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The Influence of the Thermodynamic Cycle Parameters on the Performance of Refrigerators and Freezers

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THE INFLUENCE OF THE THERMODYNAMIC CYCLE PARAMETERS

ON THE PERFORMANCE OF REFRIGERATORS AND FREEZERS

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ABSTRACT

The designs of domestic refrigerators and freezers have gone through major cosmetic changes to enhance customer appeal, but the principal parameters of the thermodynamic cycle have not changed much since the advent of these devices. In this paper the results of a parametric study are presented which show the effects of the main thermodynamic cycle parameters on the performance of refrigerators and freezers with particular reference to the variations of the ambient temperature over a wide range. The results show that the coefficient of performance and the refrigeration capacity decrease sharply as the ambient temperature rises. Test results support these findings. These variations may be reduced quite significantly by using a vaiable displacement compressor.

NOMENCLATURE

A area (m^2)
C_p specific heat at constant pressure (kJ/kg-K)
h enthalpy (kJ/kg)
$\dot{m}$ mass flow rate (kg/s)
T_c condensing temperature (K)
T_a ambient temperature (K)
T_1 compressor entry temperature (K)
$T_{2'}$ compressor discharge temperature (K) for isentropic compression
η compressor isentropic efficiency

INTRODUCTION

The refrigerators and freezers are important machines which are used to lower temperature in a closed space by removing heat and pumping it to the environment which is at higher temperature. A system diagram of a small refrigerator and the thermodynamic cycle, which describes the refrigeration process, are shown in Fig.1(a) and Fig. 1(b) respectively.

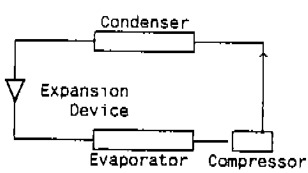


Fig. 1 (a) Block diagram of vapour compression cycle

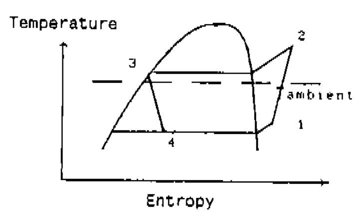


Fig. 1 (b) Temperature - Entropy diagram for the refrigeration cycle

The working fluid in slightly superheated state at point (1) is compressed to point (2) in order to raise its temperature well above the ambient temperature. Because of the resulting positive temperature gradient heat can be rejected to the environment and the state of the working fluid changes from superheated vapour to slightly subcooled liquid at point (3). The throttling process from point (3) to point (4) lowers the pressure and thus the temperature well below the temperature in the enclosed space which is being cooled. Evaporation of the working fluid from point (4) to point (1) takes place as a result of heat removal from the enclosed space. The cycle continues until the temperature in the enclosed space almost reaches the same value as the temperature of the working fluid.

Figure 1 (b) shows that heat transfer from the enclosed space would be expected to depend on the difference between the mean temperature of the working fluid from point (2) to point (3) and that of the environment. The upper cycle pressure and hence the temperature are controlled by the cycle compression ratio for a fixed pressure drop during the throttling process. But the temperature at which heat must be rejected depends on the environmental conditions. At certain times during the 24 hours cycle the temperature difference may be very small, thus adversely affecting the rate of heat transfer from the fluid to the environment per unit mass flow rate of the working fluid.

The amount of heat transfer may be increased by raising the cycle pressure ratio provided the system incorporates either a variable displacement compressor or an electronically controlled expansion valve. However, raising the cycle pressure ratio by means of the expansion valve would also reduce the flow rate, cosequently the increased heat rejection per unit mass flow rate due to higher cycle temperature might be off-set by the reduced mass flow rate of the working fluid.

THERMODYNAMIC CYCLE PARAMETERS

Referring to Fig.1(a) the performance of the refrigeration cycle can be defined in terms of the following parameters:

refrigeration capacity = heat removed by the working fluid $= h_1 - h_4$

work done on the working fluid $= h_2 - h_1$

heat rejected to the environment $= h_2 - h_3$

throttling or expansion process $= h_3 - h_4 = 0$ (constant enthalpy process)

$$\text{coefficient of performance} = \frac{\text{refrigeration effect}}{\text{work done}} = \frac{h_1 - h_4}{h_2 - h_1} \qquad (1)$$

In order to study the effects of such parameters as the cycle pressure ratio, the efficiency of the compressor, the effectiveness of the condenser as a heat exchange element, the thermodynamic properties of the working fluid, etc., equation (1) may be written as follows:

$$\mathrm{COP} = \frac{(h_2 - h_3) - (h_2 - h_1)}{(h_2 - h_1)}$$

$$= \frac{h_2 - h_3}{h_2 - h_1} - 1$$

$$= \frac{\text{heat rejected to the environment}}{\text{compressor work}} - 1 \qquad (2)$$

It is interesting to note from equation (2) that the coefficient of performance of a refrigerator or freezer reduces as either the compressor work increases, for example due to a drop in compression efficiency, or heat rejected to the environment reduces because of the rising ambient temperature.

Heat rejected from the condenser to the environment depends on many factors including the temperature and the thermal properties of the working fluid, the thermal conductivity of the condenser tube material, temperature of the environment, etc. Similarly, compressor work is a function of the cycle pressure ratio, the temperature of the working fluid at compressor entry and the efficiency of the compression process. Hence, substituting these parameters in equation (2), the following expression can be obtained:

$$\mathrm{COP} = \frac{U_o\, A\, (T_c - T_a)}{\dot{m}\, C_p \dfrac{T_1}{\eta} \left\{ \left(\dfrac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\}} \qquad (3)$$

The evaporator heat gain q_{ev} $= h_1 - h_4$

$= (h_2 - h_3) - (h_2 - h_1)$

$= q_{condenser} - W_{compressor}$

Finally,

$$q_{ev} = U_o A (T_c - T_a) - \dot{m}\, C_p \frac{T_1}{\eta} \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right\} \quad (4)$$

and compression efficiency is given by the following expression:

$$\eta = \frac{h_2' - h_1}{\text{comp. work}} = \frac{COP + 1}{COP} \cdot \left(h_2' - h_1 \right) \quad (5)$$

The effects of the important cycle parameters can be studied by solving equations (3), (4) and (5) for a range of values of the ambient temperatures and of other variables. A computer program was written in Turbo Basic to solve these equations. The program is described in the following:

COMPUTER PROGRAM

The propgram runs under MS-DOS (Microsoft Disc Operating System) on any of the IBM compatible micro computers. A flow diagram of the program is shown in Fig. 2 and its important details are given below.

Program Structure

The program ha been written in the modular form comprising four modules, namely:

Data entry module — This module allows the user to build a data file for the working fluid

Run time module — This module calls for the operational data which must be entered before running the program. It incorporates a linear interpolation routine to find the fluid properties from the stored table corresponding to the specified operational data.

Output module — This module allows the output to be directed to either the screen, or a printer/plotter.

Operation of the Program

The data entry module allows the user to enter data either from the keyboard (first time run) or to read from a data file. The properties data are stored in the computer memory in the form of a 'look up table'.

The operational data is entered every time the program is run. The fluid properties that correspond to the values of temperatures, pressures, etc. entered as the operational data are found by using a linear interpolation subroutine. The run time module calculates the performance parameters, i.e. the coefficient of performance, refrigeration capacity, etc. The results are directed to the printer by the output device.

At the end of a run the user can either terminate the program or enter another set of operational data and repeat the sequence from the run time module. This option is not shown on the flow diagram because of lack of space.

In order to run the program it is necessary to know the values of the overall heat transfer coefficient U_o, the condenser surface area A and the refrigerant mass flow rate $\dot{m}$. These values were obtained from a real life machine and are given in the following:

overall heat transfer coefficient U	$= 0.0273$ kJ m^{-2} S^{-1} K^{-1}
condenser area A	$= 0.392$ m^2
refrigerant mass flow rate $\dot{m}$	$= 1.184 \times 10^{-3}$ m^3 S^{-1}

In this investigation, U, A and $\dot{m}$ were held constant.

The variations of the specific heat (Cp) at constant pressure and of the ratio of specific heats (γ) with the saturation temperature and two values of the degree of superheat are shown in Fig. 3. The calculated values were also stored in a look up table which was used by the program.

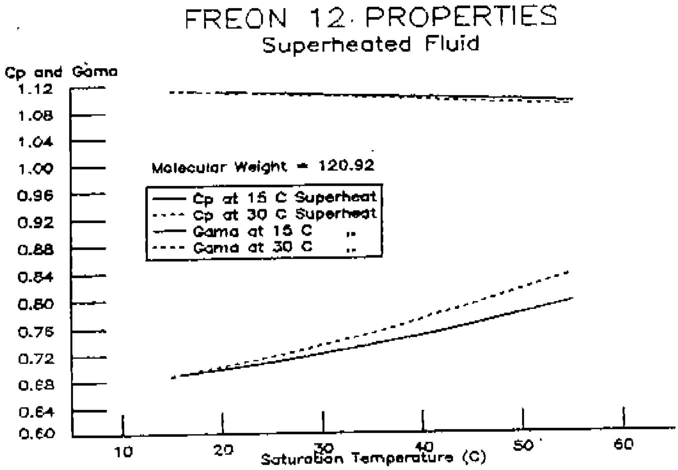


Fig. 3. The variations of the specific heat (Cp) at constant pressure and the ratio of specific heats (γ) with cycle pressure ratio and ambient temperature.

RESULTS OF THE PARAMETRIC STUDY

The results of the parametric study are shown if Fig. 4 to Fig. 7. In all cases the ordinates represent one of the two performance parameters, i.e. the coefficient of performance or the refrigeration capacity and the abcissa is used for the independent variable, i.e. the ambient temperature.

Figure 4(a) shows the effect of cycle pressure ratio on the coefficient of performance. At constant pressure, COP decreases rapidly as the ambient temperature

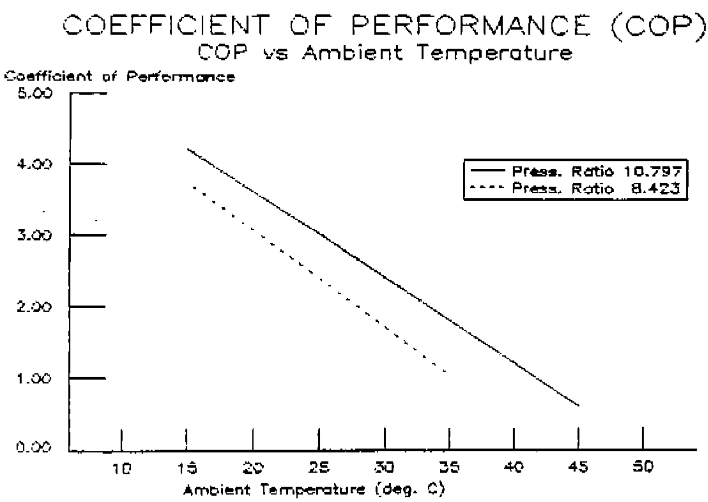


Fig. 4(a) The effect of cycle pressure ratio and ambient temperature on coefficient of performance

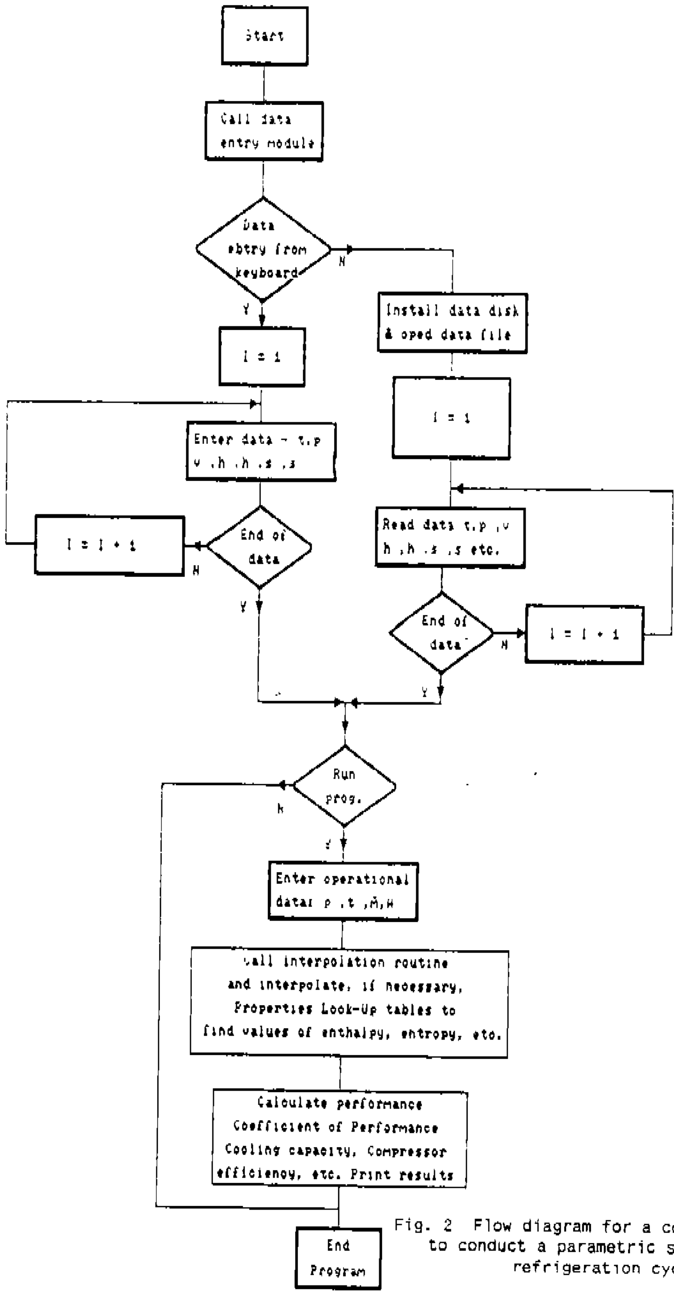


Fig. 2 Flow diagram for a computer program to conduct a parametric study of the refrigeration cycle

rises. For example at the cycle pressure ratio of 10.797, COP reduces from 4.216 to 0.614 as the ambient temperature rises from 15 °C to 45 °C, a reduction of 85.4%. It should be noted, however, that COP may be held constant by increasing the cycle pressure ratio with rising ambient temperature.

The graphs of refrigeration capacity vs ambient temperature are shown in Fig. 4(b) for the two values of the cycle pressure ratio. At constant cycle pressure ratio, the refrigeration capacity also reduces quite rapidly as the ambient temperature rises. For example, when cycle pressure ratio is 10.797 the cooling capacit reduces from 0.3758 to 0.0547 as the ambient temperature rises from 15 °Ctc 45 °C, a rection of approximately 85%.

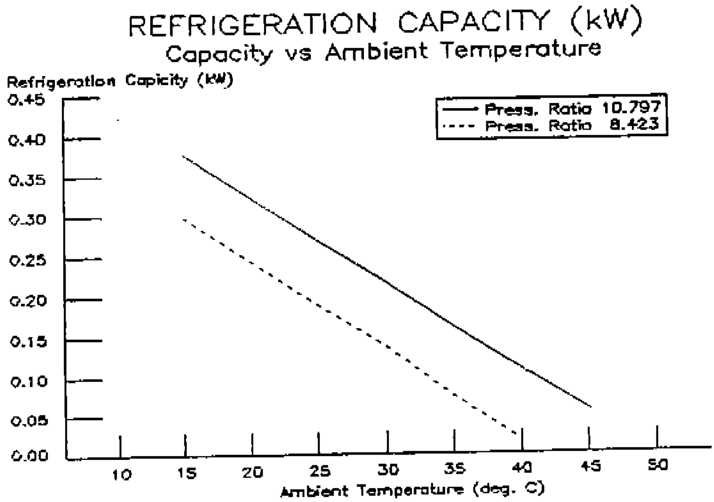


Fig. 4(b) The effect of cycle pressure ratio and ambient temperature on refrigeration capacity

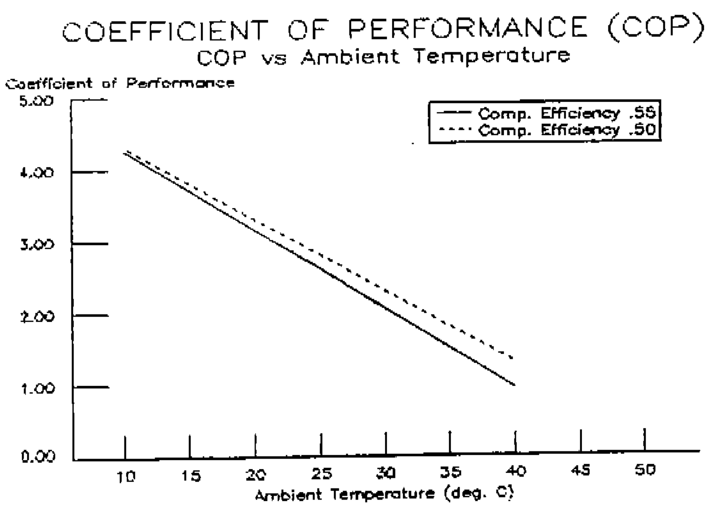


Fig. 5(a) The effect of compression efficiency and ambient temperature on coefficient of performance

The variations of COP with ambient temperature, for two values of the compression process efficiency are shown in Fig. 5(a). In this case also COP reduces quite sharply from 4.266 to 0.964 as the ambient temperature rises from 15 °C to 45 °C. This represents a reduction of 77.4%.

It is interesting to note that at the same temperature, COP reduces as the isentropic efficiency of the compression process increases. Furthermore, the amount of reduction in the value of COP becomes larger with rising ambient temperature. This observation is indeed peculiar but it can be explained quite simply by referring to the temperature - entropy diagram, Fig. 1(b). The amount of heat rejected by the condenser is represented by the area enclosed by the condensing process line and the anbient temperature line. As the compression process efficiencey reduces at a constant value of the compressor delivery pressure, point 2 moves to the right, hence the heat rejected during condensation increases. This leads to a rise in the value of COP, (eqn. 2).

The variations of refrigeration capacity with the ambient temperature are shown in Fig. 5(b) for two values of the isentropic efficiency of the compression process. Again a sharp drop in the refrigeration capacity can be seen as the ambient temperature rises. At the same ambient temperature, refrigeration capacity increaes as the efficiency of the compression process reduces.

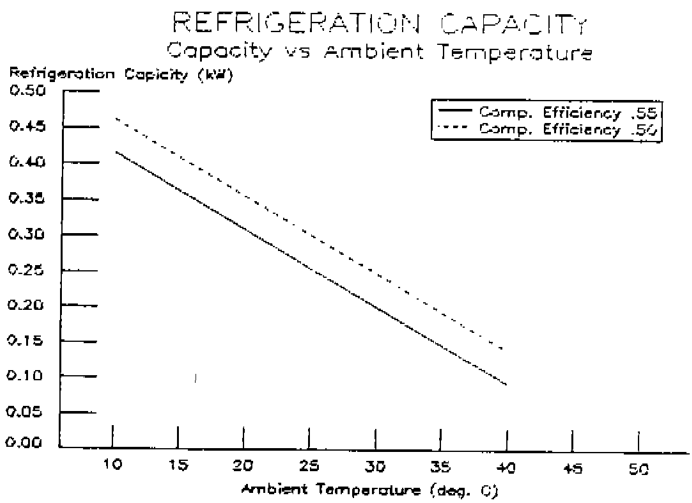


Fig. 5(b) The effect of compression efficiency and ambient temperature on refrigeration capacity

Figure 6(a) shows the variations of COP with ambient temperature for three values of the area of the condenser. At constant ambient temperature, the coeficient of performance increases as the condenser area is increased. It can be seen also that the coefficient of performance may be kept constant by increasing the condenser area but, because of its high cpital cost, a variable area condenser would not be acceptable, particularly for domestic machines. For a constant area condenser the refrigeration capacity also reduces quite sharply as the ambibient temperature rises.

THE REAL CYCLE

The domestic refrigerators and freezers are normally fitted with capillary tubes, which are inexpensive expansion devices and offer a certain degree of flow regulation to suit varying operating conditions. It must be mentioned, however, that pressure drop along the length of a capillary tube cannot be determined easily due to the continuous change of phase of the refrigerant. LI et al (1) studied the flow of R12 refrigerant through capillary tubes with particular reference to metastable conditions and Kuijpers and Jensen (2) invetigated the influence of thermal

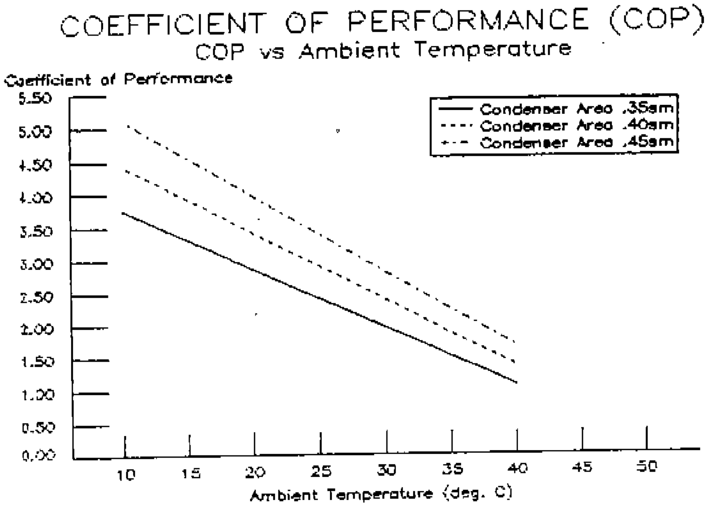


Fig. 6(a) The effect of condenser area and ambient temperature on coefficient of performance

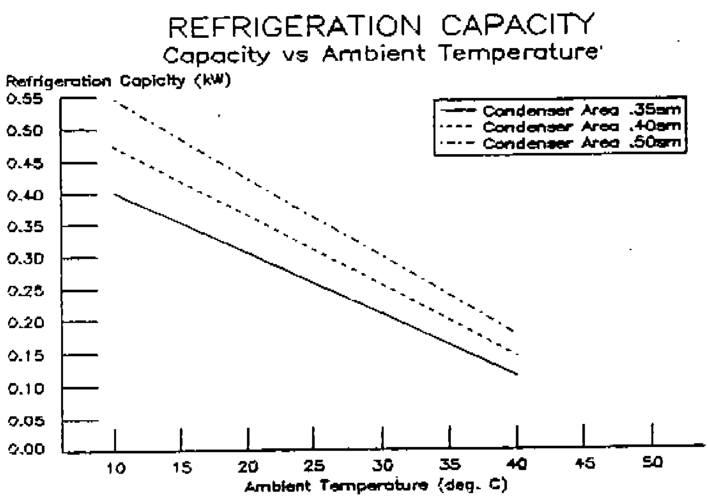
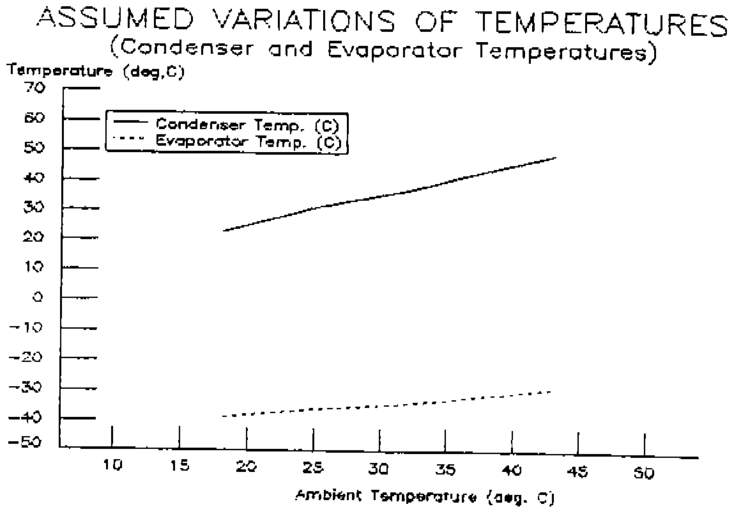


Fig. 6(b) The effect of condenser area and ambient temperature on refrigeration capacity

non-equilibrium on the mass flow rate through capillary tubes. Both papers make a significant contribution to the understanding of the nature of flow, but there is still lack of design orientated methods for predicting pressure drop as a function of tube length and the dryness fraction of the working fluid.

The overall pressure ratio vs mass flow rate characteristics that would satisfy the operational requirements of refrigerators and freezers at varying operting conditions are quite different from those offered by capillary tubes. Rising ambient temperature would impose increasing load on the machine, which, in turn would call for increased mass flow rate. The mass flow rate through a capillary tube appears to reduce as the temperture of the fluid increases due to the rising ambient temperature.



Fig, 7(a) Measured variations of the condenser and the evaporator temperature variations with the ambient temperature

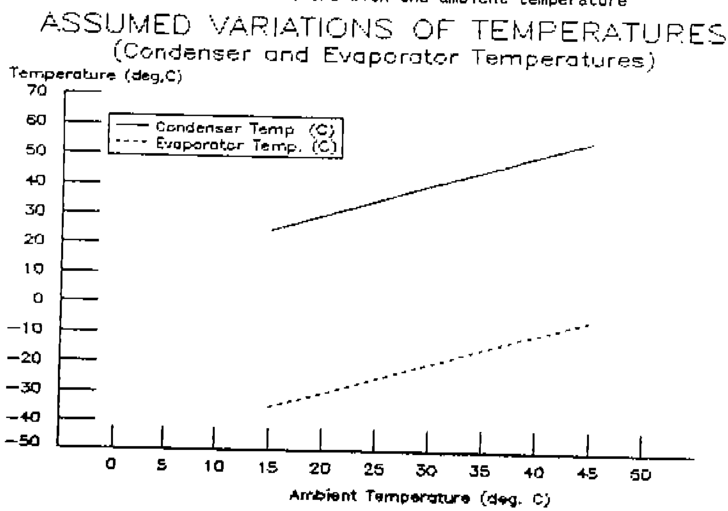


Fig. 7(b) The assumed variations of condenser and evaporator temperatures with the ambient temperature

Tests were carried out on several tropicalized refrigerators and freezers in order to examine the variationns of the condenser and the evaporator temperatures due to rising ambient teperature. A typical set of data is shown in Fig 7(a). The parametric study was then extended to investgate the effect of the assumed variations of the evaporator and the condenser temperatures as functions of the ambient temperature as shown in Fig. 7(b). The variations of COP and cooling capacity with ambient temperature were calculated by using the data from Fig. 7(b). The results are shown in Fig. 7(c).

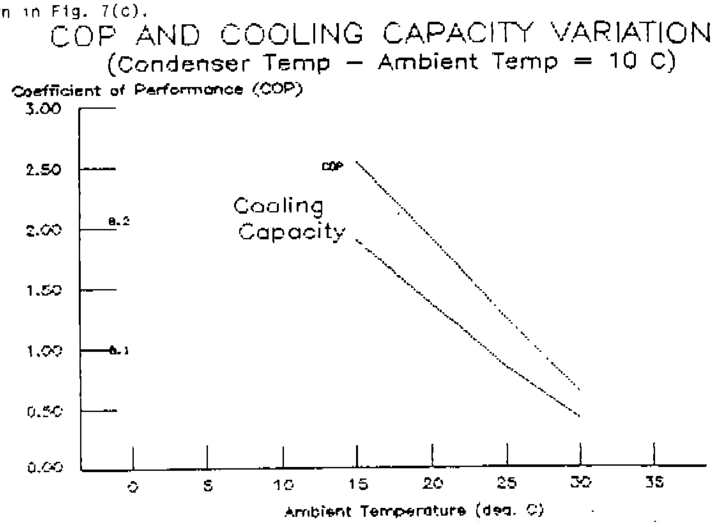


Fig 7(c) The variations of COP and refrigeration capacity calculated from the assumed variations of temperature as shown in Fig. 7(b).

It is interesting to see that in this case also COP and refrigeration capacity reduce very sharply as the ambient temperature rises. The percentage reductions in COP and cooling capacity are 87.4% and 75.7% respectively as the ambient rise from 15°C to 45 °C. These values agree closely with previously observed values, for example from Fig. 4(a) and Fig. 4(b). For the same rise of the ambient temperature, the percentage reductions in COP and cooling capacity in that case were 85% and 77.4%.

CONCLUSIONS

1. The results of a pametric study have been presented to show the influence of some of the important thermodynamic cycle parameters on the performance of domestic refrigerators and freezers.

2. It has been shown that as the ambient temperature rises from 15 °C to 45 °C, the reductions in the coefficient of perfomance and the refrigeration capacity may be as much as 85% and 75% respectively.

3. The expansion devices fitted in small refrigerating machines (capillary tubes and elecronically controlled valves) may be able to adjust the condensing pressure to be always higher than the ambient pressure, but the resulting drop in the mass flow rate would result in a significant drop in the cooling capacity.

4. The deterioration in the performance of small refrigerating machines due torising ambient temperatures may be minimised by using continuously variable displacement and speed. The design of such a compressor for small refrigerating machines is in hand and the performance of that machine will be reported at a later date.

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