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INFLUENCE OF THERMAL DILATATION UPON DESIGN OF SCREW MACHINES

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Notation

R	Rotor		Subscripts	
H	Housing		b	Bearing
C	Clearance		c	Compressor
L	Characteristic length		e	Expander
T_m	Operating temperature		h	High pressure side of machine
T_o	Ambient temperature		l	Low pressure side of machine
β	Coefficient of thermal expansion		sp	Separating plate

1. Introduction

Screw machines operate in two modes, either on dry gases or with liquid injected in the working chamber. A cross section of a typical screw compressor, in which the leakage flow paths through the clearances are indicated, is shown in Figure 1. The heat transfer rate between the working fluid and the rotors is relatively low but, especially in dry machines, leads to high rotor temperatures. However, the temperature of the working fluid is high only on a limited portion of the rotor surface, in the region of the high pressure port. Since the gas temperature around the rotor circumference is not constant at any cross section, as the rotors revolve, the local surface contact between them and the hot gases is only intermittent. However, due to the high thermal conductivity of the metal rotors, the rotor temperature remains virtually uniform across any cross sectional area perpendicular to the rotor axis, where it attains some value between the highest and lowest gas temperature at that cross section position. Accordingly, 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 distribution, thus attained, is consequently a result of a balance between the heat it receives from the gas at high temperature and rejected to the gas in regions of lower temperature. The rise in gas temperature, due to compression, thus affects both the rotor and the housing temperatures. This causes thermal distortion, which changes the size of the internal leakage paths and hence affects the machine performance. These effects are shown in Figure 2, in which estimates of how the temperature is distributed were derived by Kovacevic et al, [2002]. In this case, the resultant rotor thermal deformation is magnified 1000 times.

The efficiency of a screw machine is greatly influenced by leakage through its internal clearances. These must, therefore, be minimised for all operating conditions. In oil free machines, the internal leakage rates are greater because the clearances have to be made larger in order to allow for greater thermal expansion of the rotors, due to the lack of cooling. On the other hand, the absence of oil eliminates the high viscous drag on the rotors, associated with oil-flooded machines. Therefore, dry machines can rotate at higher rotor speeds. Consequently, since leakage is virtually independent of the rotational speed, and the gas flow rate increases with speed, the volumetric efficiency of dry compressors thereby becomes acceptable. It follows that design procedures are required to assess the relative significance of the various leakage paths within a screw machine and to allow for thermal distortion, in order to maximise the machine efficiency and reliability.

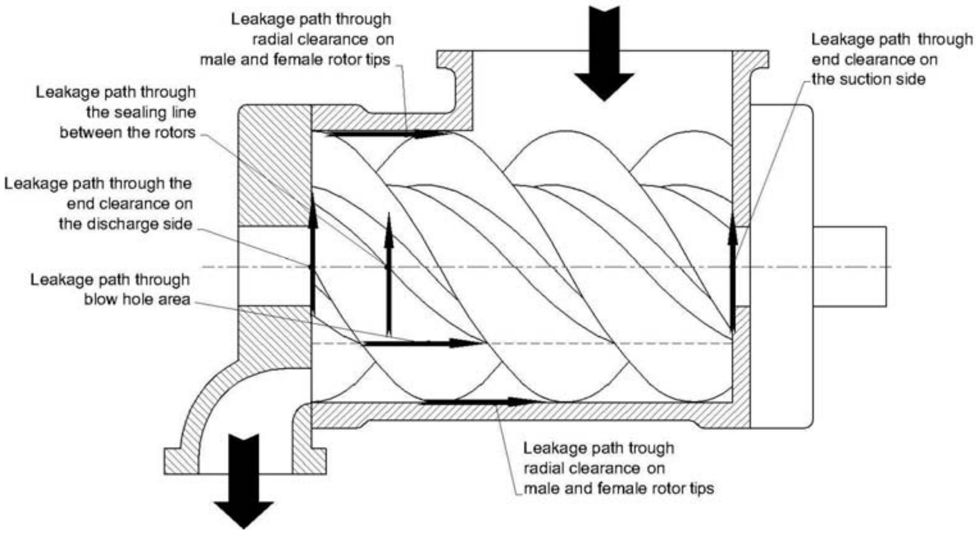

Figure 1. Leakage pathways in a screw compressor

According to [Fleming, 1994], the most significant clearance gap is that between the rotors, as shown in Figure 1. The end clearance gap on the high pressure side is of almost the same importance for machine performance. These form direct leakage paths between the working chambers at the highest and lowest pressures. Leakages through the blow hole, the radial clearances between the rotor tips and the end clearance between the rotors and the housing are driven by smaller pressure differences and are therefore less important. The size of the radial and interlobe clearances is determined from the size and tolerances of the main compressor parts. The axial clearance is, however, set during the machine assembly.

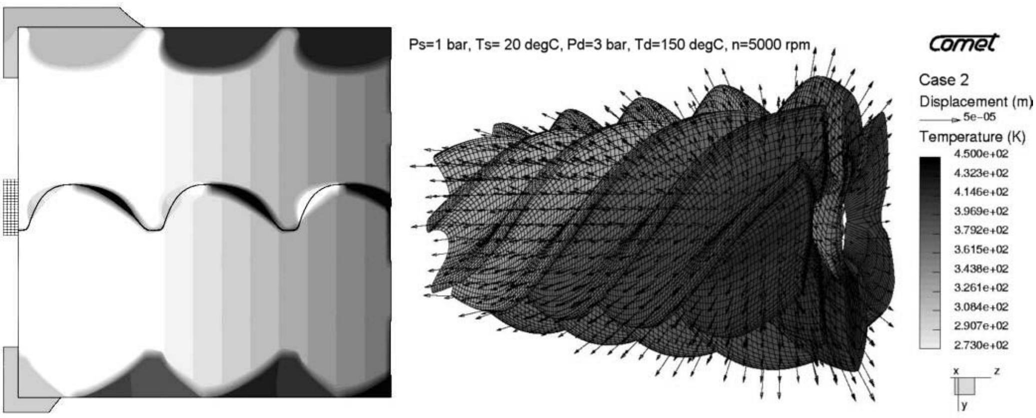

Figure 2. Temperature distribution and deformation for a dry screw compressor

The design aspects of clearance control in such machines have therefore been investigated and an initial clearance setup is suggested by the use of different materials for the various components. This procedure was derived by developing a numerical method to predict the effects of thermal dilatation on the screw machine performance.

2. Types of screw machines considered in this study

Screw machines can be used as compressors, expanders and even in a combined mode, as already reviewed by [Kovacevic et al, 2003] and [Stosic et al, 2003].

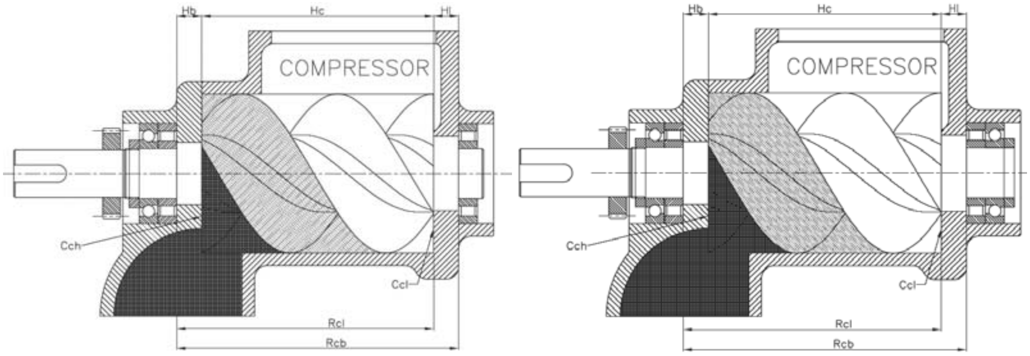

Figure 3. Two different bearing arrangements for dry screw compressor

The first machine analysed in this paper is a single dry air screw compressor. Two different rotor suspension systems, achieved by different bearing arrangements, are shown for this in Figure 3. The most common arrangement is shown on the left, where one radial and one axial bearing are positioned at the discharge end and only one radial bearing is situated at the suction end. This arrangement allows free axial dilatation of the rotors but the fact that the thrust bearing takes an axial force in only one direction can lead to problems. Namely, if at some condition the, axial force generated in the helical synchronising gears becomes greater and is of the opposite sign to that of the axial load force on the rotor, the rotor may come into contact with the housing and the compressor will fail. Therefore, manufacturers sometimes restrain the rotors on the suction side by another thrust bearing. It may be a good solution to make the differential dilatation between the rotors and housing smaller than the internal bearing clearances. Otherwise, the bearings may move in their seats, become damaged and cause incorrect location of the rotors. At first, this causes deterioration in the compressor performance. Subsequently it causes the rotors to seize.

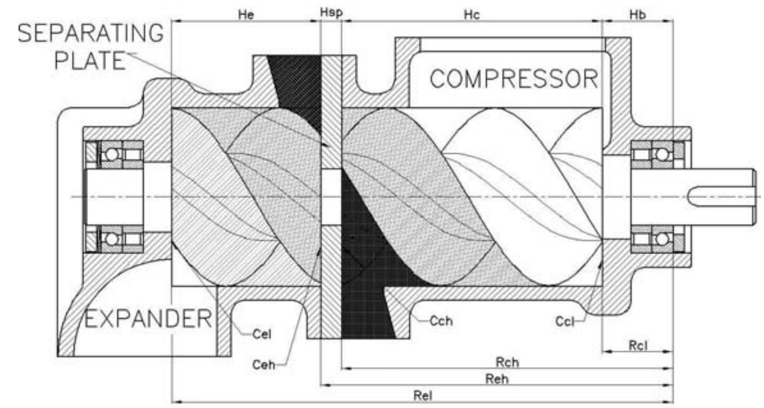

Figure 4. Compressor-expander for fuel cell application

The management of clearances in a combined compressor-expander is rather more complex. Such a device, as shown in Fig 4, has been proposed to supply compressed air to a fuel cell, where recovery of power from the expansion of the steam formed in the cell, together with the non-reacting Nitrogen, induced with the compressed air inflow, is used to increase the overall efficiency of the system. As shown by Kovacevic et al [2003], there are a number of rotor and port configurations possible for such a dual function machine. In this case, there are four ports with a separating plate placed between the compressor and the expander and the high pressure ports are located on either side of the separating plate near the centre of the casing.

This arrangement allows for almost complete balancing of the axial forces and a reduction of almost 20% in the radial forces. The mechanical losses can be reduced while the bearings can be smaller than in an ordinary screw compressor or expander. However, since the axial forces can change direction because of imbalance in the machine, it is necessary to ensure that the axial loads can be sustained in both directions. The rotors are locked in the axial bearing at the compressor suction end. At the other end, axial distortions are compensated by the use of a plate spring. The use of different materials for rotors and housings here leads to substantially different behaviour and functioning of the machine.

3. Model for clearance analysis

The basic dimensions and clearances for the compressor are shown in Figure 3. The same nomenclature is used for the combined machine shown in Figure 4. Using the main dimensions, thus described, the size of the end face clearances in the machine can be obtained by their summation, after taking account of the initially set axial clearances during assembly. Once the geometrical relations are established, it is possible to calculate changes of both the axial and the radial clearances due to thermal expansion of the rotors and the housing in the machine. All these are obtained from the relation $\Delta L = \beta \cdot L \cdot (T_m - T_o)$. Both the compressor and the combined compressor-expander are shown in the test rig in Figure 5.

Figure 5. Left: Compressor; Right: Combined machine

The temperature change of the machine elements is estimated on the basis of the temperatures and pressures at the machine ports, as explained by Kovacevic et al [2002]. The comprehensive mathematical model used to calculate these is described by Stosic et al [2005].

3.1 Dry compressor clearance calculation

When estimating the dry compressor clearances and performance, different combinations of materials were used. The rotors were always assumed to be made of stainless steel. Three materials were considered for the compressor housing, namely stainless steel, CERTAL - aluminium alloy and cast iron. The evaluated compressor has following parameters:

- Number of teeth male/female z_1/z_2 3/5
- Male rotor diameter D_1 68 mm
- Centre line distance between the rotor axes a 48 mm
- Compressor relative length L/D_1 1.1
- Working pressure P 2-3 bar
- Male rotor speed n 6000-15000 rpm

If cast iron is used for the housing and the rotors are made of steel, the increase in the discharge pressure will lead to an increased temperature, which in turn enlarges the discharge end clearance by about 10 μm at 200°C. It will therefore increase the leakage flow through the end clearance gap and the compressor performance will deteriorate. At 12000 rpm and 2.5 bar discharge, with a grey cast iron housing the compressor will deliver 0.71 m^3/min air at 213°C, consuming 2.5 kW. The volumetric efficiency of that compressor would be 28.5%. If an aluminium housing is used, the compressor will have a substantially lower discharge end clearance, as shown in Figure 6, but still the performance will drop due to the growth in the interlobe clearance. The volumetric efficiency will decrease to only 21% with a delivery of 0.53 m^3/min and a power consumption of 2.3 kW.

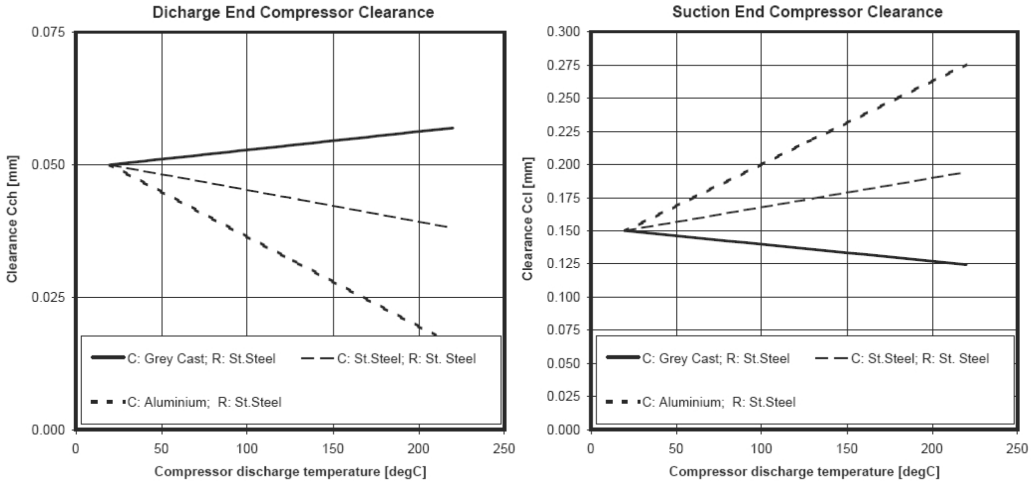

Figure 6. Estimated clearances for the single compressor

With the cast iron housing, the clearance on the suction bearing, as shown in Figure 6, is reduced by as much as 25μm at 200°C gas discharge temperature. If the suction bearing was restrained, the whole bearing seat would then move by 25 μm. If a larger dry compressor is considered, for example, 120 mm rotor diameter and 1.8 *L*/*D*, the same working conditions would move the suction bearing by approximately 100μm. Therefore, if the compressor is started and stopped frequently, the movement in the suction bearing would damage its seat and the rotor axis on the suction side would move. In such a case, the chances of the rotors seizing in the housing are large.

3.2 Clearance estimation in the combined compressor-expander

In the combined compressor – expander, shown schematically in Figure 4 and in the test rig on the right of Figure 5, the management of clearances is also very important. The proper selection of the material makes a difference to the machine performance and also to the alignment of the initial clearances. The compressor part is the same as in the case of the single compressor evaluated in 3.1. The expander has a relative length of 0.8. The separating plate is 15 mm thick. Three builds were examined, as specified in Table 1.

Table 1. Description of compressor-expander builds

Build	Compressor	Separating Plate	Expander	Rotors
1	Stainless Steel	Stainless Steel	Stainless Steel	Stainless Steel
2	Aluminium	Aluminium	Aluminium	Stainless Steel
3	Cast Iron	Aluminium	Aluminium	Stainless Steel

The results of the calculation of the changes in the axial and radial clearances on the compressor high-pressure side are presented in Figure 7. The combination of a grey cast iron compressor housing with an aluminium separating plate and expander housing gives the minimum axial and radial clearance. In this case, both decrease as the discharge temperature increases. Since the discharge temperature is proportional to the discharge pressure this means that for higher discharge pressures all major clearance gaps on the compressor side of the machine reduce and the volumetric efficiency increases. This, in turn, reduces the discharge temperature.

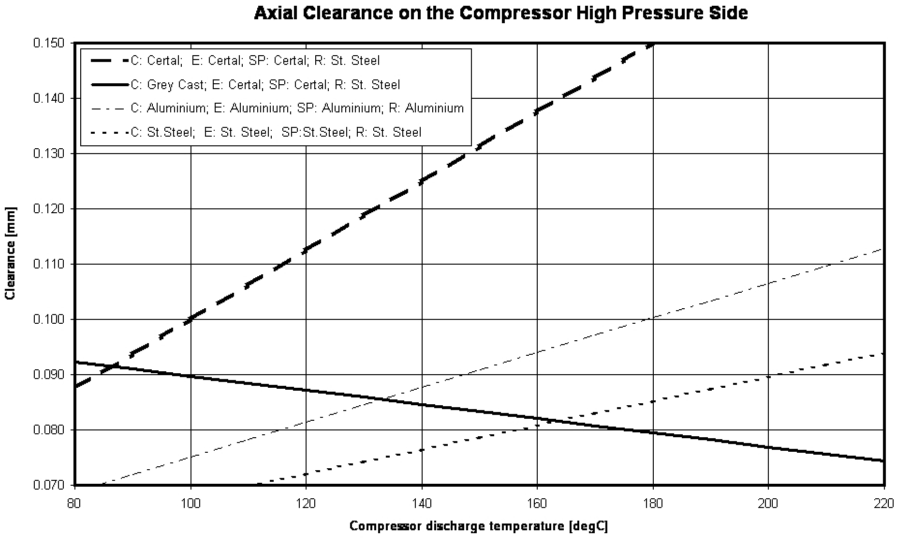

Figure 7. Estimated axial clearance on the compressor discharge in the combined machine

It can be shown that the use of an aluminium compressor housing in build 2, as described in Table 1, will produce the opposite effect. The axial clearances increase by up to 120 μm at 200^{o}C discharge temperature and the machine performance deteriorates. In order to improve the situation slightly, the axial clearance must be adjusted to virtually zero at the assembly of the machine.
More significantly a similar situations happens even if all the machine components are made of the same material, the clearances still increase as the discharge temperature rises, especially so, when aluminium components are used in place of steel, as shown in Figure 7.

4. Mapping the numerical and experimental results

Measurements of the working parameters for the combined compressor – expander machine were conducted for two cases with different speed setups, as shown in Figure 8. The dashed line represents the case with the aluminium compressor housing. It is clear that the relative mass flow decreases with temperature for all measured speeds. This is due to the sharp increase in the clearances as the temperature rises.
Clearances in the grey cast iron compressor housing are slightly reduced when the temperature rises. This reduces the leakage flow and therefore increases the volumetric efficiency at all compressor speeds, thus reducing the discharge temperature.
Since it was impossible to measure the clearances during the working process, the influence of the discharge temperature upon the compressor clearance and performance was deduced from a combination of measurements and calculations. Firstly, the machine performance was predicted by means of the SCORPATH software package described by Stosic et al [2005]. Performance predictions were therefore made for different but invariant sets of axial clearances starting from 40 μm up to 150 μm. These were then matched with the experimental results for cases where the measured and predicted discharge temperatures were equal. The axial clearances were then deduced from this.
There are some limitations to this method, since the estimated machine clearances, which are dependent on flow and temperature distribution, that are essentially three-dimensional, are derived from a one-dimensional simulation model. Firstly, thermal expansion requires knowing the temperature distribution along the rotors and housing. This has been assumed to change uniformly and linearly both through the rotor and its housing. Such an assumption was made, based on the preliminary 3D flow and solid structure calculation within the combined machine carried out by Kovacevic et al [2003]. However, this assumption may not always be accurate, especially during transient processes, such as start-up or shutdown.

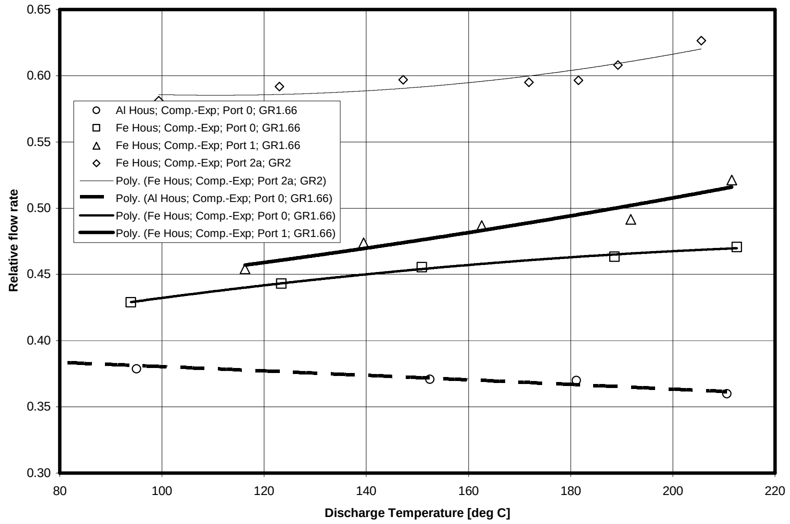

Figure 8. Measured relative flow rates of the combined machine

Additionally, the estimated temperatures of the compressor structural elements take account only of the heat generated and transferred from the working chamber.

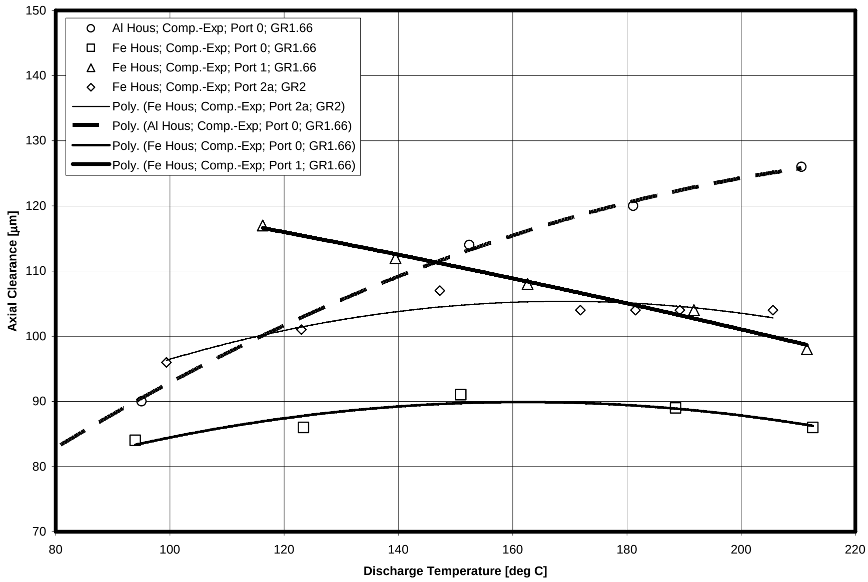

Figure 9. Discharge clearances obtained by measurements

However, the integral motor and compressor bearings generate heat, which is transferred through the rotor shafts and housing. This may also affect the clearances. Therefore, the differences between the predicted and measured discharge temperatures may, under some circumstances, be significant.

Figure 9 shows the results obtained. In the machine in which the compressor housing is made of aluminium. The axial high pressure clearance gap increases with increase in temperature while for the grey cast iron compressor housing that gap stays almost the same. This confirms the estimates made by the simple analytical method, as shown in Figure 7.

5. Conclusions

Problems associated with differential expansion in oil free screw compressors, as well as in a dry combined compressor-expander machine have been investigated. Estimates of clearance changes, based on simple linear expansion effects showed how these could be controlled and accounted for during the design of the machine to optimise its performance while avoiding the machine seizing. It was shown that, by the use of different materials for the machine components, the performance could be improved. The performance predictions, thus obtained, agree well with those obtained from tests on a machine of this type. The use of this simple expansion model as a design tool for combined screw compressor-expander units therefore appears to be valid.

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