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The Challenge of Using the Single-Screw Compressor with Steam

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THE CHALLENGE OF USING A SINGLE-SCREW COMPRESSOR WITH STEAM

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ABSTRACT

Tests are reported on a small (110 mm diameter main rotor) single- screw air compressor adapted for use with steam. A novel gaterotor design was employed with sliding pads to take up tooth flank clearances. There was also water atomisation into the suction. The test rig allowed preliminary runs to be made on nitrogen at 2990 and 5600 RPM. The best volumetric efficiency was only 57% and this was due to excessive clearances in certain locations. Runs on steam gave maximum volumetric efficiences of 30%. The additional losses were due to heat transfer effects, both steady-state and transient. Based on the understanding gained detailed recommendations are given on ways to achieve high performance in steam compression with the single-screw machine.

INTRODUCTION

For a range of process applications of heat pumps there is need for a compressor which will compress steam efficiently and reliably. Centrifugal compressors have been widely used, and provide a fully satisfactory solution, when the duty is large enough and the compression ratio is relatively low. The unsatisfied need is in the power range from 50 to 1000kW with a compression ratio of three or more.

A number of studies have been published using screw compressors for this duty including one by Degueurce et al (1) at Electricite de France with a single-screw compressor. The tests covered more than 1000 hours of running at 3000 rpm with water injection, the suction pressure being 1.36 bar and the compression ratio between 2.7 and 3.2. The volumetric efficiency was found to vary between 67% and 54% and the COP of the heat pump cycle between 6 and 4.5 with increasing pressure ratio (corresponding to isentropic compressor efficiencies of about 50%). The power consumed was about 20kW. The compressor used was designed for duty with air, and it was fitted with seals so that the bearings could still be oil lubricated. Hot water injection provided sealing between the rotors.

Other tests have been reported (2, 3) using larger twin-screw compressors consuming about 100 kW and yielding volumetric efficiencies of around 80% and isentropic efficiencies of 60 to 70%. These machines use external gearing to keep the rotors in phase. Whilst therefore they offer a solution to steam compression the view of the Authors is that the single-screw machine has the potential to be both more efficient and cheaper.

WATER INJECTION

There is an interesting contrast between liquid injection in refrigeration and heat pump duties. In both cases the greater part of the injected liquid will leave with the discharge but whereas in refrigeration this liquid is directly useful, and goes with the condensate to the evaporator, in the heat pump it is useless. Moreoever since the injected water is normally supplied at a temperature intermediate between suction and discharge it draws heat from the compressor in moving to the discharge port. If this heat is taken from the steam superheat the effect will not be deleterious, but if it results in condensation of some steam then the effect is bad. Unfortunately the latter is normally the case.

A good method of injecting water is as an atomised dispersion into the suction. In many heat pump applications, such as in drying, the entry steam will be somewhat superheated and by injecting atomised boiling water into it there will be a partial exchange of superheat for additional steam. The residual water particles in suspension will be progressively evaporated by absorbing compression superheat, thus further increasing the mass of product steam and reducing the work of compression. The water carry-over should be as small as possible.

THE TEST COMPRESSOR

The compressor available to the Authors for this study was an air compressor of 110 mm main rotor diameter made by Mitsui Seiki of Japan. The built in volumetric ratio was 4.2, coresponding to a pressure ratio with steam of 6.5, which was much higher than the desired value of about 3.

Other shortcomings of the machine for steam compression became apparent as the research proceeded; these were:

(i) The method of internally manifolding the two discharge ports into a single discharge channel involved considerable areas of metal which had suction gas on one side and discharge gas on the other. Whilst this did not matter for air compression it was highly deleterious in steam compression. The rapid condensation of steam on the discharge side exchanging heat with evaporating water on the suction side produced an effect equivalent to direct steam leakage.

(ii) Also associated with the discharge manifold were internal leakage paths which communicated directly between discharge and suction. In oil flooded air compression the resulting leakage effect was acceptable but in steam compression it was not.

(iii) A form of labyrinth was incorporated in the cylindrical surface of the main rotor at the discharge end comprising 15 V-shaped grooves approximately 1mm wide and 0.5 mm deep. It was intended, no doubt, that these grooves would reduce leakage but the reverse was the case. This was because only a small fraction of the perimeter at the discharge end of the rotor was pressurised at any one time, but as soon as leakage entered the first groove it was transmitted around the whole of the perimeter.

MODIFICATIONS TO THE COMPRESSOR

The following modifications were made to enable the compressor to operate with steam:

(i) The standard lip seal around the drive shaft was replaced by a gland seal. This change enabled the suction pressure to be above or below atmospheric.

(ii) All the bearings were changed.

The intention had been to use water lubricated ball races. Since the bearing loads in the single-screw compressor are light, and the complete avoidence of oil brings many simplifications, this solution was preferred. In the reported tests, however, there was insufficient time to procure the special races with stainless steel balls and plastic cages. An attempt to use standard ball races with water lubrication, coupled with nitrogen purging of the system to prevent rusting, proved unsuccessful. The bearings behaved satisfactorily whilst the system was clean but oxidation occurred despite the nitrogen blanketting and rust particles damaged the bearings.

As a temporary expedient, sealed-for-life bearings packed with a lithium-based, high-temperature, hydrophobic grease (Shell Alvania) were purchased and installed.

(iii) Minor modifications were made within the machine to reduce direct leakage paths from dischage to suction.

LEAKAGE PREDICTION

The leakage model for the single-screw compressor developed by Chan (4) was modified to incorporate the thermodynamic properties of steam and the geometry of the MS110 machine. When the programme was run for a steam suction pressure of 1 bar (saturated), a compression ratio of 3.61 at 2900 RPM and with uniform clearances of 0.04 mm the pressure and work profiles, given in figs.1 and 2 were obtained. The pressure profile demonstrates the effect of over-compression, which infact would have been more severe if it were not for high leakage. Uncovering of the discharge port commences at 101°. The effect of over-compression on the work requirement is graphically presented in fig.2 and makes the torque curve very 'peaky'.

For the same conditions fig.3. presents the calculated volumetric and isentropic efficiencies for a range of clearance values. The inefficiences arise only from leakage effects, no allowance being made for heat transfer effects, churning and other friction losses. Taking 0.04 mm as the clearance which could readily be achieved with a machine of this size it is apparent that the volumetric and isentropic efficiencies in this idealized situation will at best be about 50%. These figures are consistent with the performance figures obtained by Degueurce et al with a somewhat larger machine.

Hence there is a need to achieve much smaller clearances or to run at higher speeds; or both. Fig. 4. shows the calculated performance improvement obtained by increased speed. If the target isentropic ficiency is 80% it is clear that it could not be achieved by speed alone, especially when other losses are included. This conclusion becomes even more apparent when the effect of liquid water leakage rather than steam leakage is considered.

NEW GATEROTOR DESIGN

With uniform clearances approximately 40% of the total leakage occurs around the perimeter of the teeth. Moreover tight tooth clearances are very difficult to achieve due to differential contraction or expansion. If the compressor casing, main rotor and gaterotor support are made of cast iron, and the gaterotor is of plastic, there is a five-, to ten-fold differential expansion effect between the two materials. For steam compression this means that the plastic star must be machined to have large clearances at room temperature assembly

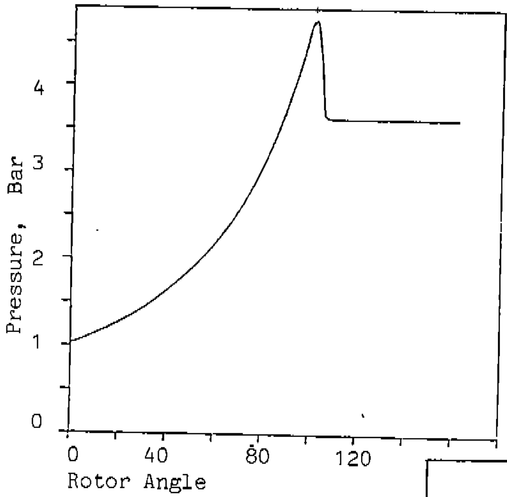

Fig.1. Calculated pressure profile with steam compression.

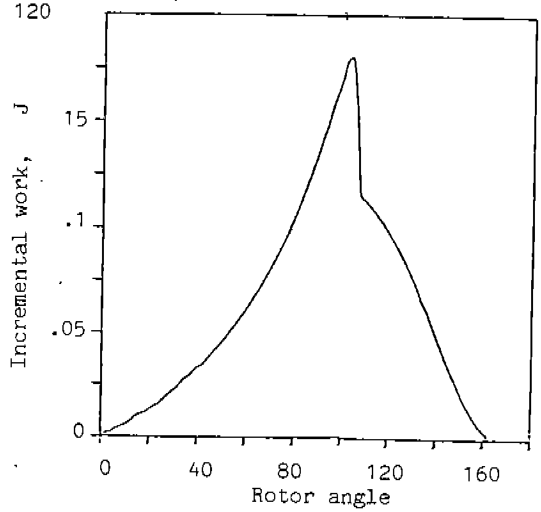

Fig.2 Work profile for steam compression

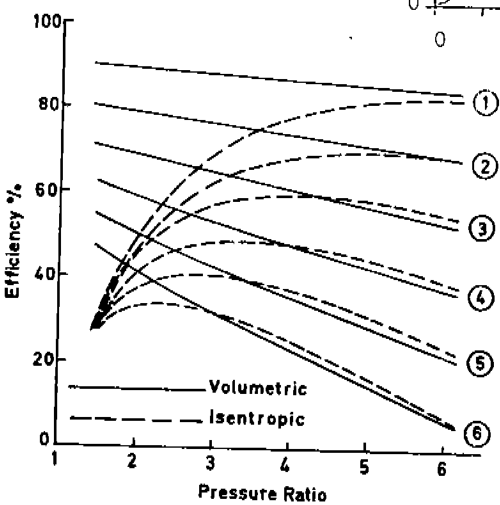

Fig.3 Calculated efficiencies for different clearance values in hundredths of a millimeter.

so that when the machine reaches operating temperature the plastic star grows to the desired size. Not only is this difficult to calculate, because it assumes a knowledge of the average temperature of each of the components, but it also assumes that they heat up at the same rate and then operate at constant temperature.

A novel design using separate sprung tooth inserts (5) has been developed by the Authors, see Fig. 5. Each insert comprises a fixed pad connected through a spring to a smaller sliding pad, the total insert being precision moulded. The fixed pad is attached to the cast iron supporting spider by pressing the integrally moulded pegs into interference holes in the spider. The insert is held by the outer, larger, peg and aligned by the inner one. The tapered portion of the fixed pad provides a rigid fulcrum for the spring. The sliding pad is partly overlapped by the fixed pad to stop it lifting, and the spring prevents it moving radially. Thus the sliding pad can move circumferentially to take up the width of the flute, though the extent of this movement is restricted by a small peg moving in an oversized hole. The peg is moulded into the under side of the overlapping portion of the fixed pad, and the hole is in the sliding part of the moving pad. Without this limitation of movement the sliding pad could be chopped by the main rotor on engagement. The inserts were made slightly over-size so that during the first few minutes of running they would be ground to the exact width and depth of the flute, and thereafter any wear would be taken up by movement of the sliding pad. Radial movement would be expected to be negligable because the pad is supported close to the perimeter of the tooth.

For the sliding pads to have a sealing function they must not only exert a positive force against the flank walls of the main rotor at all times but the reaction force created on the leading edge of the fixed pad must be positive. During most of the compression stroke, and all of the discharge stroke, the length of the trailing edge in engagement with the main rotor is greater than that of the leading edge. Hence the decision to make the trailing edge the sliding one. Moreover the sealing force was made to adjust itself automatically to the change of pressure in the flute through the mechanism illustrated in Fig.6. The slit above the sliding surfaces is always at flute pressure. If the depth of this slit A, is significantly greater than the depth B of the line seal then the total force due to hydrostatic pressure in the slit will be greater than that due to the pressure of the escaping gas in the gaps at the flute walls. This net pressure force will add to the spring force, which is required at engagement. Incidentally the centre of gravity of the sliding pad was designed to lie on the centre line through the spring and the centre of the gaterotor, thus eliminating any centrifugal effect.

The inserts were precision moulded in carbon fibre filled

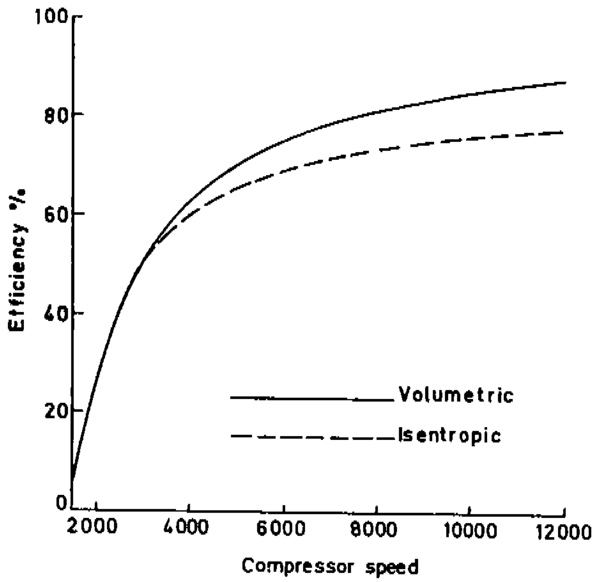

Fig.4. Calculated effect of compressor speed in RPM on performance with steam, clearances of 0.04 mm.

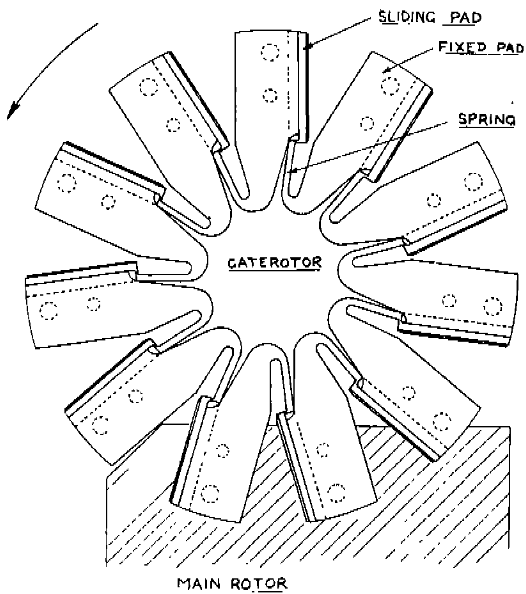

Fig.5. New design of gaterotor using separate tooth inserts.

polyetheretherketone (manufactured by ICI, trade name Victrex PEEK 450 CA 30). This material retains a tensile strength of 50 Mpa at 250 C, is chemically inert, fatigue resistant and has good wear resistance. It is not adversely affected by boiling water.

TEST RIG

Whilst the test rig was designed primarily for steam it was adapted so that it could also be used with moist nitrogen. It was appreciated that nitrogen runs would assist the interpretation of the steam results.

The steam leaving the compressor passed through a cyclone to separate discharged water. After passing through a flow straightener and a standard orifice flow meter it was divided, part going to a condenser, and the remainder to a constant pressure expansion valve. The expanded superheated steam could then return to the compressor suction or its superheat could be reduced to the desired extent by passing a fraction through a packed column down which boiling water flowed. The work of compression was removed in the condenser, and the fraction of steam passed to the condenser effectively controlled the suction pressure of the compressor for a controlled discharge pressure.

A boiling water atomiser was attached directly to the compressor suction. It comprised a stainless steel tube 100mm diameter and about 450mm long along the centre line of which were mounted six water sprays. These were specially constructed swirl-spray nozzles with an orifice size of 0.5mm designed to produce droplets of diameter 20-50 μ, and each having a throughput of about 0.1ℓ/m when fed with boiling water at about 6 bar. It had previously been calculated that with droplets of this size a high proportion would persist in suspension as the steam traversed the angular paths leading into the flutes of the main rotor.

In addition to the atomiser there was provision for direct injection of boiling water through a pair of holes in the compressor casing, these being uncovered very early in the compression process.

When runs were made on nitrogen, the expanded nitrogen was cooled by counterflow with water in the packed column, this water being cooled by diverting it through the shell of the condenser.

RESULTS

The first tests made on the compressor, with the new gaterotors, were with nitrogen at suction pressures of approximately 1 and 2 bar at room temperature, different compression ratios and water injection rates. During these

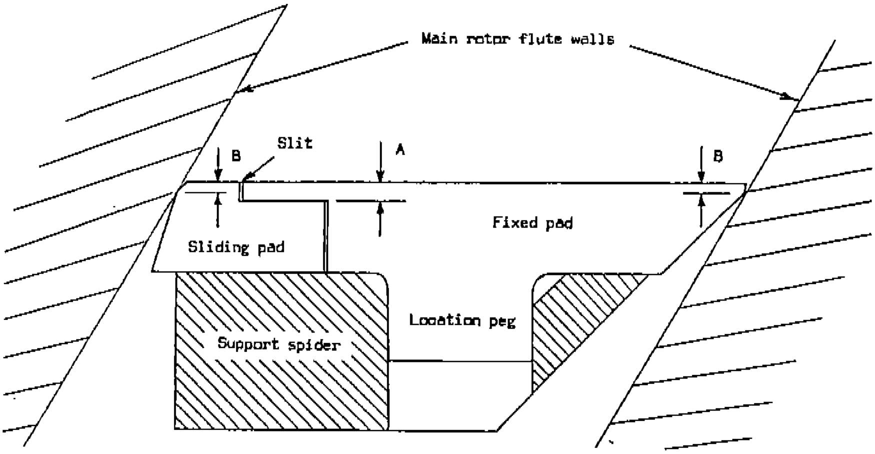

Fig.6. Section through gaterotor inserts illustrating the pressure force used for sealing

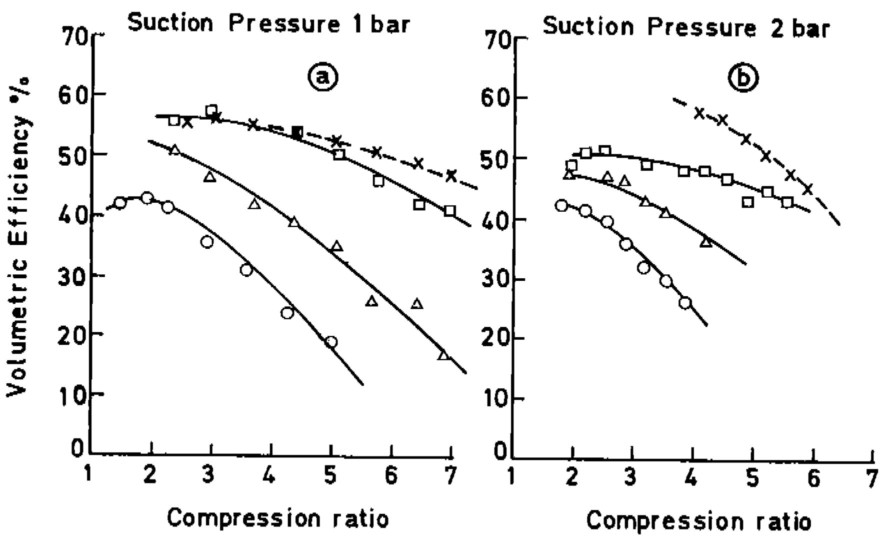

Fig.7. Measured volumetric efficiencies with nitrogen compression The top curve in each case is for 5660 RPM, the other curves are at 2990 RPM.

experiments the greater part of the water was injected through the compressor casing. Most measurements were made at 2990 rpm but a few experiments were made with the highest water injection rate at 5600 rpm. The resulting volumetric efficiences are given in figures 7a and 7b. It had been hoped to obtain meaningful isentropic efficiencies by measuring the motor power and applying correction factors for motor and transmission losses, but these were so large that the nett power was uncertain.

The results were disappointing. Refering to the results at 2990 rpm, the lower suction pressure, and a water injection rate of 1.9 ℓ/min, the volumetric efficiency is seen to be only 20% at a compression ratio of 5. At lower pressure ratios the efficiency does not rise as much as it should due to the effect of over-compression. The compressor was run briefly at a compression ratio of 2.5 without water injection and the volumetric efficiency dropped to 17%. These findings indicated that some at least of the clearances in the machine were very much larger than the design value of 0.04 mm, especially as the gaterotor tooth losses should have been small.

This explanation is supported by the fact that increasing the water injection rate had a very beneficial effect. For a water injection rate of 7.6 ℓ/min and a compression ratio of 4 the volumetric efficiency was about 60%. On the other hand it is surprising that the efficiency achieved at the higher compressor speed was not substantially better. The explanation may lie in the fact that at the higher speed, with a constant water injection rate, the water/gas ratio was halved. It may, therefore, be more appropriate to compare the high speed run with the 2990 rpm run with a water injection rate of 3.8 ℓ/min, in which case the effect of the speed increase is very considerable.

The results for a suction pressure of approximately 2.0 bar show similar trends though the maximum efficiency achieved with the 7.6 ℓ/m injection rate were rather less and the advantage gained by speed increase was rather more.

When the first tests were made on steam at a compressor speed of 2990 rpm no product steam flow was registered. A water injection rate of about 7.5 ℓ/min was being used and its temperature was only about 80 C due to heat losses in the pump. It was apparent that all the shaft work together with the latent heat of the entering steam was going into water heating.

The position of the injection holes was such that injection started after only 5° rotation and continued to 75°. Hence it was not possible to inject water having a temperature higher than about 100°C, otherwise it would have flashed on injection down to this value. The best that could be done therefore was to add a heater to bring the injected water to this temperature

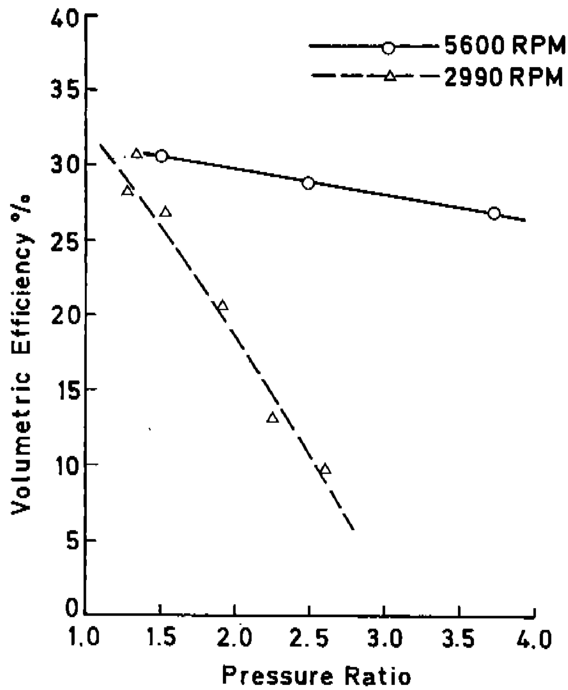

Fig.8. Measured volumetric efficiencies for steam compression

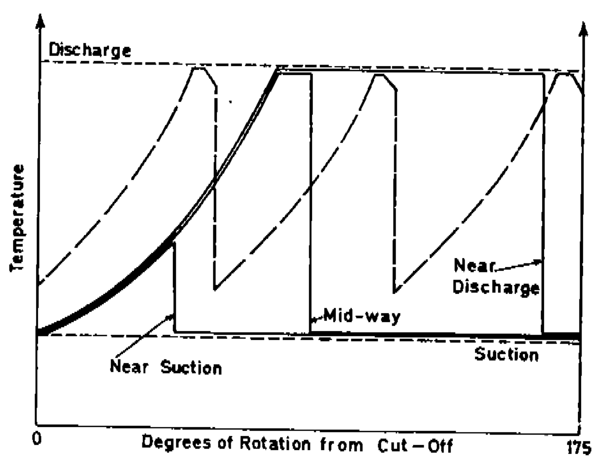

Fig.9. Temperature profiles experienced by points on the main rotor surface (full lines), and at an intermediate point on the casing (hatched line).

and to use as little water as possible. Fig. 8 gives the volumetric efficiency values for steam with an injection rate of 2.7 ℓ/min. A significant advantage was achieved by running at high speed but even so the volumetric efficiency was only about 30%.

These results will now be analysed to see whether such poor performance is inevitable or whether better figures should be possible.

DISCUSSION OF RESULTS

(i) Gas or vapour leakage

To explain the 17% volumetric efficiency achieved in the dry run with nitrogen would have required an average clearance of about 0.08 mm. In view of the close fit of the gaterotor teeth other clearances would need to be about 0.1 mm or other undetected leakage paths found. The compressor was stripped down, therefore, and carefully measured.

Two explanations of additional leakage were found:

(a) The clearance between the main rotor and casing was found to be 0.05 mm in the central region but to be 0.10 mm at the discharge end.

(b) The slot in the cylindrical casing through which the gaterotors protrude was found to be unnecessarily long at the discharge end. Thus with the covers of the gaterotor housings removed one could see through the slot not only the flutes in the main rotor but also 3 grooves of the labyrinth. Since the gaterotor housings operate at suction pressure this meant that the labyrinth grooves were providing direct leakage to suction.

Thus the poor performance with nitrogen was accounted for. The top curve in fig.7a, with a water injection rate of 7.6 ℓ/min corresponds to dry compression with clearances of 0.04 mm. Had all the clearances been of this size the injected water would have raised the volumetric efficiency much further. Unfortunately time was not available to modify the compressor to remove the faults.

(ii) Steam and boiling water leakage

It is now appropriate to see how far the inferior results with steam compression could be understood, firstly by allowing for the effects of the additional leakage paths, and then by applying corrections for heat transfer effects.

In previous paper (5) in which compression in the presence of liquid refrigerants was discussed, it was shown that liquid leakage through the same clearances would lead to mass flowrates about seven times higher than for vapour. the increase was due partly to the far higher density but also because the drag exerted by the moving surfaces was far more significant with the liquid phase. On the otherhand not all of this liquid leakage would necessarily be deleterious. If, for instance, the liquid underwent adiabatic expansion only about 15% would evaporate, hence the overall effect would be similar to vapour leakage. However if the leaked liquid found itself in good thermal contact with a metal surface at near discharge temperature then much more evaporation would occur based on heat drawn from the metal.

On this basis liquid leakage through the lip clearance will not be too serious because the boiling water will be largely in contact with the plastic gaterotor inserts, whilst that occurring at the ends of the flutes into the labyrinth grooves will be very serious because most of it will evaporate in contact with a large area of hot metal. The boiling water leaking between the edges of the flutes and the cylindrical casing, and that leaking around the gaterotor teeth will be subject to complex temperature fluctuations and these will now be examined.

(iii) Transient heat transfer effects

Large areas of the main rotor and of the cylindrical casing will be exposed to cyclic temperature variations. Fig.9 shows by full lines the imposed temperatures due to the compression process at three points on the surface of the main rotor. Near to the suction end the surface sees only a small part of the compression process but half way along it sees the full cycle, though spending half its time at suction pressure. Near discharge the surface spends most of its time in contact with discharge steam. In a 3000 rpm machine this cycle will take 0.01s. The dotted line shows the corresponding temperature cycle for a point mid-way along the compressor casing. Here the transients occur with 3 times the frequency, and whilst considerable areas see wide fluctuations none experience the full temperature span from suction to discharge.

To interpret the significance of these transients in terms of equivalent leakage consideration will first be given to the middle area of the main rotor and it will be assumed that metal a millimeter or so below the surface has taken up an intermediate temperature. During the second half of the compression process, and during the short time at discharge conditions, the steam in the flute will have a saturation temperature higher than the metal surface temperature. It will therefore condense progressively, the rate of heat transfer being determined by the thermal conductivity of the resulting condensate film and the temperature and thermal diffusivity of the underlying metal. The surface metal will heat up. The condensate will be subject to centrifugal force but since in about a thousandth of a second it is swept by a gaterotor tooth little movement will be possible.

Depending on the clearance of the tooth some or most of the condensate film will be swept away. That which survives the passage of the tooth will find itself in a steam atmosphere at suction pressure whilst it is in contact with metal at a much higher temperature so, rapid evaporation will ensue.

This transient heat transfer process was modelled for points on the surface of the main rotor, using the following assumptions:

(a) The surface heat transfer coefficient is given by the Nusselt equation and is inversely proportional to the thickness of the water film. The initial film thickness could be varied between zero and 0.025 mm.

(b) Unidimensional conduction was assumed into the metal whose thickness was taken as infinite. the mean starting temperature of the metal was taken as 10 C below the discharge saturation temperature. The conduction equation was solved by finite difference methods. The backward difference solution was found preferrable due to the potential instability introduced by the surface condition.

(c) A pressure profile similar to that given in fig.1 was used to define the saturation temperature applying during the cycle for one flute. The programme also gave the surface area over which condensation would occur.

In this way the programme calculated the total mass of condensate produced in a single cycle for one flute and compared it with the mass of steam initially contained in the flute. Thus it calculated the deterioration of volumetric efficiency arising due to transient condensation.

The model indicated that with a starting water film thickness of 0.025 mm the loss of volumetric efficiency would be 7.8%, for 0.013 mm 14%, and for zero film thickness 61%. The temperature wave penetrates the cast iron to a depth of about 0.6 mm. The stored heat would be available to evaporate the water left on the rotor after passing the gaterotor tooth. Since the time for evaporation is relatively long, and since nucleate boiling is likely, most of the water will evaporate.

Calculations were also made for the case where the gaterotor and casing surfaces were coated with Victrex PEEK plastic to depths of 0.015 and 0.035 mm. For the three wetting conditions given above the loss of volumetric efficiency dropped to 6%, 4% and 3.5% with little difference due to the extra thickness.

(iv) Steady-state heat transfer

In addition to the transient heat transfer effects there are steady state conduction losses due to internal manifolding but also due to the temperature gradients in the main rotor and casing.

Taking the case of steam compression from 1 to 3.61 bar (a change of saturation temperature of 40 C - ignoring superheat) it is estimated that conduction in the manifold would reduce both the volumatic and isentropic efficiency by about 10%. For the main rotor there is an average cross-sectional area of cast iron of 3.85 x 10 m and a mean path length of 0.06m. If the full temperature difference of 40 C applies then the heat flux will be about 130W. A similar estimate for the casing gives a heat flux of 120W, and therefore the total steady-state conduction is 250W. for this duty (compressor running at 2950 rpm) the ideal work requirement is only 1.5kW. This heat conduction would contribute a further 12% loss to the volumetric efficiency but a 17% loss to the isentropic efficiency.

RECOMMENDATION FOR AN IDEAL STEAM COMPRESSOR

Attention will be directed to a single-screw compressor of the size of the MS110 machine because any machines of larger size incorporating the same principles will have much higher efficiencies and be more attractive. The volumetric ratio of 4.5 will also be used and performance will be predicted for an operating pressure ratio of 6.0 since, again, operation with a lower volumetric ratio and pressure ratio will be more favourable.

The steam compression model indicates that if the average clearance could be maintained at 0.02 mm the volumetric and isentropic efficiencies (allowing for dry vapour leakage only) would be 85% and 82% respectively, whilst for average clearances of 0.01 mm they would be 92% and 90% respectively. The lower clearance is believed possible around the edges of the gaterotor teeth if the sprung inserts described in this paper are adopted. To maintain close tolerances between the main rotor and the casing, and having in mind the heat transfer problem, the Authors would favour bonding a very thin layer of an elastomer, such as silicone rubber, to the inside of the casing. In this way an interference fit cou d be achieved, and the passage of very small dirt particles would not necessarily cause damage. Water arising from condensation would lubricate the rubbing contact.

Leakage at the discharge end of the flutes would be tackled by removing the labyrinth but leaving a parallel extension about 1 mm long backed by a lip-seal pressing against the discharge end of the rotor very near to the perimeter.

To reduce heat transfer effects (and to eliminate rusting) the main rotor would be coated with unfilled Victrex PEEK with a depth of about 0.08 mm.

The atomised water injector would be retained at the compressor suction but the other water injection holes would be eliminated. Thus the wetting of the surfaces to prevent wear would be ensured at the suction end by the injected water, and at the discharge end by condensation plus the water ploughed forward by the main rotor.

Each triangular discharge port would have its own discharge flange placed as close as possible. Manifolding would be done externally. The short channels leading to the discharge flanges would be internally insulated, and a thermally insulating washer would be inserted between the flange and the external discharge manifold. Thus the discharging steam would have minimum thermal contact with the compressor casing which would operate at near suction temperature. The main rotor could be bored out to reduce longitudinal conduction.

Since the effects of both the transient and steady state heat transfer losses would be smaller at the higher operating speed it is considered that a machine built along the lines described could have a volumetric efficiency of at least 80% and an isentropic efficiency of 75%. For larger machines values of at least 90% and 80% respectively would not be unreasonable goals.

ACKNOWLEDGEMENTS

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