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DETERMINATION OF REFRIGERANT COMPRESSOR PERFORMANCE

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INTRODUCTION

According to the ASHRAE, Guide and Data Book [1], refrigerant compressors should be tested 'To find the performance of the plant in which the compressor has been used.' The performance of the machine has been defined as 'An evaluation of the ability of the machine to accomplish the given task. It is a design compromise between the physical limitations of: (a) the refrigerant, (b) compressor and (c) motor to provide (i) the most refrigerating effect for the least power input, (ii) the greatest trouble-free life, (iii) the lowest cost and (iv) a wide range of operating conditions.'

The capacity (which is proportional to the displacement) and performance factor are both measures of compressor performance. The capacity is the refrigerating effect which may be produced by the compressor. It is equal to the difference in fluid enthalpies between the evaporator outlet (or compressor inlet, if there is no intermediate heat exchanger) and the saturated liquid enthalpy at the compressor outlet pressure. The performance factor of an open-type compressor is given by the equation:

$$\text{Performance factor}^1 = \frac{\text{Capacity}}{\text{Power input to the shaft}} \quad (\text{open type})$$

For a hermetic sealed compressor however, the performance factor would include the motor efficiency, since the power input at the shaft cannot be determined.

The Indian Standards Institution prescribes the 'run-round the cycle test', to determine the capacity and coefficient of performance. This is therefore the method used in the present experimental set-up. The method tests the compressor installed in a complete refrigerating circuit. Calorimetric measurements are made at the evaporator to determine the refrigerating effect. The compressor refrigerating

¹ Note that the performance factor as defined here is the reciprocal of that specified by the ASHRAE guide [1].

capacities are determined at various evaporator temperatures. The results so obtained are used to compute the performance factor and volumetric efficiency, which may be exhibited graphically.

The Indian Standards [2] test code further prescribe that the compressor tests should be carried out in two ways, independent of each other. The first is the Principal test while the second is a Confirmatory test. Three different possible principal tests and five possible confirmatory tests, have been specified. The principal tests are referred to as methods A, B and C. Method A uses a secondary fluid calorimeter in the suction line. Method B incorporates a flooded refrigerant calorimeter in the suction line, while method C uses a dry system refrigerant calorimeter. In method B, which is employed in the present set-up, the calorimeter (evaporator), is flooded with the liquid refrigerant upto a definite level and heated directly with electrical heater rods immersed in the liquid. The calorimeter is, of course, properly insulated to make energy gains from the outside negligible. The calorimeter is thus simple to fabricate. Besides this, the refrigerating capacity can be measured with greater accuracy as compared with methods A and C, since all of the electrical input goes into the vaporization of the refrigerant.

The Indian Standards also prescribe that the calorimeter should be pre-calibrated to determine a heat-leakage factor, F_1 , as given by the expression

$$F_1 = \phi_i (t_r - t_a) \quad (1)$$

where ϕ_i = energy input as heat/min,
 t_a = ambient temperature, and
 t_r = liquid refrigerant temperature.

The test should be performed with the liquid refrigerant in the evaporator upto the normal operating level and heated so as to maintain its temperature, about 14°C above the ambient. The electrical energy input should be maintained constant within ± 1 percent during the test. If the hourly readings of the liquid refrigerant

temperature do not vary by more than 0.6°C, it may be assumed that thermal equilibrium has been obtained. The readings obtained are then used to calculate F_1 , as defined by Eq. (1).

During the actual compressor test, the Standards require the determination of the following quantities:

- (i) The refrigerant pressure at the evaporator outlet;
- (ii) The pressure of the liquid refrigerant entering the expansion valve.
- (iii) The ambient temperature, and
- (iv) The electrical input to the calorimeter.

The data thus obtained help in computing the refrigerant mass flow rate, \dot{m}_f , as given by the equation:

$$\dot{m}_f = [\phi_h + F_1(t_a - t_r)] / (h_{g5} - h_{f3}) \quad (2)$$

where ϕ_h = rate of energy input to the calorimeter (kcal/min),
 h_{g5} = enthalpy of saturated vapour at the evaporatory pressure, and
 h_{f3} = enthalpy of liquid refrigerant before throttling. See Fig. 1-1.

The quantities t_a and t_r are the ambient and refrigerant temperatures, during the test. The refrigerating capacity is finally given by the equation:

$$RF = \dot{m}_f (h_{g1} - h_{f3}'), \quad (3)$$

h_{g1} being the enthalpy of the vapour at the compressor inlet and h_{f3}' , the saturated liquid enthalpy at the compressor exit pressure.

The Confirmatory Test: Of the five methods prescribed by the Indian Standards, the method selected for the following set up employs a refrigerant liquid quantity flow meter placed between the condenser and the throttling valve. The flow-meter consists of two long well-insulated hollow pipes, fitted with gauge glasses over their entire lengths. The liquid from the condenser can enter either of the two pipes, as regulated by valves fitted at the top. The outlets of the pipes are also connected with a common line through shut-off valves. The liquid flow rate may be determined by noting the drop in liquid level in one of the pipes over a specified period of time. This flow-rate should agree closely with that determined during the principal test.

2. APPARATUS

A schematic diagram of the apparatus

employed is shown in Fig. 2-1 and the control panel in Fig. 2-2. The test circuit is a simple compression refrigeration system using the accessories and confirmatory test liquid flow meter as specified by the Indian Standards.

A compressor of nominal capacity - 2 tons, A, manufactured by Freezeking India Ltd., is used with R-12 as the refrigerant. At both the suction and discharge ends of the compressor, needle valves have been included to regulate the flow through the system. Precalibrated 24 gauge BSW copper-constantan thermocouples have been inserted at both the inlet and exit of the compressor.

The condenser, is a two-pass water-cooled system of maximum capacity about 3 tons, manufactured by the American Refrigeration Co., Calcutta. This oversized condenser has been deliberately employed to ensure that the emerging refrigerant is subcooled. The rate of cooling water flow through the condenser is measured accurately with a precalibrated rotameter of capacity 0-36 lit/min and least count 0.8 lit/min. Further, the temperatures and pressure respectively, at the inlet and outlet of the condenser are measured with precalibrated thermocouples and a pressure gauge calibrated with a dead-weight tester.

Following the condenser is a drier-filter combination, after which the liquid refrigerant enters the flow meter used for the confirmatory test, as mentioned earlier. The flow meter is made of two 7.5 cm nominal internal diameter mild steel pipes, 1.25 m long. These dimensions were chosen to satisfy the ISI requirement that the liquid refrigerant collecting inside one of the pipes shall rise a minimum of 150 mm in 2 minutes. Gauge glasses fitted on each pipe are used to note the liquid level. The pipe ends covered with flanges, provide connections to the filter-drier combination at the top, and the throttling valve at the bottom. The pipes are also thoroughly lagged over their lengths with 5 cm thick fibre-glass insulation. At the outlet, a thermocouple and pressure gauge have been connected for appropriate measurements. The flow meter has itself been precalibrated by fitting the pipes with distilled water of known density to various specified heights and noting the volume of the liquid in each case.

From the flow meter, the liquid enters a two-ton thermometric expansion valve, fitted in the circuit as shown. In addition, a needle-valve has been fitted in series with the thermostatic valve, to obtain the desired pressure variation in the evaporator.

The calorimeter (evaporator) is made of a cylindrical brass shell, 22.9 cm diameter and 6 mm thick. The capacity of the shell is large enough to accumulate sufficient refrigerant liquid to act as a liquid receiver as well. The refrigerant is heated by means of five electric heater rods, each 100 cm long and 1 kW heating capacity, inserted in mild steel 1.25 cm internal diameter tubes, brazed to both the end-plates of the calorimeter. The rate of refrigerant flow and hence the liquid level in the calorimeter are so maintained that the pipes carrying the heater rods are always immersed in the liquid. The whole calorimeter is insulated with 7.5 cm thick fibre-glass insulation. The pipes connecting the calorimeter to the expansion valve are also similarly insulated.

The outlet of the calorimeter is connected to the compressor inlet through a copper tube, covered with a heating pad and insulated on the outside. This device ensures that the vapour reaching the compressor is superheated. A single-phase precalibrated energy meter is so connected as to measure the power input to the calorimeter and the heating pads, which are independently regulatable.

For further details regarding the apparatus, Ref. [5].

3. EXPERIMENTAL PROCEDURE

The compressor is driven with a three-phase induction motor, the power input to which may be measured with a pre-calibrated three-phase energy meter. After the compressor is started, the expansion valve and/or the needle valve in the circuit is adjusted to give an appropriate flow rate. The calorimeter is then heated, by controlling the voltage impressed across the heater rods with a variac operating on 230 V, 50 cycles mains. When the pressures and temperatures at various points remain constant for at least 15 minutes, the following readings are taken:

- (i) The pressure gauge readings at all the indicated points (Fig. 2-1),
- (ii) The length of time needed for 10 revolutions of the 3-phase, energy meter connected in the induction motor circuit (power input to the motor),
- (iii) The time needed for 10 revolutions of the single-phase energy meter fitted in the heater circuit (evaporator input),
- (iv) The rise in level of the liquid in one of the pipes of the flow meter over a period of 2 minutes, when the outlet valve of the pipe and the inlet valve are simultaneously opened.

- (v) The thermocouple readings at all the points, using a potentiometer, and
- (vi) The rate of condenser water flow as indicated by the rotameter.

The experiment and observations are then repeated for various expansion valve settings, and similar readings are obtained at different evaporator temperatures. The data are used to calculate the following quantities:

- (a) The mass flow rate of the refrigerant as given by Eq. (2),
- (b) The energy input to the motor in kW and hence the compressor shaft power input, from a knowledge of motor characteristics,
- (c) The coefficient of performance, COP, as given by

$$\text{COP} = \frac{\text{(Refrigeration effect)}}{\text{(Shaft power input)}}$$

$$= \text{RF/P,}$$

The relative coefficient of performance is the ratio

$$(\text{COP})_R = \text{COP}/(\text{COP})_S,$$

$(\text{COP})_S$ being the ideal coefficient of performance based on isentropic work input, as determined from the p-h diagram,

- (d) The specific capacity or the refrigeration effect per kW of shaft energy input, and
- (e) The volumetric efficiency, from a knowledge of the mass flow rate, evaporator temperature and compressor speed, bore as well as stroke.

4. RESULTS AND DISCUSSION

The performance curves of the compressor (Model FK-450, manufactured by Freezeking India Ltd.) are exhibited in Fig. 4-1 which is a plot of refrigeration capacity and COP against evaporator temperature. While calculating the refrigeration capacity, the cooling effect obtained due to liquid sub-cooling at the condenser has been omitted because its magnitude depends upon the capacity of the condenser and cooling water temperature, independently of compressor performance. It is seen that the refrigeration capacity increases with increasing evaporator temperatures, as one may readily expect. The COP too as is to be expected, increases with increasing evaporator temperatures.

Chaikovsky, Shmiglya and Savkov [3] have performed experiments on compressors and have plotted curves of refrigerant capacity and specific refrigeration effect against evaporator temperatures. The

curves exhibited by them, Fig. 4-2, are similar to those obtained from the present tests. A quantitative agreement between the present results and those of [3] cannot be expected since the compressors used are of different models and makes in these cases.

Figure 4-3 exhibits the condenser cooling effect as a function of evaporator temperature. The curve shows a rising trend, since both the energy input to the compressor shaft and the refrigerating effect increase with increasing evaporator temperatures.

Figure 4-4 shows the measured mass-flow rate as a function of evaporator temperatures. For a constant compressor speed of 500 RPM at which all the measurements were made, the volumetric efficiency is proportional to the product of the mass flow rate, and the specific volume of the vapour, saturated at the evaporator pressure. Hence, the quantity $m_{ev} v_{g2}$ is also exhibited in Fig. 4-4, showing clearly that both the mass flow rate and the volumetric efficiency of the compressor increase with increasing evaporator pressures.

BENDIXEN [4] has determined the volumetric efficiencies of uniflow and counter-flow compressors as a function of evaporator temperatures. Though one may expect on theoretical grounds that the volumetric efficiencies of counterflow compressors should be greater than those of uniflow compressors, Bendixen's results show that the observed difference in the volumetric efficiencies of the two types of compressors is within a few percent. Unfortunately, his plots of volumetric efficiency have been against suction superheat instead of evaporator temperatures. Evidently, the volumetric efficiency depends more strongly on evaporator temperature and pressures than on suction superheats especially since the superheat is limited to a few degrees in most compressors. It is therefore believed that the present plot is of greater relevance in determining the compressor performance than those provided by Bendixen.

An error analysis based on the least counts of the various instruments used in the test indicate that the random errors in measurements do not exceed the maxima specified below:

- (i) Power measurements: ± 0.1 percent
- (ii) Refrigerant flow rate: ± 0.5 percent
- (iii) Refrigeration capacity: ± 5.3 percent.

The results obtained seem therefore to be satisfactory for all practical purposes. Test-rigs of this type may therefore be

used to test the performance of open-type commercially available compressors of small capacity.

REFERENCES

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- [2] Indian Standards Test Code for Refrigerant Compressors, No. 5111, 1969.
- [3] CHAIKOVSKY, SHMIGLYA and SAVKOV, The Volumetric Efficiency of Medium Capacity Refrigerating Compressor, Prog. Ref. Science and Tech.
- [4] IVAR BENDIXEN, (A/S Atlas Denmark), The Volumetric Efficiency of the Refrigerant Compressor, Prog. Ref. Sc. and Tech.
- [5] PRASAD, K., Fabrication of Compressor Test Rig, M.Tech Thesis, IIT Kanpur, 1974.

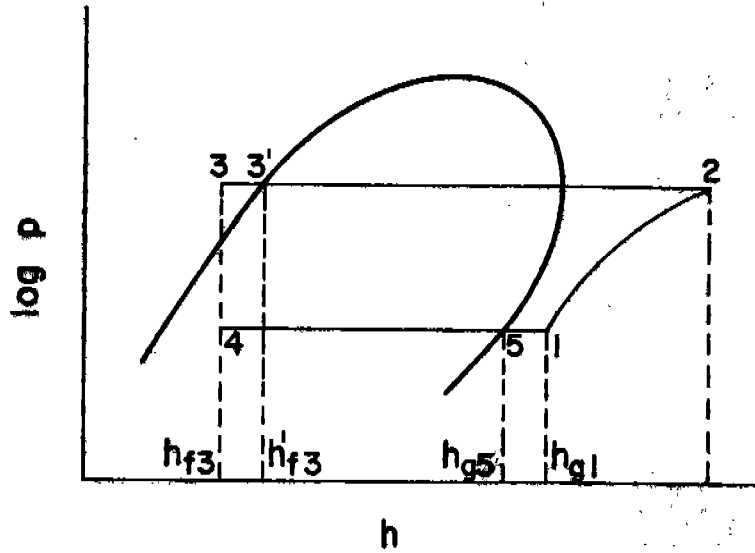
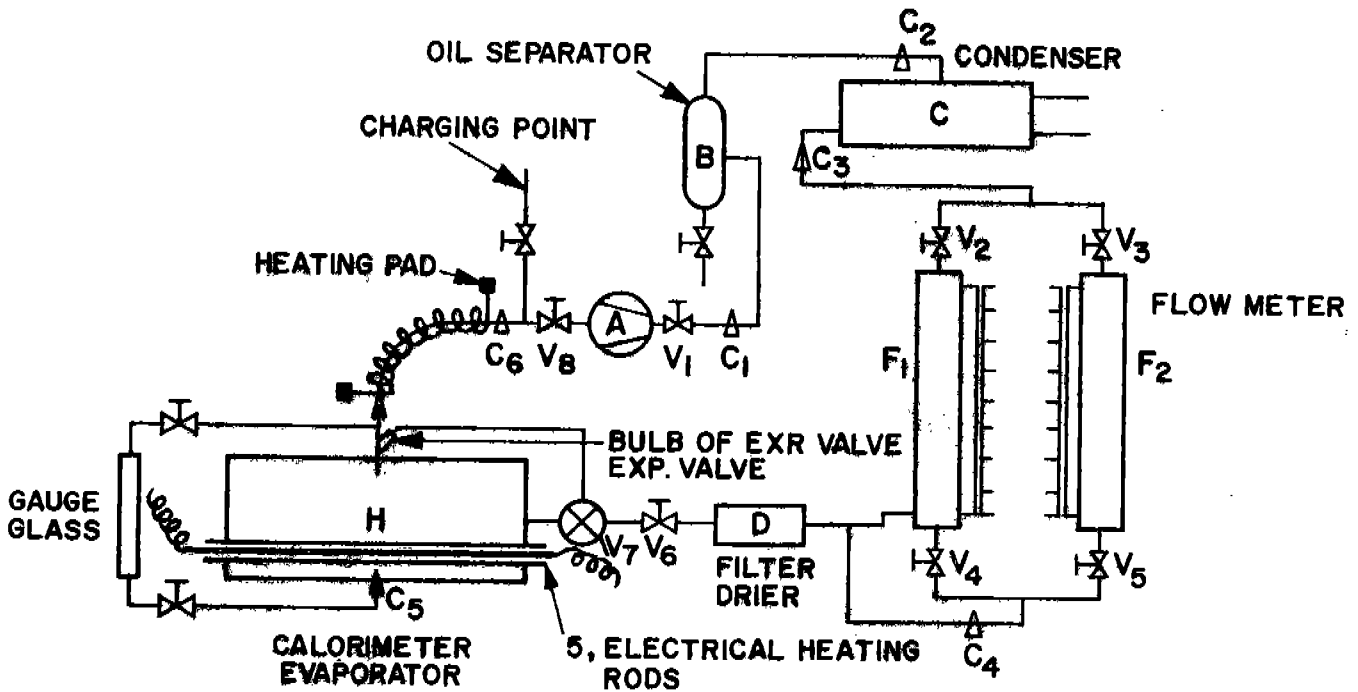
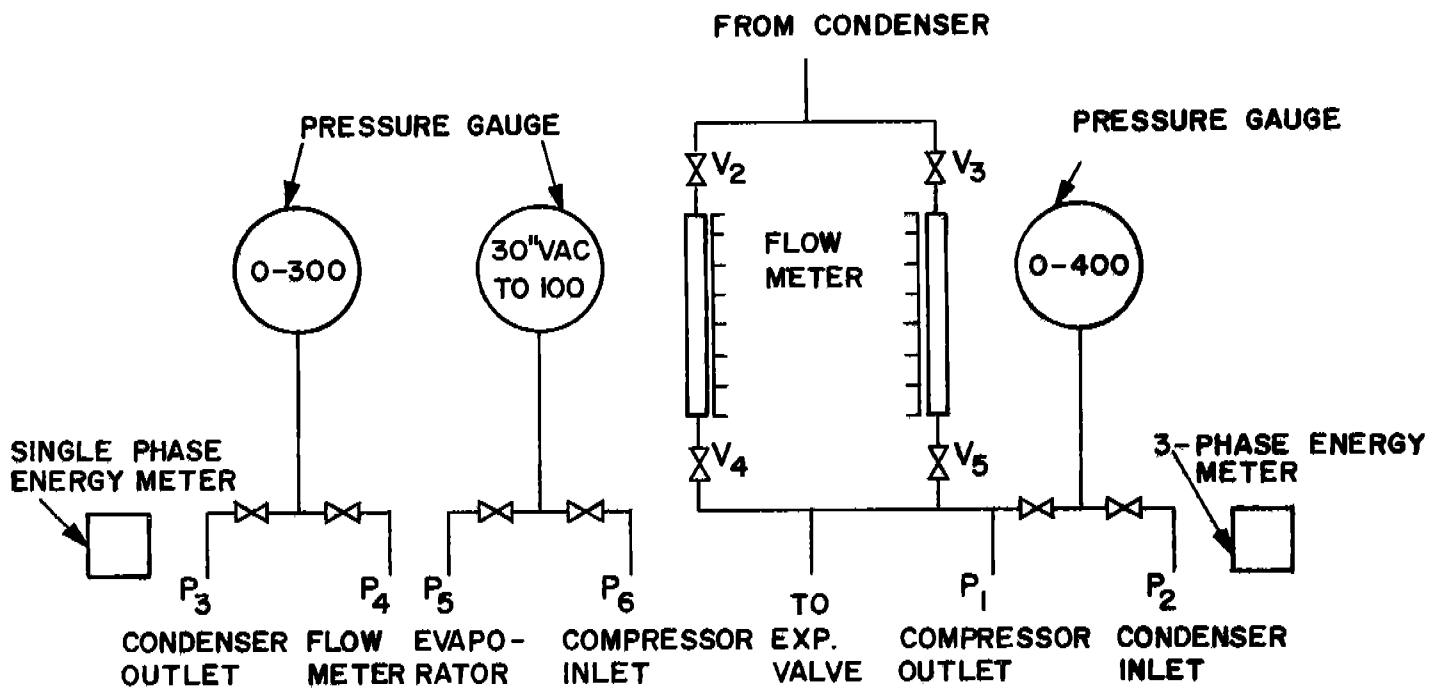


FIGURE 1-1



SCHEMATIC DIAGRAM PLANT LAY-OUT

FIGURE 2-1



SCHAMATIC DIAGRAM OF CONTROL PANEL

FIGURE 2-2

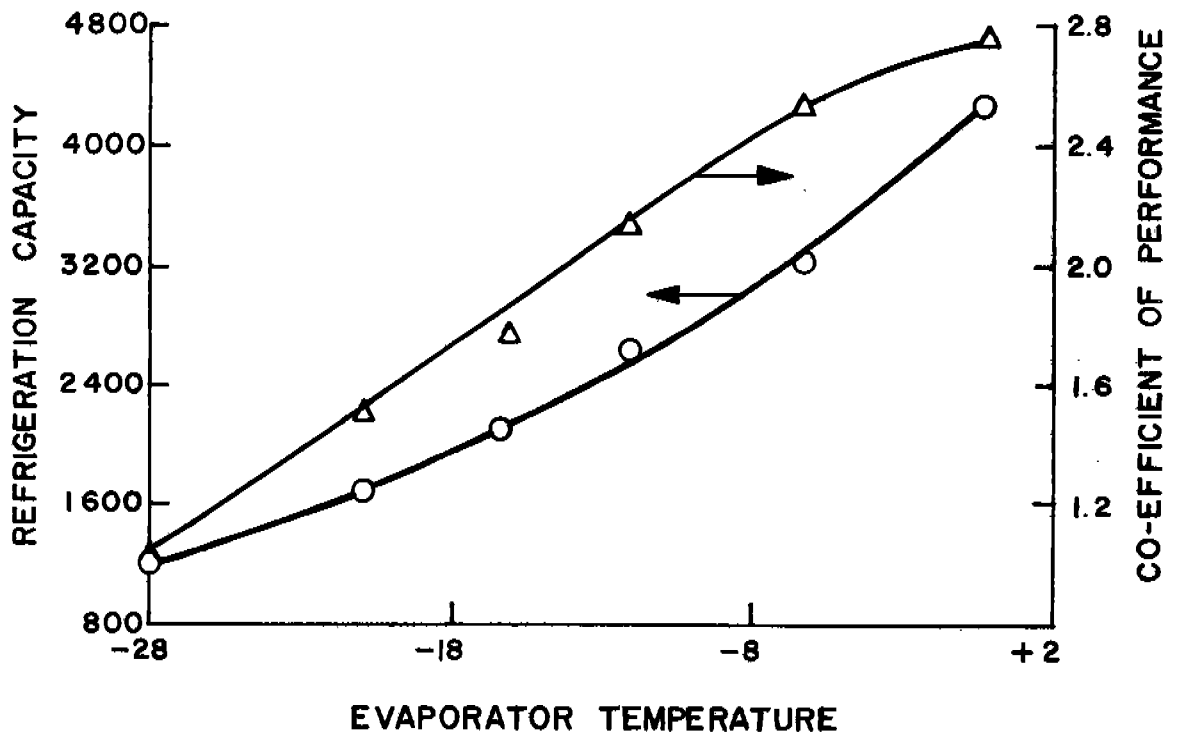
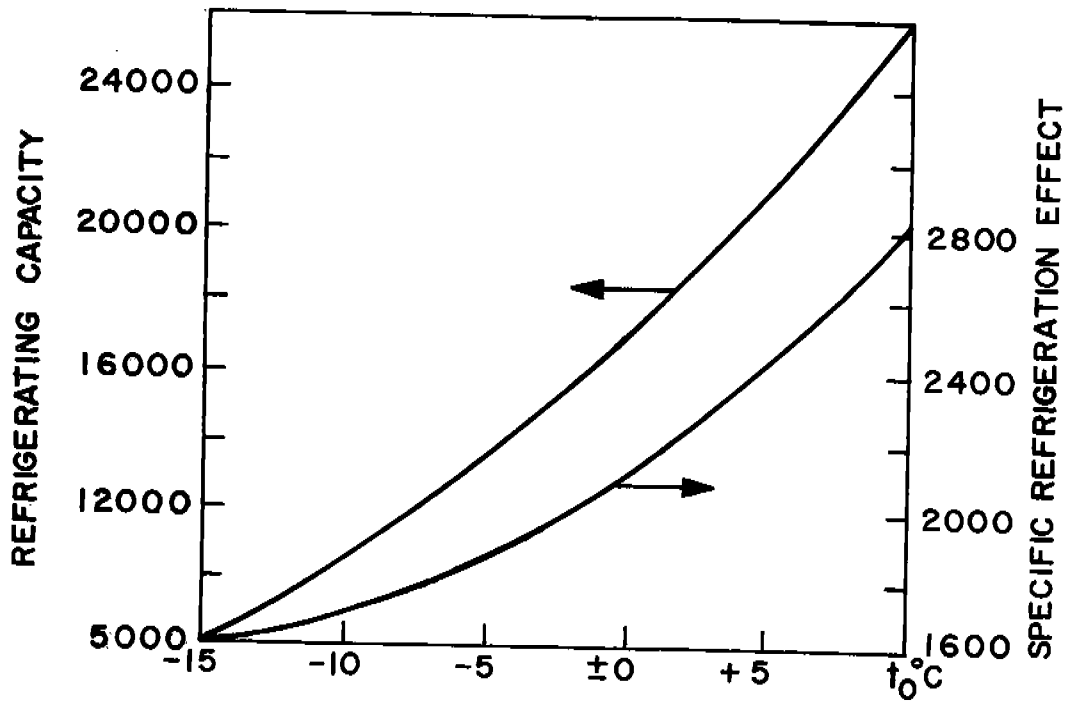


FIGURE 4-1



$t_c = 50$ C.C = 4.66%

RESULTS OF CHAIKOVSKY ET AL [3]

FIGURE 4-2

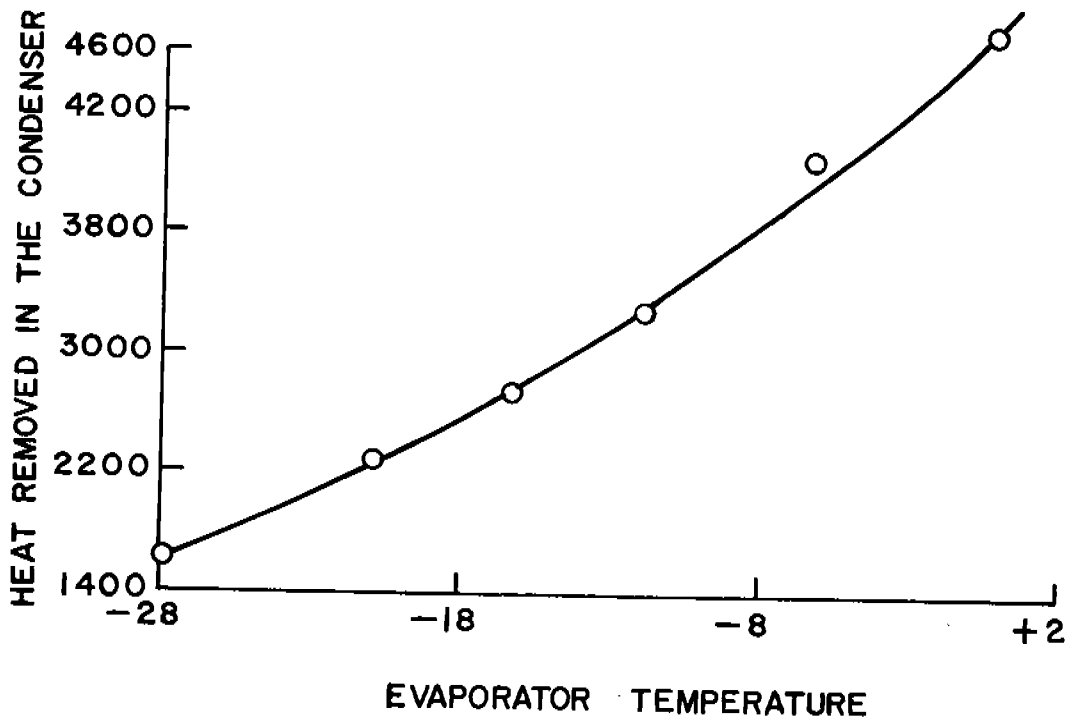


FIGURE 4-5

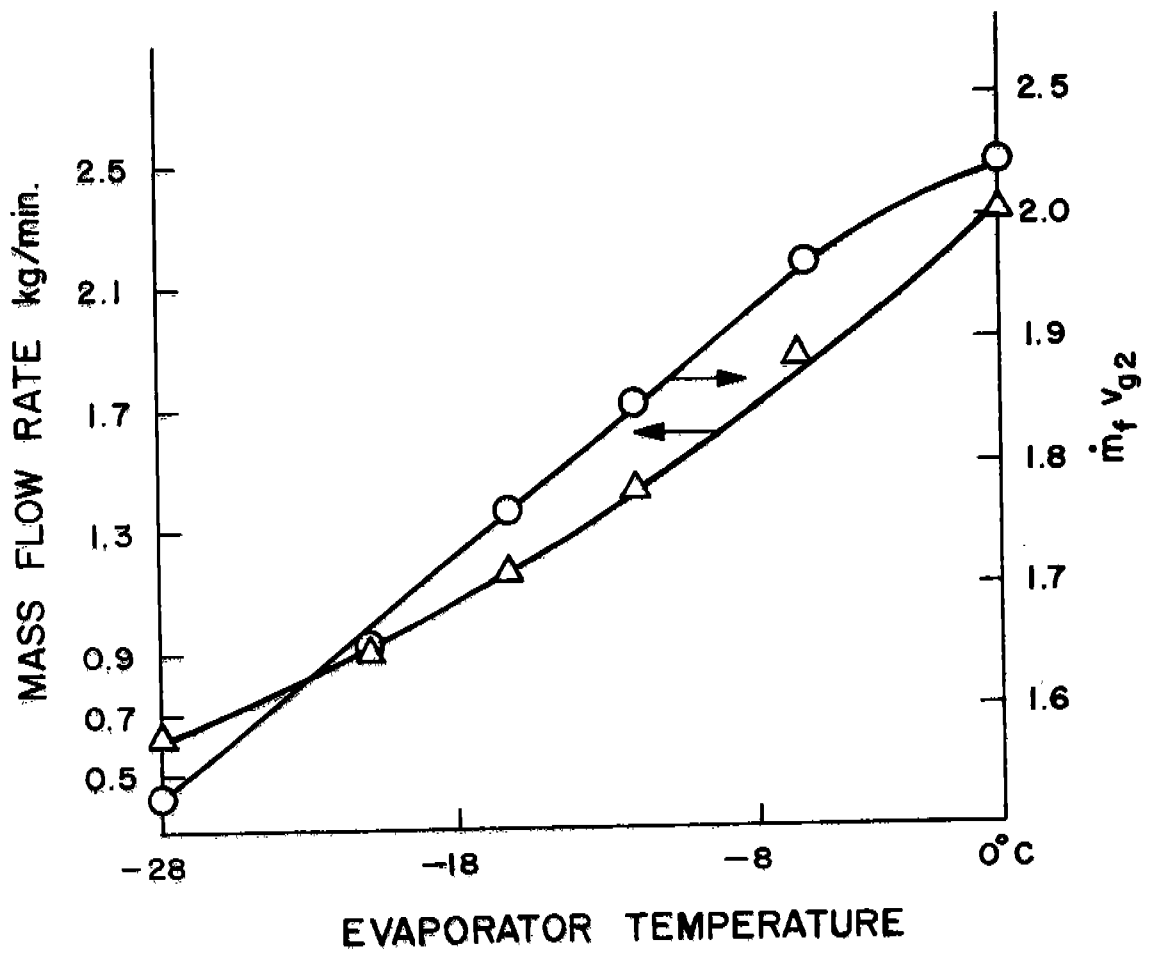


FIGURE 4-4