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PERFORMANCE CHARACTERISTICS OF A RECIPROCATING REFRIGERANT COMPRESSOR OVER A RANGE OF SPEEDS

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ABSTRACT

Performance characteristics of a two cylinder open reciprocating compressor are presented over the speed range 300 to 900 r.p.m. Data is included on the concentration of lubricating oil in the discharge vapour. The adiabatic efficiency and the 'rational efficiency' of the compressor are presented. A newly defined parameter, the 'rational efficiency for compression and heat rejection', is proposed as an appropriate figure of merit for a compressor to be used in a refrigeration system which rejects heat to the environment. The displacement utilisation efficiency is given. Valve lift and cylinder pressure diagrams and a number of parameters derived from these are presented and discussed.

INTRODUCTION

In this paper the main performance characteristics and some figures of merit are presented over a range of speeds for a compressor with R-12 refrigerant under the following nominal test conditions:

Suction saturation temperature:	-10°C
Discharge saturation temperature:	40°C
Suction temperature at intake pipe:	5°C
Mean ambient temperature:	22°C
Speed range:	300 to 900 r.p.m.

The essential aspects of the load stand which was used, and the measurement methods, were described in a previous paper (1). One significant improvement in the testing technique was that two independent measurements of the refrigerant mass flow rate were made. The primary method was based on the measurement of the pressure drop across an orifice plate in the suction line. A damping chamber and a baffle plate were installed to smoothen flow pulsations so that a mercury manometer could be used as the differential pressure measurement device. The secondary flow measurement method consisted of a heat balance which was described previously (1). Good agreement was achieved between the two flow measurement methods.

The emphasis in this paper is on the presentation of the main performance characteristics as follows: 1. refrigerant mass flow rate. 2. shaft power input. 3. discharge temperature. It has been common practice in the industry to present data for the cooling effect or the heating effect of a compressor, rather than the mass flow rate, and to omit data on the discharge temperature. Many end users appreciate this form, as the direct selection of a compressor for a given nominal duty is facilitated. However, the three main characteristics mentioned contain all the necessary information at a specified operating condition. Presentation of these brings the characteristics of the compressor, as an entity, into focus. Accurate values of the heating or cooling effect can be readily determined from the main characteristics with the aid of a pressure - specific enthalpy chart. Data are presented, too, on the oil concentration in the discharge vapour. These may also be required by a system designer, firstly, because the entrained oil has an effect (which was small for the compressor tested) on the thermodynamic properties of the refrigerant and, secondly, because the oil concentration may serve as a measure of the effectiveness of the lubricant distribution system within the compressor over a range of speeds. In evaluating refrigerant thermodynamic properties, as part of the analysis of test results, the influence of the entrained oil was taken into account, using the theory described by Hughes et al. (2).

Parameters are presented to quantify the utilisation of shaft power and the utilisation of volume displacement, as these are the two main aspects of a compressor's performance. The use of rational efficiency values, based on the Second Law of Thermodynamics, is advocated for quantifying shaft power utilisation. Valve lift and cylinder pressure diagrams are presented to illustrate valve behaviour and as the basis for evaluation of the indicated power requirements attributable to valve throttling.

PERFORMANCE CHARACTERISTICS

Mass Flow Rate

The mass flow rate of refrigerant (or, more correctly, of the refrigerant/oil mixture) was found to vary almost linearly with speed over the range tested, fig 1. Bars through the data points on fig. 1 illustrate the calculated precision of the mass flow rate data, based on the known or estimated accuracies of the measurement devices involved.

Shaft Power Input

Shaft power increased linearly with speed over the range 300 to 900 r.p.m., fig. 2.

Discharge Gas Temperature

The discharge temperature showed a small linear increase with increasing speed over the range tested. This characteristic is shown in fig. 3.

Oil Concentration in the Discharge Vapour

It was found that a considerable time interval at a given operating condition was necessary before reasonably consistent measurements of the oil concentration could be obtained. For instance, much higher oil concentration values were observed shortly after start up of the plant. Even with special care, the data showed a lot of scatter, fig. 4.

The steady state oil concentration was clearly low at low speeds. It reached a maximum of perhaps 0.4% in the mid speed range and appeared to decline at higher speeds. The value of nearly 0.6% shown on fig. 4, at about 800 r.p.m., was considered to be spurious. Further testing was necessary to clarify the exact nature of the oil concentration versus speed characteristic.

In the concentrations measured in the tests over the speed range, entrained oil did not greatly affect the thermodynamic properties of the refrigerant passing through the compressor, as it was well superheated. In the tests, the superheat at entry to the compressor was 15 K. However, the effect on the properties would have been significant if the 'apparent superheat' was less than about 5 K.

UTILISATION OF SHAFT POWER

Indicated Work Quantities

The indicated work per cycle, which corresponds with the total area enclosed by a pressure/volume curve such as that shown in fig. 8, was determined by numerical integration and, hence, the mean effective pressure (MEP) was evaluated. That part of the area enclosed by the curve and below a horizontal line representing the mean suction pressure can be described as the 'indicated suction work'. In the same way, that part of the enclosed area which is above the mean discharge pressure can be described as the 'indicated discharge work'. These are the quantities of 'pumping work' which are necessary per cycle in order to induce refrigerant into the cylinder and to discharge it. The necessity for pumping work arises from the pressure losses across the suction and discharge valves.

The 'suction effective pressure' (SEP) and the 'discharge effective pressure' (DEP) are defined in equations 1 and 2. Both are analogous to the familiar mean effective pressure. Values are shown on fig. 8.

$$\text{suction effective pressure} = W_{si}/V_{sw} \qquad 1.$$

$$\text{discharge effective pressure} = W_{di}/V_{sw} \qquad 2.$$

where

V_{sw} = swept volume per cylinder [m^3]
W_{si} = indicated suction work [J]
W_{di} = indicated discharge work [J]

By multiplying the indicated suction work or the indicated discharge work by the number of cycles per second, the suction pumping power or the discharge pumping power can be calculated.

$$\text{suction pumping power} = W_{si}N \qquad 3.$$

discharge pumping power $= W_{di} N$ 4.

where
N = number of cycles per unit time $[s^{-1}]$

The indicated power, the discharge pumping power and the suction pumping power are plotted against compressor speed in fig. 5. All three were found to increase linearly with speed.

Mechanical Efficiency
Mechanical efficiency, i.e. the ratio of indicated power to shaft power, was found to vary little with speed. It showed a small increase from about 92% to about 94% over the speed range. In the following sections power utilisation efficiency data based on the indicated power are presented. These are readily referred to the shaft power input by multiplying the values by the mechanical efficiency.

Indicated Adiabatic Efficiency
Adiabatic efficiency (also called the isentropic efficiency) is defined by the following equation:

$$E_{is} = \frac{\dot{m}\,(h_{2s} - h_1)}{P_i} \qquad 5.$$

where
E_{is} = indicated adiabatic efficiency
h_1 = specific enthalpy of the refrigerant evaluated at the compressor suction condition [J/kg]
h_{2s} = specific enthalpy evaluated after an assumed isentropic compression process to the discharge pressure [J/kg]
$\dot{m}$ = the actual externally measured mean mass flow rate of the compressor [kg/s]
P_i = indicated power [W]

Values of E_{is} are plotted in fig. 6, along with rational efficiencies which are defined in the following sections.

Second Law Rational Efficiency
The concept of availability analysis has been described in the literature (3, 4). For specified entry and exit states of a fluid passing through a steady flow system and for a specified temperature of the surroundings the minimum work which must be done *on the fluid* is given by the change in the flow availability function between the two states. The value of this function depends on the temperature of the surroundings as well as on the thermodynamic state properties. Furthermore, the minimum work quantity can be achieved only if the flow process between the states is reversible (internally and externally). The approach is readily applied to compressors.

Indicated Compressor Rational Efficiency
The ideal work and a rational efficiency based on availability analysis can be defined for the compression process as follows:

$$W_{ideal} = b_2 - b_1 \qquad 6.$$

$$= h_2 - T_0 s_2 - (h_1 - T_0 s_1) \qquad 7.$$

$$E_{ir} = \frac{\dot{m}\, W_{ideal}}{P_i} \qquad 8.$$

where
W_{ideal} = the ideal work of compression (done on the fluid) between the actual suction and discharge states [J/kg]
b = specific flow availability function of the refrigerant [J/kg]
= $h - T_0 s$
T_0 = temperature of the surroundings [K]

s = specific entropy of the refrigerant [J/kg K]
E_{ir} = indicated compressor rational efficiency.
Subscripts 1 and 2 refer to the suction and discharge states respectively.

The rational efficiency defined by equations 7 and 8 depends on the temperature of the surroundings. It is important, therefore, that this temperature is stated as one of the test conditions whenever a rational efficiency is quoted. Values of E_{ir} are plotted in fig. 6. These data quantify the effectiveness of indicated power utilisation in transforming the thermodynamic state of the refrigerant from the suction to the discharge condition in a steady flow process, or, the effectiveness with which indicated work is converted to availability of the refrigerant. No information is contained within the E_{ir} data, or implied, on the suitability of either the suction or discharge states of the compressor to the overall purposes of the plant. Such information would follow, however, if the availability analysis were extended to include other regions of the plant. The characteristics of a compressor in producing particular discharge states when operating with specified suction states and specified discharge pressures must be described separately, i.e. by presentation of the discharge temperature characteristics.

Indicated Rational Efficiency for Compression and Heat Rejection

In a refrigeration plant the discharge pressure must be higher than the refrigerant saturation pressure corresponding to the temperature of the surroundings. Within this constraint, and for a given condenser, a lower condensing temperature and pressure will result if part of the heat rejection to the surroundings occurs in the compressor. For this reason, and because the work of compression is reduced when there is heat rejection, compressors which involve a high degree of heat rejection from the refrigerant are to be preferred for refrigeration applications.

If the compressor discharge vapour is at a temperature above that of the surroundings, then, unnecessary work has been expended in the compression process. The extent of the unnecessary work depends on the characteristics of the compressor and on the characteristics of the condenser. In a condenser of finite heat transfer area, the condensing pressure and the discharge pressure of the compressor will be higher than the saturation pressure corresponding to the temperature of the surroundings. However, even with an ideal condenser of infinite heat transfer area and zero pressure drop, the discharge temperature of the compressor at the saturation pressure corresponding to the temperature of the surroundings might be higher than that temperature. This would invariably be the case for current technology reciprocating refrigeration compressors.

On the basis of the above discussion a new type of rational efficiency, which depends on the characteristics of a refrigeration compressor combined with a condenser, can be defined.

$$E_{ircr} = \frac{\dot{m}(b_{tosat} - b_1)}{P_i} \qquad 9.$$

where

E_{ircr} = indicated rational efficiency for compression and heat rejection
b_{tosat} = specific flow availability function of saturated refrigerant, evaluated at the temperature of the surroundings, T_o [J/kg]
= $h_{go} - T_o s_{go}$
and also
= $h_{fo} - T_o s_{fo}$
h_{fo} = specific enthalpy of saturated liquid refrigerant at the temperature of the surroundings, T_o [J/kg]
h_{go} = specific enthalpy of dry saturated vapour refrigerant at T_o [J/kg]
s_{fo} = specific entropy of saturated liquid at T_o [J/kg]
s_{go} = specific entropy of dry saturated vapour at T_o [J/kg]

Given the compressor suction condition, the indicated power and the temperature of the surroundings, the rational efficiency for compression and heat rejection, E_{ircr}, can be evaluated for a specified suction pressure, suction temperature and discharge pressure. This parameter represents the performance of the compressor, combined with a condenser which operates at the specified pressure and de-superheats, condenses and subcools the refrigerant to the temperature of the surroundings.

Where the compressor is to be used to provide cooling only, the value of $E_{irc r}$ can be quoted to quantify its performance in conjunction with a condenser which would cause it to operate at a specified discharge pressure. This parameter incorporates information on the suitability of the discharge state to the purpose of the plant. Values are plotted in fig. 6.

$E_{irc r}$ could also be evaluated experimentally for the special case where the discharge pressure is the saturation value corresponding to the temperature of the surroundings. This would represent the rational efficiency of the compressor combined with an ideal condenser - this figure could be of interest where the highest possible coefficient of performance was to be achieved by using a highly oversized condenser.

Situations sometimes arise where, in addition to the cooling effect, some useful heating is provided by the plant. In these cases, as for heat pump compressors, the indicated rational efficiency, E_{ir}, and the discharge temperature should be quoted in order to describe the merit of the refrigeration compressor in utilising its shaft power input, when operating between a specified suction state and discharge pressure. If useful heating is provided directly from the compressor, e.g. from a cooling water jacket, then a modified definition of E_{ir}, to include the availability transfer corresponding to this heating effect, would be necessary.

Discussion of the Indicated Power Utilisation Efficiencies

The indicated adiabatic efficiency E_{is}, the indicated rational efficiency, E_{ir}, and the indicated rational efficiency for compression and heat rejection, E_{ircr}, all showed small linear decreases with increased speed, fig. 6. The indicated adiabatic efficiency was a little lower than the rational efficiency, while E_{ircr} was considerably lower than both. All three characteristics versus speed had roughly the same slope.

The indicated rational efficiency, which had a value of about 60% in the middle of the speed range, was of particular significance in the context of heat pump applications of the compressor. Forty percent of the availability of the indicated power was lost, due mainly to the irreversible throttling processes at the suction valves and the consequent suction and discharge pumping work requirements. Part of the loss in availability would also have been due to the heat loss from the compressor to the surroundings and due to the fact that heat transfer processes within the compressor were irreversible.

The approach adopted in presenting the indicated rational efficiency as a performance parameter of the compressor has scope to trace the availability of the refrigerant, and availability transfers, throughout the cycle of the reciprocating compressor, and thus attribute the overall loss in availability to specific quantified causes. This was outside the scope of the work described here and would have required better test data on the thermodynamic states of the refrigerant within the cylinder. The more traditional parameter, the isentropic efficiency, does not have scope for this type of analysis, even if the data were available.

In the context of refrigeration applications, the values of E_{ircr} showed that if the compressor were combined with a condenser, which operated at the discharge pressure used in the tests and de-superheated, condensed and subcooled the discharge vapour to the liquid state at the temperature of the surroundings, about 50% of the availability of the indicated power would be lost. This loss would be due mainly to the factors mentioned above, but, part would also be due to the inappropriateness of the discharge pressure, which, in the tests, was higher than the saturation pressure corresponding to the temperature of the surroundings, and, to the inappropriateness of the discharge temperature.

UTILISATION OF VOLUME DISPLACEMENT

The displacement utilisation efficiency, which is also known as the volumetric efficiency, quantifies the utilisation of the volumetric displacement of the piston and is defined as follows:

$$\eta_v = \frac{\dot{m}\, v_s}{N\, V_{sw}} \qquad 10.$$

where

η_v = displacement utilisation efficiency, or, volumetric efficiency

v_s = specific volume of the refrigerant at the suction condition $[m^3/kg]$

Values of the displacement utilisation efficiency, η_v, are plotted in fig. 7. It was found to remain almost constant over the range of compressor speeds, at about 66%. If the compressor were operated over this speed range for the purposes of capacity control, there would be no further net displacement utilisation losses due to such operation.

Indicated Volumetric Efficiency

This parameter is calculated from measurements on an indicator diagram such as that shown in fig. 8.

$$\eta_{vi} = \frac{V_1 - V_4}{V_{sw}} \qquad 11.$$

where

η_{vi} = indicated volumetric efficiency

V_1, V_4 = volumes at the points of suction pressure equalisation, from the indicator diagram, e.g. at points 1 and 4, fig. 8. $[m^3]$

The indicated volumetric efficiency, η_{vi}, showed a significant decrease from about 89% to about 81% over the speed range tested, fig. 7. Furthermore, these values were considerably higher than the actual displacement utilisation efficiency values. The differences were due to effects which may have included heat transfer, backflow through the valves, leakage past the valves and the piston, and consequences of cyclic refrigerant solubility in the lubricating oil.

CONCLUSIONS

The fundamental characteristics of a compressor have been presented for a particular suction condition and discharge pressure over a range of operating speeds. The main performance parameters have also been stated. The purpose of the paper was not simply to provide data which related to a particular compressor, but, to focus on the essentials of compressor performance and provide a basis for discussion on the ways in which data describing the essentials can be presented.

ACKNOWLEDGEMENT

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REFERENCES

1. McGovern, J.A. 1986. "Performance testing of an open reciprocating refrigeration compressor over a range of speeds". Proceedings of the 1986 International Compressor Engineering Conference - at Purdue, edited by Hamilton, James F. and Cohen, Raymond. West Lafayette, Indiana: Purdue University. pp.242-253.

2. Hughes, D.W., McMullan J.T. and Morgan, R. 1982. "Determination of the thermodynamic properties of refrigerant mixtures". Proceedings of the 1982 Purdue Compressor Technology Conference, edited by Cohen, Raymond. West Lafayette, Indiana: Purdue University. pp.214-221.

3. Moran, Michael J. 1982. Availability Analysis. New Jersey: Prentice Hall.

4. Kotas, T.J. 1985. The Exergy Method. London: Butterworths.

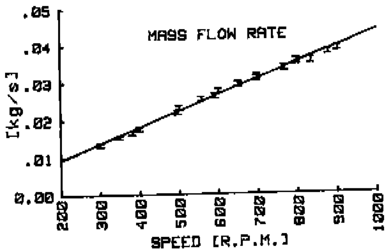

Fig. 1

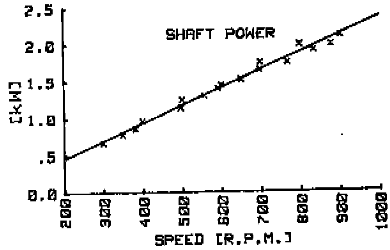

Fig. 2

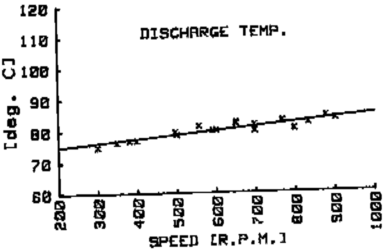

Fig. 3

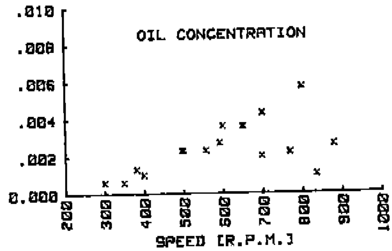

Fig. 4

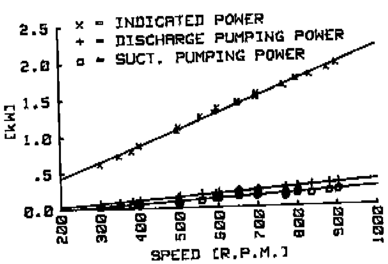

Fig. 5

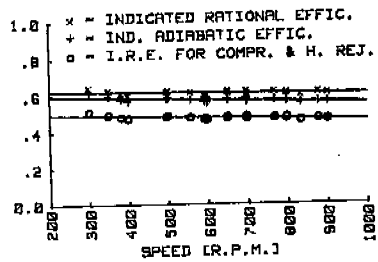

Fig. 6

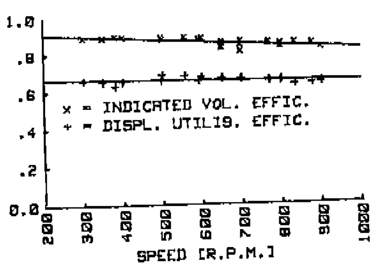

Fig. 7

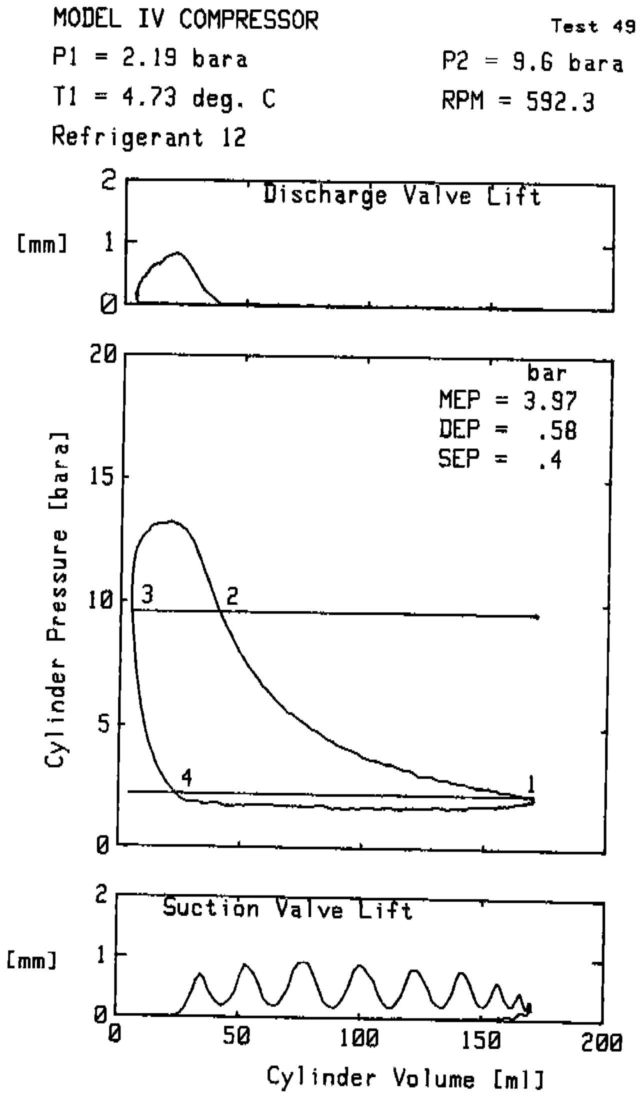

Fig. 8 Diagrams of valve lift and cylinder pressure versus volume. The horizontal lines, 3-2 and 4-1 on the pressure diagram represent the mean pressures in the discharge and suction pipes respectively. The points of suction pressure equalisation, 4 and 1, are where the cylinder pressure equals the suction pressure. Similarly, at the points of discharge pressure equalisation, 2 and 3, the pressure within the cylinder is equal to the mean discharge pressure.