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LUBRICANT RELATED PROBLEMS WITH HEAT PUMPS

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ABSTRACT

Oil is required in refrigeration and heat pump systems for lubrication of the compressor, expansion valve and other moving parts. However, the oil is often highly miscible with the refrigerant, in proportions which depend on pressure and temperature, as well as oil and refrigerant type. As some of the oil circulates around the system with the refrigerant, it follows that at all times a proportion of the refrigerant will be dissolved, as a liquid, in the oil. Not only does this oil-refrigerant miscibility give rise to dilution of the oil, hence impairing its lubricating properties, but it also leads to a reduction in evaporator capacity. This is because, at the evaporator outlet, some of the refrigerant remains in the liquid state, dissolved in the oil, and is thus unavailable for evaporation to carry latent heat. This effect on its own would not be too serious, were it not that the coefficient of performance (COP) is also impaired due to oil circulation. This results from the fact that, while evaporator capacity is reduced, compressor power is largely unaffected by the oil circulation rate. This of course implies a reduced COP. In a heat pump design, COP is usually all-important, and this paper examines some of the effects of oil circulation on heat pump performance. Particular attention is drawn to designs using oil-flooded rotary sliding-vane compressors, where oil circulation is usually much

greater than with reciprocating compressors.

BACKGROUND

In this laboratory we have been concerned for some time in the design and development of vapour compression heat pumps for a range of applications, and it has usually been our experience that performance is not quite as good as would be expected from the compressor manufacturer's performance charts. There are many reasons for this, but the short-fall in performance is not usually very great. However, recent tests using a rotary sliding-vane compressor showed up an abnormally wide and unacceptable discrepancy between observed performance and the manufacturer's data under certain conditions. Another factor was the consistent measurement of liquid line temperatures several degrees above saturation. These readings, indicating superheat, were somewhat puzzling as we could clearly see liquid refrigerant in the sight glass. After these observations, the accuracy of our measuring instruments was thoroughly checked, and the tests repeated, still with the same results.

At this point we realised that the effects of oil-refrigerant miscibility could be more important than we had thought previously, and a fairly trivial calculation using Raoult's Law showed that the presence of oil in small amounts could

change the saturation temperature to the observed extent. From here our investigations followed in the footsteps of such as Bambach (1955) and Spauschus (1963).

OIL-REFRIGERANT INTERACTION

Oil is miscible with fluorocarbon refrigerants to a varying degree. In the case of liquid R12 it is totally miscible, the mixture forming a single phase at all temperatures and pressures. The effects produced by the oil in a heat pump system can be summarised as follows:

1. It changes the working fluid from a pure refrigerant with well known properties to a poorly understood mixture, with properties which depend on the oil type and concentration.
2. It can affect the heat transfer processes in the evaporator and condenser. This effect can be either to improve or impair heat transfer, depending on oil concentration (Green, 1963).
3. The boiling point of the mixture is elevated above that of the pure refrigerant, according to Raoult's Law.
4. The heat carrying capacity of the mixture is reduced during evaporation because the oil holds a proportion of the refrigerant in the liquid phase, thereby reducing the latent heat pick-up. As will be shown later, this effect implies a reduction in the coefficient of performance (COP).

It follows from these points that the presence of oil in a heat pump circuit could have a significant effect on its performance. The following analysis is an attempt to predict and quantify the likely effects.

THEORY

A study of the effects of oil circulation on air conditioner capacity was made by Cooper and Mount (1972). They made use of the earlier work by Bambach (1955) on the miscibility of R12-oil mixtures. The theory presented here is basically a further extension of this work, adapted to study the effects of oil circulation on COP.

The heat capacity of the evaporator is given by (for pure refrigerant)

$$Q_r = \rho_{S_r} \dot{V}_D (h_{2_r} - h_{1_r}) \eta_{V_r} \quad (1)$$

For a refrigerant-oil mixture this becomes

$$Q_m = \rho_{S_m} \dot{V}_D (h_{2_m} - h_{1_m}) \eta_{V_m} \quad (2)$$

Experiments show that volumetric efficiency may be affected by oil circulation, but as a first approximation we can assume that $\eta_{V_m} = \eta_{V_r}$

thus

$$\frac{Q_m}{Q_r} = \frac{\rho_{S_m}}{\rho_{S_r}} \frac{(h_{2_m} - h_{1_m})}{(h_{2_r} - h_{1_r})} \quad (3)$$

or

$$\frac{Q_m}{Q_r} = f_\rho \cdot f_h \quad (3a)$$

Now, $\rho_{S_m} > \rho_{S_r}$ but $(h_{2_m} - h_{1_m}) < (h_{2_r} - h_{1_r})$

therefore it is not possible to say whether Q_m is greater or less than Q_r without some calculation. The denominator of equation 3 is readily found using published refrigerant properties, but determination of the numerator requires knowledge of the oil-refrigerant solubility. Bambach (1955) has produced a solubility equation for R12 - paraffinic oil mixtures, so we will make use of his equation by considering this particular combination.

As the mixture leaves the evaporator, some refrigerant remains trapped in the oil, as liquid. This reduces the amount of refrigerant available to carry latent heat.

If the following quantities are defined:

x = mass of oil/total mass
(oil fraction)

ω = mass of refrigerant in the oil/
mass of liquid mixture

then we can write

z = mass of liquid mixture/total mass
(liquid fraction)

$$z = x/(1 - \omega)$$

Hence $(1 - z)$ is the vapour fraction.

A value of ω can be found for a particular temperature and pressure from the equations given by Bambach:

For temperatures below 0°C :

$$\log_{10} P = 4.9972 - 0.558\omega^{-\frac{1}{2}} - \frac{1177.67 - 98.753\omega^{-\frac{1}{2}}}{T} \quad (4)$$

$$= A$$

For temperatures above 0°C:

$$\log_{10} P = A - (T-273.16)[0.002338(\omega-0.6)^2 - 0.000075] \quad (4a)$$

The enthalpy of the oil can be approximated by integrating its specific heat, and taking, for consistency, -40°C as the reference temperature. The specific heat of a typical paraffinic oil is given by:

$$C_p = 1.754 + 0.0038t \quad (5)$$

$$\text{thus } h_o = 1.754t + 0.0019t^2 + 67.12 \quad (6)$$

The liquid mixture enthalpy entering the evaporator will be:

$$h_{1m} = (1-x) h_{1RL} + x h_{1o} \quad (7)$$

At the evaporator exit the liquid mixture enthalpy will be:

$$h_z = (1-\omega) h_{2o} + \omega h_{2RL} \quad (8)$$

and the enthalpy of the vapour is that of pure refrigerant vapour. Thus the enthalpy of the total mixture including vapour will be:

$$h_{2m} = z h_z + (1-z) h_{2RV} \quad (9)$$

Subtracting equations 7 and 9:

$$(h_{2m} - h_{1m}) = x (h_{2o} - h_{1o}) + z\omega (h_{2RV} - h_{1RV}) + (1-z) (h_{2RV} - h_{1RL}) \quad (10)$$

The three terms of equation 10 represent the enthalpy changes in: the oil, the refrigerant which remains in the liquid state, and the refrigerant which evaporates, respectively.

An average value for the mixture density ρ_m can be found as follows.

For the liquid mixture:

$$\frac{1}{\rho_z} = \frac{\omega}{\rho_{sRL}} + \frac{1-\omega}{\rho_{so}} \quad (11)$$

The density of the total mixture is then given by:

$$\frac{1}{\rho_{sm}} = \frac{z}{\rho_z} + \frac{1-z}{\rho_{srv}} \quad (12)$$

The density of a typical paraffinic oil at temperature t_s is:

$$\rho_{so} = 932.47 - 0.6298 t_s$$

The refrigerant liquid and vapour densities can be found from the relevant published data. (The effects of volume contraction and the heat of mixing of refrigerant and oil have been neglected in this analysis as being small, following Bambach.)

Equation 3 may now be evaluated using equations 10 and 12, to find the ratio Q_m/Q_r , which is the evaporator capacity correction factor due to oil circulation.

Now let us consider the effects of oil circulation on COP. In order to do so, we must make an assumption about the effect of the oil on compressor power. The presence of oil in the compressor suction line will undoubtedly affect the density of the suction fluid, thus affecting the mass flow. However, some of the refrigerant will be dissolved in the oil, as liquid, and because of the changing equilibrium of oil-refrigerant solubility during the compression process, the precise influence of oil circulation on compressor power cannot be determined theoretically. Nevertheless, two reasonable, alternative assumptions are possible:

1. Compressor power is unaffected by oil circulation.
2. Compressor power varies in proportion to changes in suction density due to oil circulation.

Neither of these assumptions is likely to be correct, but as will be seen, they are both sufficiently accurate for the purposes of this analysis.

Heat pump COP is given by (ignoring heat losses):

$$COP_H = \frac{Q_E + W}{W} \quad (13)$$

Thus the pure refrigerant COP is:

$$COP_{Hr} = \frac{Q_r + W_r}{W_r} \quad (14)$$

and with oil circulation:

$$COP_{Hm} = \frac{Q_m + W_m}{W_m} \quad (15)$$

To compare COP_{Hm} and COP_{Hr} , we need to know Q_m and W_m in terms of Q_r and W_r . Q_m can be found from equation 3. For case (1), constant compressor power, $W_m = W_r$, therefore:

$$\frac{COP_{Hm}}{COP_{Hr}} = \frac{(Q_m + W_r)}{W_r} \cdot \frac{W_r}{(Q_r + W_r)} \quad (16)$$

Substituting from equation 3a:

$$\frac{COP_{Hm}}{COP_{Hr}} = \frac{f_p f_h (COP_{Hr} - 1) + 1}{COP_{Hr}} \quad (17)$$

For case (2), density-proportional compressor power,

$$W_m = f_p W_r, \text{ therefore:}$$

$$\frac{COP_{Hm}}{COP_{Hr}} = \frac{(Q_m + f_p W_r)}{f_p W_r} \frac{W_r}{(Q_r + W_r)} \quad (18)$$

Again, substituting from equation 3a:

$$\frac{COP_{Hm}}{COP_{Hr}} = \frac{f_h (COP_{Hr} - 1) + 1}{COP_{Hr}} \quad (19)$$

Equations 17 and 19 can be solved to obtain COP_H correction factors (representing "worst" and "best" cases respectively) given the refrigerant and oil properties, and the oil fraction in the system.

COMPUTER MODEL

A computer program based on the foregoing theory was used to study the effects of oil on heat pump performance. The refrigerant considered was R12, and the oil assumed to be a paraffinic type. The program calculated the pure refrigerant COP_{Hr} given

evaporating and condensing temperatures, evaporator superheat, liquid subcooling, compressor isentropic efficiency and drive motor efficiency. The COP_{Hr} figures

obtained in this way were probably somewhat optimistic, but as the COP_H correction factor (equations 17 and 19) is only a weak function of COP_{Hr} , no great loss of accuracy occurred.

It should be noted that, with an oil-refrigerant mixture, the definition of superheat is strictly meaningless, as there is always some liquid present. However, "apparent" superheat can still be observed, therefore superheat will be defined here as:

(bulk fluid temperature) - (saturation temperature of pure refrigerant at the existing pressure)

This definition is correct for pure refrigerant, and is the same as would be measured in practice in the presence of oil.

Figure 1 shows the effects of oil circulation and suction superheat on evaporator capacity. These curves are similar to those produced by Cooper and Mount, and show clearly how, contrary to expectation, evaporator capacity is reduced at low superheat levels, with even a small

amount of oil.

The effect of oil circulation on the COP_H correction factor is very similar to that on evaporator capacity, as shown in Figure 2. Figure 3 illustrates the dramatic effect of oil circulation at low suction superheat levels.

It must be emphasised that the COP_H correction factors obtained from these curves apply to the pure refrigerant COP_H which would be realised at the same superheat. In a heat pump with a fixed source temperature, an increase in superheat reduces the evaporating temperature, thereby also reducing the COP_H . A pure refrigerant heat pump thus, in theory, performs best with no suction superheat. However, when oil is present, as in a real heat pump, this effect is opposed by the influence of the oil-refrigerant solubility which tends to reduce COP_H at low superheat. Interaction of these two effects could be expected to produce an optimum superheat level for a particular heat pump. We have, in fact, often observed this effect in experimental work.

Figure 4 shows COP_H plotted against suction superheat, as predicted by the oil-refrigerant solubility effects model. These curves clearly show that the optimum superheat level rises with increasing oil circulation.

The effect of liquid subcooling on the COP_H correction factor is very small, except at high oil concentrations, as shown in Figure 5.

DISCUSSION

Heat pumps using reciprocating compressors normally have very low oil circulation (typically less than 5%), so that effects on COP_H are not serious. However, certain other types of compressor, notably the oil-flooded, rotary, sliding-vane type, require a high oil circulation ratio, possibly more than 10% in some cases. The effect on COP_H can then become quite significant, and some thoughts need to be given to ways of reducing the problem. This does not appear to be a simple task, but here are some possible lines of approach.

1. Reduction of the oil/refrigerant ratio. This will certainly reduce the effect on COP , and the oil charge should always be kept to a minimum for this reason. However, the oil is necessary for lubrication and sealing with most current compressors, and

reduction of oil circulation below a certain level will lead to other problems.

superheat levels on existing heat pump designs.

2. Optimisation of suction superheat. Figure 4 shows the optimum superheat effect produced by oil circulation, and it is probably worth experimenting to find the best superheat setting on a heat pump installation.
3. The use of an alternative refrigerant with a lower solubility in the oil (or a different oil, perhaps). This is possibly the most attractive approach, and we are investigating this further at present.
4. Reduction of oil circulation through the evaporator, by means of an oil separator in the compressor discharge pipe. At first sight this seems to be the obvious solution to the problem. However, if this approach is tried, another problem, possibly equally serious, is then encountered. The oil separator removes liquid from the vapour stream. Unfortunately, not only is the oil removed, but also a considerable amount of liquid refrigerant which has been dissolved in the oil. (We have found that up to 75% of the separated liquid can be refrigerant, which means that up to 30% of the refrigerant can be removed with a 10% oil charge). Not only is this liquid a rather poor lubricant (it is usually returned directly to the compressor), but the effects of "short-circuiting" a considerable proportion of the total refrigerant flow back to the low pressure side of the compressor (where much of it flashes off and has to be re-compressed), can only lead to reduced performance. Our own experimental work has confirmed this. It seems, therefore, that an oil separator is not the simple solution it appears to be.

- (5) Oil problems may be expected to be greater when rotary vane compressors are used, due to the higher oil circulation required.
- (6) There is no obvious, effective solution to the problem, although the use of alternative refrigerant-oil combinations seems the most promising approach.

NOMENCLATURE

COP_H	heat pump coefficient of performance	
C_p	specific heat at constant pressure	$\text{kJ kg}^{-1} \text{ } ^\circ\text{C}^{-1}$
f_p	density correction factor	
f_h	enthalpy correction factor	
h	specific enthalpy	kJ kg^{-1}
P	pressure	bar
Q	heat flow	kW
T	absolute temperature	K
t	temperature	$^\circ\text{C}$
\dot{V}_D	compressor displacement	$\text{m}^3 \text{ s}^{-1}$
W	compressor power	kW
x	oil fraction in total mixture	
z	liquid mixture fraction in total mixture	
ρ	fluid density	kg m^{-3}
η_V	volumetric efficiency	
w	liquid refrigerant fraction in liquid mixture	

SUBSCRIPTS

1	evaporator inlet
2	evaporator outlet
E	evaporator
L	liquid
m	total refrigerant-oil mixture
o	oil
r	pure refrigerant
s	compressor suction
v	refrigerant vapour
z	liquid mixture in total mixture

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CONCLUSIONS

The influence of oil on heat pump performance can be summarised as follows:

- (1) Oil is necessary for lubrication of the compressor, expansion valve, and other moving parts.
- (2) Oil circulation reduces evaporator capacity, the effect being more pronounced at low suction superheat levels.
- (3) Oil circulation reduces COP, in a manner similar to the effect on evaporator capacity.
- (4) The effect of oil on COP can explain the existence of optimum suction

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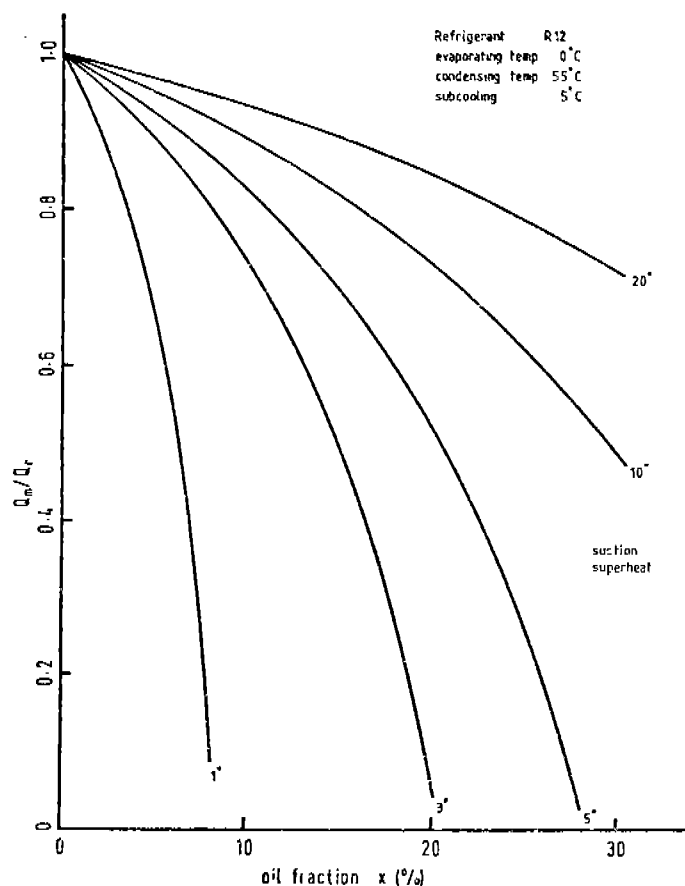


Fig 1 Effect of oil and superheat on evaporator capacity.

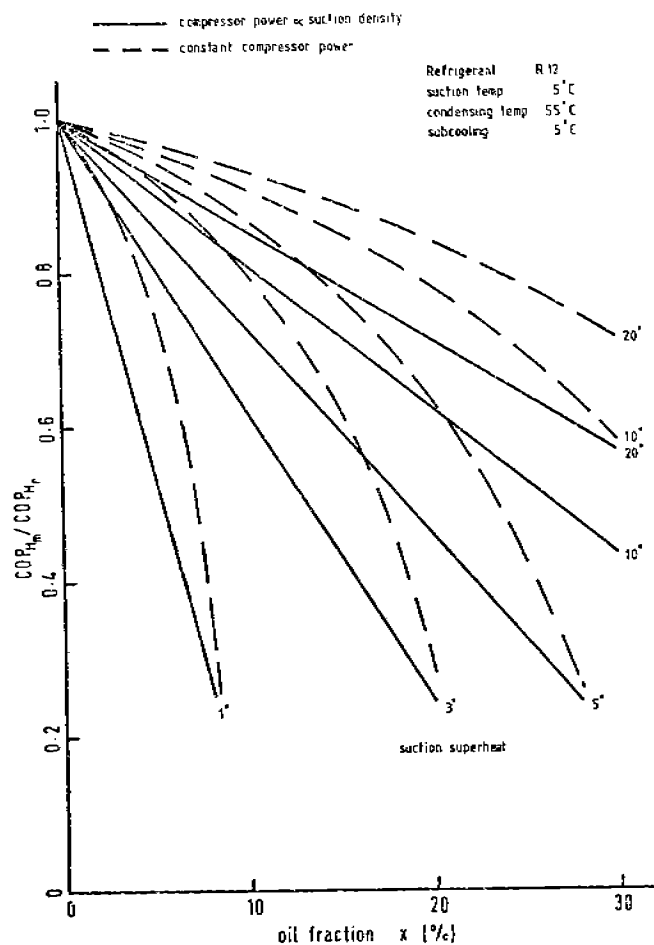


Fig. 2 COP correction factor as a function of oil fraction

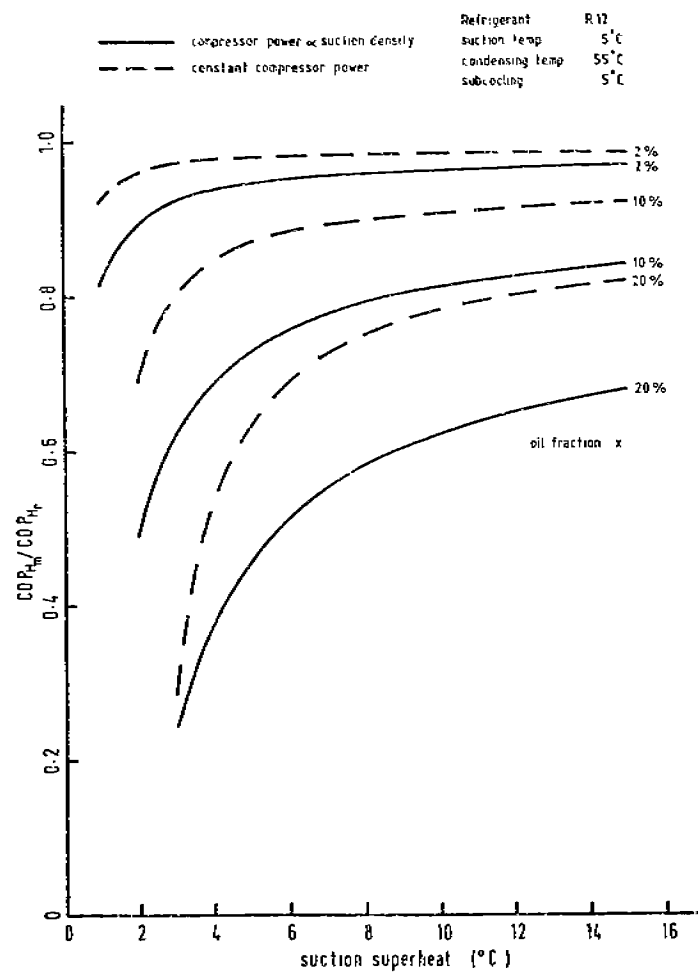


Fig. 3 COP correction factor as a function of suction superheat.

Heu

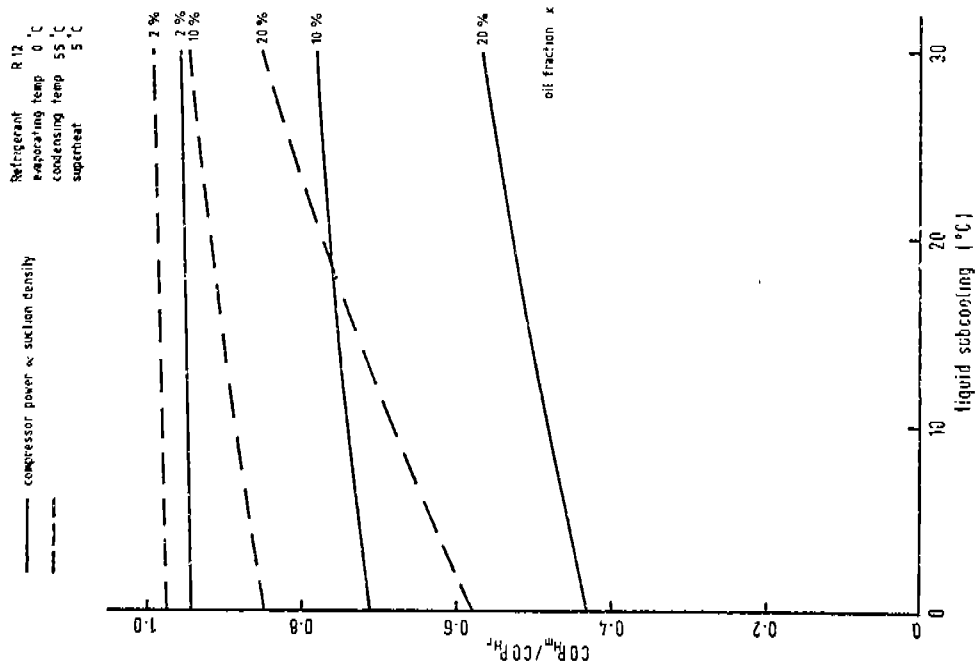


Fig. 5 Effect of liquid subcooling on COP correction factor

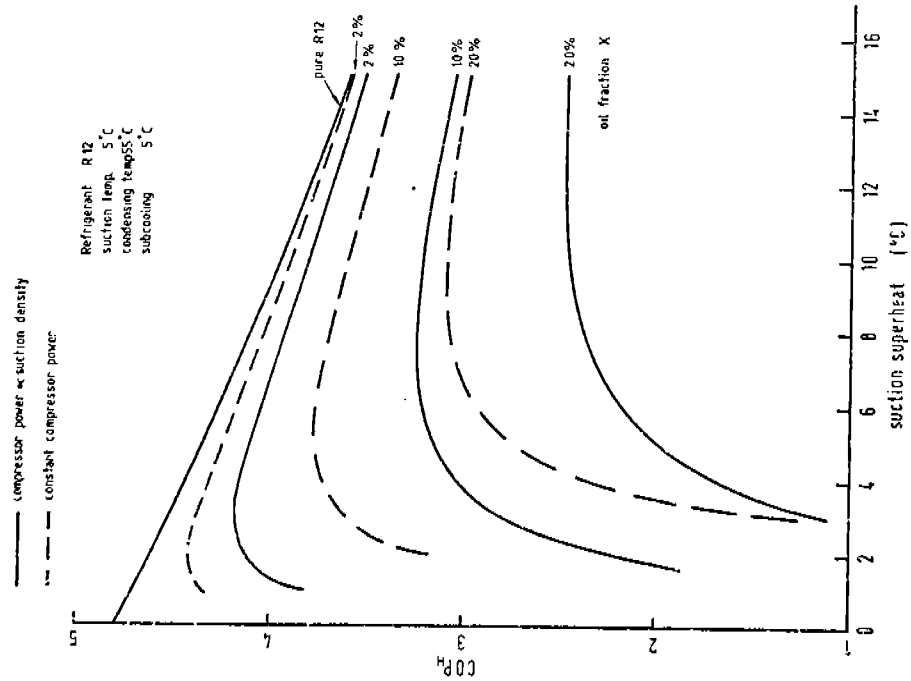


Fig. 4 Effect of superheat and oil fraction on COP_{Rh}