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A PRELIMINARY EVALUATION OF THE GROLL ROTARY VANE COMPRESSOR

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ABSTRACT

The principles of operation and design of the prototype of a novel design of vane compressor with significant advantages over the conventional sliding vane type are described. Its performance is given and compared to that of equivalent commercially available sliding vane and reciprocating compressors used in refrigeration systems. Its volumetric efficiency on average was 19% greater than that of the sliding vane compressor and 38% more than that of the reciprocating machine. The adiabatic efficiency was on average 15% greater than that of the sliding vane and about equal to that of the reciprocating compressor. Due to its high rotational speed it is more compact than the reciprocating compressor. It is smooth and quiet in operation and no mechanical faults were encountered in 432 hours of operation.

INTRODUCTION

Since January 1987 an investigation of a novel rotary vane compressor invented by Herr R. Groll of Munich, West Germany, has been carried out at the Thermo-Fluids Engineering Research Centre of City University, London, on behalf of Rotary Compression Systems (RCS) S.A. of Switzerland, who are holders of the design patents. The work involved the establishment of suitable test facilities, the design, manufacture and development of a machine capable of sustained running without mechanical failure, the development of computer software to support the design process and the evaluation of the performance of the machine as a refrigerant compressor once mechanical reliability was achieved. This paper presents a summary of the work carried out so far, the efficiencies achieved to date and how these compare with those of alternative existing commercial compressors.

THE GROLL COMPRESSOR

The Groll compressor [1] incorporates the same process of compression as a conventional sliding vane compressor but contains features which overcome some of the inherent problems. It comprises a rotor (1a) which rotates eccentrically within a circular stator (1b) as shown in Fig 1. The rotor is held in position within the stator by endplates. The rotor has longitudinal bores, the edges of the bores being tangential with the rotor radius. Within these bores are rollers (1c) with longitudinal slots cut in them. At the centre of the hollow rotor, a shaft (1d) is mounted which is central to the stator bore. Rods (1e) on this central shaft, are mounted which are fixed to the vanes (1f) located in the roller slots. The vanes are fixed to the rods so that the vane tips are adjacent to the stator bore. When the rotor revolves, the vanes are driven to rotate by the movement of the rotor. Because the vanes are fixed by the rods they revolve about the centre of the stator. Thus they remain normal to the stator surface as they rotate and their tips can have the same radius as the stator. Since the rotor is mounted eccentrically within the stator, the movement of the vanes with respect to the rotor is not radial. Therefore in order that the rotor can rotate, the rollers within the rotor which contain the vanes rock about their own centres. Friction between the vane tips and the casing, characteristic of sliding vanes, is eliminated by setting a small clearance between the vanes and the stator. The leakage through this clearance can be made less than that across the equivalent line of contact of a free sliding vane by making the vane much thicker to create a flow path of very high resistance.

The offset of the rotor centre from the stator centre can be greater than the difference in their radii by cutting a scallop in the stator. A large area of near contact between the rotor and stator is thus formed to make a very good seal between the high and low pressure regions (1g). Such a scallop is not possible in the sliding vane machine as it leads to radial bounce of the vanes. Because of the

scallop, the discharge port can remain open until all the high pressure gas is discharged. The high pressure pocket of trapped gas in the sliding vane compressor is thereby eliminated.

The kinematics of the mechanism are an inversion of the familiar slider - crank chain, as shown in Fig 2. This may be better understood if the rotor is considered to be stationary. OC in Fig 2 then becomes the crank and the vane - roller the slider. The mechanism is then the equivalent of the pivoted cylinder system sometimes used in model steam engines to avoid the need for small end bearings in the pistons as shown in Fig 3.

PROTOTYPE COMPRESSOR DETAILS

The compressor tested had six vanes in a rotor 101 mm diameter and 102 mm long with its centre offset from that of the casing by 4 mm. The vanes were 8 mm thick and the casing diameter was 108 mm. This gave a sealing scallop angle of 55° , a swept volume of 201 cu cm per revolution and a built in volumetric compression ratio of 2.1:1. The compressor was mounted on a horizontal test bed and connected directly to a two speed AC electric motor via a transmission dynamometer of the strain gauge type. The unit formed the compression stage of a refrigeration system which included a water cooled evaporator and condenser.

Lubrication of the vanes and the rollers in which they slide was through oil entrained in the refrigerant. In order to obtain reliable estimates of the refrigerant thermodynamic properties from temperature and pressure measurements at the compressor inlet, a coalescent oil separator was fitted to the discharge side of the compressor which removed virtually all the oil from the refrigerant in two stages. The first stage, which extracted the bulk of the oil, was of the centrifugal type. The separated oil was fed through a control valve directly to the main bearings and to the vane support cams via drilled oil ways. The partially refined refrigerant then passed through a secondary separation stage consisting of a fine filter membrane. The filtered oil was throttled and returned directly to the suction side of the compressor through the inlet port downstream of the temperature and pressure tapping points. The oil from each section of the separator was returned to the compressor through separate water cooled heat exchangers of the tube in tube type.

DEVELOPMENT PROGRAMME

The programme began with the design and construction of a four vane experimental machine and a simple refrigeration circuit in which to test it. The first design was based largely on sliding vane practice. This was rapidly improved as the unnecessary restraints of conventional compressor technology were discarded.

A computer program to carry out a stress analysis of the entire mechanism was then developed and this was used to aid in the design of a six vane prototype which replaced the four vane machine. At the same time an additional computer program was written to analyse the effect of changes of any geometrical parameter in the compressor design on the swept volume, volume ratio and port inlet and exit losses. A new and more extensive test rig was built for this machine.

The test rig was extensively instrumented with pressure transducers and high quality semiconductor temperature sensors each of which were individually calibrated. To improve the accuracy of the heat balances in the heat exchangers, the water temperature changes across them were obtained from differential measurements. Water flow measurements were made with turbine flow meters each individually calibrated. All signals were logged directly into a buffer file in an IBM AT type microcomputer using solid state circuits and associated power packs designed and manufactured at City University. The data was displayed on the monitor and, when satisfactory steady state conditions were obtained, was processed with the aid of THERPROP, a software package which has been developed at City University over a number of years. This contains a data bank of over 100 fluids which includes all the common refrigerants and subroutines for evaluating thermodynamic properties which are based on the Lee-Kesler equations of state for non-polar fluids. Heat balances thereby obtained showed average discrepancies of less than 5% and any results with discrepancies of greater than 7% were rejected.

RESULTS AND DISCUSSION

The compressor ran for a total of 432 hours and the results at 3,000 rpm are shown in Figs 4 to 7. The plotted points are the average of 91 recordings at 17 settings and were chosen to correspond as closely as possible to typical refrigerator operating conditions. The range of operation was as large as possible with the available cooling water temperatures.

As can be seen, both volumetric and adiabatic efficiencies fall as the pressure drop increases and causes both greater leakage and friction. Since the settings chosen correspond to simultaneous changes in both pressure ratio and difference, deviations from linearity are greatest where increases in pressure difference were attained while simultaneously decreasing the pressure ratio and vice versa. The very high volumetric efficiencies achieved at the low pressure drops confirm the advantages of zero clearance volume and low leakage rates which were predicted for this machine. However, the relatively low adiabatic efficiencies indicate that friction losses in this design were rather high.

These results are compared with those of equivalent reciprocating and rotary sliding vane compressors, taken and adapted from commercial brochures, in Figs 8 to 10. The improvements in evaporator duty show that the gain in volumetric efficiency of the Groll compressor over that of the sliding vane machine is too large to be allotted only to its zero clearance volume. It follows that, as was anticipated, the leakage across the vanes must be less. The overall mean arithmetic average of all points recorded shows a gain of 19% over the sliding vane machine and 38% over the reciprocating compressor.

The coefficient of performance graphs indicate that the adiabatic efficiency, while not high, is superior to that of the sliding vane machine and virtually equal to or better than that of the reciprocating compressor in all but the highest pressure differences. There, the inherently lower friction of the reciprocating machine shows to best advantage. There is however, little doubt that with further development, the friction losses of the Groll compressor could be reduced. Outright superiority in all respects over existing alternatives would then be achieved. For all points recorded, the overall mean arithmetic average COP and hence adiabatic efficiency is 15% greater than that claimed for the sliding vane and 2% greater than that of the reciprocating compressor. The latter figure is however too small to be significant.

It should be noted that the Groll compressor is extremely robust and despite a number of mechanical failures in the early development, none caused total destruction and in all cases repairs were effected with very few replacement parts. Consequently the entire test programme was completed on a single machine with very little down time due to mechanical breakdown. The entire test series recorded in this report, in addition to extensive test running at 1500 rpm, was carried out without failure.

Also due to absence of blade chatter, the Groll compressor runs very quietly with vibration levels sufficiently low to permit a 50p coin to balance on its edge on top of the compressor for extended operating periods.

CONCLUSIONS

Prospects for the Groll compressor appear to be most encouraging. By virtue of its high rotational speed it shares the advantage of compactness with the conventional sliding vane machine but is smoother and quieter in operation and far less likely to incur vane failure. The prototype machine performance is significantly better than that of developed sliding vane machines and equivalent in adiabatic efficiency to reciprocating compressors. There is little doubt that further improvements are possible as the development programme proceeds but even at this stage it appears to be commercially viable.

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REFERENCES

1. Groll, R. C. Rotary Piston Unit, European Patent 0102555 issued 2nd Dec 1987.

Fig. 1

Fig.2

Fig.3

GROLL COMPRESSOR PROTOTYPE 6 VANE

Fig.4

GROLL COMPRESSOR PROTOTYPE 6 VANE

Fig.5

GROLL COMPRESSOR PROTOTYPE 6 VANE

Fig.6

GROLL COMPRESSOR PROTOTYPE 6 VANE

Fig.7

Fig. 8

Fig. 9

Fig.10