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Extending the Operation Range of Dry Screw Compressors by Cooling Their Rotors

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ABSTRACT

Dry screw compressor are limited to maximum pressure ratios of approximately 3:1 due to the high gas temperatures resulting from the compression process, which cause them to deform To attain higher pressure ratios the compressors must be cooled. The main features of heat transfer in screw compressors are reviewed qualitatively and resulting from this, an analytical and experimental study was carried out to validate a novel but simple means of rotor cooling, involving local injection of very small amounts of a volatile liquid. It is shown that this enables oil free compressors to operate up to much higher gas temperatures and hence, extends the pressure ratio over which a single oil free screw compressor stage can operate.

1. INTRODUCTION

Screw compressors are simple volumetric machines, which consist essentially of a pair of meshing helical lobed rotors, contained in a casing. Together, these form a series of working chambers. Admission of the gas to be compressed occurs through the low-pressure suction port which is formed by opening the casing, shown in the top left of Fig 1. Exposure of the space between the rotor lobes to the suction port, as their front ends pass across it, allows the gas to fill the passages formed between them and the casing. Further rotation then leads to cut off of the port and progressive reduction in the trapped volume in each passage, until the bottom ends of the passages between the rotors are exposed to the high-pressure discharge port. The gas then flows out through this at approximately constant pressure.

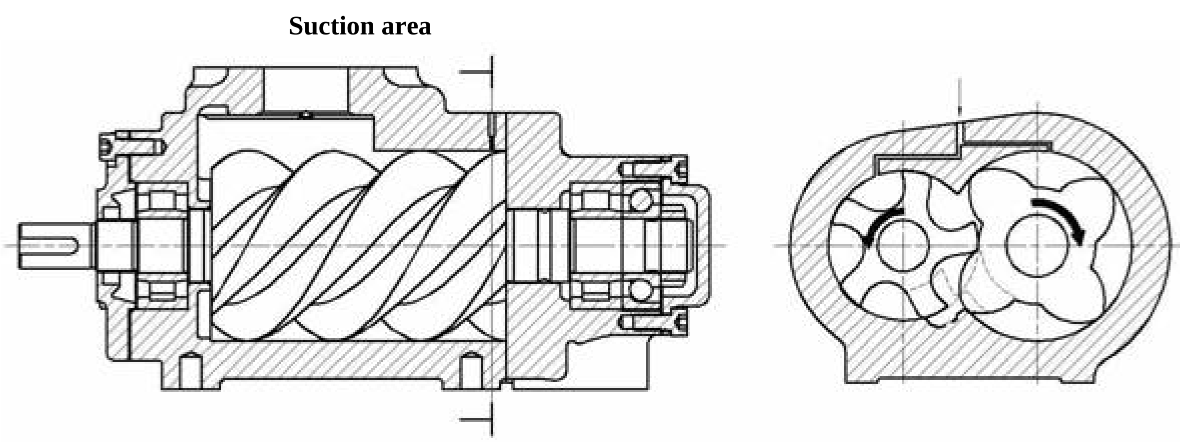

Figure 1: Screw Compressor Main Components

Machines of this type are normally classified into two main types. These are oil injected or oil free. The bulk of screw compressors manufactured are of the oil injected type. In these, a relatively large mass, but small volume, of oil is admitted into the compressor after admission of the air or gas is complete and remains in contact with the gas being compressed, as a dispersed liquid, which is then discharged with the gas. After leaving the compressor, the oil is separated from the gas, cooled and then reinjected. The driving force for reinjection is the pressure difference between the discharged gas and that trapped in the compressor immediately after suction is complete. The oil serves three purposes; namely as a lubricant between the meshing rotors, as a sealant of the clearances between the rotors and between the rotors and the casing and as a coolant of the gas being compressed. Because of this latter effect, gases can be compressed to pressure ratios of up to about 15:1 in one screw compressor stage without an excessive temperature rise.

In contrast to this, dry screw compressors operate without any liquid injection and the temperature of the compressed gas rises rapidly as the discharge pressure is increased causing the rotors to expand. Since the flow of gas through the compressor is not symmetric their rotors and housing tend to deform. Thus, they cannot operate with a high pressure ratio without cooling. Hitherto, simple cooling methods have not been available and current practice is to limit their compression ratio for air to approximately 3:1. The aim of this investigation was to identify the principles of heat transfer in screw compressors to determine whether a simple means of cooling could be devised to raise this to a higher value.

2. HEAT TRANSFER IN SCREW COMPRESSOR ROTORS

As a preliminary consideration, it is important to appreciate that the rate of heat transfer from the compressed gas to the compressor parts is small and, in energy terms, comprises less than one percent of the power input to the machine. *Brok et al, 1980* claim that the heat transfer between gas and compressor elements is negligible. *Recktenwald et al, 1986* disagree, but they still accept that the significance of this heat transfer in the compression process is small. Nevertheless, the build up of temperature within the compressor parts during a large number of compression cycles can be substantial.

Next, it must be understood that the gas temperature is high only in the high pressure region of the screw compressor. This occurs in quite a limited portion of the rotor length, approaching the high pressure port, in the region shown in Fig 1. Hence the high temperature domain where heat is transferred from the gas to the rotors is limited to only a small portion of the rotors and housing.

Since the rotors revolve at high speeds while residing only briefly within areas of different gas temperatures during one cycle, the rotor temperature becomes virtually uniform across any cross sectional area perpendicular to the rotor axis and its value is somewhere between the highest and lowest gas temperature at that cross section area. Accordingly, due to the relatively high thermal conductivity of the compressor components, which are made of metal, the main means of heat transfer within the machine is by conduction along the rotors from the high pressure to the low pressure ends. The rotor body temperature attained is thus a result of a balance between the heat received from the gas at high temperature and its rejection to the gas in regions of lower gas temperature. Therefore, rotor cooling is required only in the compressor high temperature region to keep the rotor temperatures at an acceptable level.

When these considerations are taken into account, an effective means of cooling the rotors could be to inject a small quantity of flashing liquid into the casing, at the high pressure port end in any circumferential position. Preferably, the injection point should be on the opposite side of the casing to the port, as shown in Fig 1. By this means, the liquid can be injected without the need for a pump, in the same manner as oil is injected in oil flooded compressors. The liquid, which is preferably water, would then impinge on the rotors at such a rate that it would instantly evaporate on coming into contact with them.

Some of these characteristics and the effects of such a cooling procedure can be observed in Figures 2 and 3, which give the results of an analysis of the compression process in a dry screw machine based on a mathematical model of the heat and fluid flow processes within it. These show how the compression chamber volume, heat transfer area, pressure and gas and rotor metal temperature vary with the shaft rotational angle. The heat transfer between the gas and the rotors is calculated by use of Anand's simple expression, which though recently criticized, is widely used for heat transfer between gas and metal parts in volumetric machines, $Nu=0.023Re^{0.8}$.

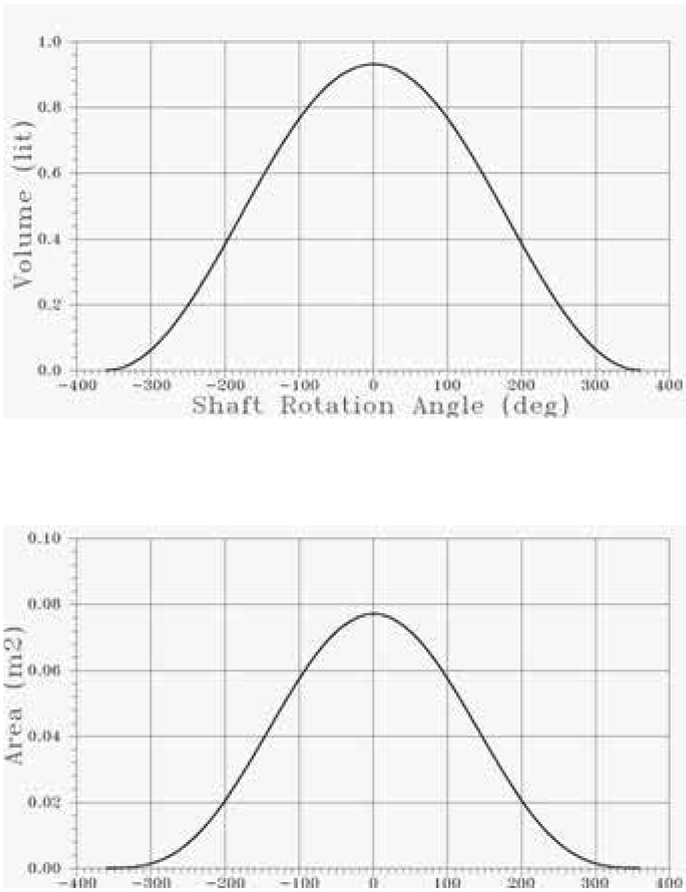

Figure 2: Screw compressor volume and rotor heat transfer area

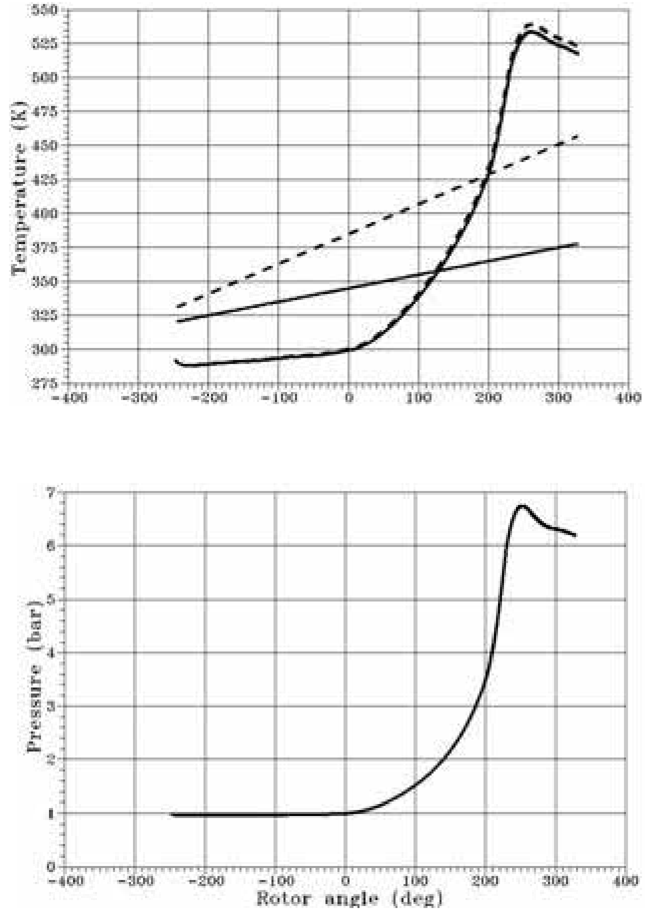

Figure 3: Pressure and temperature

The results shown correspond to a compressor with a male rotor of 102 mm and a length/diameter ratio of 1.55 rotating at 10,000 rpm, compressing air from 1 to 6 bar. In this case the power input was estimated at 47 kW, but the heat transfer rate to the rotors at the high temperature region was only about 100 W. The rotor cooling rate was estimated by a trial and error procedure, after a guess linear rotor temperature distribution was corrected to balance the heat transferred to the rotors at high temperature regions and from the rotors at low temperature regions. As can be seen, the gas temperature, which is shown by a full line in the case of the cooled rotor and by a dotted line when the rotor is not cooled, attains a peak value of 260^oC. The rotor temperature distribution is shown by two lines, the higher one, when there is no rotor cooling, has a maximum value of 180^oC and the lower one, for the cooled rotor has a maximum of only 110^oC. In this case, about 1/3 of the heat transferred to the rotors is removed by the water jet impinging on the high temperature area of the rotor. For the calculated conditions the water flow was 0.0132 g/s, while the air flow was 0.184 kg/s, giving the mass ratio of water injected against the air flow of 72 ppm. This is well within the region of humid but not wet air. This implies that the injected water evaporated completely, using its latent heat to cool the rotors. Under these circumstances, the liquid would be recovered from the compressed gas after cooling in the compressor aftercooler, which is a common element of dry compressor plant. It should be noted that the water injection rate is so small that the rotor cooling hardly affected the gas discharge temperature.

3. EXPERIMENTAL INVESTIGATION

A compressor test rig at City University Compressor Centre Laboratory, which meets Pneurop/Cagi requirements for screw compressor acceptance tests, was used for this investigation. An ordinary dry screw compressor, coupled to its drive shaft through a gearbox, was modified to allow water cooling by an impinging jet, with injection in the moderate pressure region at the discharge end of the compressor rotors, as shown in Fig 1. By this means, the need for a water pump was avoided but vapour compression of the evaporated liquid was less than if the injection had been performed in the minimum pressure region. The compressor was driven by an electric motor of 100 kW maximum output, which may operate at variable speed, controlled by means of a frequency converter. This permits the testing of oil-free screw compressors with discharge rates of up to 16 m^3/min.

The compressor rotors and the water injection hoses are shown in Figs 5 and 6.

Approximately 200 sets of measurements were recorded, with the discharge pressure varied between 3 and 6 bar absolute. The flow of water injected was kept low to ensure full evaporation.

Measured values were used to calculate compressor flow, power and specific power and water injection rate.

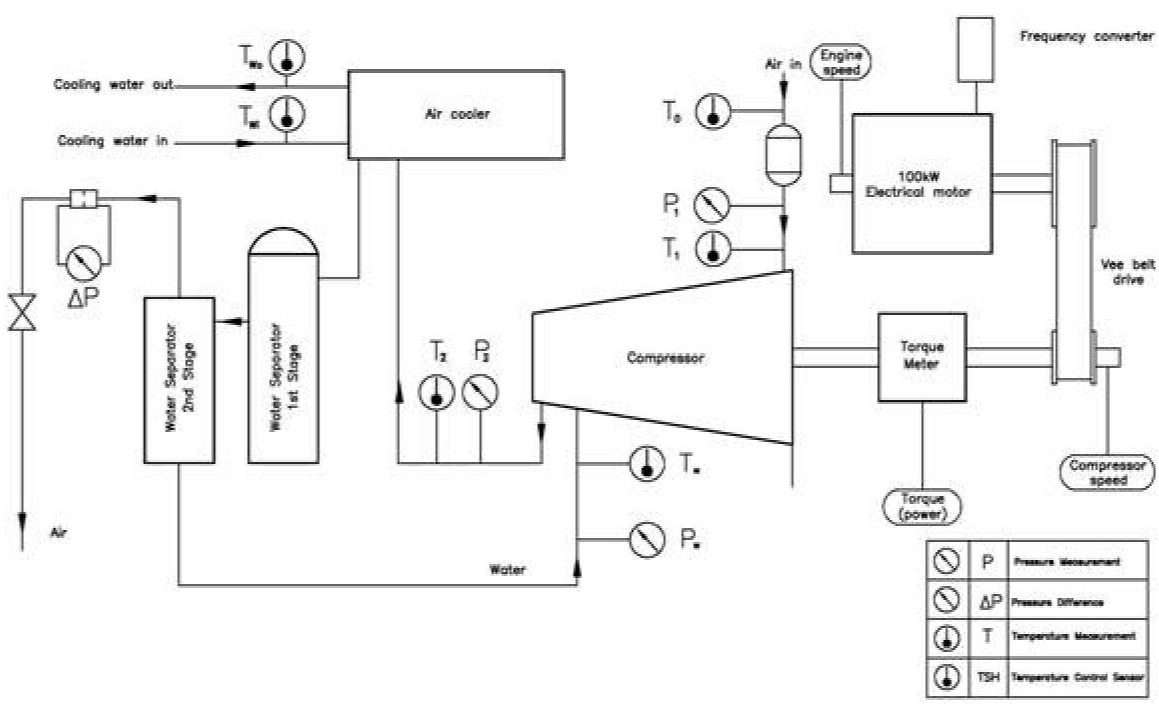

Figure 4: Test rig for measurement of the water injected screw compressor

The following is a typical result for a pressure ratio 5.82, when the compressor was running continuously at an average speed of 10016 rpm. In this case, the measured power input was 47kW and the measured air discharge temperature was 247^{o}C. Such a pressure ratio and discharge temperature were far beyond the design limits of the machine and would have caused either rotor seizure or severe damage if the rotors had not been cooled. Moreover, the compressor worked well at the higher pressure ratios. The design and construction of this machine was such that it was considered unsafe to run it at pressures above 6 bar but the results of these tests were so favourable that further tests are planned to be carried out at even higher pressure ratios on an improved version of it.

Figure 6: Compressor rotors, housing removed

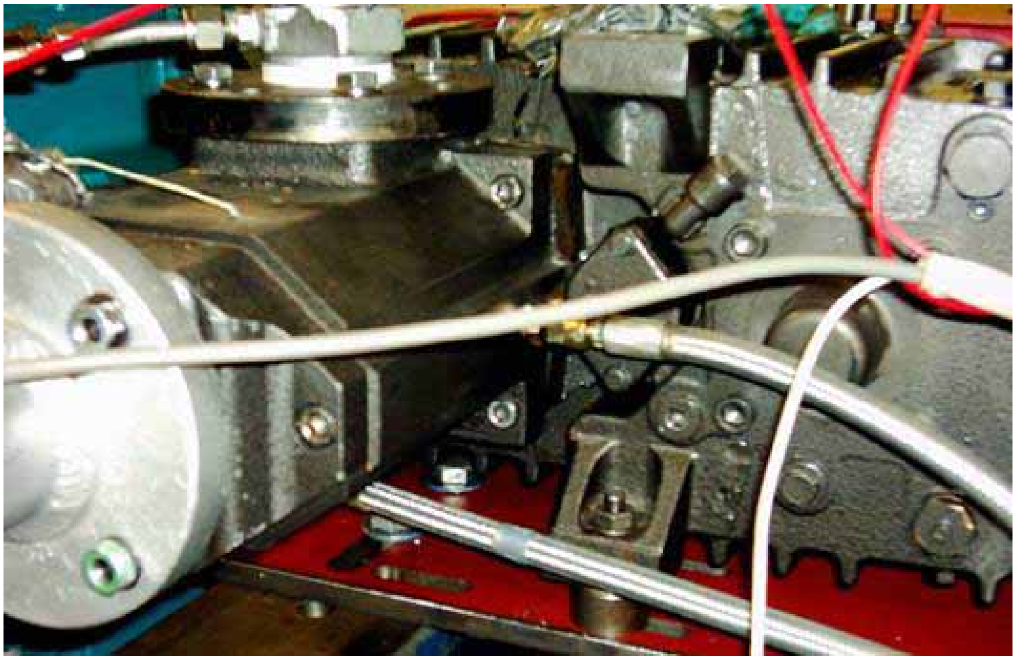

Figure 6: Test compressor with the water injection hoses

4. CONCLUSIONS

The result of this investigation is that it has been shown that only minor modification is required for oil free compressors to be made to operate with much higher pressure ratios in a single stage.. The cost of their manufacture for high pressure applications would thereby be reduced and their efficiencies also increased. Additionally, it would be possible to use the same principle to enable twin screw machines to expand gases with much higher entry temperatures than is presently thought to be possible.

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