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INVESTIGATION OF THE THERMODYNAMICS
OF A RECIPROCATING COMPRESSOR.

Ole Jensen, M.Sc., Lab. of Refrigeration.
Techn. Univ. of Denmark.

INTRODUCTION

The recent development in mathematical simulation models of reciprocating compressors has made clear the demand for data and correlations of the various detailed processes taking place in the compressor. Examples of such are: Heat transfer in suction and discharge plenums, valve passages, and cylinders, and also between the compressor and the ambient, frictional pressure drop and pressure pulsations in suction and discharge manifolds. The correlation of the heat transfer rates makes knowledge of temperatures of the corresponding parts of the compressor desirable, and an analytical model should be able to predict these. Compressor body temperatures are therefore also studied in this investigation. But first more knowledge of reciprocating compressor thermodynamics must be achieved. This investigation is a modest contribution hereto.

DATA FOR THE TEST COMPRESSOR.

The compressor is tested with R-12 and is a two-cylinder, belt drives machine. This conventional construction was chosen because of its simplicity and accessibility. It is furthermore relatively easy to describe thermodynamically. The ratio of the surface areas to the swept volume is high, and therefore also the internal and external heat transfer can be varied relatively much in relation to e.g. mass flow rate.

Compressor data
Cylinder dimensions: D = 85 mm (3,35 in.)
S = 65 mm (2,56 in.)
Design speed interval: 310-465 rev./min.

Displacement (2 cyl.): 682 cm³ (= 41,6 qu.in.)
Swept volume at 420 rev.: 17,2 m³/h (=10 qu.ft/min)
Clearance volume per cyl.: 14 cm³ (= 0,85 qu.in.) ~ 4,1%.

Surface areas
Outside: 0,67 m² (= 1039 sq.in.)
Inside:
Suction channel 470 cm² (= 73 sq.in.)
Discharge 147 cm² (= 22,8 sq.in.)

Fig. 1 Section through the test compressor.
Surface temperature measuring points are indicated by the figures. No. 5 and 8 are on the outside cylinder surface, no. 12 in the oil and no. 4 on the discharge side of the cylinder cover.
Valves
Two suction and two discharge valves per cylinder. Valve reed thickness: 0.4 mm
(≈ 0.016 in.)

TEST STAND AND MEASUREMENTS

The compressor test stand is shown on fig.2. A precise measurement of the refrigerant mass flow rate can be obtained in the liquid phase of the refrigerant. All the circulating refrigerant is therefore condensed.

The flow is measured with specially calibrated Rotameters. The test stand permits a good control of the vapour temperature at the compressor inlet by means of the brine-temperature and flow rate, by a constant pressure valve controlling the evaporator pressure and by a liquid-vapour heat exchanger. Special care with by-pass has been taken to avoid droplets of oil or refrigerant to enter the compressor. This is done by inserting four oil and liquid separators in the refrigerant circuit.

A special arrangement is made to measure the piston leakage. The aim was not to study this, but merely to control, that the assumption of negligible piston leakage was and remained correct. The principle of the measurement is shown in fig.3. Before a measurement the pressure in the vessel is reduced to a valve reasonably lower than the desired pressure in the crankcase. With the constant pressure valve the pressure in the crankcase is kept constant. The measurement can run as long as the pressure in the vessel is lower than in the crankcase.

Pressure and temperature in the vessel are measured and the integrated leakage mass can then be calculated. The rate of piston leakage is in the order at 1% of the refrigerant mass flow.

THE INSTRUMENTATION

The cylinder pressure is measured with a piezo-electric pick up installed in the top of the cylinder. The signal is lead to a charge amplifier that is connected to the y-axis of an oscilloscope. The pressure pick up is mounted in a two-way switching...
The absolute value of the cylinder pressure is measured by exposing the pickup to a reference pressure, e.g. atmosphere or evaporator pressure.

Adaptor, that makes it possible to obtain a reference line for the pressure signal, as shown on fig.4. As a p-v diagram was preferred, a piston way transducer of the capacitive type was coupled to the crankshaft and the signal connected to the x-axis of the oscilloscope.

The temperatures of the compressor was measured on 8 places, as indicated on fig.1. For the measurement Ni-CrNi thermocouples were used, with the cold junction in distilled water-ice, and the hot junction in 2-3 mm deep grooves on the surface. The temperature of the gas was measured on 4 places (fig.5). The hot junction was inserted in the gas stream in hypodemic tubes filled with oil; an example is shown on fig.6.

The signals are measured and recorded on a 24-channel potentiometric recorder. The stationary vapour pressures were measured by precision bourdon-tube pressure gauges. The power consumption of the compressor was determined by measuring the torque of the motor and its speed.

### Test Conditions

Four independent variables were chosen:

- Suction pressure
- Discharge pressure
- Vapour temperature at compressor inlet
- Compressor speed

The table shows their tested values.

**Speed:** 210, 420, 840 rev/min.

Usually 4-6 tests were necessary to cover these ranges of suction vapour superheating. This demands a number of tests of 100-120 with only one refrigerant. It would be desirable to run tests with two other refrigerants, and of course the number of tests for these can be much lower.

#### Table 1

<table>
<thead>
<tr>
<th>Table 1</th>
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<tr>
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<td>Condenser Pressure Temperature</td>
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<td>85 21 70</td>
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<td>8</td>
<td>114 32 90</td>
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<td>142 41 105</td>
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![Fig.4](image1.png) **Fig.4** The vapour temperature was measured at the points no. 1, 2, 4 and 5. The suction vapour is heated both from the cylinder and the discharge chamber. About 1/5 of the total suction chamber surface area is in the cylinder cover.
THE INDIVIDUAL LOSSES IN THE EFFICIENCES

The losses in volumetric efficiency can be divided into four single losses, originating from:

1) Clearance volume
2) Suction manifold heat transfer
3) Cylinder heat transfer
4) Suction valve losses.

An analysis of the test results is made according to this division.

The clearance volume loss is measured from the p-v diagrams as shown on Fig. 7.

The suction manifold superheat is directly measured by thermocouples. The loss is calculated as $\eta_s = v_3/v_2$.

The two losses, 3) and 4), are lumped together and are calculated indirectly from Fig. 7.

The total volumetric efficiency is then

$\eta_v = \eta_c \cdot \eta_t$

where $\eta_s$ is the individual clearance volume efficiency.

Fig. 8 shows, for two test series, $\eta_s$, $\eta_t$, $\eta_c$ and $\eta_v$ with the vapour superheating at the compressor entrance as the independent variable. The standard deviation of this calculation of $\eta_v$ from the direct calculation is about 1.5%. In the lower part of Fig. 8 is for the same tests the total volumetric loss shown as the sum of the individual losses. The increase in $\eta_v$ at higher superheatings is due to the increase in $\eta_t$, which again mainly is due to a reduction in the loss caused by cylinder heat transfer and valves.
Fig. 9 The total loss in indicated efficiency shown as the sum of the individual losses.

The indicated efficiency is influenced by the valve losses and the suction vapour heating. It appears from fig. 10 that the valve losses are independent of the vapour superheating at the suction stop valve, but the suction vapour heating in the machine decreases with increasing vapour superheating before the machine and causes thereby an increase in \( \eta_i \).

Suction vapour heating includes here also dissipation of energy from suction valve. The individual efficiency of it is calculated from

\[
\eta_h = \frac{v_1}{v_3}
\]

The standard deviation of this calculation of \( \eta_i \) from the value calculated from

\[
\eta_i = \frac{m \cdot \Delta h_{ad}}{W_i}
\]

<table>
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<th>kcal</th>
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Fig. 10 The vapour enthalpy increase in the machine. For the measuring points see fig. 5. Also shown is \( W_i \):
Indicated power and \( W_k \): Total power input to the compressor, both per lb (kg) R-12. The isentropic enthalpy increase is calculated from \( t_1 \).
is for the test shown on fig. 9 about 1%. The increase in 

The individual losses will naturally differ much from one machine to another, depending on the valves and the suction vapour heat transfer. But it is useful to notice that \( n_2 \) can be determined well from those two losses.

**THE VAPOUR ENTHALPY INCREASE IN THE MACHINE**

Useful information can be gained by study of the vapour enthalpy increase in the machine under different operating conditions. The enthalpy increase takes place in three main sections: the suction manifold, the cylinders, and the discharge manifold. Thermocouples were inserted between these sections, and at the compressor inlet and outlet, as indicated on fig. 5.

The results from a test with constant discharge and suction pressure corresponding to saturated refrigerant temperatures of appr. 32°C (90°F) and -30°C (-23°F), and with the suction vapour superheating as the independent variable is shown on fig. 10. The suction manifold heat transfer causes an enthalpy increase \( h_2-h_1 \) of about 3-4 kcal/kg (5-7 BTU/lb), which corresponds to a temperature increase of 20-30°C (35-55°F). It is smallest at the highest temperatures, because of the smaller temperature difference between the compressor walls and the vapour (see fig. 11).

The enthalpy increase during the compression is also smallest at the highest temperatures, because of both the smaller temperature difference and the increased heat transfer to the surroundings, as indicated in the figure. At low superheating is \( q_k = q_f \) but at high suction vapour temperatures is \( q_k \) so high, that \( h_5-h_1 \) is smaller than the isentropic enthalpy change.

The refrigerant mass flow rate is seen to decrease at higher suction temperatures even though \( n_f \) is known to increase.

**THE TEMPERATURE OF THE COMPRESSOR**

The temperature of the external surface of the compressor was measured at 8 points as indicated on fig. 1. Fig. 11 shows results from the same tests as shown on fig. 10, and, for the comparison, the discharge vapour temperature. The four cylinder temperatures differ as much as 10°C (18°F) from each other. No. 5 and 8 are measured on the side opposite to the suction side, and they are 5-7°C (9-13°F) higher than the corresponding temperatures at the end of the compressor. The measurement side is also exposed to the piston guide pressure. The average temperature of the suction and discharge pressures is at low suction temperatures lower than the lowest cylinder temperature, and at the highest suction temperatures almost equal to the highest cylinder temperatures. The temperature of the internal surface of the cylinder is approximately the same as of the external surface, as the external heat transfer coefficient is low. It has been found to 14 kcal/m²h°C at 420 o/min, but depends upon the compressor speed, e.g. at 210 o/min is it 10 kcal/m²h°C.

The high temperature difference between the cylinders and the crank house causes a considerable vertical heat transfer. The lowest of the crank house temperatures is, however, not much influenced of it, as it only varies 4°C (7°F) when the cylinder temperature varies 20°C (40°F). It follows mainly the oil temperature (\( t_{12} \)).

The mean cylinder temperature as a function of the pressure ratio (fig. 12) shows that this depends more of the discharge pressure than of the suction pressure. However, the constant suction vapour superheating implies different temperatures of the suction vapours, which must be taken in consideration.
The cylinder temperature is not of major interest in itself, but it is important for the suction manifold heat transfer, and should therefore also be predicted by an analytical model of the compressor. It is also important for the calculation of the heat transfer from the compressor to the surroundings.

THE HEAT TRANSFER TO THE SURROUNDINGS

The heat transfer to the surroundings $Q_k$ depends on several factors: The ratio of internal and external surface area, the rate of internal heat transfer, the compressor speed or mass flow rate. Data obtained from one machine must therefore be correlated carefully before further use.

$Q_k$ is important specially for compressors with a relatively large external surface. For the compressor tested was at 420 o/min $V_r/F_u = 25$ m$^3$/h/m$^2$, which is a very low value.

$Q_k$ was calculated from

$W_k = Q_k = m (h_5 - h_4)$

$Q_k$ is much smaller than the other two quantities, which explains the scatter of the results shown on fig.13.

Fig.12 The average cylinder temperature at constant suction superheat $t_1 - t_0$. The suction temperatures are therefore different for different suction pressures. This explains that it looks nearly independent of the suction pressure.

Fig.13 Heat transfer from the compressor to the surroundings per kg (lb) R-12 circulated.

Fig.15 Temperature increase $t_2 - t_1$ of the vapour in the suction chamber.
The suction manifold heat transfer

An important problem in thermodynamic modeling of a compressor is the suction manifold heat transfer. The internal heat transfer consists mainly of heat transfer from the discharge manifold and the cylinders to the suction manifold, but it is at present not possible to predict the magnitude of it. To enable this, experimental data are needed.

The heat transfer was determined by measuring the temperature $t_1$ and $t_2$ (fig.1) and calculate it from

$$Q_s = m \cdot c_p (t_2 - t_1)$$

The results can be correlated in several ways. Here the directly measured temperature difference $t_2 - t_1$ is presented as a function of two different independent variables (fig.15 and 16). The dependence of $(t_2 - t_1)$ of $(t_1 - t_0)$ results in the increase of $n_v$ with $(t_1 - t_0)$ in spite of an decrease in $m$.

The refrigerant mass flow rate

It is well known, that the mass flow rate depends strongly on the pressure ratio, as fig.19 shows. Fig.20 shows how it decreases when the suction vapour superheat increases, because of the larger specific vapour volume, while $n_v$ increases because of less increase of superheat in the machine. At $p_5 = 10$ ata and $p_1 = 1$ ata refrigerant vapour condensed in the cylinder at $t_1 - t_0 < 25-30^\circ$C. The cylinder wall temperature was lower than the saturation temperature at these conditions.

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Fig.16 Temperature increase $t_2 - t_1$ of the vapour in the suction chamber at constant, suction vapour superheat, $t_1 - t_0$.

Fig.17 R-12 mass flow rate decrease at increasing suction superheat. Note vapour condensation in cylinders at $p_5/p_1 = 10/2$ ata.

Fig.18 R-12 mass flow rate at different pressure ratios and a constant suction superheat, $t_1 - t_0$. 
NOMENCLATURE

\( c_p \): Specific heat at constant pressure

\( F_u \): External surface area of compressor.

\( \Delta h_{ed} \): Isentropic enthalpy difference

\( \dot{m} \): Mass flow rate

\( \dot{m}_s \): Clearance volume mass flow rate

\( p \): pressure

\( Q_k \): Heat from compressor to the surroundings

\( q_k \): \( Q/\dot{m} \)

\( t \): temperature

\( V_k \): Swept volume of compressor

\( v \): Specific volume of R-12

\( W_f \): Frictional power

\( W_i \): Indicated power

\( W_k \): Total power

\( \varepsilon \): Clearance volume ratio.

\( \eta_h \): Thermal indicated efficiency \((v_1/v_3)\)

\( \eta_i \): Indicated efficiency

\( \eta_c \): Clearance volume volumetric efficiency

\( \eta_s \): Suction chamber volumetric efficiency \((v_1/v_2)\)

\( \eta_t \): Thermal volumetric efficiency \((v_1/v_3)\)

\( \eta_v \): Volumetric efficiency

SUBSCRIPTS

Referring to the vapour (see fig. 5)

0: At the evaporator (assumed is \( p_0 = p_1 \))

1: At compressor suction stop valve

2: At suction valve (fig. 6)

3: At end of suction stroke

4: At discharge valve

5: At discharge stop valve

Referring to compressor temperatures, see fig. 1.