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PERFORMANCE TESTING OF AN OPEN RECIPROCATING REFRIGERATION
COMPRESSOR OVER A RANGE OF SPEEDS

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ABSTRACT

A load stand has been built and used to determine the performance of an open type reciprocating refrigeration compressor over a range of speeds.

The load stand and instrumentation are described and sample test readings are given. Measured performance characteristics for the compressor over a range of speeds are presented. Aspects of the load stand are discussed and conclusions are drawn from the results of the compressor tests.

SYMBOLS

- m mass flow rate of refrigerant through the compressor, kg/s
 Q rate of heat transfer in the heat exchanger, W
 h_i specific enthalpy of refrigerant at measurement location i , J/kg
A,B,C,D load stand control valves
a,b,c,.. measurement locations on the load stand
M.E.P. mean effective pressure, bar
TDC top dead centre position
BDC bottom dead centre position
 η_{mech} mechanical efficiency
 η_{vol} volumetric efficiency

INTRODUCTION

This paper describes work which has been carried out at Trinity College Dublin as part of a research programme entitled "Capacity Control of Reciprocating Piston Compressors for Heat Pump or Refrigeration Applications by Speed Variation".

A load stand has been built according to the principles outlined in a previous paper (1) and has been found to operate satisfactorily. Work is continuing in developing the instrumentation and data logging facilities and in automating the test procedures.

DESCRIPTION OF THE LOAD STAND AND INSTRUMENTATION

The load stand is shown schematically in Fig. 1. It is a hot gas bypass type, employing a single heat exchanger, in which part of the total refrigerant flow is fully condensed to the liquid state and then mixed with the remainder. The suction and discharge pressures and the suction superheat temperature are controlled by means of throttle valves.

Control of Operating Conditions

The compressor is belt driven by a standard induction motor and speed is varied by means of an electronic frequency inverter.

The saturation temperature in the heat exchanger is determined by the temperature and flow rate of the cooling water and by the rate of heat rejection from the refrigerant. This rate is approximately equal to the indicated power of the compressor. For the water flow conditions described in the sample results the sensitivity of the heat exchanger saturation temperature to compressor indicated power is about 5 K/kW. Thus it changes little with the operating conditions of the compressor.

Discharge pressure is controlled by valve A while valve D is used to control the compressor suction pressure. The settings of valves B and C determine the suction superheat temperature. Valve B in the vapour bypass line provides a pressure difference across valve C and ensures adequate liquid feed to that valve which is located above the condenser level. Adjustment of valve B is necessary only when major changes in operating conditions are made, e.g. for most operating conditions it can be left fully open, but, it must be closed partially at very low compressor speeds. Valve C is a fine metering valve and controls the flow rate of liquid from the heat exchanger. It was found necessary to install two valves, one for coarse and one for fine adjustment, to obtain adequate metering over a wide range of operating conditions. These valves are fitted with micrometer handles so that their settings can be noted and repeated. The four main control valves A,B,C,D are interactive to some extent and an adjustment in one may require a compensating adjustment in the others.

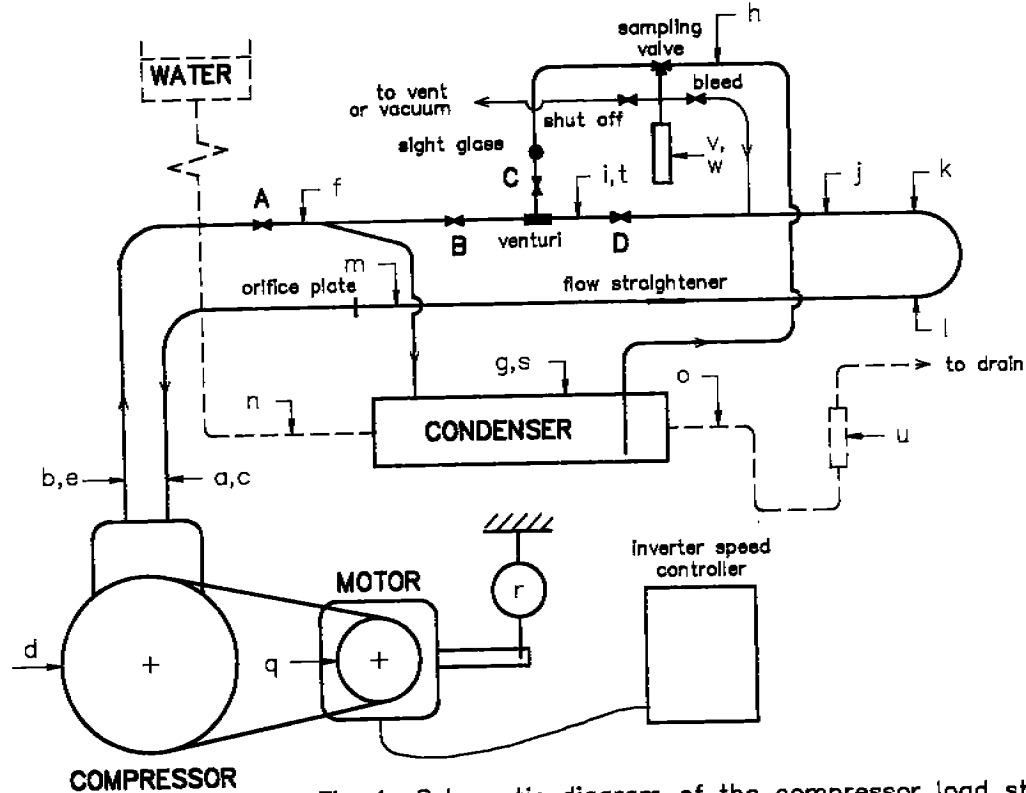


Fig. 1: Schematic diagram of the compressor load stand. Lower case letters indicate measurement locations: refer to sample readings for descriptions and values.

Measurement of Torque and Speed

The electric motor is mounted on a cradle which is pivoted about the axis of rotation. The motor reaction torque is measured by a spring balance attached to a radius arm on the cradle. Shaft power is computed from the motor torque and speed, and thus includes any power loss in the drive system.

Motor speed and compressor speed are measured by the use of thin serrated rims attached to the belt pulleys. Slotted opto switches which straddle these rims provide square wave pulse trains. Rotational speed is computed from the pulse count of each opto switch over a period of one second. A third opto switch provides a top dead centre marker in conjunction with a tag attached to the compressor pulley.

Measurement of Temperatures and Pressures

Temperatures on the refrigerant and water circuits are measured by thermocouple junctions attached to the pipework by adhesive tape. All pipework is insulated to minimize heat losses or gains and to ensure accurate thermocouple readings.

There are four main pressures on the refrigerant circuit: suction pressure, discharge pressure, heat exchanger pressure and mixing pressure. Gauges are installed to give visual indications of these pressures and, in the case of the suction and discharge pressures, transducers are installed for data logging purposes.

Mass Flow Rate of Refrigerant

The mass flow rate of refrigerant through the compressor is determined by a heat balance on the condenser, combined with a heat balance for the liquid and vapour mixing process. The rate of heat transfer to the cooling water is obtained from its flow rate and temperature rise. The mass flow rate of refrigerant through the compressor is given by

$$m = \frac{Q}{h_f - h_h} \left(1 + \frac{h_j - h_h}{h_h - h_j} \right)$$

where m = refrigerant mass flow rate through compressor
 Q = rate of heat transfer to cooling water
 h = specific enthalpy at measurement position
indicated by subscript

Water flow rate is read visually from a variable area flow meter.

Oil Concentration Measurement

The mass concentration of oil in the refrigerant coming from the compressor is measured by a weight over volume method. A sample of up to about 12 ml of refrigerant is drawn into an evacuated glass vessel of known weight. The volume of refrigerant is read from a scale and it is then allowed to evaporate slowly by opening a bleed valve to the suction line. When there appears to be only oil left in the vessel the bleed valve is closed and the

sample vessel is vented to atmosphere. It is then connected to a vacuum pump and evacuated to ensure the evaporation of any remaining refrigerant. The vessel can then be removed and weighed to determine the mass of oil.

Valve Lift Measurement

Eddy current type proximity transducers are used to measure valve lift (see Figs. 2a and 2b).

The suction valve transducer was mounted almost flush with the inner valve plate surface. As the compressor characteristics were not to be altered the choice of location was very restricted and compromises had to be accepted. The transducer was installed in a position a little to the side of the valve port and directly under the web separating the suction and discharge plenums in the cylinder head. A clearance hole was drilled through the valve plate and the threaded transducer was mounted, using thread locking fluid, in a tapped hole through the web. At this location the web was modified slightly to increase its thickness by deposition of cast iron weld. It was not possible to provide a recess of one diameter all around the tip of the transducer, as recommended in the manufacturer's data sheet. The maximum recess which did not interfere with the valve seat was used instead. A PTFE insert was press-fitted into the recess and was milled so that the groove around the valve seat continued into the insert, maintaining the original geometry. The clearance volume of the cylinder was thus unaffected.

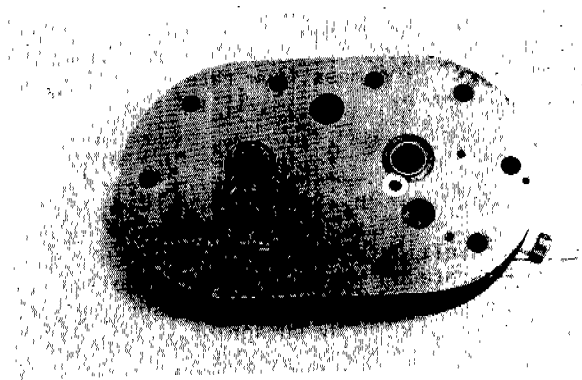


Fig. 2a: View showing the tip of the suction valve displacement transducer surrounded by the white PTFE insert. The pressure port for the piezoelectric transducer is also visible.

The discharge valve lift transducer was mounted over the mid-point of the valve. Radial clearance for the tip was provided by drilling a large hole through the spring-loaded valve back plate. Reinforcement of the back plate was necessary as a consequence. A

hole had also to be drilled through the tie bar between the two valve mounting bolts. The threaded transducer was supported in a small tapped boss fitted to the cylinder head in the discharge plenum. A lock nut was used.

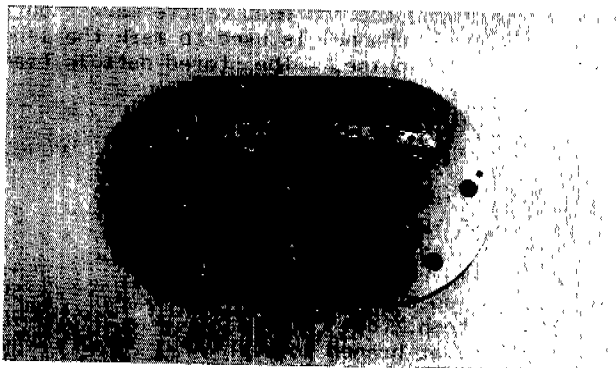


Fig. 2b: View of the plenum side of the valve plate: the valve back plate and tie bar on the right hand side have been modified. The hole through the valve plate for the suction valve transducer can also be seen.

The manufacturer's calibration data did not apply to either of the transducers due to insufficient radial clearances at the tips and due to the fact that the target area on the suction valve was less than ideal. For this reason the transducers were calibrated after they had been mounted on the cylinder head and with the valve plate bolted to the head. The gasket between the two was in place when this was done so that its thickness was taken into account in the calibration. All ten bolts were tightened to full torque. Access to both valves for displacement measurement during the calibration was from the cylinder side of the assembly. Polynomials were fitted to the measured lift versus output signal data.

Cylinder Pressure Measurement

A miniature piezoelectric pressure transducer (6.3mm dia.) was installed in a hole drilled into the edge of the valve plate. A 3.3mm hole was drilled from the cylinder side of the valve plate as the pressure port (see Fig. 2a).

Data Logging

A computer interfacing system with secondary addressing, multiplexing and A to D facilities communicates with a host microcomputer on an IEEE bus. The thermocouples, opto switches and pressure transducers are connected to this unit. It is intended to replace the spring balance for the measurement of motor

torque by a load cell. An orifice plate and flow straightener have been installed in the suction line. These will be used in conjunction with a pulsation damping chamber and a differential pressure transducer as the primary method for the measurement of refrigerant flow rate.

A four channel digital storage oscilloscope is used to capture valve lift and cylinder pressure data over one cycle. This is triggered by a pulse from the top dead centre marker on the compressor pulley. One channel is used to mark the length of the cycle using the same IDC pulse. The stored data is transferred to the host microcomputer on the IEEE bus.

SAMPLE TEST READINGS

a.	suction pressure/(bar a)	2.18
b.	discharge pressure/(bar a)	7.4
c.	suction temperature/(deg C)	4.9
d.	compressor speed/(r.p.m.)	599.6
e.	discharge temp./(deg C)	70.5
f.	refr. temp. at branch to heat exchanger/(deg C)	67.4
g.	heat exchanger surface temp./(deg C)	25.1
h.	refr. liquid line temp./(deg C)	17.8
i.	temp. of refr. after mixing/(deg C)	15.5
j.	refr. temp., suct. line, 1st/(deg C)	3.7
k.	refr. temp., suct. line, 2nd/(deg C)	3.5
l.	refr. temp., suct. line, 3rd/(deg C)	3.6
m.	refr. temp., suct. line, 4th/(deg C)	4.1
n.	temp. of water into cond./(deg C)	9.2
o.	temp. of water out of cond./(deg C)	14.9
p.	ambient temp./(deg C)	20.0
q.	motor speed/(r.p.m.)	1247.7
r.	motor torque/(Nm)	10.7
s.	heat exchanger pressure/(bar g)	5.2
t.	refr. press. after mixing/(bar g)	4.3
u.	water flow rate/(l/min)	3.21
v.	refr. sample vol./(ml)	11.25
w.	mass of oil/(g)	.0196

SAMPLE RESULTS

Operating Conditions

suction saturation temp./(deg C)	-10.2
discharge saturation temp./(deg C)	29.7
suction vapour temp./(deg C)	4.9
compressor speed/(r.p.m.)	599.6

Performance Data

refr. mass flow rate/(kg/s)	.0307
shaft power/(kW)	1.4
discharge temp./(deg C)	70.5

mechanical efficiency 1)
volumetric efficiency 2)
oil concentration/(kg/kg)

.95
.77
.0013

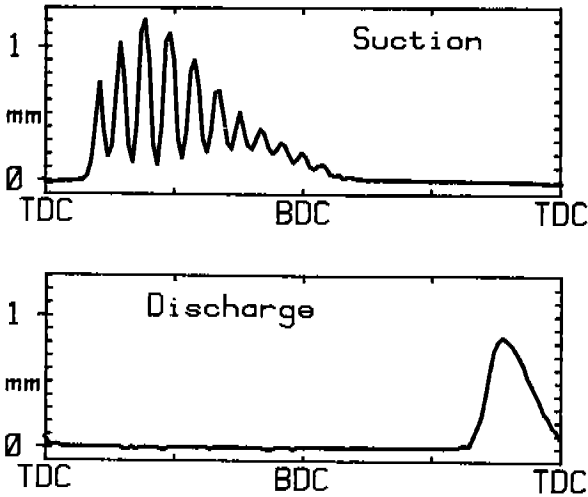


Fig. 3a: Valve lift versus time diagrams for one crank revolution

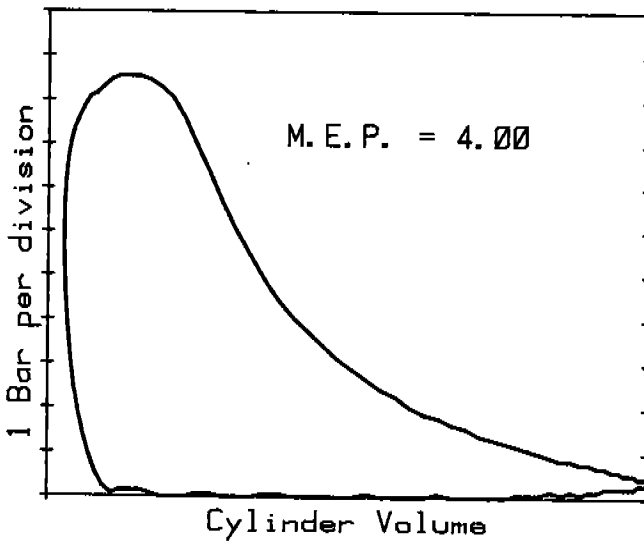


Fig. 3b: Cylinder pressure versus volume diagram derived from the measurement of pressure versus time for one crank revolution

Notes:

- 1) $\text{Emech} = (\text{indicated power}) / (\text{shaft power})$
- 2) $\text{Evol} = (\text{vol. flow rate at suct. condn.}) / (\text{swept volume rate})$

LOAD STAND PERFORMANCE

The load stand has been found to operate satisfactorily over a range of compressor operating conditions, although the precise limits have yet to be determined.

Under the present system of manual control a desired steady operating condition can be achieved in less than 50 minutes. In one test of 74 minutes duration, during which no adjustments were made, the deviations in the operating conditions were as follows:

suction temp:	-0.2 deg C
suction press:	+0.01 bar a
discharge press:	-0.01 bar a
compressor speed:	-1.2 r.p.m.
ambient temp.:	-0.7 deg. C

The thermodynamic processes which take place within the load stand and compressor are shown in Fig. 4

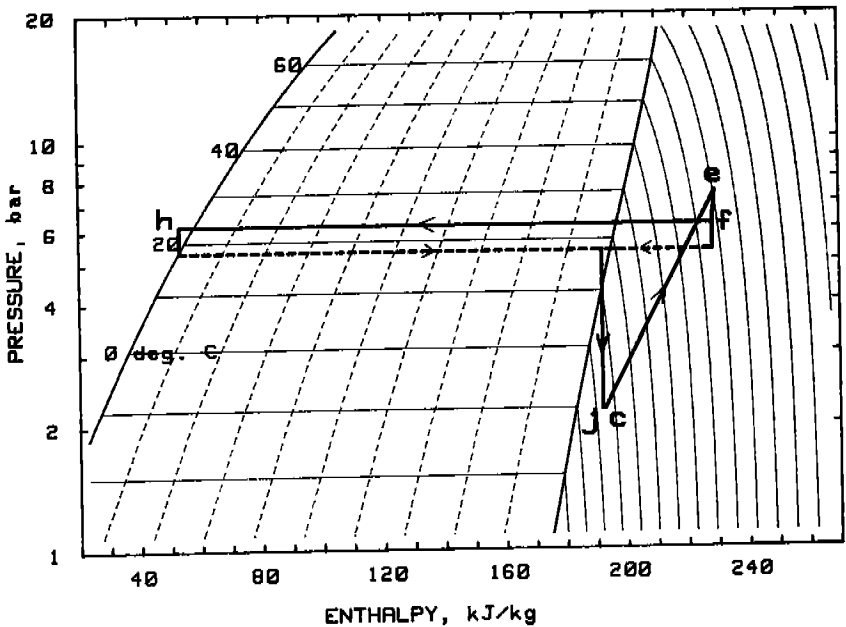


Fig. 4: Load stand and compressor processes: refrigerant R-12

Thorough mixing of the liquid and vapour streams is essential to the operation of the load stand. Originally the liquid line coming from valve C was connected as the vertical limb of an inverted I-junction to the horizontal vapour line between valves B and D. Detailed monitoring of temperatures at close intervals along the pipework led to the conclusion that liquid could build up in the bottom of the vapour line and, after a period, could flood through valve D. This effect introduced a considerable time lag between any adjustment of valve C and the corresponding change in the suction temperature at the compressor. Valve D had been mounted with its stem in the horizontal plane and this contributed to the problem also. This valve has an eccentric entry port which could form a liquid dam in the bottom of the pipe if not installed correctly.

Three steps were taken to overcome the accumulation of liquid:

- (i) A venturi was installed in the vapour pipe. The line from valve C (in fact two lines, one from the coarse and one from the fine adjustment valve) was connected to the throat of the venturi.
- (ii) The liquid metering valves, C, were moved as closely as possible to the venturi to minimize erratic lodgement of liquid in the short connecting pipes.
- (iii) The stem of valve D was moved into the vertical upwards position.

These measures had the desired effect. The sample readings illustrate the fact that the refrigerant in the pipe after valve D (measurement j) is superheated and that further temperature changes in the long suction line (k,l,m,a) are small and can be attributed to heat pick-up.

COMPRESSOR PERFORMANCE OVER A RANGE OF SPEEDS

The performance results at three different speeds are summarized in Fig. 5 over the range 300 to 600 r.p.m. Mass flow rate varies in direct proportion to compressor speed. Shaft power also varies roughly in proportion to speed. Mechanical and volumetric efficiency vary little over the range.

Further testing is necessary to establish a comprehensive set of performance figures, but, these preliminary results indicate that capacity control by speed variation would not involve any extra inefficiencies due to compressor characteristics, over at least this limited speed range.

The data on oil concentration vary considerably and further investigation will be required to determine if systematic errors are present. All measured values are below one half of one percent and it can be concluded that the influence of oil on the

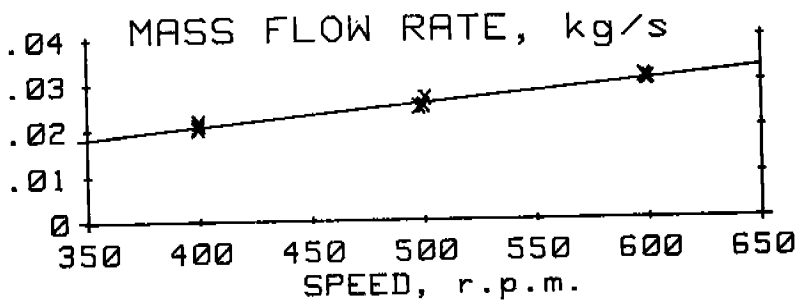
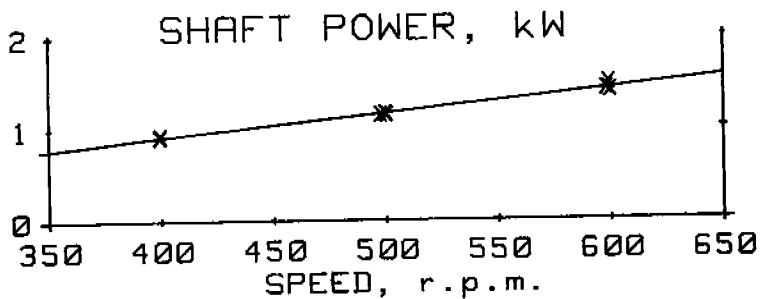
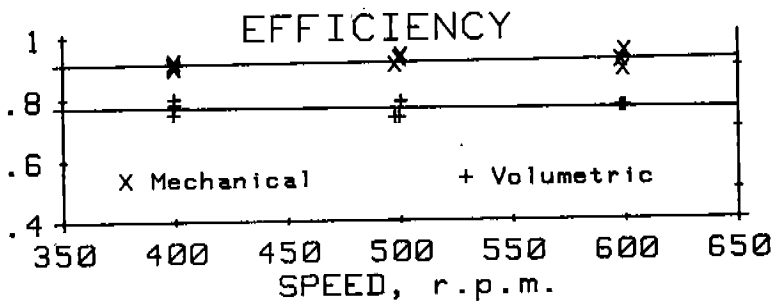
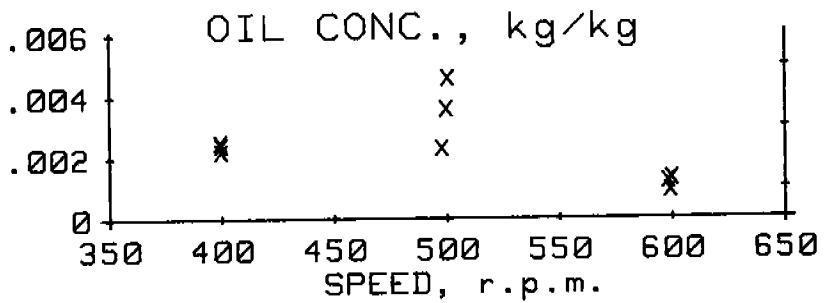


Fig. 5: Compressor test results

thermodynamic performance of a refrigeration system using this compressor would not be large.

CONCLUSIONS

A load stand to determine the performance of reciprocating compressors at different speeds over a wide range of operating conditions has been developed and has been used to test a two cylinder open type compressor. The load stand performs satisfactorily and initial results suggest that speed variation over the range 400 to 600 r.p.m., for the compressor tested, involves no significant losses in performance. In fact the mean shaft work per unit mass was found to decrease slightly at reduced speed.

ACKNOWLEDGEMENTS

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REFERENCES

1. J.A. McGovern, "Analysis of a Refrigerant Compressor Load Stand Incorporating Hot Gas Bypass and a Single Full Condensation Heat Exchanger" in Proc. of the 1984 International Compressor Engineering Conference - at Purdue, ed. Raymond Cohen.

RECENT STATUS OF TROCHOIDAL TYPE COMPRESSORS FOR HEAT PUMPS IN GERMANY

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H. Kruse

ABSTRACT

The development of trochoidal type heat pump compressors will be discussed. Evaluating the theoretical design parameters advantages and disadvantages of various principal versions of those machines will be considered in detail. Based on this consideration the hardware development of some trochoidal type compressors will be described as has been made in various German companies. Finally the design of the 1 : 2 version will be presented and test results will be dealt with. Resulting from this evaluation, conclusions regarding the recent status of trochoidal compressors will be drawn as far as the behaviour in heat pump application is concerned.

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