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ABSTRACT

When testing the refrigeration performance of a compressor in a laboratory, the equipment which best simulates a real system is a calorimeter. However, for some large compressors it is not always practical to use calorimeters so, flowrater devices such as orifice plates, venturi or mass flow meters are used.

This paper identifies the differences which have been found using orifice meters and calorimeters for determining the refrigeration capacity of compressors.

In the event of there being any refrigerant property errors, they will cancel out in the calorimeter method because two enthalpy differences are used as a ratio. However, in the flowrater method any refrigerant property errors have a direct effect as specific volume in the mass flow calculation, and again as enthalpy change in the capacity calculation.

INTRODUCTION

As standards of efficiency and control of refrigeration equipment improve, the need for more precise data on compressor performance increases.

To determine the refrigeration capacity of a compressor, evaluation of mass flow rate is needed and also the determination of the specific enthalpy change between the compressor inlet and the evaporator inlet obtained by means of refrigerant property data. This paper will compare the differences between two methods for determining capacity and will discuss the importance of refrigerant property accuracy.

CALORIMETER METHOD

The calorimeter (Figure 1) measures capacity by means of a heat balance. The compressor suction pressure is adjusted by controlling the liquid refrigerant at the expansion device, and the temperature of the refrigerant vapour entering the compressor is adjusted by varying the electrical heat input. The discharge pressure is adjusted by varying the temperature and flow of the condensing medium, or by a pressure control device in the discharge line. The mass flow rate of the refrigerant, as determined by the test, is given by the formula

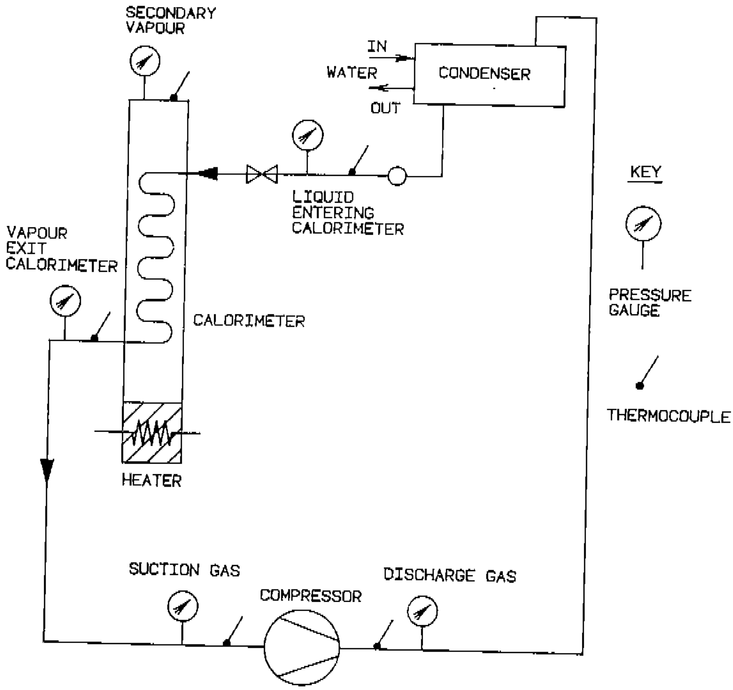
$$m_f = \frac{Q_i + F_l(t_a - t_s)}{h_{g2} - h_{f2}} \qquad [1]$$

where Q_i = Heat input to the calorimeter W
F_l = Heat leakage factor W/K
t_a = Average ambient temperature K
t_s = Temperature of secondary fluid K
h_{g2} = Specific enthalpy of evaporated refrigerant leaving calorimeter J/kg
h_{f2} = Specific enthalpy of refrigerant liquid entering expansion valve J/kg

and the refrigeration capacity at the test condition is given by

FIGURE 1

SECONDARY FLUID CALORIMETER



$$Q_c = m_f(h_{g1} - h_{f1}) \quad [2]$$

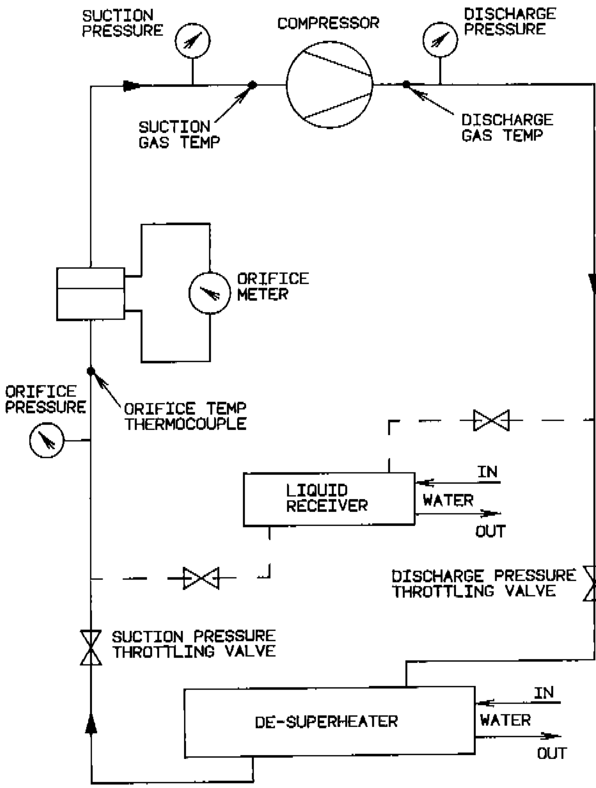
where
Q_c = Refrigeration capacity of the compressor W
m_f = Mass flow rate of refrigerant measured on a calorimeter kg/s
h_{g1} = Specific enthalpy of refrigerant entering the compressor at the specified test condition J/kg
h_{f1} = Specific enthalpy of refrigerant liquid at the temperature of saturation corresponding to compressor discharge pressure specified in the test condition J/kg

REFRIGERANT VAPOUR FLOWRATER

A flowrater, for measuring refrigerant volume flow rate consists of an orifice plate, venturi, nozzle or other device.The flowrater can be installed in the suction or discharge pipeline of a closed circuit comprising of the compressor, a pressure reducing device and a means for reducing vapour superheat. The heat of compression is normally removed by a water cooled heat exchanger.

The circuit used in the present test is shown in Figure 2. Adjustment of the refrigerant charge and throttle valves in the main circuit is used to attain different conditions. The liquid receiver is used as a storage vessel for refrigerant. Hot discharge gas can be passed to it or liquid refrigerant can be injected into the main circuit from it to alter the refrigerant charge in the main circuit. This only happens when changing conditions and the receiver is closed

FIGURE 2
REFRIGERANT VAPOUR FLOWRATER



off from the main circuit during tests. This method gives very stable conditions, avoiding perturbations which can occur when liquid is injected to de-superheat the gas, as is the case in other methods.

The refrigerant mass flow rate m_v is measured with an orifice plate, nozzle or venturi constructed and installed in accordance with agreed standards and this paper reports on measurements made with a square-edged orifice plate with D and D/2 tappings designed and installed in accordance with ISO 5167 (1), BS 1042:Section 1.1 (2) and BS 1042:Section 1.2 (3) which is for pipes between 25mm and 50mm diameter. The orifice was located in the suction line of the compressor. A discharge line device would be too small, falling outside the standard. Under most operating conditions, no liquid is present anywhere in the main circuit. With a vapour temperature of 20°C at the compressor suction and values close to this at the meter there is negligible opportunity for liquid presence to affect the results.

Once the pressure and temperature upstream of the metering device and the pressure difference across the device have been measured, the mass flow can be derived from the following formula

$$m_v = C.E.e.\frac{\pi}{4}.d^2.\sqrt{\frac{2\,\Delta p}{v_1}} \qquad [3]$$

where m_v = Refrigerant mass flow rate from orifice test kg/s
C = Coefficient of discharge
E = Velocity of approach factor, $E=(1-\beta^4)^{-1/2}$
β = Diameter ratio, d/D
D = Upstream internal pipe diameter mm
e = Expansibility (expansion) factor
d = Diameter of orifice mm
Δp = Differential pressure bar
v_1 = Specific volume of refrigerant volume of refrigerant volume at orifice conditions m^3/kg

and the capacity, Qf is given by the following equation

$$Q_f = (h_{g1}-h_{f1})m_v \qquad [4]$$

An alternative to using an orifice plate is to use a direct mass flow meter whose input is a direct function of mass flow only. This means that pressure, temperature and other properties do not affect the output. One such meter, and the one that is referred to in this paper, is the coriolis type meter with which the measurement of the torque required to produce coriolis force in the fluid gives a measure of mass flowrate.

Using the mass flow meter in either the suction vapour line, discharge vapour line or the full liquid line of a calorimeter, the measured value for mass flow rate can be used directly in equation 4 to give capacity.

BACKGROUND TO THE INVESTIGATION

The calorimeter method and the refrigerant vapour flowrater method are both principle methods for the determination of refrigeration capacity as defined in ISO 917 (4) and ASHRAE 23-78 (5). The calorimeter method is best applied to smaller capacity compressors - below say 30 kW. It closely simulates the actual refrigeration system and so, with low heat leakage factors, low pressure drops between the calorimeter and the compressor, the measured electrical power input is very close indeed to the actual refrigeration capacity. This means that any uncertainties in deriving mass flow rate arising from refrigerant property errors are effectively eliminated from the final result, and so the calorimeter should be the preferred method. However there are limitations which make the use of flowraters desirable:-

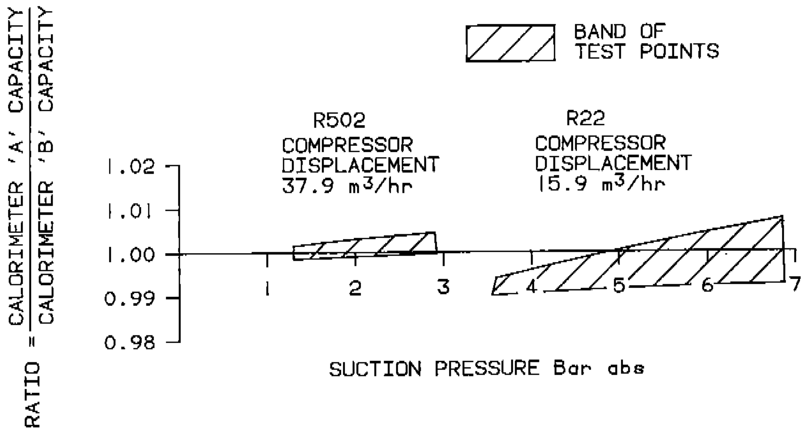
- calorimeters become very large and expensive above capacities of about 30kW - hence for large compressors a flowrater becomes a more economical solution.
- the running costs are about 1/3rd for a flowrater.
- stabilization times are lower for a flowrater meaning that testing will be quicker and hence less costly.

As part of a programme for maintaining a check on instrumentation calibration, a number of compressors have been tested on different test stands in order to verify capacity measurement accuracy.

For this purpose each test consisted of performance measurements at a number of points over the operating envelope of the compressor. This data was then fitted using the CRATE fitting technique previously described at Purdue(6). This technique not only smoothes the data but avoids the need to test the same operating point on different pieces of equipment. It gives fitted data which is within 0.5% of the measurement at each point. A sample of the error report is given in Table 1 at the end of this paper. Using this method it has been found with a compressor tested on different calorimeters that the final capacity data is within ±1% over its operating envelope. Figure 3 shows the results of testing compressors on two different calorimeters of similar design.

FIGURE 3

CALORIMETER VERIFICATION



Consistent results have also been found with flowraters, with measurements made on a single compressor on different test stands yielding the same results. In particular, changing the orifice plate, which alters the pressure differential, diameter ratio and coefficient of discharge, still gives consistent results.

Throughout all of this work, instrumentation calibration has been thoroughly implemented at regular intervals and always at the beginning and the end of a set of tests.

FIGURE 4

CAPACITY RATIO AS A FUNCTION OF VOLUME FLOW

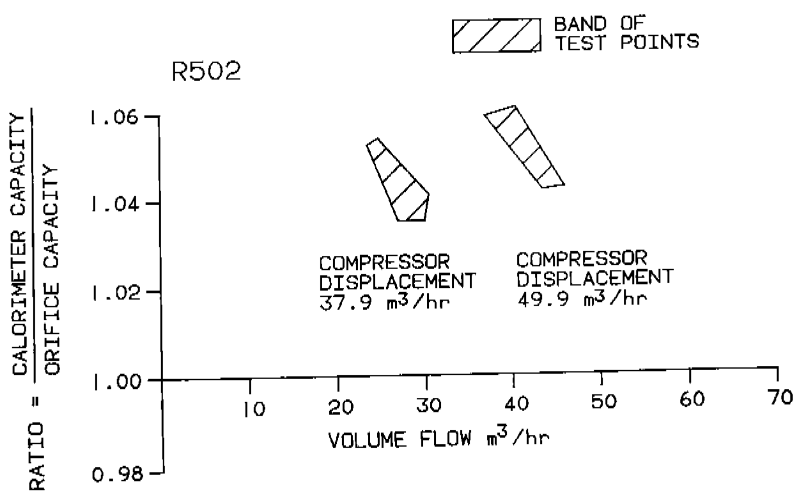
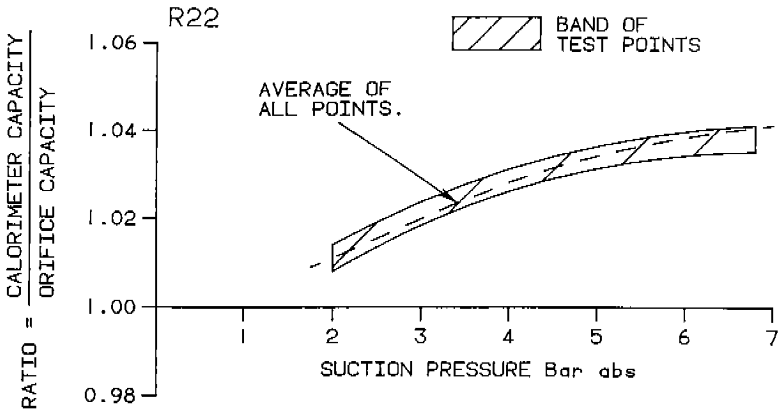


FIGURE 5

CAPACITY RATIO DERIVED FROM TEST



COMPARISON BETWEEN FLOWRATER AND CALORIMETER

Whilst consistent data were obtained between different calorimeters and between different flowraters, differences were observed between the two methods. Using the CRATE technique it has been possible to compare fitted data obtained from flowrater tests with those from calorimeter tests. Figure 4 shows the capacity ratio of flowrater/calorimeter as a function of volume flow. This is for two compressors of different displacements with R502. From this it can be seen that there does not exist a single relationship between capacity ratio and volume flow as compressors of different displacements will give similar capacity ratios at very different flows.

It was found that the data correlated best when plotted against suction pressure. This is shown for a compressor of 49.9m3/hr displacement running on R22 at a constant suction temperature of 20°C in Figure 5. The difference between the calorimeter and the flowrater capacities are greatest at high suction pressures and least at low suction pressures. The band of test points is shown and the average ratio is drawn as a dashed line.

A similar relationship was found for R502 at a constant suction temperature of 20°C and is shown in Figure 6. This graph is derived from test data taken from four different compressors using three calorimeters and two different orifice flowraters. The band of test points is wider for four compressors than for a single R22 sample, but still well defined. The details of the compressors and the test ranges used to generate this graph are given in Table 2 at the end of this paper.

COMPARISON OF MASS FLOW METER AND FLOWRATER

A comparison has been made between an orifice plate in the suction line and a coriolis type mass flow meter in the discharge line of the same refrigerant vapour test stand. Data was taken from both the orifice plate meter and the mass flow meter at the same conditions and the two sets of test results were fitted using CRATE. The results are given in Figure 7. The range of the mass flow meter was limited by the compressor characteristics and the high presure drops across it. Also at flows below 0.1 kg/sec, considerable scatter was observed on the readings and consequently only readings above 2.3

FIGURE 6

CAPACITY RATIO DERIVED FROM TEST

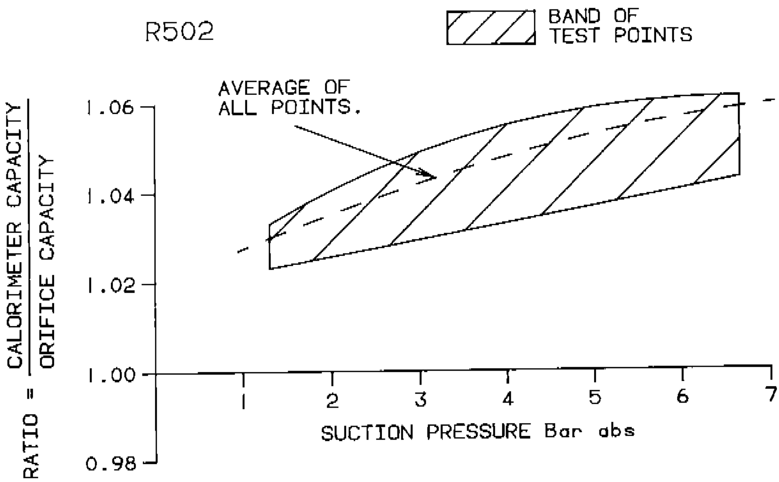
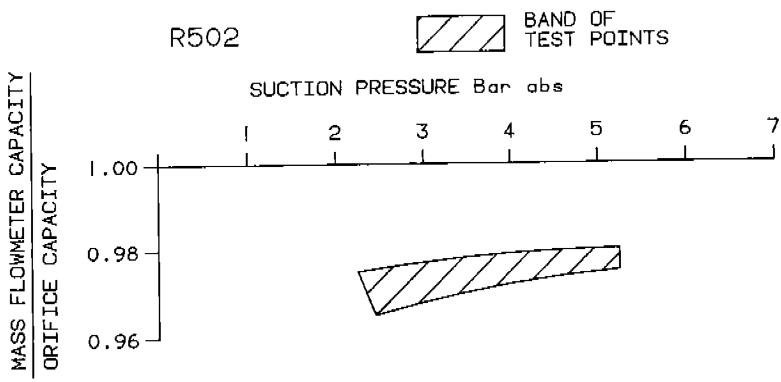


FIGURE 7

CAPACITY COMPARISON, MASS FLOW METER

COMPRESSOR DISPLACEMENT 49.9 m^3/hr



bar are used. (The meter capacity is 1 kg/sec.). The very high accuracies specified for mass flow meters only appear to apply for liquids and there is no means of performing a direct calibration with refrigerant vapour. We can say that the mass flow meter readings tend to support the orifice readings and were observed to be slightly lower.

DISCUSSION OF RESULTS

Differences have been observed between calorimeter and flowrater capacity measurements which are not a function of mass flow, volume flow or orifice diameter. Plotting the data against suction pressure gives consistent results. In examining the reasons for the differences, the following have been considered.

Coefficient of discharge and Expansibility factor

There is an overall uncertainty on the orifice meter result which can be estimated from the formula given in ref(1). The present work was performed with a pipe diameter of 38.4mm and orifice plates of bore 15mm, 20mm and 25mm (a pipe diameter of 26.1mm and an orifice plate of 18mm bore were used for the $18m^3/hr$ compressor). Using realistic estimates for uncertainties of measurements and 1.0% for fluid density in these experiments gives an overall uncertainty of approximately 1% according to this formula. However ref (3) includes an additional uncertainty which must be applied for pipes of less than 50mm diameter. This is between 2% and 3% and indicates that knowledge of the behaviour of these smaller meters is less precise.

In effect, at least part of the observed offset could be accounted for by this, but as different sizes give the same result it must be assumed that whether coefficient of discharge is significant or not, another factor exists.

Calorimeter heat loss

By selecting a suction gas temperature of 20°C, the calorimeter is operating with a secondary fluid temperature within a few degrees of ambient. The heat loss correction is very small indeed, and can be effectively eliminated from the discussion.

Refrigerant properties

With the calorimeter method, the enthalpy difference $(h_{g2}-h_{f2})$ in eq.1 is used to determine the mass flow, and $(h_{g1}-h_{f1})$ is then used in calculating capacity (eq.2). Any consistent errors in enthalpy change will cancel out and will not effect the final capacity result.

With the flowrater method the specific volume of superheated vapour is used to calculate mass flow in eq 3 and the enthalpy change is used to calculate capacity in eq 4. An error in either of these properties would contribute directly to an error in capacity. It is therefore important to specify the source of the refrigerant property data, and in this case the Du-Pont equations (7) have been used throughout. The differences observed between this data and the IIR tables (8) is minute, but the overall tolerance on the individual properties over different ranges is not specified. The maximum observed differences are at high suction pressures where the superheat is low and the potential for fitting errors within the refrigerant data is high.

Effects of oil

With the compressors used in these tests, the oil circulation is known to be of the order of 1% by mass. Oil circulation will tend to reduce the capacity measured with the calorimeter making the true calorimeter capacity even larger. However, it should be noted that in the calorimeter/calorimeter verification experiment (Fig. 3) that an oil separator was used in one case. This would reduce the oil circulation by an order of magnitude. Thus the effect of the oil is shown to be small.

An oil separator was also used on the flowrater tests. The effect of the very small quantities in circulation is not known.

Mass flow meter results

The mass flow meter records a lower mass flow than the orifice (Fig. 7) although any oil in circulation should be included. In other words the calorimeter/mass flow meter capacity ratios would be larger than shown in Figures 5 and 6.

Pulsation

It is known that pulsation will affect the orifice meter results, and whilst pulsation cannot be completely eliminated with positive displacement compressors on a flowrater, the heat exchanger and the pipework volumes are quite large on the flowraters used for these tests. Also, compressors of the 2 cylinder and 3 cylinder type have been included in the programme (Fig. 6) and these would be expected to have different pulsation characteristics. It is therefore reasonable to assume that pulsation is not the major cause of the difference.

Pulsation could however be affecting the mass flow meter results, but this has not yet been investigated.

Calibration and accuracy

Measurement errors will tend to be greatest at lower suction pressures and so there is more uncertainty in the results in this region. Whilst this could lead to questioning of the slope of a line drawn to represent the data, the trend has been consistent with the four compressors tested.

CONCLUSIONS

Evidence over a number of tests indicates an offset of up to 6%, varying with suction pressure, between calorimeter and orifice meter flow measurement for refrigerant compressor capacity.

The offset shown in Figures 5 and 6 could arise because of uncertainties in the coefficient of discharge of the orifice meter, and this would mean that similar offsets would not necessarily be seen with other sizes of orifice line or other devices (e.g. venturi). If, however, the offset is due to other factors such as refrigerant properties, then it could occur with other sizes.

Flow meter methods rely on accurate specific volume and enthalpy difference data for refrigerants, but the calorimeter method does not. The differences observed may result from several factors, and the absolute accuracy of the specific volume and the enthalpy difference values being used is brought into question. Therefore a calorimeter method should be used as a reference where possible.

This paper reports on tests over a wide range of conditions (Table 2), as tests at single rating points would produce offsets of limited use. Although it has been possible to suggest reasons for the differences between the two methods, it is not possible to make a definitive statement, but merely to raise several questions.

Compressor testing standards call for results of two different test methods to be within ±4% for ISO 917 and ±3% for ASHRAE 23-78 and it is shown that this could only be attained with a flowrater when a relation between flowrater and calorimeter capacity measurement has been established.

REFERENCES

(1)ISO 5167-1980 Measurement of fluid flow by means of orifice plates, nozzles and venturi tubes inserted in circular cross-section conduits running full.

(2)BS 1042 : Section 1.1 : 1981 Measurement of fluid flow in closed conduits. Specification for square-edged orifice plates, nozzles and venturi tubes inserted in circular cross-section conduits running full.(Technically equivalent to ISO 5167)

(3)BS 1042 : Section 1.2 : 1984 Measurement of fluid flow in closed conduits. Specification for square-edged orifice plates and nozzles (with drain holes, in pipes below 50mm diameter, as inlet and outlet devices) and other orifice plates and Borda inlets.

(4)ISO 917-1974 Testing of refrigerant compressors

(5)ASHRAE 23-78 Methods of testing for rating positive displacement refrigerant compressors

(6)Lawson, S. and Millett, H., "Rating Technique for Reciprocating Refrigeration Compressors". Purdue International Compressor Conference, 1986

(7)Du-Pont Thermodynamic Properties of "Freon" 502 Refrigerant. Technical Bulletin T-502-SI
Du-Pont Thermodynamic Properties of "Freon" 22 Refrigerant. Technical Bulletin T-22-SI

(8)IIR Thermodynamic and physical properties of R502, 1982
IIR Thermodynamic and physical properties of R22, 1982

TABLE 1

CRATE Error report

THIS DATA IS FITTED TO THE 'TEST POINTS' CONTAINED IN FILE A:9 20.1.86
using fitting routine CRATE (pressures)

REFRIGERANT R22 No. POINTS FITTED 12

PS	PD	VOL EFF		KW		ISEN. EFF.	% ERROR	
		TEST	FIT	TEST	FIT		FLOW	POWER
2.008	11.908	0.689	0.689	6.160	6.194	0.607	-0.040	0.547
3.550	11.920	0.791	0.795	8.200	8.225	0.615	0.505	0.311
5.830	11.890	0.854	0.850	9.100	8.974	0.566	-0.539	-1.380
7.363	11.983	0.866	0.866	8.750	8.696	0.508	-0.042	-0.612
2.001	15.221	0.618	0.616	6.600	6.454	0.600	-0.350	-2.208
2.973	15.313	0.710	0.711	8.350	8.446	0.621	0.115	1.146
4.973	15.313	0.787	0.791	10.480	10.606	0.614	0.445	1.198
7.323	15.323	0.829	0.829	11.200	11.305	0.571	-0.048	0.941
3.531	24.261	0.626	0.630	10.700	10.736	0.615	0.647	0.340
4.241	24.291	0.669	0.666	12.070	12.093	0.621	-0.506	0.188
5.857	24.247	0.721	0.716	14.440	14.327	0.622	-0.675	-0.784
7.327	24.237	0.738	0.742	15.650	15.643	0.615	0.502	-0.043
		************************************				RMS	0.435	0.998

TABLE 2

Test range for R502

Compressor disp.m³/hr	Cond temp °C	Evap temp °C	Discharge pressure bar abs	Suction pressure bar abs
49.9	30 to 60	-20 to 5	13.2 to 26.0	2.9 to 6.7
37.9	35 to 45	-35 to -20	14.9 to 18.8	1.6 to 2.9
37.9	30 to 55	-40 to -20	13.2 to 23.4	1.3 to 2.9
18.0	45	-25 to 2	18.8	2.4 to 6.1