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EXPERIMENTAL INVESTIGATION OF THE INFLUENCE OF LUBRICATING OIL ON HEAT PUMP PERFORMANCE

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SUMMARY

Experimental results are presented which demonstrate the effect of lubricating oil on the COP and evaporator performance of refrigeration and heat pump systems. These are compared with the predictions of a simple theoretical analysis and it is shown that the agreement is satisfactory.

It is demonstrated that evaporator heat extraction can be degraded by as much as 50 per cent and the COP by up to 30 per cent when compared to the performance indicated by the properties of pure refrigerants for systems containing up to 15 per cent oil.

An interesting phenomenon is also demonstrated, wherein there is a clearly identifiable optimum setting for the evaporator superheat control, and the way in which this varies with oil concentration is indicated.

KEY WORDS Heat pumps Heat pump performance Oil in heat pump systems

INTRODUCTION

It has been shown that the presence of lubricating oil in a refrigeration or heat pump system influences performance both through its effect on the heat transfer coefficients (Green and Furse, 1963; Hughes *et al.* 1982b; Mawhinney, 1981), and through the effects of the solubility of the refrigerant in the oil (Cooper and Mount, 1972; Hughes *et al.*, 1980; Hughes *et al.*, 1982a). The present authors have already reported investigations on both effects elsewhere (see references above), and the purpose of this paper is to present the results of some experimental studies on the solubility related effects.

EXPERIMENTAL FACILITIES

Two different experimental test beds were used in these investigations. The first was a test heat pump designed to evaluate the thermodynamic characteristics of both components and systems in an operational environment. The second was a more specialized experimental facility designed specifically to investigate the influence of oil on system performance. This latter facility has been described in more detail elsewhere (Hughes *et al.*, 1982b), and only a brief description will be given here. The test rig was centred around a straight tube-in-tube evaporator with transparent sections allowing visual inspection of the refrigerant flow conditions. Facilities were included for controlling the flow of lubricating oil to the evaporator, and for its removal and separation at the evaporator discharge. This equipment allowed detailed examination of evaporator heat transfer coefficients and total heat flows under a range of operating conditions and oil-refrigerant ratios, and permitted extension and verification of the results collected on the other test rig.

The primary equipment for these experiments, however, was our main heat pump thermodynamic cycle test facility. This was an air-source, water-sink machine sited in a large controlled environment laboratory (Buick *et al.*, 1976). The laboratory air represented the atmosphere and the water circuit represented the heating load. By the use of control systems on both the air and water sides, stable, known conditions were available for evaluation of the heat pump performance. During the period of these experiments the apparatus was set up to allow substitution of different compressors, the same evaporator and condenser being used throughout.

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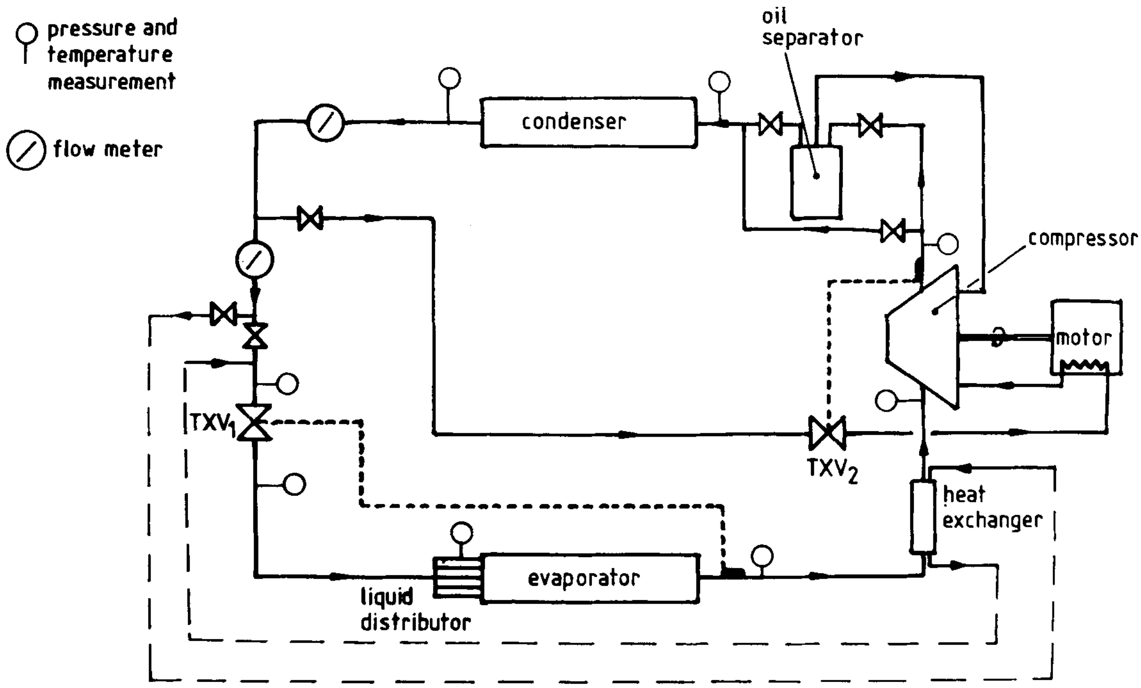

Figure 1. Schematic diagram of tests pump

The arrangement of the equipment for testing a rotary sliding vane compressor is shown in Figure 1, and it can be seen that facilities are included for liquid separation in the compressor discharge line, interstage injection to the compressor, and liquid–suction heat exchange. Refrigerant pressure and temperature were monitored at points around the system, as were the refrigerant mass flow rate, and the water flow rate and temperature differences in the external heat exchange circuits.

Initially, the temperatures were monitored using platinum resistance thermometers, but a number of serious operational problems were encountered which led to their use being discontinued. These arose because the data recording was being done continuously and automatically over lengthy periods of time. Thus, many problems which would not normally manifest themselves, if manual recording techniques were used, began to make their presence felt, resulting in temperature measurement errors that were too large to allow reasonably accurate determination of either heat transfer or refrigerant flow conditions. These effects included self heating, current supply instabilities, amplifier drift, and the general problem of resolving small differences between large signals. Random differences between the platinum resistance elements themselves were also a constant source of calibration problems.

Both thermocouples and thermistors were tried as alternatives. Thermistors, although inherently more sensitive than PRT's, required individual calibration and, because of their extreme non-linearity, were very difficult to use for quantitative work. (Recently, however, matched thermistors have become available at low prices and it is possible that this objection may be removed. We are, at present investigating the possibility of using matched thermistors for heat pump measurements again.) However, for the experiments being reported here, only thermocouples remained as an acceptable method of temperature measurement. In order to reduce the amount of wiring, all thermocouples were connected to the same reference junction, which was an ice–water bath agitated by a constant stream of air bubbles. The use of a common reference junction requires the electrical isolation of the test junctions from each other—a point that is all too often overlooked, though not for long.

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DATE 180980
RUN  5
REMARKS  1410 PHOTOGRAPHS 1-10 ON FIRST FILM 1 DEG C SUPERHEAT

COMPONENT              HEAT FLOW TO WATER(KW)
CONDENSER                2.59
EVAP SECTION 1           0.14
     SECTION 2           0.35
     SECTION 3           1.02
     SECTION 4           0.63
DESUPERHEATER            0.00
MOTOR COOLER             0.36
OIL COOLER               0.00
EVAPORATOR TOTAL         2.14

COMPONENT             HEAT(KW)   WT1   WT2    RT1   RT2   LMTD   HEAT TRANS COEFF.   PRESSURE DROP(BAR)   MEASURED T-SAT.T
EVAP SECTION 1          0.14    13.34 13.57  6.72  6.37  6.91          0.44              0.019                 2.69
     SECTION 2          0.35    13.57 14.12  6.37  4.88  8.22          0.90              0.008                 2.40
     SECTION 3          1.02    14.12 15.75  4.88  5.68  9.65          2.25              0.013                 2.13
     SECTION 4          0.63    15.75 16.75  5.68  7.41  9.71          1.37              0.043                 3.32
     BEND                                                                                0.120
                                                                                         -----
                                                                                         0.203

WATER SIDE HEAT TRANS. COEFF.           REF. SIDE HEAT TRANS. COEFF.(KW/SQ.M-K)   REYNOLDS   PRANDTL
   4.94                                      0.54                                   4472.      8.5
   4.97                                      1.22                                   4518.      8.4
   5.03                                      4.55                                   4648.      8.1
   5.11                                      2.10                                   4809.      7.8

COND WATER FLOW                         5.0 L/MIN
EVAP WATER FLOW                         8.9 L/MIN
REF FLOW                                0.022 KG/S
REF VOL. FLOW                           1.110
OIL FLOW                                0.000 KG/S
ELECT. POWER IN                         1.21 KW
COND TEMP IN                           43.1 DEG C
     TEMP OUT                          50.4 DEG C
EVAP TEMP IN                           16.8 DEG C
     TEMP OUT                          13.3 DEG C
TOTAL HEAT IN                           2.14 KW
TOTAL HEAT OUT                          2.95 KW

COP(HEAT OUT/POWER IN)                  2.44
COP((POWER IN+HEAT IN)/POWER IN)        2.77
COP(HEAT OUT/(HEAT OUT-HEAT IN))        3.64

PRESSURE BEFORE EXP.VALVE     12.88 BAR
TEMP.BEFORE EXP.VALVE         49.72 DEC C    52.38
PRESSURE AT EVAP.EXIT          3.31 BAR
TEMP.AT EVAP.EXIT              7.41 DEG C
ENTRANCE QUALITY               0.30
ENTRANCE ENTHALPY             84.95 KJ/KG
EXIT ENTHALPY                191.88 KJ/KG
EXIT SUPERHEAT                 5.23 DEG C
OIL CONCENTRATION              0.000
OVERALL LMTD                   7.90 DEG C
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Figure 2. Sample printout from test heat pump data reduction program

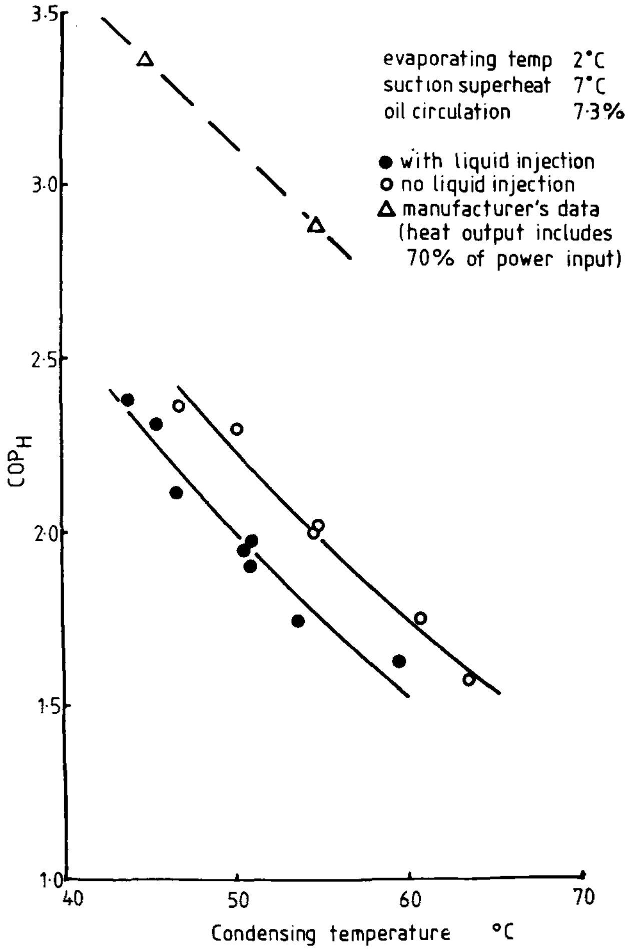

Figure 3. Comparison of compressor performance with manufacturer's performance chart—COP for heating

Pressure was monitored using strain gauge pressure transducers which gave an output voltage proportional to pressure. These were factory calibrated and arrived with calibration charts giving their non-linearity, hysteresis and sensitivity. It was found that the zero errors of these transducers varied with time and temperature (probably due to creep of the strain gauge diaphragm), and consequently they had to be recalibrated before each experimental run. This was easily achieved by using two-way valves to vent the transducers to atmosphere, enabling single point calibration readings to be taken. It was estimated that, after allowing for zero error, reference calibration and amplifier drift, these transducers were accurate to better than 1 per cent.

Fluid flows were measured using axial flow turbine meters, power was determined by thermal wattmeters, and the compressor shaft power was measured for open compressors using a dynamometer constructed in our own laboratories.

Data were collected using a Motorola 6800 based microcomputer data logger which recorded the experimental measurements on floppy disk and then transmitted them to the university mainframe computer for processing. Thus, data were recorded quickly and at variable cycle rates down to about once every ten seconds, though once per minute was the normal sampling rate.

The raw data were processed to provide information about the performance of the system as a whole and second law analyses allowed the behaviour of each component to be analysed. A sample printout from the data reduction program is shown in Figure 2.

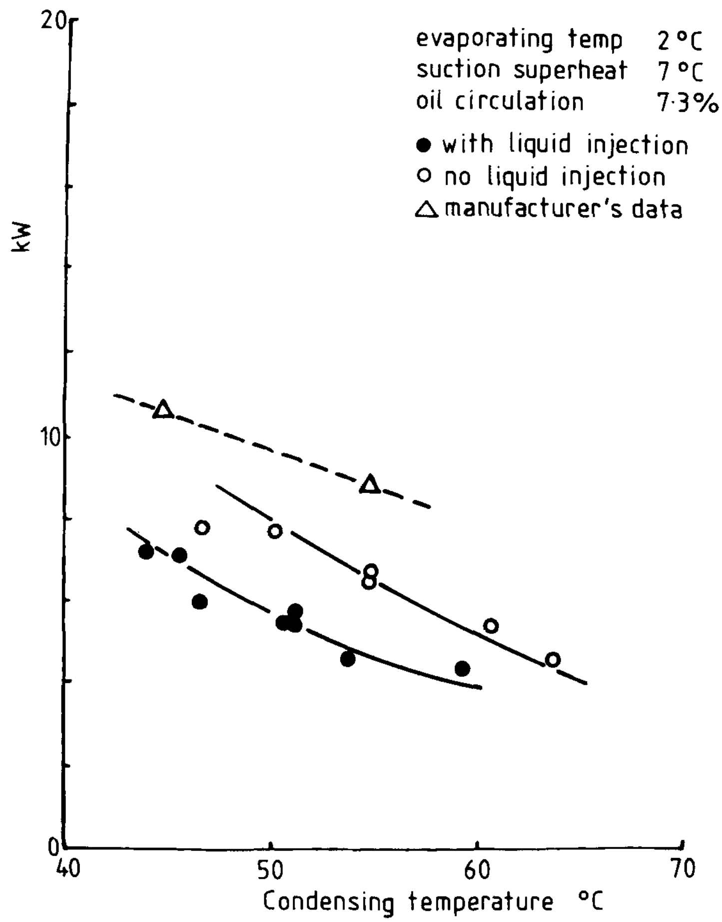

Figure 4. Comparison of compressor performance with manufacturer's performance chart—heat extraction

DATA REDUCTION

The data reduction program contained all of the calibration and correction factors needed to convert the instrument readings to the appropriate physical units. It also had access to refrigerant state point property subroutines to calculate the refrigerant enthalpy, entropy, saturation temperature and density at each of the measured points around the cycle. From these data, the heat flow in each element of the system could be calculated, together with a second law analysis which gave the irreversibility in that component. In components where heat transfer to the external water system was taking place, it was possible to compare these indirect values with experimental measurements.

From the outset, inconsistencies were observed between the external measurements and the internal thermodynamics as determined from this analysis. These were

(a) The observation of liquid line temperatures and evaporator temperatures significantly higher than the saturation temperatures read off refrigerant charts (or calculated by our refrigerant properties programs).

(b) The heat changes in the evaporator and condenser as calculated from the water measurements were smaller than those derived from the refrigerant state point properties.

(c) The measured heat pump COP was about one third less than the value estimated from the manufacturer's compressor performance data.

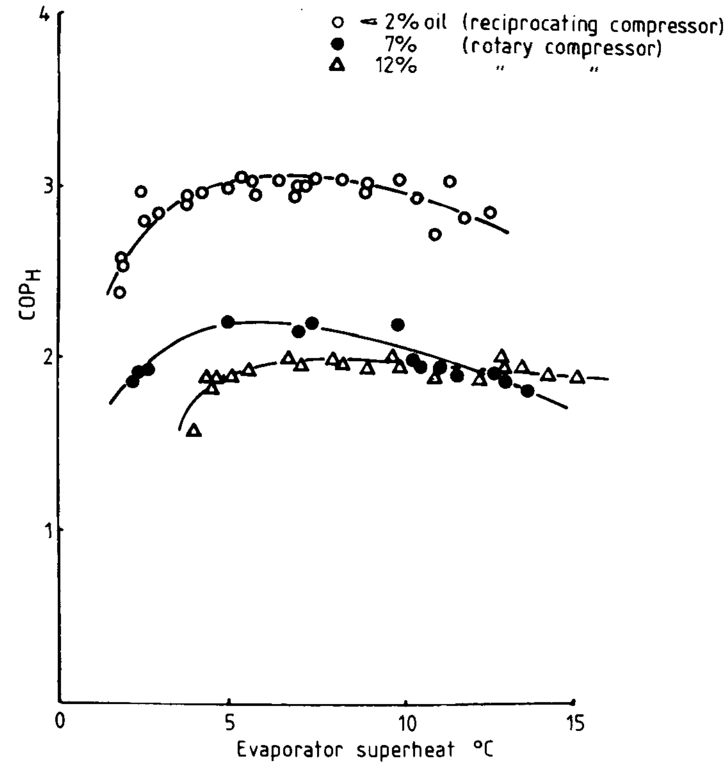

Figure 5. Variation of COP with evaporator superheat for different oil concentrations

These observations were rather puzzling at first and instigated an extensive period of equipment assessment and calibration which finally led us to have confidence in our instrumentation and experimental approach, but left the irregularities unexplained. Eventually, in a blinding glimpse of the obvious, it was realized that the lubricating oil in an oil flooded compressor was having a serious effect on the system performance, and that our analysis would have to be restructured to allow for this. A simple calculation based on Raoult's Law produced the levels of temperature elevation that we were observing, and provided the justification for launching a much more extensive study of these effects.

Preliminary results from the theoretical side of this study were described by Hughes *et al.* (1980), and this paper sets out to report some experimental results and their relationship with the theoretical development.

RESULTS

The first observation, and the one that instigated the detailed assessment in the first place, was that the compressor performance observed in our experiments was consistently below that claimed by the manufacturer. It was also typically around 25 per cent of Carnot efficiency, whereas 30–40 per cent is much more common. A comparison of the measured performance and that expected from the manufacturer's performance chart is shown in Figures 3 and 4. Figure 4 shows that the reduction in performance is arising through a reduction of evaporator heat transfer by about 30 per cent compared to that claimed by the manufacturer. After a length period of investigation, we eventually managed to identify the probable cause of this discrepancy. We believe that the manufacturer was using the standard gas-loop test method in which the compressor discharge refrigerant is cooled and throttled back to the required suction conditions. The compressor performance is then calculated using pressure and temperature

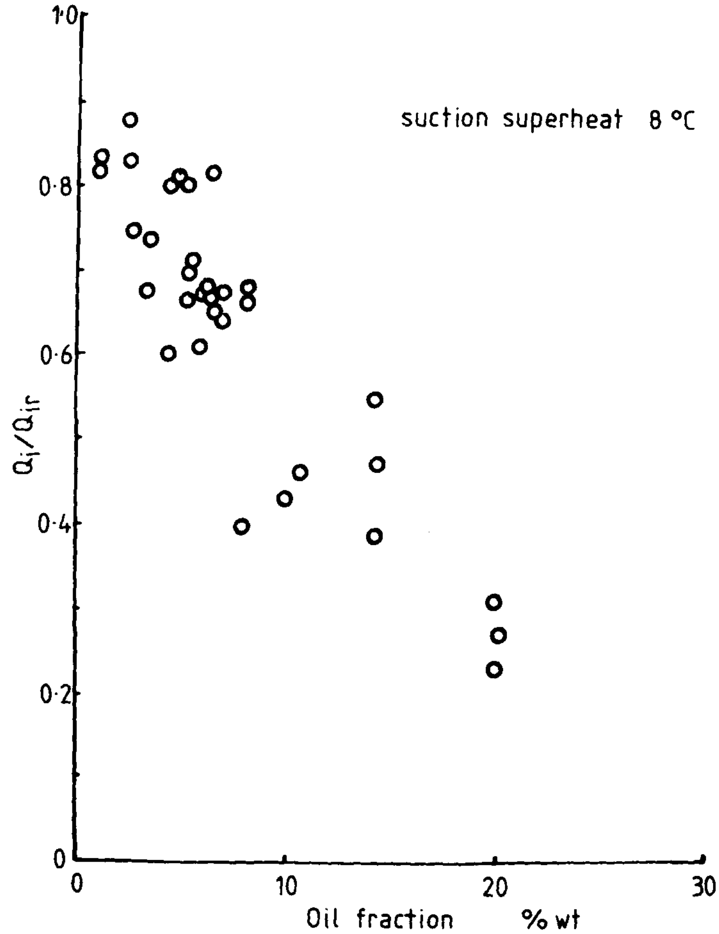

Figure 6. Variation of ratio of measured heat input to that found from pure refrigerant properties with oil concentration for fixed evaporator superheat conditions

measurements and the properties of pure refrigerant as provided by the refrigerant producers. At no point in the cycle does the refrigerant exist in the liquid state, and, even more seriously, there is no measurement of actual heat transfer—indeed, there cannot be measurement of evaporation or condensation heat transfer rates as the processes do not take place. Our test unit performed calorimetric measurements on the whole heat pump or refrigeration system, and therefore allowed assessment of compressor performance under actual operating conditions. Thus, the fundamental difference in approach ensures that in our facility the effects of factors such as oil-refrigerant solubility will be apparent, whereas in the gas-loop system they cannot be seen, as the crucial final step in performance assessment is made using the properties of pure refrigerant rather than those of the complicated fluid mixture actually present in the machine.

This degradation in the COP has been discussed elsewhere (Hughes *et al.*, 1980) and we wish to present here some experimental measurements for comparison with the analysis given there. Before making any comparison between theory and experiment, one feature of the analysis is worth stressing. In direct contrast with what would be expected from pure refrigerant data, an optimum value of the evaporator superheat was predicted, at which the COP was a maximum. (For a pure refrigerant system the best COP would be expected at zero evaporator superheat.) Figure 5 shows the effect very clearly, and illustrates how an increasing oil concentration both reduces the COP and increases the optimum evaporator

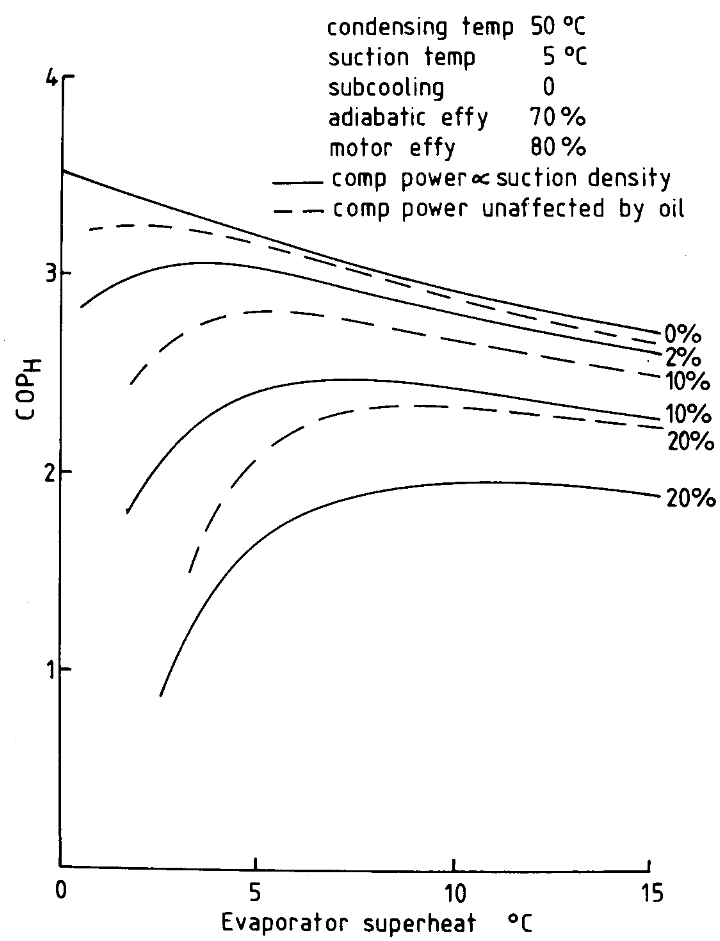

Figure 7. Theoretical calculation of variation of COP with evaporator superheat for a range of oil concentrations, Solid line indicates the assumption that compressor power is proportional to the suction density; dotted line indicates that compressor power is taken as constant

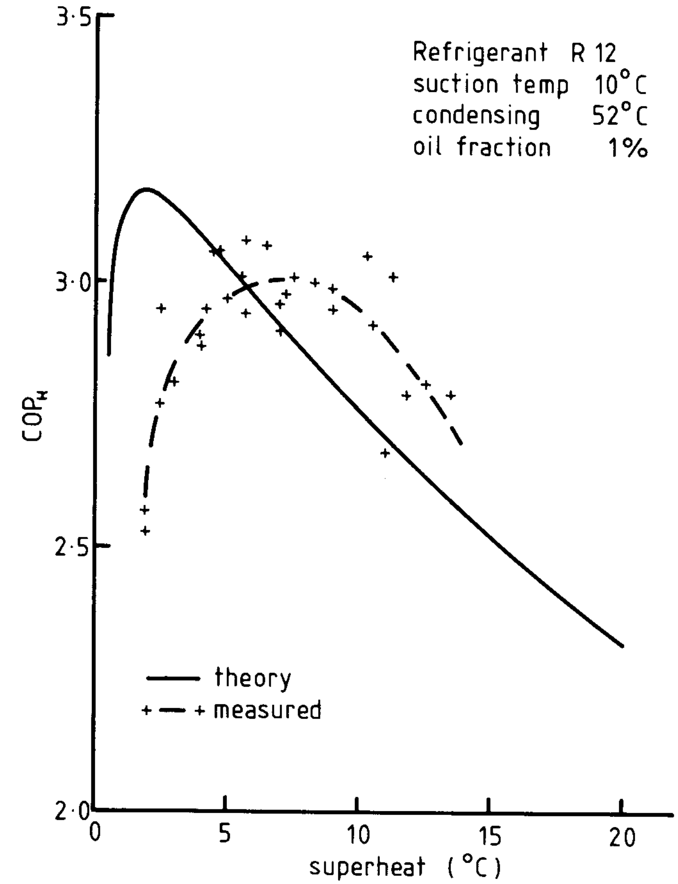

Figure 8. Comparison of theory with experiment—oil concentration <2 per cent

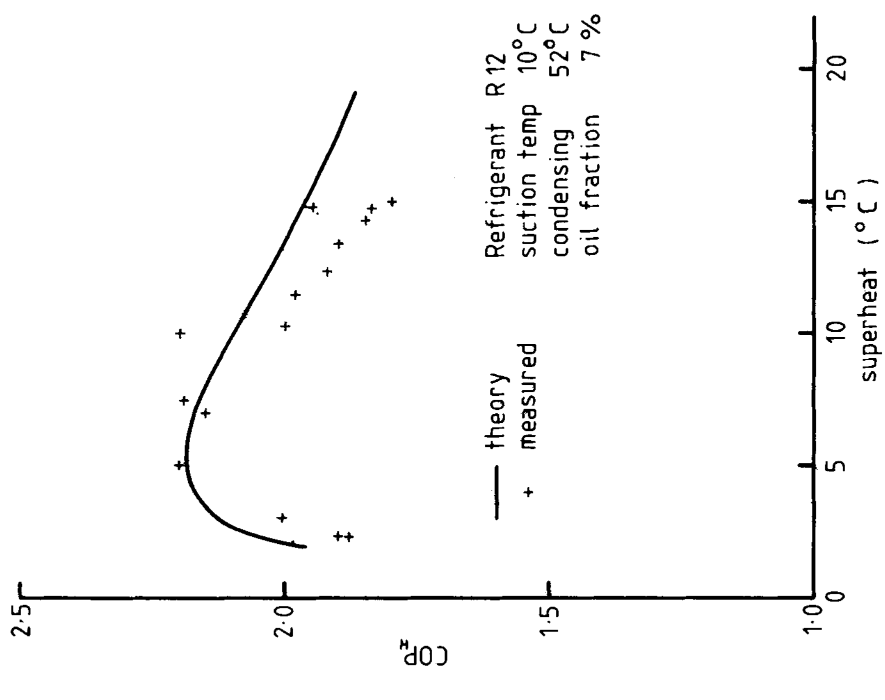

Figure 9. Comparison of theory with experiment—oil concentration 7 per cent

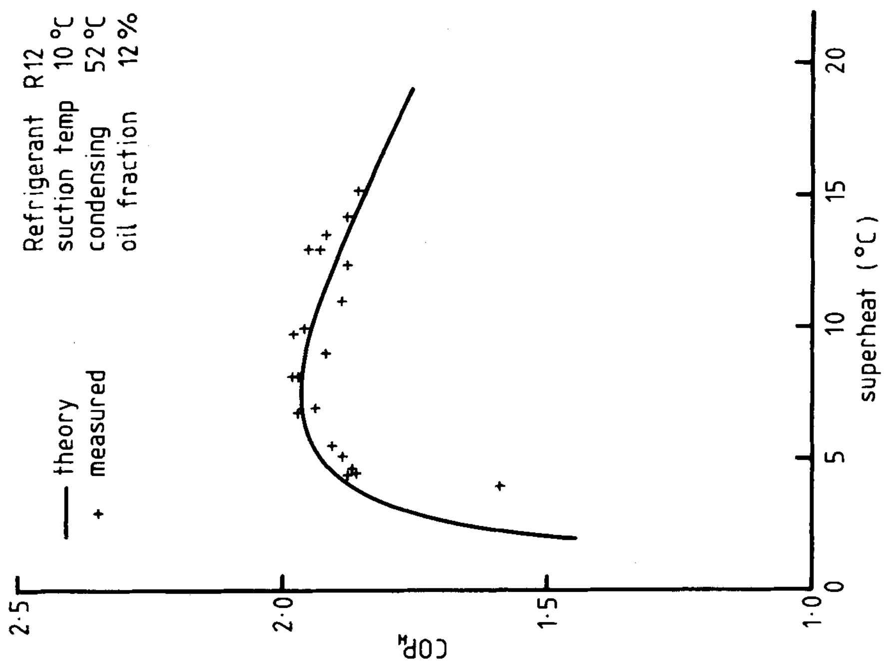

Figure 10. Comparison of theory with experiment—oil concentration 15 per cent

superheat setting. Curiously, for normal heat pump operation, where the suction temperature is dictated by that of the source, this means that the optimal evaporator temperature is depressed by the increased oil concentration. The general shape of the dependence is the same in each case, showing a rapid rise at low superheat settings, and a long declining tail.

The evidence for this behaviour is supported by other experiments which we have reported elsewhere (Hughes *et al.*, 1982a) and one graph is of particular interest here. Figure 6 shows how the ratio of the actual heat extraction at the evaporator to that determined from the measured state point properties, assuming pure refrigerant, varies with the oil fraction for a constant evaporator superheat of 8°C. It is striking that a 10 per cent oil concentration leads to a 50 per cent reduction in evaporator capacity under these conditions.

COMPARISON WITH THEORY

The theoretical analysis of the influence of oil on evaporator performance referred to earlier (Hughes *et al.*, 1980) uses refrigerant–oil solubility data, along with pure refrigerant thermodynamic data and physical data for both fluids, to determine the effect of the oil on evaporator capacity and on system COP. A diagram heat pump model is used with the assumption that the oil has no effect on compressor performance. The theory also uses an alternative assumption—that the compressor power is proportional to the bulk density of the suction mixture, but with the volumetric efficiency still unaffected. It is likely that neither of these assumptions is correct, but in the absence of any other evidence, they seem to be reasonable, and there is some evidence that the first assumption may approximate to the truth. The system was analysed for a range of superheat settings between 1 and 15°C, and for oil concentrations up to 30 per cent. The refrigerant was R12, and the theory assumed Bambach's (1955) paraffinic oil solubility data. Although the oil used in the experiments was Shell Clavus 68, it is known that the solubility of R12 in this oil is very similar to that assumed in Bambach's solubility data.

The predictions of the theory, illustrating the comparison of the two compressor power assumptions are shown in Figure 7, and the comparison between theory and experiment (for the suction density dependent compressor power version) is presented in Figures 8–10. In view of the lack of refinement of the model the agreement is remarkably good, and most encouragingly, does show that the general shape of the curves predicted by the theory is correct.

The most important limitations on the validity of the model lie in the unknown contribution of non-equilibrium effects, and in our lack of understanding of the influence of oil on compressor performance. These aspects are presently being investigated in a continuing research programme sponsored by the Commission of the European Communities under its Second Energy R & D Programme.

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