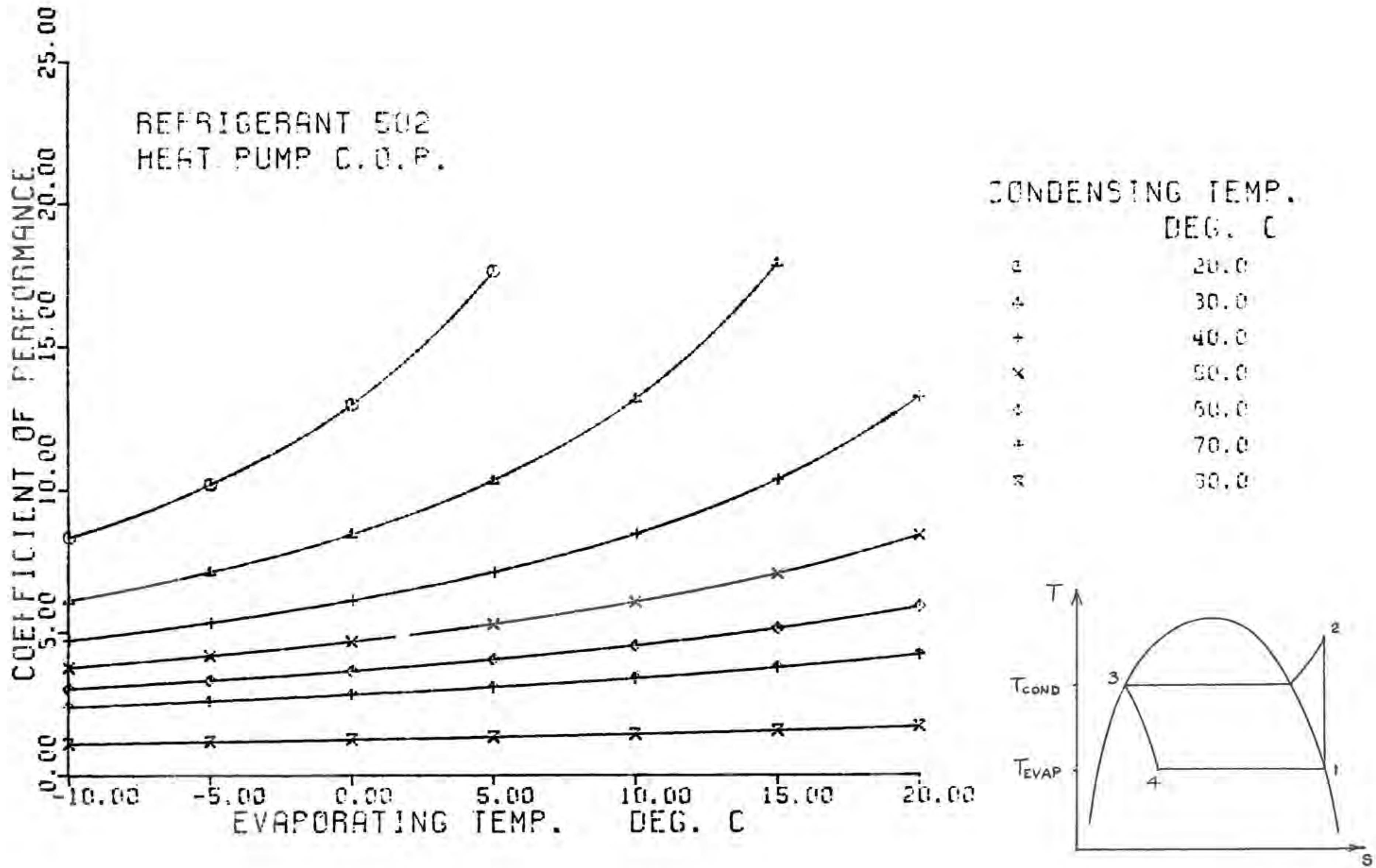

REFRIGERANT 502
HEAT PUMP C.O.P.
COEFFICIENT OF PERFORMANCE
0.00
5.00
10.00
15.00
20.00
25.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C
CONDENSING TEMP. DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0
T
T_{COND}
T_{EVAP}
1
2
3
4
S

fig. 7-4

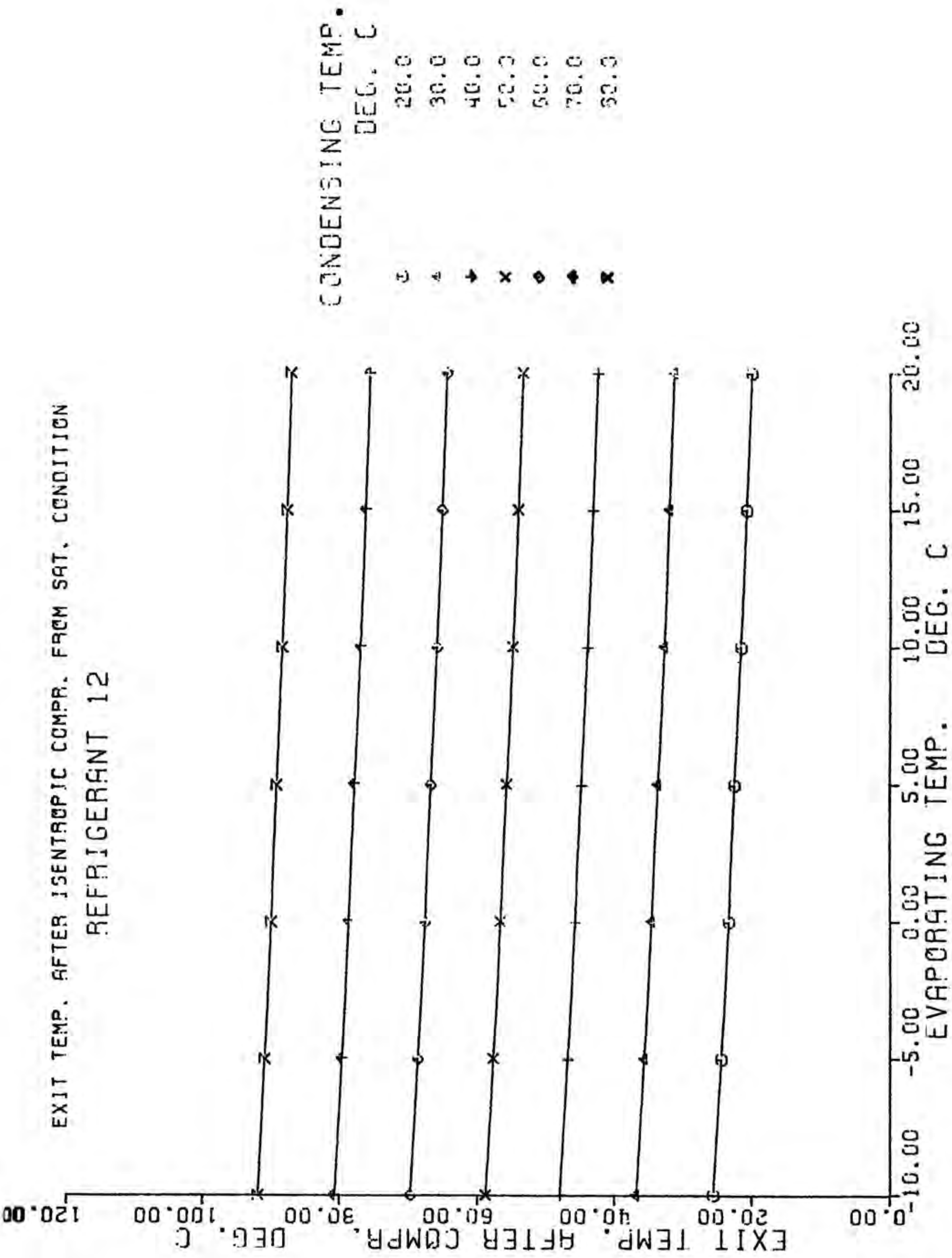

EXIT TEMP. AFTER ISENTROPIC COMPR. FROM SAT. CONDITION
REFRIGERANT 12
CONDENSING TEMP.
DEG. C
20.0
30.0
40.0
EXIT TEMP. AFTER COMPR. DEG. C
0.00
20.00
40.00
60.00
80.00
100.00
120.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C

fig. 7-5

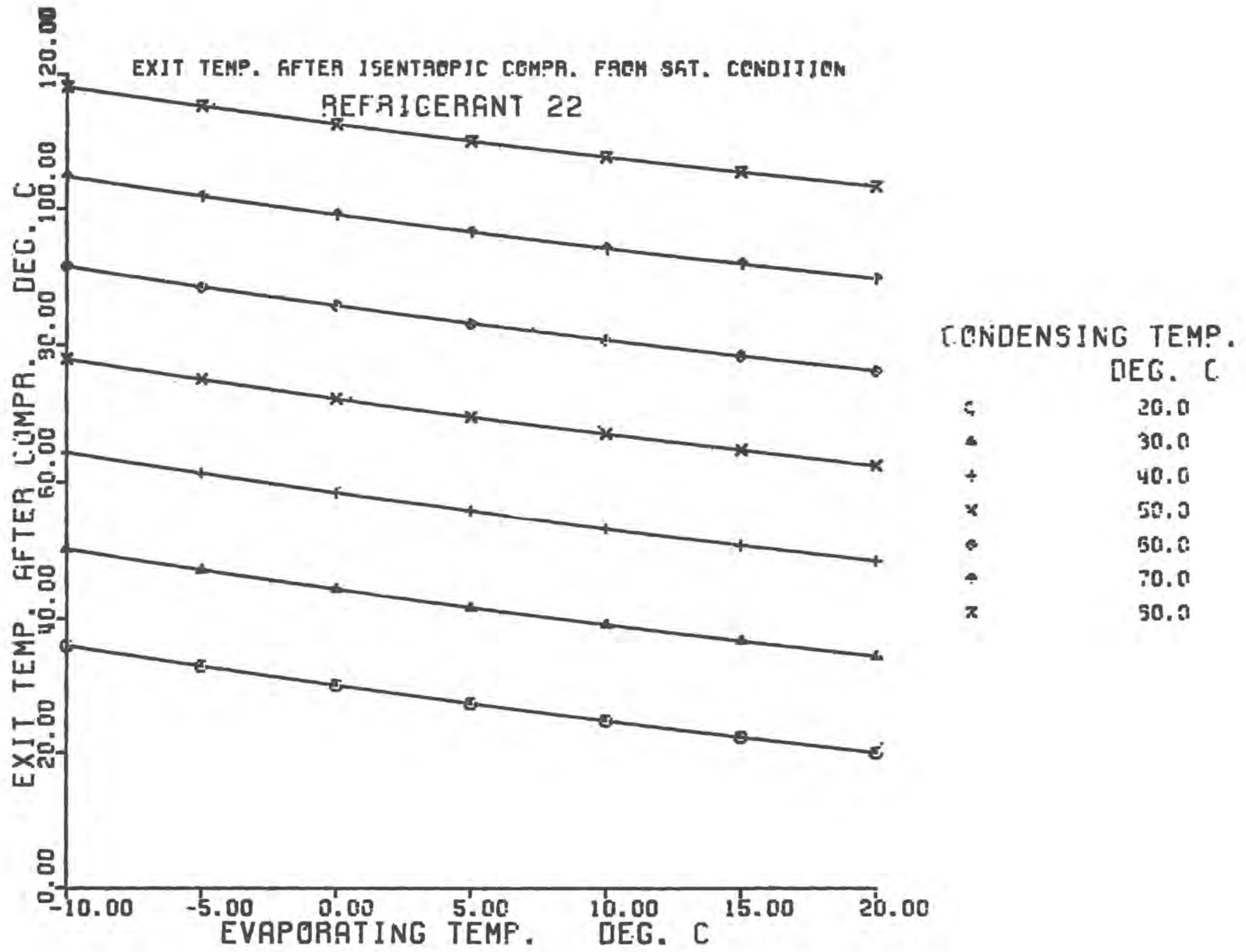

EXIT TEMP. AFTER ISENTROPIC COMPR. FROM SAT. CONDITION
REFRIGERANT 22
EXIT TEMP. AFTER COMPR. DEG. C
0.00
20.00
40.00
60.00
80.00
100.00
120.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C
CONDENSING TEMP.
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
50.0

fig. 7-6

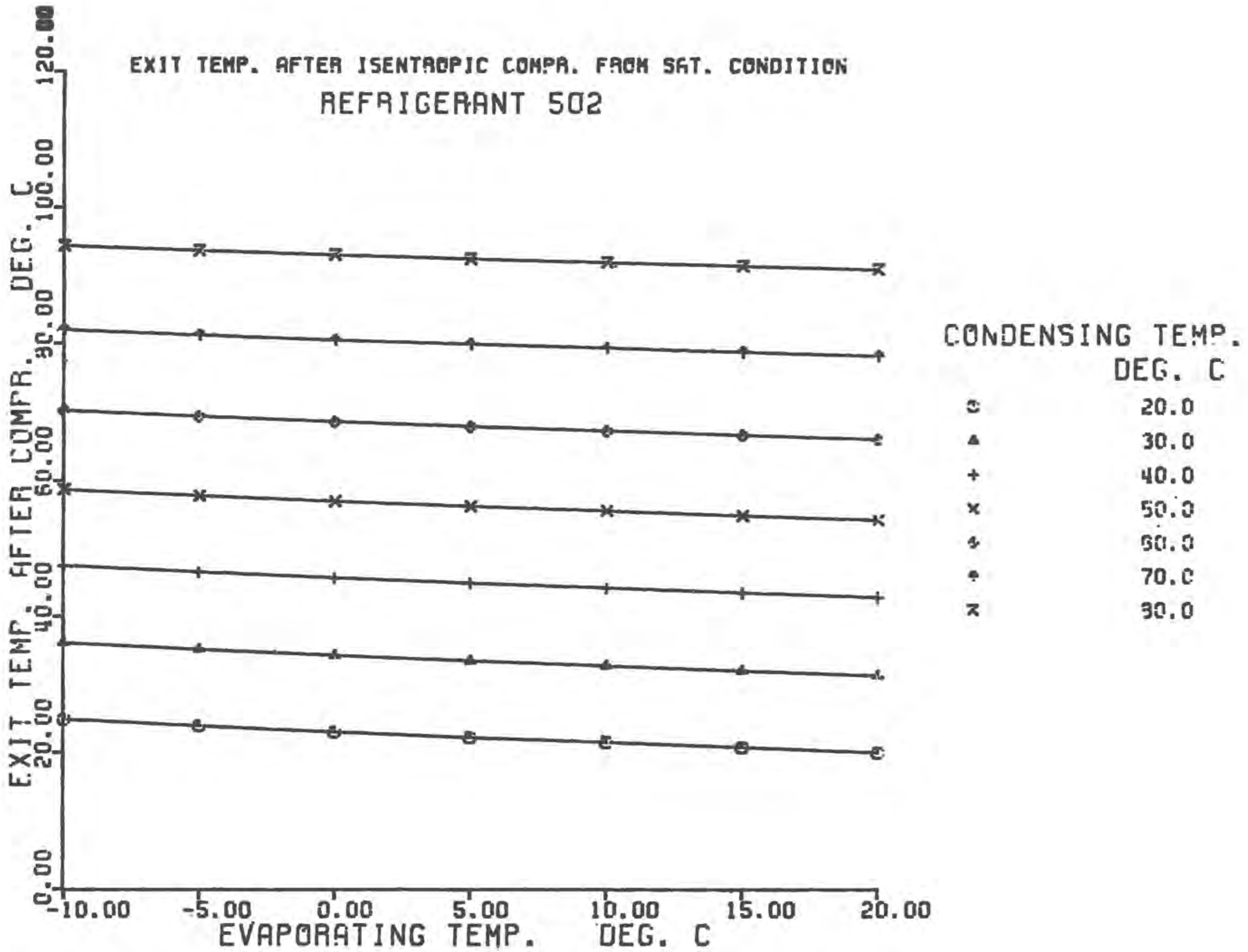

EXIT TEMP. AFTER ISENTROPIC COMPR. FROM SAT. CONDITION
REFRIGERANT 502
EXIT TEMP. AFTER COMPR. DEG. C
0.00
20.00
40.00
60.00
80.00
100.00
120.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C
CONDENSING TEMP.
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0

fig. 7-7

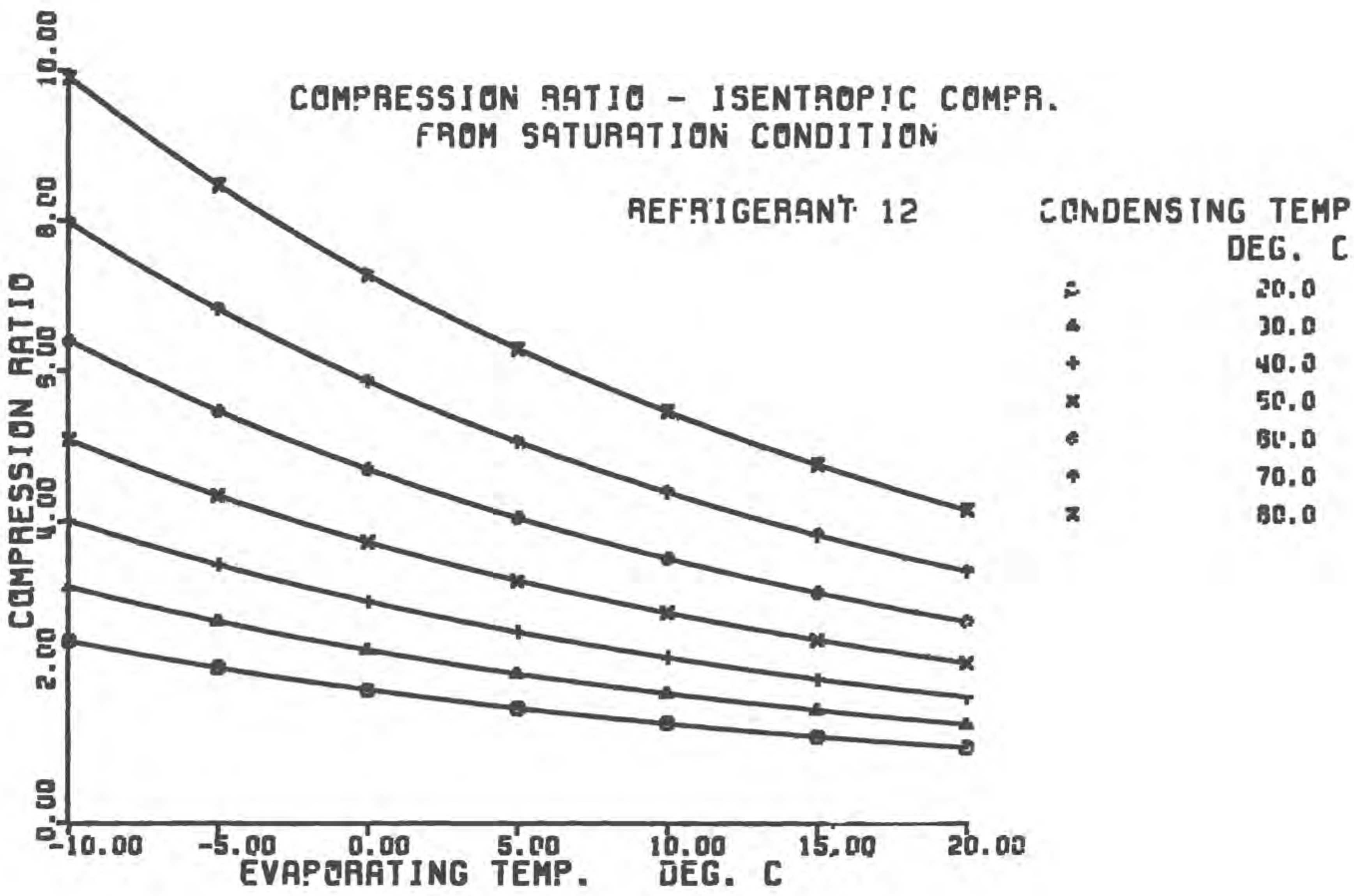

COMPRESSION RATIO - ISENTROPIC COMPR.
FROM SATURATION CONDITION
REFRIGERANT 12
CONDENSING TEMP
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0
COMPRESSION RATIO
0.00
2.00
4.00
6.00
8.00
10.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C

fig. 7-8

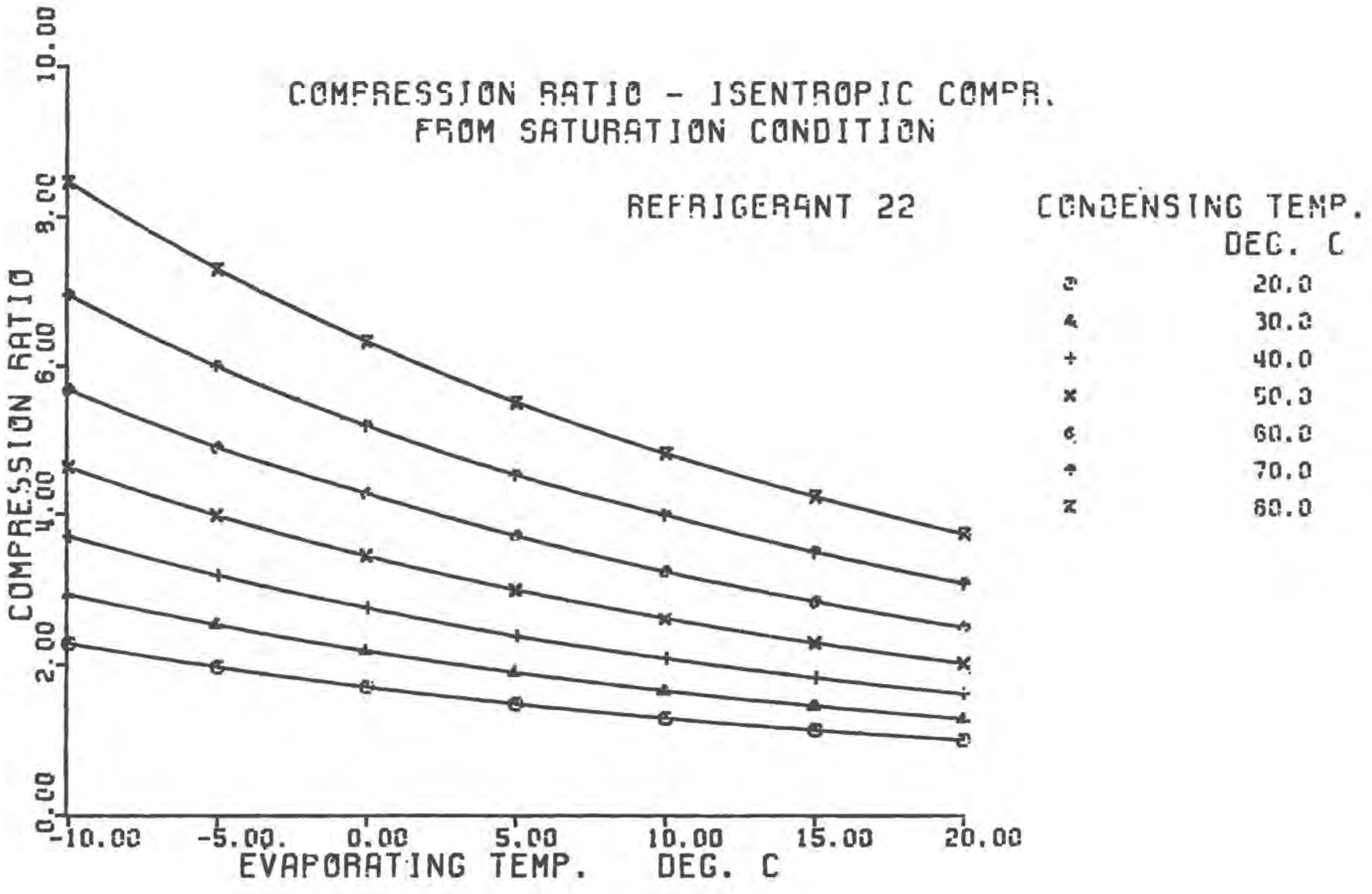

COMPRESSION RATIO - ISENTROPIC COMPR.
FROM SATURATION CONDITION
REFRIGERANT 22
CONDENSING TEMP.
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0
COMPRESSION RATIO
0.00
2.00
4.00
6.00
8.00
10.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C

fig. 7-9

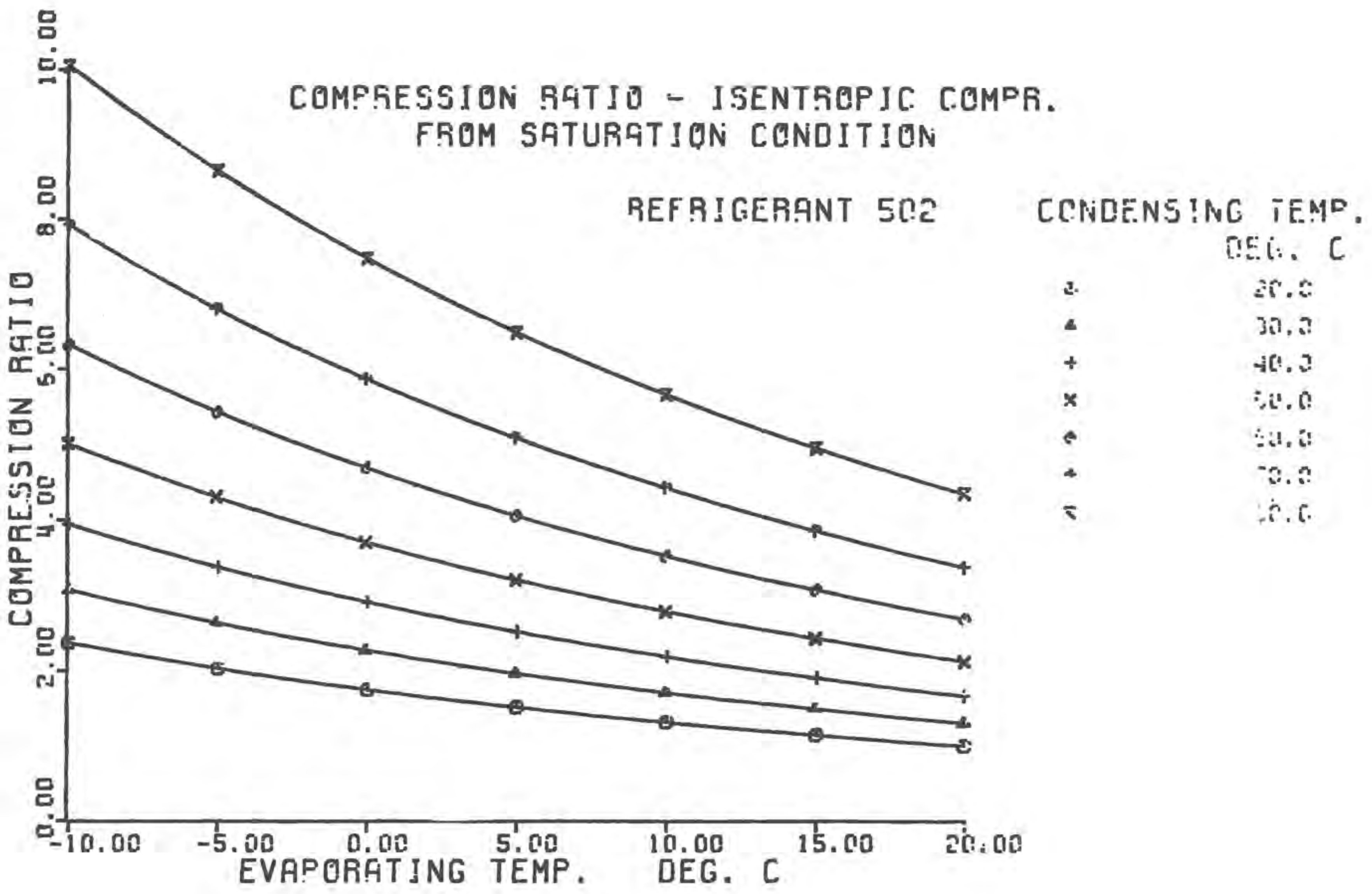

COMPRESSION RATIO - ISENTROPIC COMPR.
FROM SATURATION CONDITION
REFRIGERANT 502
CONDENSING TEMP.
DEG. C
20.0
30.0
40.0
50.0
COMPRESSION RATIO
0.00
2.00
4.00
6.00
8.00
10.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C

fig. 7-10

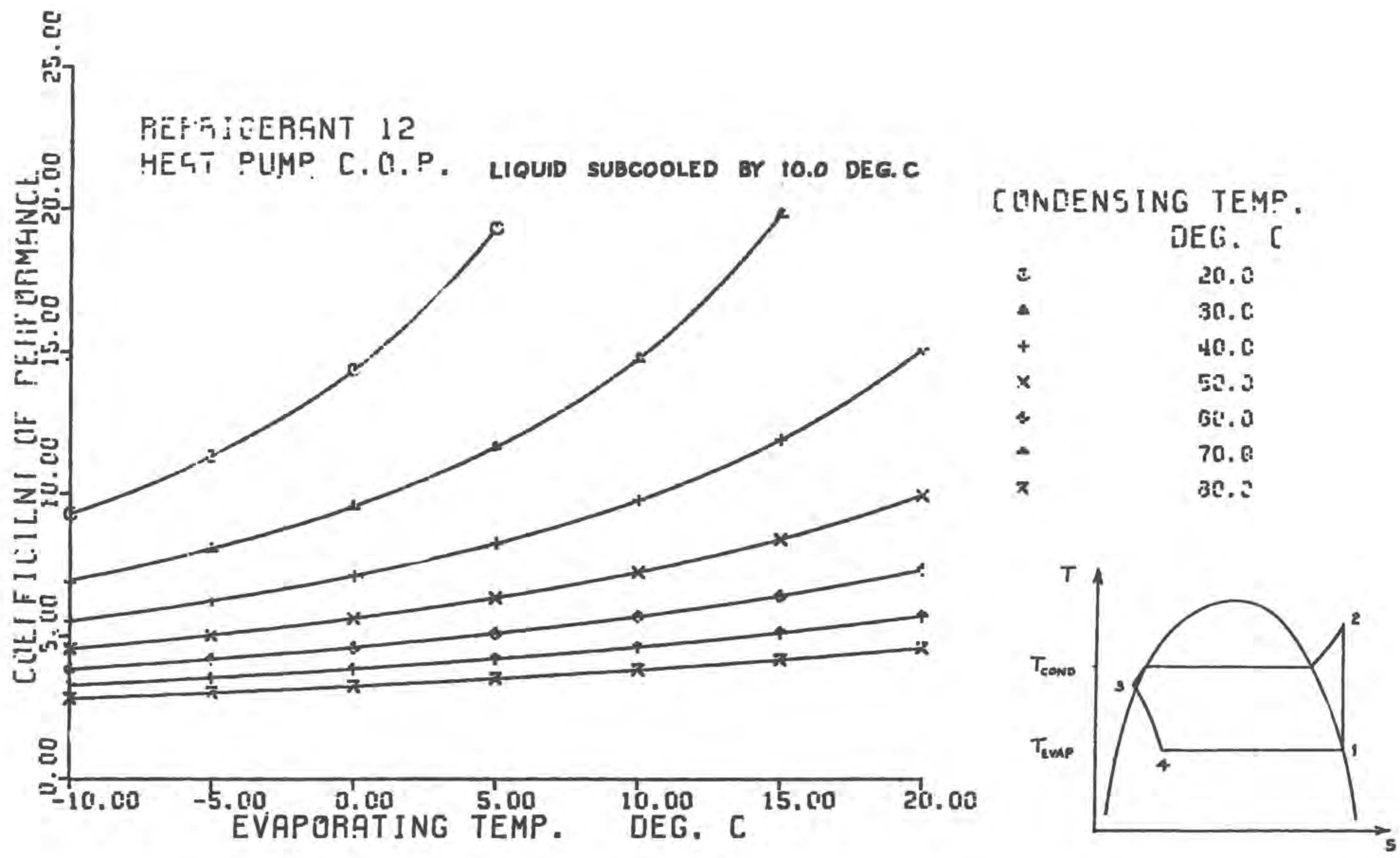

REFRIGERANT 12
HEAT PUMP C.O.P.
LIQUID SUBCOOLED BY 10.0 DEG.C
COEFFICIENT OF PERFORMANCE
EVAPORATING TEMP. DEG. C
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
25.00
CONDENSING TEMP.
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0
T
S
T_{COND}
T_{EVAP}
1
2
3
4

fig. 7-11

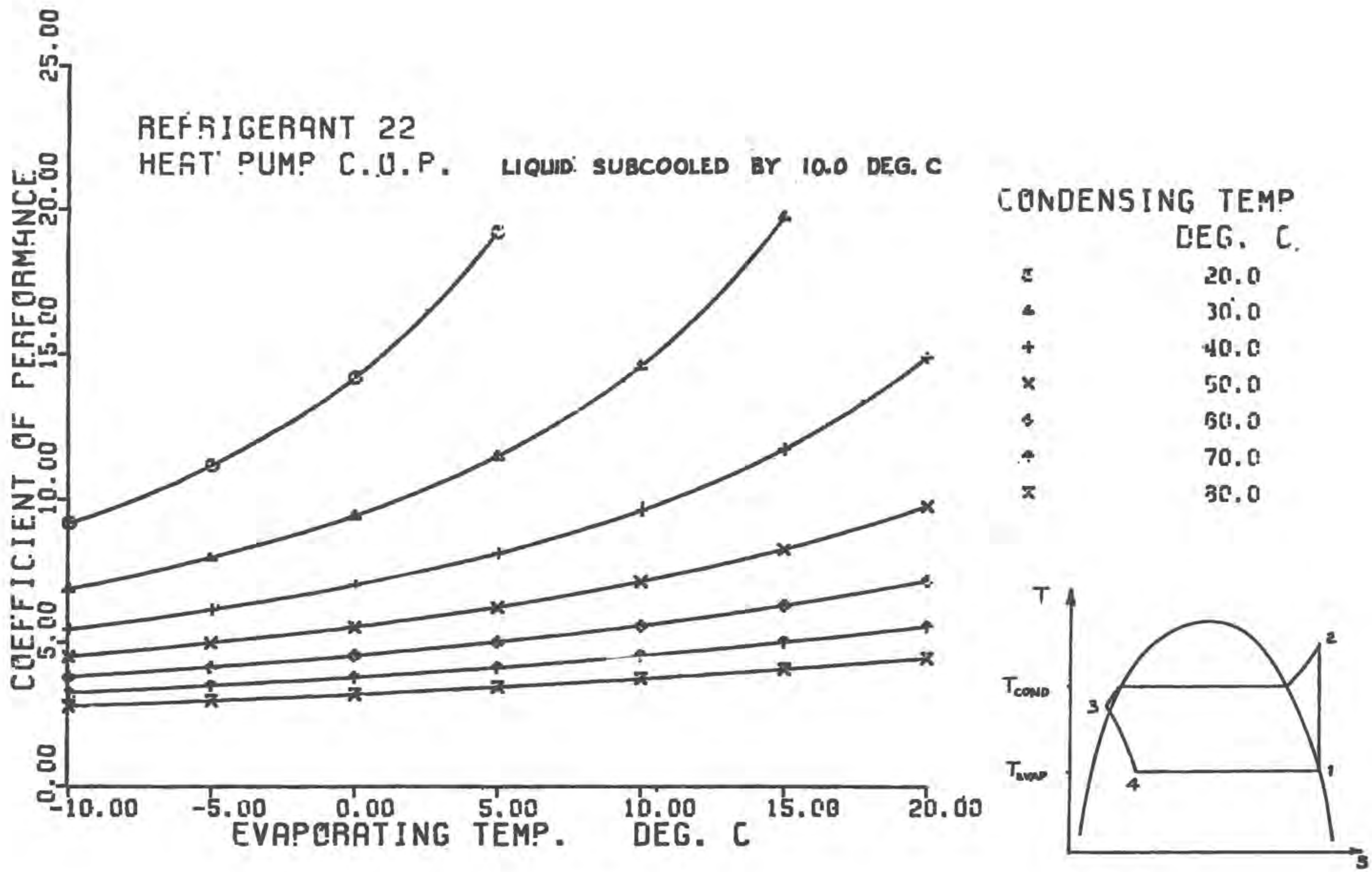

REFRIGERANT 22
HEAT PUMP C.O.P.
LIQUID SUBCOOLED BY 10.0 DEG. C
CONDENSING TEMP
DEG. C.
20.0
30.0
40.0
50.0
60.0
70.0
80.0
COEFFICIENT OF PERFORMANCE
0.00
5.00
10.00
15.00
20.00
25.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C
T
T_{COND}
T_{EVAP}
1
2
3
4
S

fig. 7-12

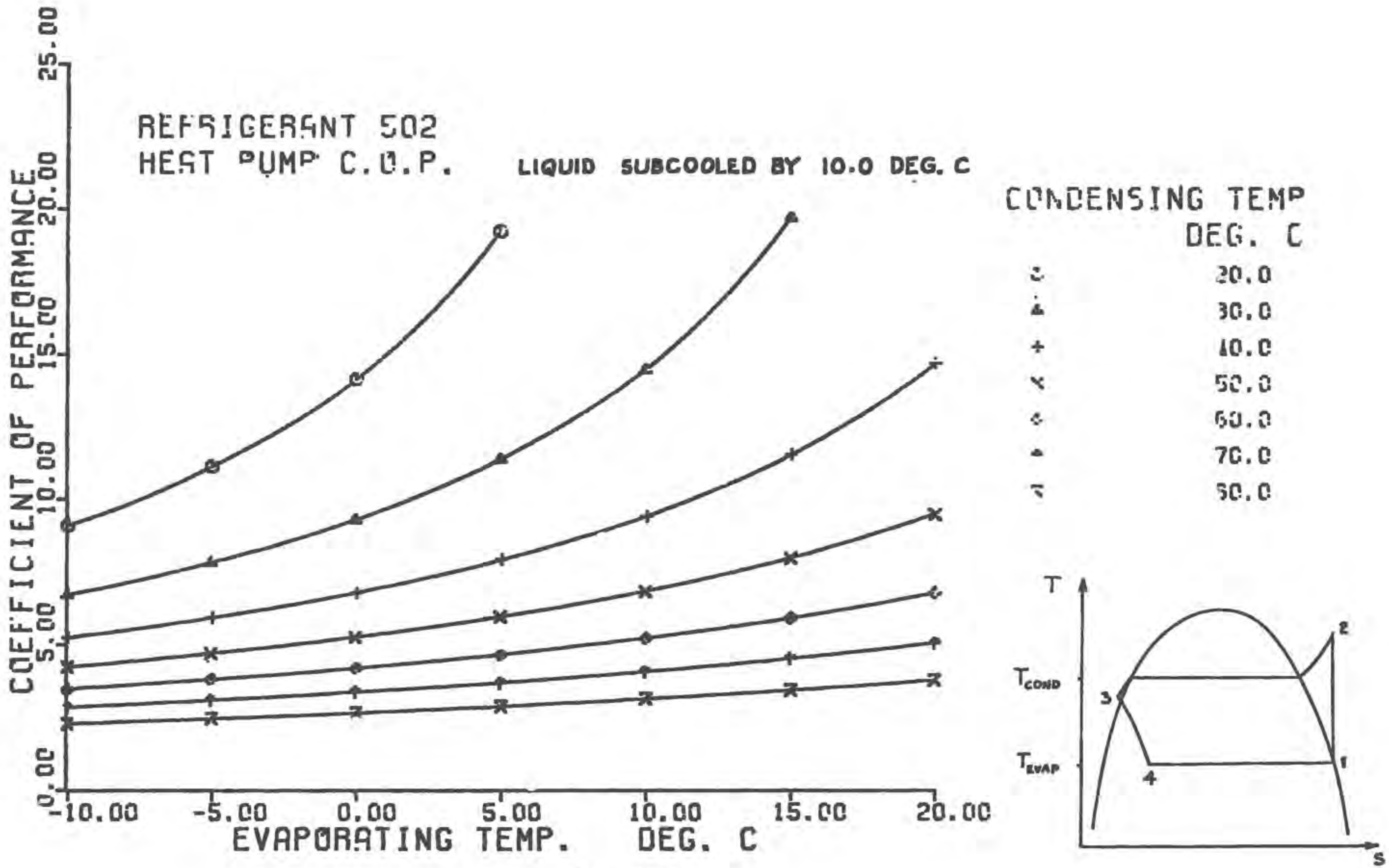
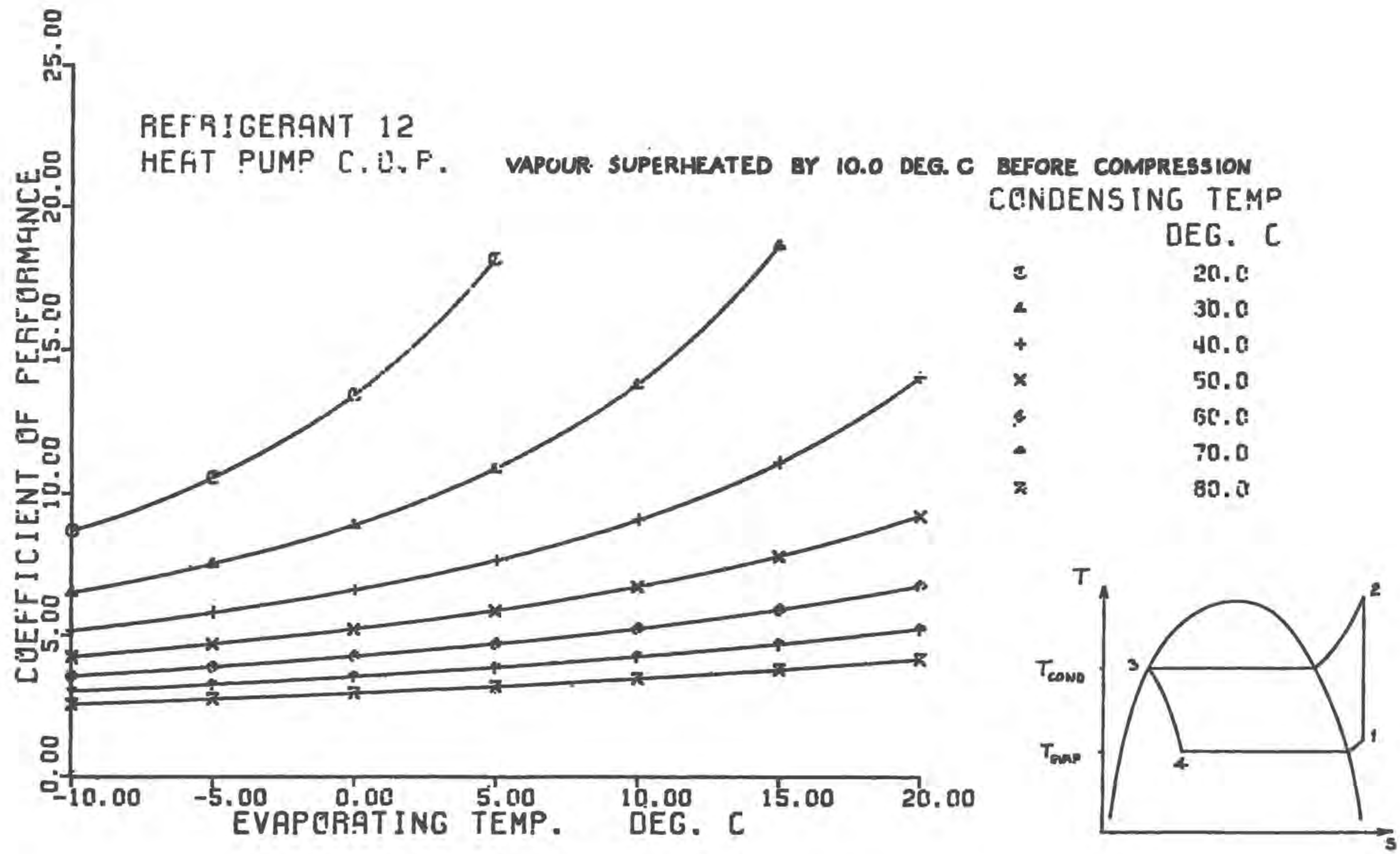

REFRIGERANT 502
HEAT PUMP C.O.P.
LIQUID SUBCOOLED BY 10.0 DEG. C
CONDENSING TEMP
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0
COEFFICIENT OF PERFORMANCE
0.00
5.00
10.00
15.00
20.00
25.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C
T
T_{COND}
T_{EVAP}
1
2
3
4
S

fig. 7-13



REFRIGERANT 12
HEAT PUMP C.O.P.
VAPOUR SUPERHEATED BY 10.0 DEG. C BEFORE COMPRESSION
CONDENSING TEMP
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0
COEFFICIENT OF PERFORMANCE
0.00
5.00
10.00
15.00
20.00
25.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C
T
T_{COND}
T_{EVAP}
1
2
3
4
s

fig. 7-14

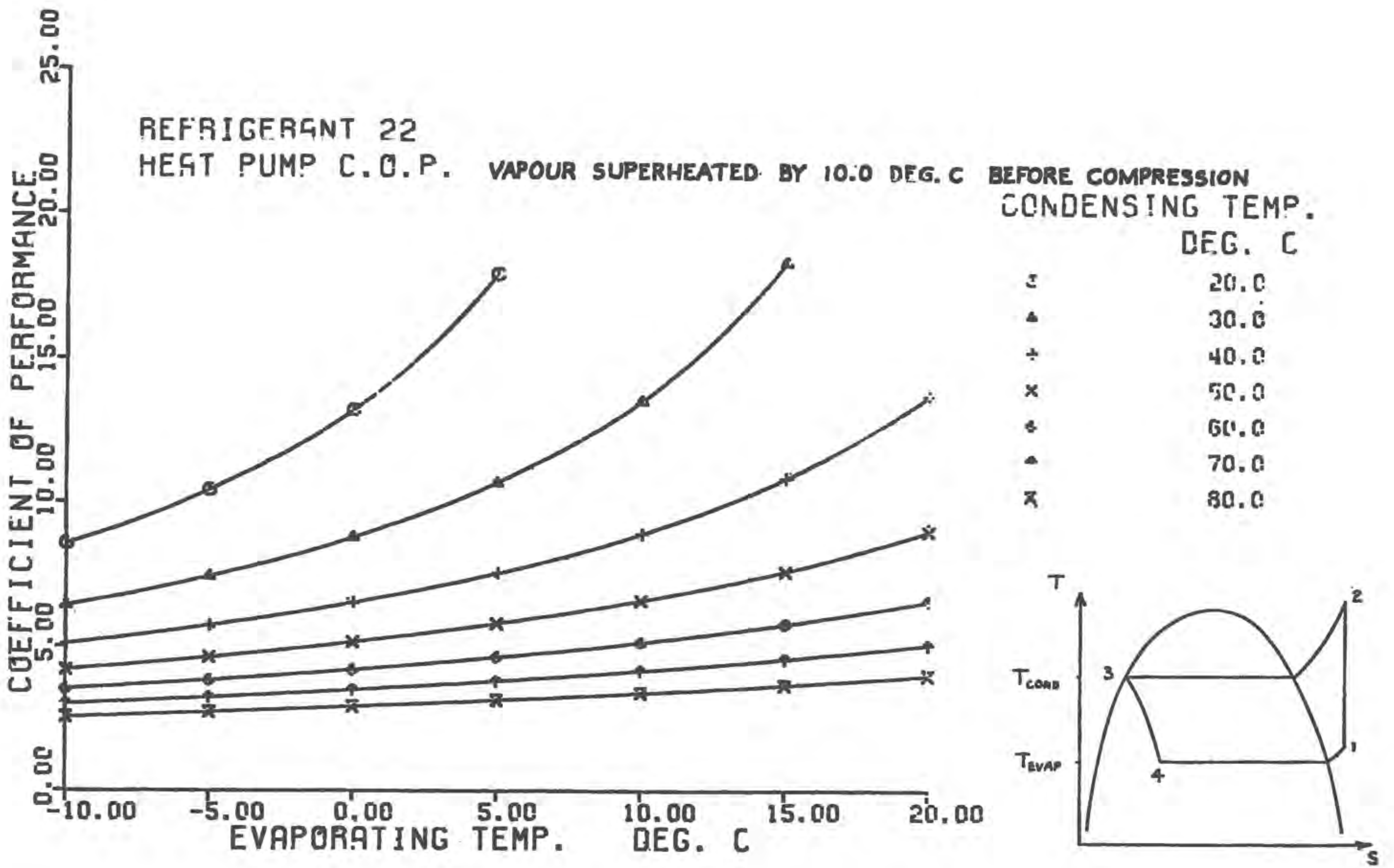

REFRIGERANT 22
HEAT PUMP C.O.P. VAPOUR SUPERHEATED BY 10.0 DEG. C BEFORE COMPRESSION
CONDENSING TEMP.
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0
COEFFICIENT OF PERFORMANCE
0.00
5.00
10.00
15.00
20.00
25.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C
T
S
T_{COND}
T_{EVAP}
1
2
3
4

fig. 7-15

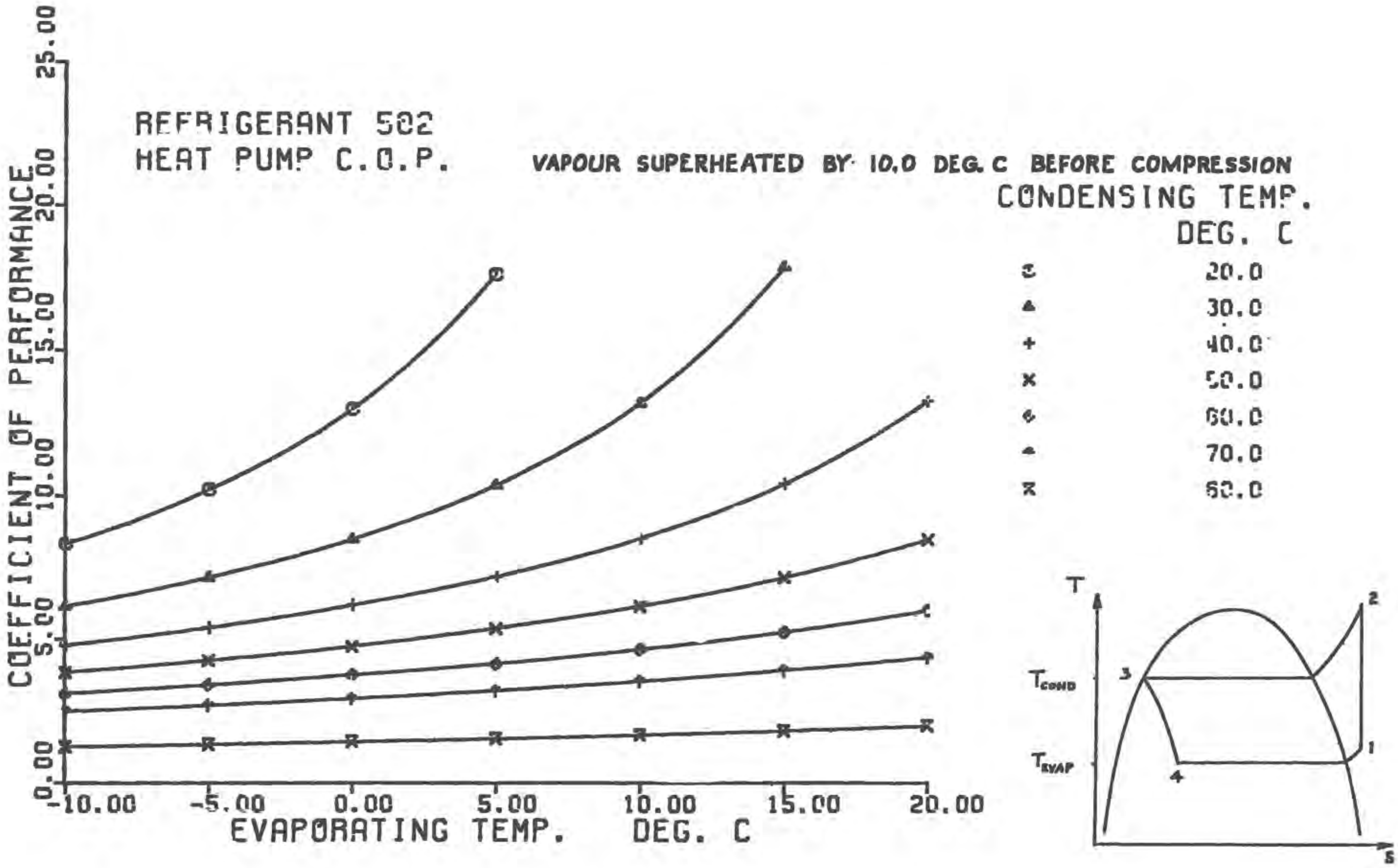
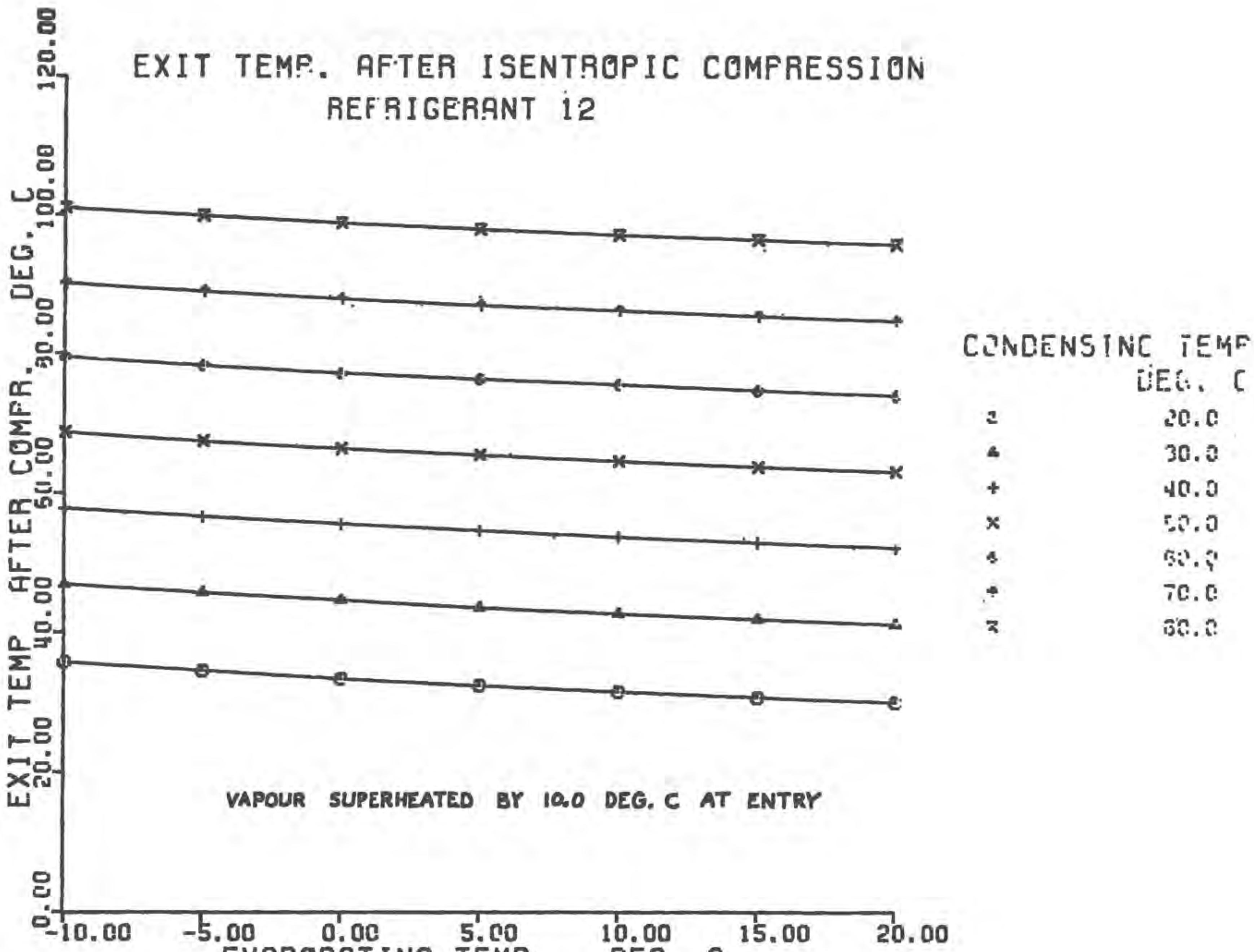
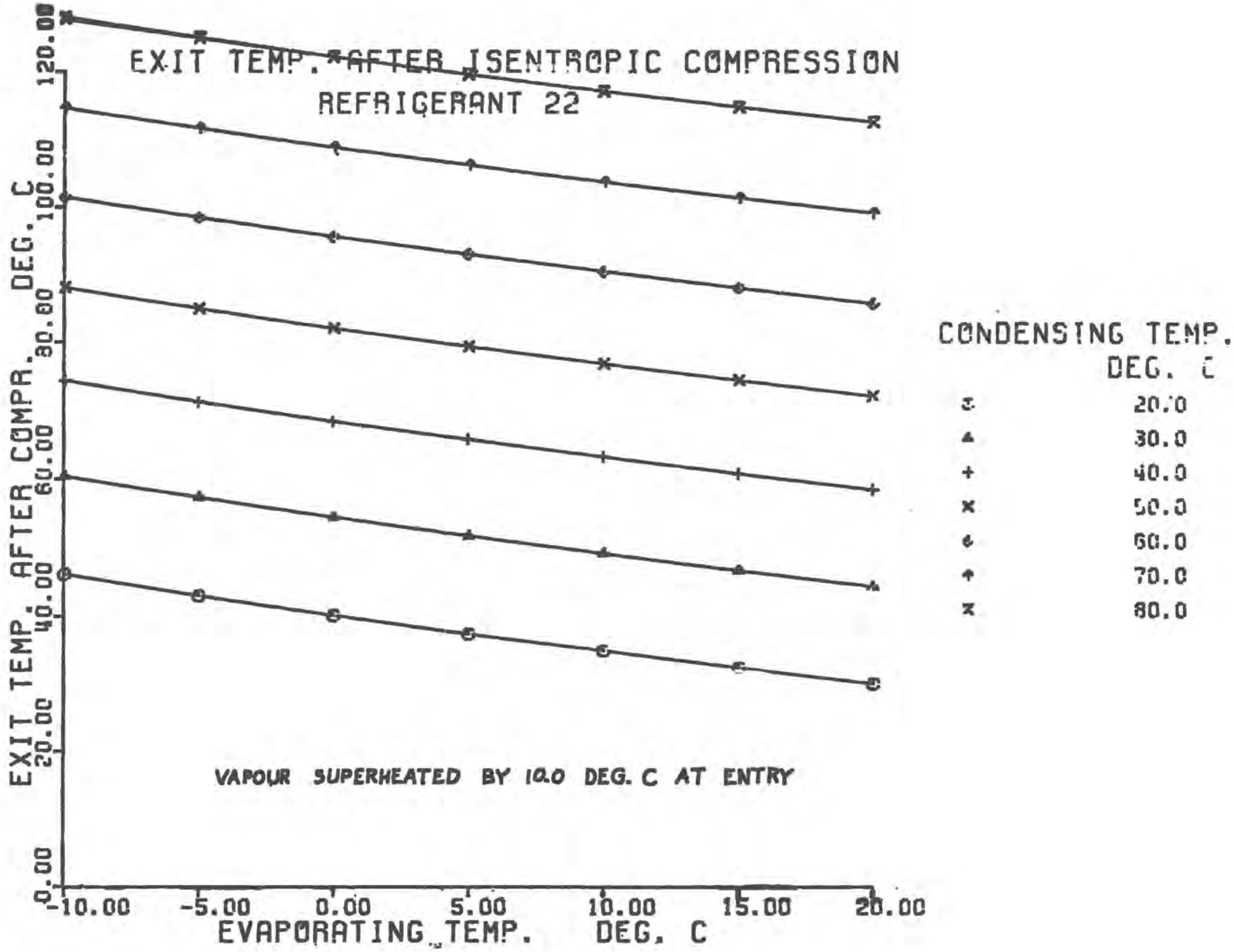

REFRIGERANT 502
HEAT PUMP C.O.P.
VAPOUR SUPERHEATED BY 10.0 DEG. C BEFORE COMPRESSION
CONDENSING TEMP.
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0
COEFFICIENT OF PERFORMANCE
0.00
5.00
10.00
15.00
20.00
25.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C
T
T_{COND}
T_{EVAP}
1
2
3
4
s

fig. 7-16



EXIT TEMP. AFTER ISENTROPIC COMPRESSION
REFRIGERANT 12
CONDENSING TEMP
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0
VAPOUR SUPERHEATED BY 10.0 DEG. C AT ENTRY
EXIT TEMP AFTER COMPR. DEG. C
EVAPORATING TEMP. DEG. C

fig. 7-17



EXIT TEMP. AFTER ISENTROPIC COMPRESSION
REFRIGERANT 22
EXIT TEMP. AFTER COMPR. DEG. C
0.00
20.00
40.00
60.00
80.00
100.00
120.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C
VAPOUR SUPERHEATED BY 10.0 DEG. C AT ENTRY
CONDENSING TEMP.
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0

fig. 7-18

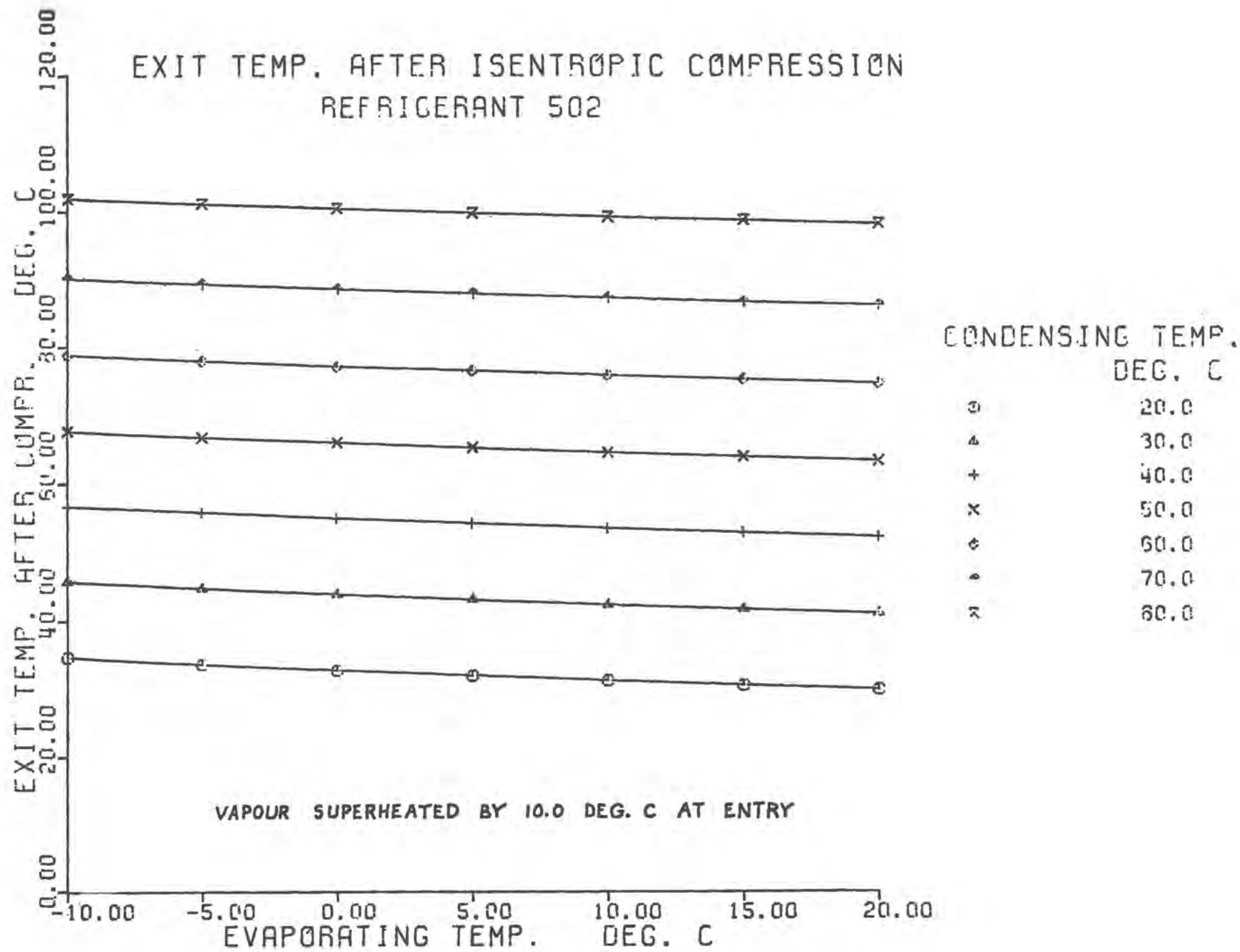

EXIT TEMP. AFTER ISENTROPIC COMPRESSION
REFRIGERANT 502
EXIT TEMP. AFTER COMPR. DEG. C
0.00
20.00
40.00
60.00
80.00
100.00
120.00
-10.00
-5.00
0.00
5.00
10.00
15.00
20.00
EVAPORATING TEMP. DEG. C
VAPOUR SUPERHEATED BY 10.0 DEG. C AT ENTRY
CONDENSING TEMP.
DEG. C
20.0
30.0
40.0
50.0
60.0
70.0
80.0

fig. 7-19

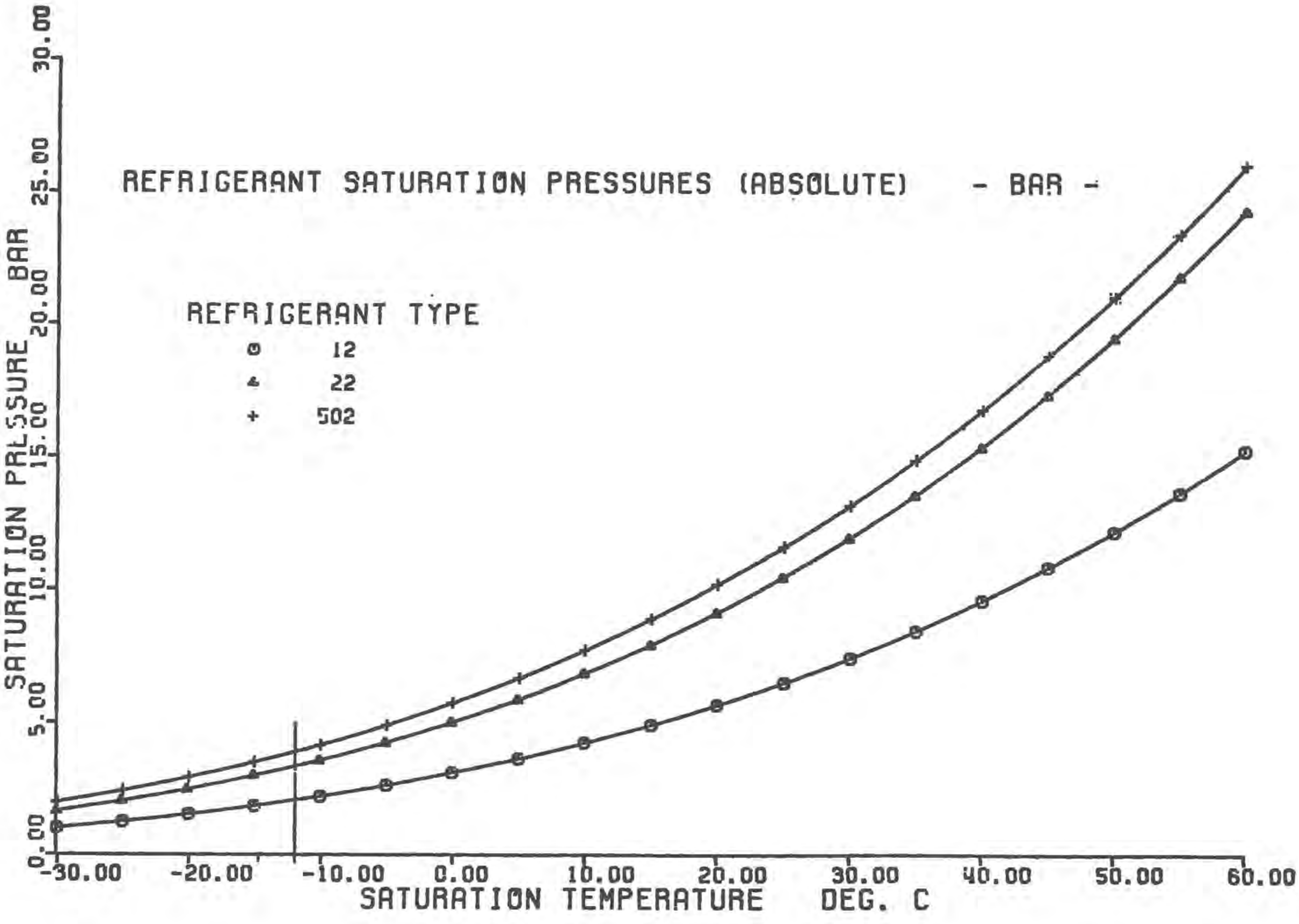

REFRIGERANT SATURATION PRESSURES (ABSOLUTE) - BAR -
REFRIGERANT TYPE
12
22
502
SATURATION PRESSURE BAR
0.00
5.00
10.00
15.00
20.00
25.00
30.00
-30.00
-20.00
-10.00
0.00
10.00
20.00
30.00
40.00
50.00
60.00
SATURATION TEMPERATURE DEG. C

fig. 7-20

8 CONCLUSIONS

At this stage of the research program it is not possible to give conclusive results. The performance obtained with the Klimatkyla unit is very poor, though the reasons for this have not yet been fully analysed. The measured air flow rates are considerably lower than the design values quoted by the manufacturer. An increase in air flow might give a slight increase in C.O.P., but the main problem seems to be that the heat transfer units are just too small.

Progress in developing the simulation program is quite satisfactory as the main structure of the simulation model has been completed and only requires refinement.

The comparison of refrigerants is still in progress, but a considerable amount of data has already been assembled.

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