The analysis of leakage in a twin screw compressor and its application to performance improvement

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The performance of a helical screw compressor is influenced more by the internal gas leakages than by any other thermo-fluid aspect of its behaviour. Six separate types of leakage path can be identified. Only the cusp blow holes have a constant geometry; every other path has a geometry and resistance to flow which varies (periodically) in a manner unique to it. The pressure difference driving the gas along a leakage path also varies (periodically) and does so in a manner that is not the same for every leakage path. This is quite obviously a complex problem requiring insight in modelling the thermo-fluid behaviour and the solution of a large number of simultaneous equations.

The distribution of leakage through the various leakage paths within the machine is important for the improvement of the compressor performance. A method of determining the aggregate leakage through each path individually over a complete compression cycle is required to enable this study to be conducted. The authors have constructed a mathematical model of the complete compressor thermo-fluid process which is suitable for this purpose, its macropredictions having been verified against measured data derived from a test compressor. The nature of its micropredictions and their verification, that is for each leakage path, are the subject of the paper proposed here. Analytical techniques are proposed and experimental methods are discussed. The influence of different rotational speeds on the leakage is considered. Also discussed is the manner in which the leakage distribution prediction could be used to optimize a compressor design.

Key words: twin screw compressor, gas leakage, thermo-fluid, leakage paths, thermodynamic model

NOTATION

A	flow area (m^2)
c_1	coefficient to modify the flow area
c_2	coefficient to modify the fluid flow speed
i	fluid enthalpy (J/kg)
$L_{sealing}$	sealing line length (m)
m	fluid mass (kg)
p	gas pressure (N/m^2)
s	fluid entropy (J/kg K)
W	fluid flow speed through the flow area (m/s)
δ	gap clearance in the compressor (m)
ρ	fluid density (kg/m^3)
φ	rotation angle of the male rotor (rad)
ω	angular speed of the male rotor (rad/s)

1 INTRODUCTION

The helical screw compressor is a positive displacement compressor, the working cavity of which is enclosed by the housing bores, housing end plates and the helical surfaces of the male and female rotors. As the rotors rotate, the volume of the working cavity varies from zero to its maximum and from its maximum to zero periodically in a manner determined uniquely by the geometry of the compressor. As a consequence of this periodic variation, the compressor completes its suction, compression and discharge processes.

Due to the geometry of the mating parts and the need for clearances between them, the compressor has several leakage paths as follows: across the contact line between the male and female rotors, across the rotor tips and end sealing lines, and through the cusp blow holes and the compression start blow hole. The summation of the leakages through all these paths has a large influence on the performance of the compressor. Leakages are of two kinds. The first is leakage from the enclosed cavity or discharge chamber to the suction chamber or the cavity still connected to the suction chamber, which reduces both the volumetric and indicated efficiencies. The second is the leakage from the enclosed cavity or the discharge chamber to the following enclosed cavity, which reduces the indicated efficiency, but has no direct influence on the volumetric efficiency.

Each leakage path has a different influence on the performance of the compressor. It is very important to know the leakage through each leakage path and the percentage by which it can reduce the efficiencies for the purpose of prioritizing design procedures in general and for improving the rotor lobe profile. For example, reducing the blow hole area and the contact line length may be expected to increase the performance, but very often for a given rotor profile the reduction of the blow hole area results in an increase of the contact line length. If the influence of the contact line length on the efficiencies is (say) much smaller than that of the blow hole area, a new profile may be generated which has a relatively large contact line length, from which a very small blow hole can be obtained.

The calculation of the leakage of a given leakage path at small time increments throughout the working process is complex, because during the operation of a compressor the states of the working medium before and after the leakage path, the leakage area and the length of the path, etc., vary periodically and continuously. In this paper a powerful thermodynamic model for the simulation of the working process of a refrigeration helical screw compressor, which has been verified by comparing its output with many test results, is used to make the quantitative leakage calculations. For many working conditions, including different rotational speeds, the leakage of each leakage path and the percentage by which it can reduce the volumetric efficiency

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and the indicated efficiency are calculated and presented. These results are used to improve the profile and to optimize the geometrical characteristics. The greatest attention should be paid to those leakage paths that have the greatest influence on performance. Some suggestions and the results of optimization are presented in the paper.

2 LEAKAGE PATHS AND THE MATHEMATICAL MODEL

2.1 Compressor geometry and leakage paths

In the working process simulation program all the leakage paths and the leakage rates through those paths are considered. The working cavity of a helical screw compressor is bounded by the housing bores, housing end faces and the helical surfaces of the male and female rotors. The clearances needed to accommodate the relative sliding movement between the rotors and between the rotors and the housing casing and end plates result in many leakage paths through which the working fluid leaks into, or out of, a working cavity.

Considering a given cavity volume, the following are the leakage paths through which the working fluid leaks in or out:

Path 1: the contact line between the male and female rotors. The working fluid leaks to suction pressure from the considered enclosed cavity across the contact line between the male and female rotors. At any rotational angle, the leakage area can be obtained by the contact line length times the average clearance between the rotors. The average clearance between the male and female rotors is calculated by the cutter blade calculation program, another program developed by the authors (**1**). If the wrap angle of the male rotor is large enough the fluid may leak to the considered cavity from a high-pressure cavity at the beginning of the compression process or it may leak to a low-pressure cavity from the considered cavity at the end of the compression process or discharge process due to the overlap of rotor threads.

Path 2: the sealing lines between the rotor tips and the housing bores. The working fluid leaks to the following cavity or suction pressure from the considered cavity across the following sealing lines between the rotor tips of the male and female rotors and the housing bores. It also leaks to the considered cavity from the leading cavity or discharge region across the leading tip-to-bore sealing lines. It should be noted that the clearances between the housing bores and the rotor tips are usually different for the male and female rotors.

Path 3: the cusp blow hole. The working fluid leaks to the following cavity or suction pressure through the following cusp blow hole, which is a small triangular-shaped area formed by the housing cusp on the high-pressure side and the male and female rotor tips. It also leaks to the considered cavity from the leading cavity or discharge pressure through the leading cusp blow hole.

Path 4: the compression start blow hole. The working fluid leaks to suction pressure through the compression start blow hole, which is defined in reference (**2**) by the authors and is a blow hole along the housing cusp on the low-pressure side. It exists at the beginning of compression for about 30–40° of male rotor rotation.

Path 5: the clearance between the end plate and the rotor end face at the suction end. The working fluid leaks to suction pressure across the rotor end face at the suction end during the early stage of the compression process.

Path 6: the clearance between the end plate and the rotor end face at the discharge end. At the discharge end the situation of leakage across the rotor end face is more complicated than at the suction end. The working fluid always leaks to the following cavity from the considered cavity. In addition, during the end of the compression process and during the discharge process it can leak to suction pressure directly, so for part of the process parallel leakage paths exist. This makes the leakage flow particularly difficult to model since the distribution of the leakage flow among the possible paths is not known.

Some of the above leakage paths are shown in Fig. 1. In common arrangements two or three enclosed cavities exist between the suction and discharge ports, which are simultaneously in compression. Not all the leakage paths exist for the whole period of the compression and discharge processes. Further, for the duration of their existence the leakage areas of most paths vary in a complex manner with male rotor angle. As a consequence, taking all the leakage paths into account, as is done in the authors' working process simulation program, a complex mathematical model results. Figure 2 is a schematic diagram of leakage paths.

2.2 Distribution of leakage

In the study of any leakage problem, a sensible way to begin is by examining the flow areas available for leakage. When the areas vary, as they do in the case of the twin screw compressor, the average values may perhaps be chosen. It is instructive to compare the

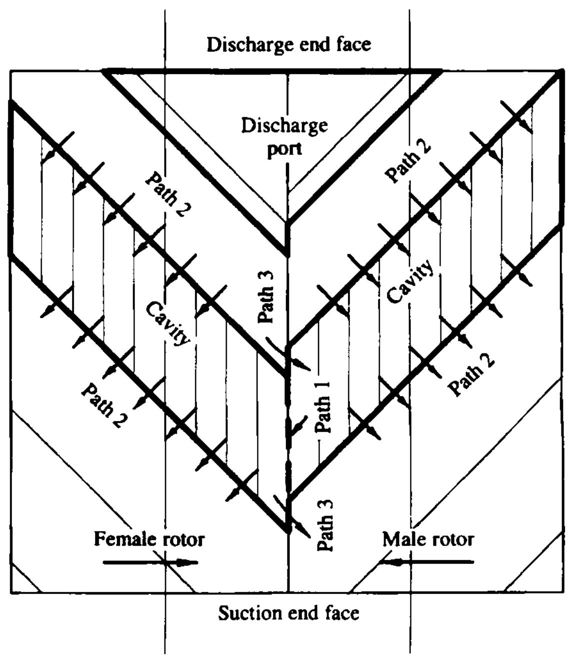

Fig. 1 The leakage paths of an enclosed cavity

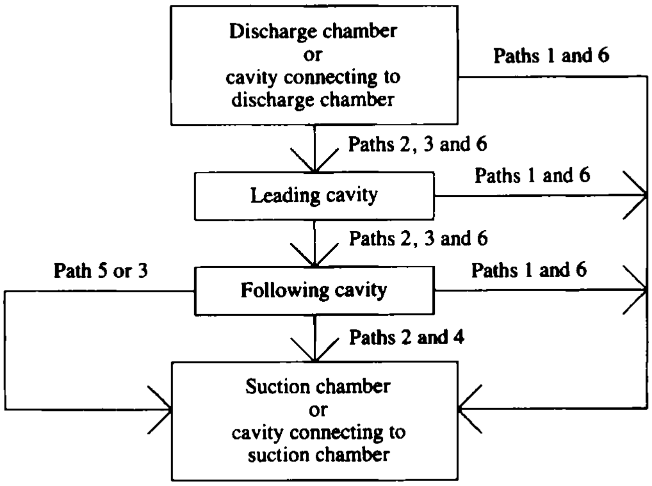

Fig. 2 The schematic diagram of leakage paths

average leakage areas with the net flowrates (that is derived from a complete period of operation) predicted for each path using the authors' comprehensive thermodynamic model of the working process. This may be done by examining Table 1.

If the paths had been represented by six playing cards, rows two and three could be the ranking orders before and after a shuffle. Most striking is path 4 which has by far the biggest flow area and very nearly the lowest leakage. This is explained by the short-lived nature of path 4, the compression start blow hole which, though enormous in (relative) area, exists for only 30–40° of male rotor rotation. Furthermore, it exists only at the beginning of compression when the pressure difference driving the flow is very small. This explains its comparative unimportance. It also illustrates well the need for accurate determinations of the geometric characteristics as functions of rotor rotation if a good thermo-fluid model is to be constructed. Equally clearly, the model must predict the pressure–volume and pressure–rotor angle behaviour accurately.

In science, any theoretical postulation or development, especially if it contains simplifications, requires to be verified or tested in some way. This is usually done by comparing the predictions of 'the theory', the model, with measured values of the quantities concerned. In the work reported here only the predictions of the macro-behaviour of the compressor have been examined in this way: that is total mass flowrate, volumetric efficiency and the *trend* of total efficiency. For a machine similar to that considered here, predicted bearing reactions have also been shown to agree reasonably well with measured values, which suggests that the predicted pressure–volume diagram is also reliable.

2.3 Modelling of the flow in the leakage passages

Every leakage path contains a complicated flow phenomenon. The flow directions, the flow areas and the shapes and lengths of the leakage paths are different for different paths, and they change with the angle of rotation of the male rotor. In reality the fluid leaking through the leakage paths is gaseous refrigerant mixed with droplets of oil and liquid refrigerant (when the injection of liquid refrigerant is on duty). Heat exchange between these components will occur during the leakage process, but its relative importance should be small since the dwell time is small due to the high velocities reached in the gap, frequently sonic. For all leakage paths the presence of oil in an oil injected machine will, as intended, reduce the leakage to some degree. The leaking fluid is subjected to wall friction at the boundaries of the leakage paths in all cases. This maintains a 'smear' of oil on the boundary surfaces which results in a smaller cross-section for flow than exists in a dry machine. In addition, the gas flow through any leakage path should be considered as compressible since the Mach number is often large and sometimes equals one. Simplifications are needed to turn such a complicated phenomenon into a mathematical model. The following modelling steps are used to achieve the simplification:

Table 1 Ranking order of magnitude—largest to smallest

Path	1	2	3	4	5	6
Average area	Sixth	Second	Third	First	Fourth	Fifth
Mass flowrate	Third	Second	First	Fifth	Sixth	Fourth

1. The flow through every leakage path is treated as one-dimensional flow.
2. The refrigerant gas, oil and liquid refrigerant are assumed to be separate fluids, and only the gas leaks through leakage paths. Oil and liquid refrigerant remain in the cavity into which they were injected and pass into the discharge region with the gaseous refrigerant during the normal discharge process.
3. The vapour flow through every leakage path is determined by a reversible adiabatic calculation modified by the use of an empirical coefficient, as shown in equation (3) below.

The gas flow speed is calculated by the following one-dimensional, isentropic, compressible flow energy conservation differential equation:

$$W\,\mathrm{d}W + \mathrm{d}i = 0 \tag{1}$$

The maximum flow speed through any leakage path can approach the local sound speed if the pressure difference over the path is large enough. The local speed of sound can be calculated by the following differential equation:

$$W = \sqrt{\left(\frac{\partial p}{\partial \rho}\right)_s} \tag{2}$$

In order to calculate the flowrate through any leakage path, the following continuity equation is needed:

$$\frac{\mathrm{d}m}{\mathrm{d}\varphi} = \frac{c_1 c_2 \rho W A}{\omega} \tag{3}$$

In equation (3), the minimum flow areas for the blow hole (path 3) and the compression start blow hole (path 4) are calculated by the authors' geometrical characteristic calculation program (**3**). The definition of Singh and Patel (**4**) is used to calculate the cusp blow hole areas in the program which are expressed in terms of the male rotor rotational angle. The flow areas for the contact line (path 1), the rotor tip sealing lines (path 2), the rotor suction end face (path 3) and the rotor discharge end face (path 6) are calculated by the following equation:

$$A = \delta \times L_{\text{sealing}} \tag{4}$$

where the sealing line length $L_{sealing}$ is given as a function of the male rotor rotational angle by the geometrical characteristic calculation program.

2.4 Empirical flow coefficients

In equation (3), two modification coefficients are introduced, c_1 and c_2. Coefficient c_1 provides an area change to account for the difference between the area calculated for a compressor at rest and uniform in temperature, and the area under load. Coefficient c_2 is applied to account for irreversibilities reducing the flow speed calculated on the basis that the flow is isentropic and one dimensional, since neither condition holds in reality.

The intention is that coefficient c_1 accounts for three quite distinct physical effects as follows:

(a) the tendency of the oil in the compressor to seal the leakage gaps;
(b) the changes caused to nominal clearances by the cavity pressure under load;
(c) the changes caused to nominal clearances by the temperature distribution of the rotors and housing under load.

The influence of the above factors on the blow hole area and the compression start blow hole area is very small, so that coefficient c_1 for the areas of these two leakage paths could be set to one. However, the influence of the above factors on the leakage area of the other paths could be considerable. Although the authors have developed computer programs such as the force analysis program and the rotor deflection program, which could be used as part of a scheme to determine clearances under load, much more work needs to be done since the determination of clearances under load involves many other factors, such as the calculation of the thermal expansion of the rotors and housing, and the clearances and deformations in the radial and thrust bearings, etc. This is clearly a major task and one to which the authors intend to apply themselves in the future. Different operational conditions result in different clearances under load. Only at the stage where the clearances under load have been determined could coefficient c_1 become a function of the total oil supply rate alone. In fact, experiments show that once the oil supply rate is increased above a certain level, no further improvements of significance are observed in the volumetric efficiency. The oil flowrate is then chosen to achieve a desirable gas discharge temperature; that is the cooling effect of the oil then has priority over the sealing effect. However, injecting quite large quantities of oil into the compressor cavity is a relatively inefficient way of reducing discharge temperature. This is done much more efficiently by injecting relatively small quantities of liquid refrigerant since it flashes by borrowing the substantial amount of latent heat required from the cavity gas. Designers who opt for this method of cooling can reduce the oil flowrate to 'semi-dry' levels, as they are sometimes known, and in so doing reduce power consumption by reducing viscous friction losses.

For the machines for which measured clearances are known, the measured clearances are used in equation (4); otherwise the nominal design clearances are applied. For the contact line between the male and female rotors, an average clearance normal to the rotor helical surface, calculated by the authors' cutter blade calculation program (**1**, **5**), is used.

Since each leakage path has its own unique geometry it is reasonable to expect that c_2 will have a value unique to each path. The leakage paths across the rotor tips are like an orifice plate if sealing strips exist; the leakage paths across the rotor suction and discharge end faces are like a narrow passage between two parallel plates and the others are like convergent–divergent nozzles. The authors considered two procedures for the determination of the coefficients. The first treats each path as having its own coefficient, the second uses the same coefficient for all paths. In the first procedure, it is difficult to decide on different values for the coefficients used for different leakage flow paths. Although the measured or theoretical data for standard orifices and turbine nozzle flow coefficients are available, they relate to dry gas or steam only and to precise geometries, which are not the same as those in the leakage paths. In the second procedure, every path has the same coefficient which is determined in this work by comparing the predicted volumetric efficiency with the measured volumetric efficiency for a medium-sized compressor running in mid-range conditions. In this paper, coefficients are determined by the second procedure, and the agreement between the predicted and measured results for a range of compressors and running conditions is good, although it is to be expected that the predicted leakage distribution will be less good than that derived from a pattern of dissimilar coefficients if a realistic pattern were available, which of course at present it is not.

Furthermore, at the present stage of development of this work it is not possible to evaluate c_1 and c_2 individually. They have to be evaluated and used as a product. Separating them is an objective of the next stage of the work. The product $c_1 c_2$ is determined on a medium-sized machine which has a medium ratio of diameter to length, medium volume and pressure ratios and medium rotational speed, and then the results predicted when the same value of the product coefficient is used through each leakage path are checked against measured results for machines of various sizes (163–510 mm range of rotor diameter; only machines having equal rotor diameters are considered) and various working conditions. The maximum difference between the predicted and measured values of volumetric efficiency is about 5 per cent for R22.

The product coefficient $c_1 c_2$ is influenced by the working fluid and the profile used. For a different working fluid and/or a different profile, different coefficients require to be derived from measured data. Hence, the generality of application of the method extends to compressors of different sizes running under a wide range of operating conditions, but not to different working fluids or compressors with different lobe profiles.

3 COMPUTER PROGRAM OF WORKING PROCESS SIMULATION

The total number of papers published on twin screw compressors is not great when compared with the published work on certain common engineering components such as turbines or transformers, for example.

Papers on general thermodynamic behaviour programs (**4**, **6**, **7**) and on the sophistication with which detailed aspects of thermodynamic behaviour are treated, such as oil injection and economizers (**8–10**), have been published in the last ten years. The authors have also described a comprehensive computerized mathematical model for simulating the working processes of a refrigeration helical screw compressor (**11**). Its flow diagram is shown in Fig. 3.

In the model, oil injection directly into the working cavity and oil draining from the bearings, liquid refrigerant injection, flash of refrigerant dissolved in the oil, vapour charge from the superfeed, different refrigerants and partial loading, etc., are considered simultaneously and separately. Twelve subroutines have been developed to calculate the thermodynamic properties of the refrigerants, in which the thermodynamic property equations for real gases are used (the Martin–Ho equations). The program can be used to predict the performance of any refrigeration helical screw compressor, operating under a variety of operating conditions with different working fluids, providing they are single fluids. The program is being extended to handle the non-azeotropic mixtures currently being introduced in industry. The high predictive accuracy of the program has been confirmed by comparing the predictions with test results (**11**). The mathematical model and the corresponding computer program have been modified recently to take the friction losses in the compressor into account. The detailed description for the mathematical model and procedure for the determination of the other important coefficients besides c_2 will be publshed in a separate paper by the authors.

A geometrical characteristics calculation program has been completed by the authors. The program calculates all the geometrical characteristics, expressing several of them as functions of the male rotor rotation angle, especially those that are required for running the working process simulation program. The start angle is the angle where the cavity volume equals zero. Details of this kind are in a separate paper concerning the computer aided geometrical analysis of the geometrical characteristics published by the authors (**3**).

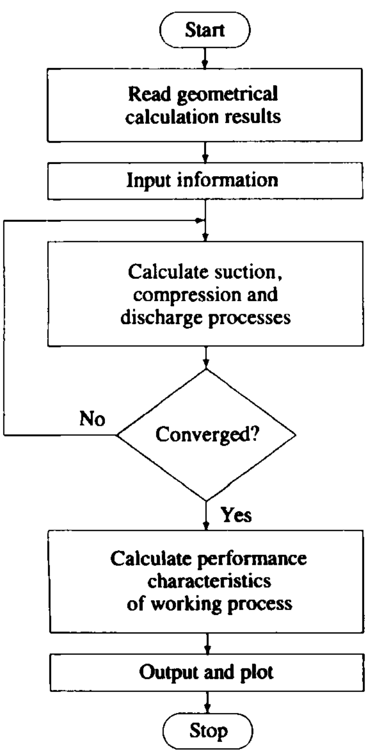

Fig. 3 Process calculation flow diagram

In addition to the results of the geometrical characteristics program, which are read by the working process simulation program automatically, the following data are needed and must be entered by the user:

(a) the rotational speed of the male rotor,
(b) the rotor tip clearances, rotor end clearances and the average clearance between the two rotors,
(c) refrigerant (there are four refrigerants, R12, R22, R134a and R717, that can be chosen),
(d) operating conditions—evaporating and condensing temperatures or suction and discharge pressures,
(e) gas state in superfeed before injection and flowrate or nozzle coefficient (if superfeed is on duty),
(f) liquid refrigerant state before injection and flowrate or nozzle coefficient (if liquid refrigerant injection is on duty),
(g) oil pressure and temperature before injection, flowrate or nozzle coefficient, and mass fraction of refrigerant dissolved in the oil,
(h) flash coefficient, suction port area modification coefficient and discharge port area modification coefficient.

The program calculates pressure, specific volume, temperature, entropy, specific internal energy, mass and sound speed in the cavity volume, and the average gas speeds in the suction and discharge ports. All these calculated results are expressed as functions of the rotation angle of the male rotor, so that plots of them versus the rotation angle or cavity volume can be drawn. The program also calculates the temperature increase of the oil injected during the compression and discharge processes. For convenience the following characteristics of the working process are calculated and easily output:

(a) the gas mass flowrates through the superfeed and slide-valve by-pass port,
(b) the liquid refrigerant mass flowrate through the injection port,
(c) the oil mass flowrate through the injection port,
(d) the mass flowrates of the unflashed liquid refrigerant injected directly or dissolved in the injected oil,
(e) the leakage mass flowrate through *any* leakage path,
(f) theoretical capacity, real capacity and volumetric efficiency,
(g) indicated power, isentropic power and isentropic indicated efficiency,
(h) other useful information.

The simulation program used in this paper has been specially designed in order to calculate the leakage rates leaking in to and leaking out of a chosen cavity volume, and as a consequence is a powerful tool for the leakage analysis of a twin screw compressor.

4 INFLUENCES OF LEAKAGE PATHS ON COMPRESSOR EFFICIENCIES

In order to investigate the influences of the leakage paths on compressor behaviour, a twin screw compressor for which performance data are available was chosen. The specification of the compressor is as follows:

(a) lobe profile: SRM D standard,
(b) lobe combination: 4 + 6,
(c) bore diameters (equal): 163.2 mm,
(d) male rotor wrap angle: 322°,
(e) length–diameter ratio: 1.93,
(f) volume ratio for axial discharge port: 5.0,
(g) volume ratio for radial discharge port: 3.6.

The running conditions are:

(a) refrigerant: R22,
(b) Condensing temperature: 298.15 K (corresponding condensing pressure: 10.44 bar),
(c) evaporating temperature: 263.15–233.15 K (corresponding pressure ratios: 2.95–9.95),
(d) superheat degrees: 30 K,
(e) oil pressure before injection: 10 bar,
(f) oil temperature before injection: 313.15 K,
(g) running speeds: 3000 and 3500 r/min.

The clearances of paths 1, 2, 5 and 6, which should be input to calculate the leakage areas when the simulation program is run, are those that have been set and checked by measurement in the test compressor.

Figure 4 shows the simulated and tested results of the specified compressor for the specified working conditions. The results are for two rotational speeds of the male rotor (driven), that is 3000 and 3500 r/min. For the two rotational speeds the predicted and test volumetric efficiencies show the same trends and almost the same values. The curves of predicted indicated efficiency and test total efficiency have the same shape. The ratio of the total efficiency to the indicated efficiency gives the mechanical efficiency of the compressor. For a range of pressure ratio and one speed, it is very nearly constant. This is expected since the rotational speed is constant and the mechanical efficiency depends predominantly on it. The higher rotational speed results in a higher volumetric efficiency for the whole pressure ratio range considered. Higher indicated and total efficiencies are also shown for most of the pressure ratio range considered. Since an increase in speed reduces the relative leakage losses but increases the viscous friction loss, it may be concluded from Fig. 4 that the efficiency gain due to reduced leakage outweighs the reduction due to viscous friction since a net increase in both the measured total efficiency and the calculated indicated efficiency is shown.

The working fluid can leak *out* from the considered cavity to the following cavity or to the suction pressure (the suction chamber or the cavity connected to the suction chamber), or it can leak *into* the considered cavity from the leading cavity or from the discharge pressure (the discharge chamber or the cavity connected to the discharge chamber). Table 2 shows the leaking outflow rates from the considered cavity, the leaking inflow rates to the considered cavity and the net leaking outflow rates from the considered cavity through the different leakage paths. The rotational speed and the evaporating temperature for Table 2 are 3000 r/min and 253.15 K respectively. As shown in Table 2, the highest individual leakages occur across the male and female tip sealing lines (path 2), but the net leaking outflow rate is not the maximum due to a very high inflow rate. The leakage rates through the cusp blow hole (path 3) are similar to those that occur across the male and female tip sealing line, but the net leaking outflow rate is much smaller due to a relatively higher inflow rate. The leaking outflow rate across the contact line (path 1) is much smaller than through path 2 or path 3, but since the leaking inflow is very small path 1 has the maximum net leaking outflow rate. The situ-

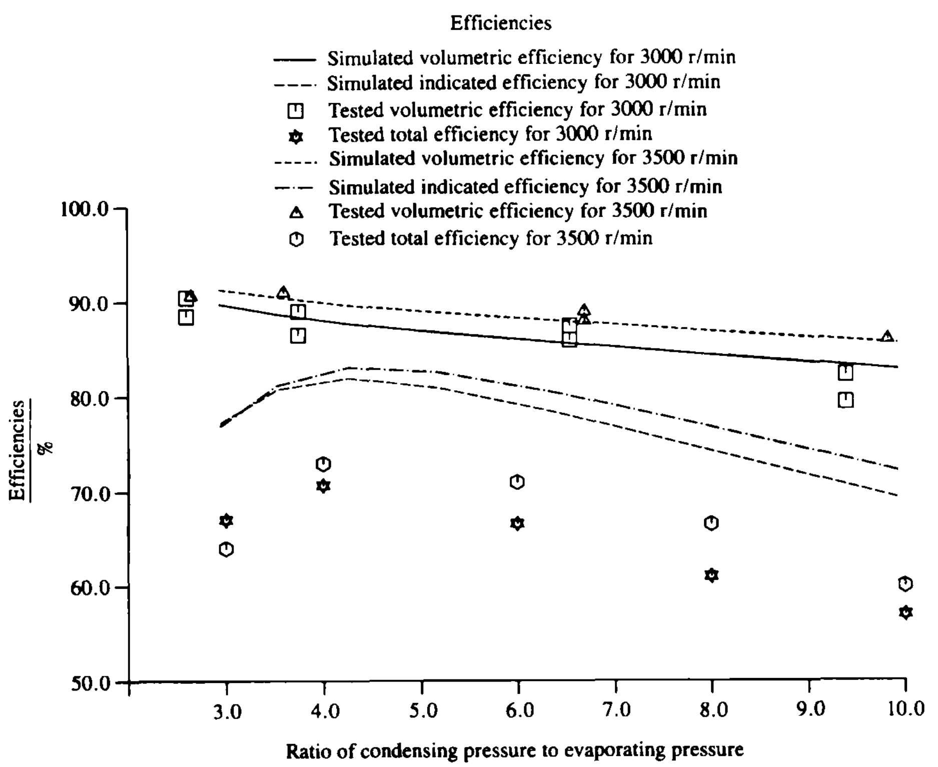

Fig. 4 Predicted and tested efficiencies

Table 2 Leaking flowrates through each leakage path* (rotational speed: 3000 r/min; evaporating temperature: 253.15 K)

	Leaking outflow rate kg/min	Leakage inflow rate kg/min	Net leaking outflow rate kg/min
Path 1	5.61	0.27	5.34
Path 2	16.83	13.79	3.04
Path 3	13.06	12.00	1.06
Path 4	2.42	0.00	2.42
Path 5	0.97	0.00	0.97
Path 6	3.80	2.84	0.96

* Quantities predicted by the model are rounded off to four significant figures to retain descrimination of small changes in model output. This does not imply the precision with which the model predicts reality.

ation is similar for leakage through the compression start blow hole (path 4) and the rotor end clearance at the suction pressure end (path 5), which have zero leaking inflow rates and, as a result, have large net leaking outflow rates. The leakage through the rotor end clearance at the discharge pressure end (path 6) is small compared with most of the others, due mainly to the smaller clearance set here.

Figure 5 shows the influence of each leakage path on both volumetric and isentropic indicated efficiencies predicted by the model for the same machine and running conditions as for Table 2. Worthy of note is the very different influence on the two efficiencies a particular leakage path can have. Figures 5, 6 and 7 have been constructed by first using the model to calculate efficiencies with each leakage set in turn to zero. The model is then run normally with all leakages effective. Efficiency reductions are predicted which are represented in these three figures. The cusp blow hole (path 3), which has a small net leaking outflow rate, has a small effect on the volumetric efficiency, which it reduces by 0.95 per cent, but a big effect on the isentropic indicated efficiency, which it reduces by 3.68 per cent. Basic thermodynamics explains this. All leakages are irreversible and cause a loss of energy. This is shown as reduced energy efficiency and is predicted by the model. Internal leakages (leaking to the following enclosed cavity) reduce the indicated efficiency and external leakages (leaking to the suction pressure) reduce both the indicated efficiency and the volumetric efficiency. Although the contact line (path 1) has the most important influence of the six paths on the volumetric efficiency, which it reduces by 4.77 per cent, its influence on the isentropic indicated efficiency is less important, coming in order of importance after the rotor tip sealing lines (path 2) and the cusp blow hole (path 3). The leakages through the compression start blow hole (path 4) and the rotor end clearance at the suction pressure end (path 5) have a considerable influence on the volumetric efficiency, but have no significant influence on the isentropic indicated efficiency as the leakages take place at the very beginning of the compression process when the pressure in the cavity is very low. The reason why they result in such a big reduction of the volumetric efficiency is due to the comparatively large area of the compression start blow hole and the very big rotor end clearance at the suction end. The leakage through the rotor end clearance at the discharge pressure end (path 6) has an unimportant influence on both the volumetric efficiency and the isentropic indicated efficiency, due to the small clearance maintained here.

For different evaporating temperatures, that is different pressure ratios of discharge to suction, the leakage table is similar. Figure 6 shows the reduction of the volumetric efficiency for every leakage path for the same machine and rotational speed but for two different evaporating temperatures. The evaporating temperatures for Fig. 6 are 263.15 and 243.15 K, which correspond to pressure ratios of 2.94 and 6.39, respectively. The other running conditions are the same as for Fig. 5. As shown in Fig. 6, the increase in the pressure ratio has the largest influence on the leakage through path 1, the contact line between the rotors. This is as expected

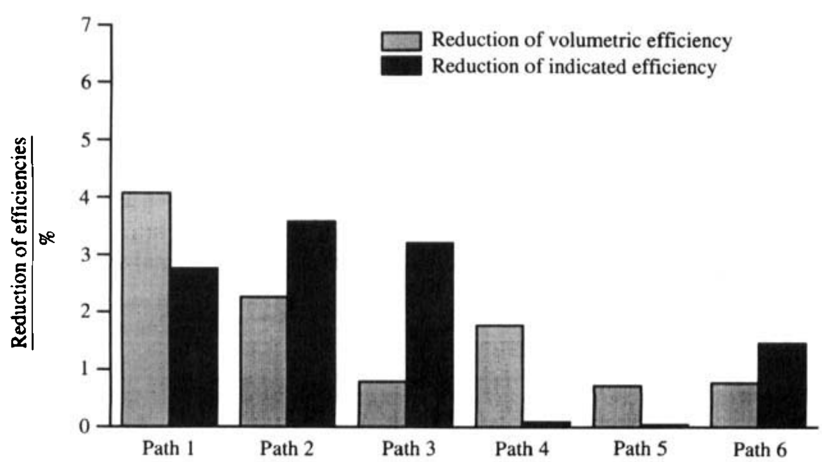

Fig. 5 Reduced efficiencies: leakage compared with zero leakage path by path (rotational speed: 3000 r/min; evaporating temperature: 253.15 K)

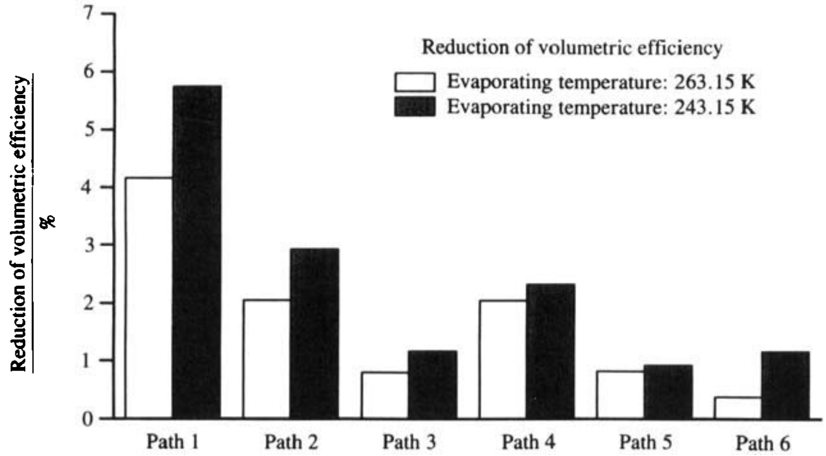

Fig. 6 Reduced volumetric efficiency: leakage compared with zero leakage path by path (rotational speed: 3000 r/min)

when it is considered that the maximum pressure difference available at each male rotor angle drives the leakage flow through path 1.

Figure 7 shows the influence of each leakage path on both volumetric and isentropic indicated efficiencies for the same machine and running conditions as for Fig. 5 but for a different rotational speed, 3500 r/min. Compared with 3000 r/min (Fig. 5), it is clear that all leakage paths have less of an influence on the volumetric and isentropic indicated efficiencies. The outflow rates leaking from the considered cavity, the inflow rates leaking into it and the net leaking outflow from it are shown in Table 3. Compared with Table 2, these leaking rates remain almost unchanged, but as the theoretical capacity of the machine increases from 112.09 to 130.77 kg/min due to the increase of the rotational speed, the *relative* leaking rates are decreased considerably. As a result, the volumetric efficiency is increased from 87.70 per cent (corresponding to 3000 r/min) to 89.61 per cent (corresponding to 3500 r/min). Figure 8 shows the predicted and measured efficiencies as functions of the rotational speed of the male rotor. When running at 3500 r/min the compressor has both higher volumetric and isentropic indicated efficiencies than when running at 3000 r/min. Increasing the rotational speed of the machine always results in an increase of the volumetric efficiency, but the influence of the rotational speed on the isentropic indicated and total efficiencies is a little more complicated. It is a topic that is beyond the scope of this paper since it concerns the viscous friction

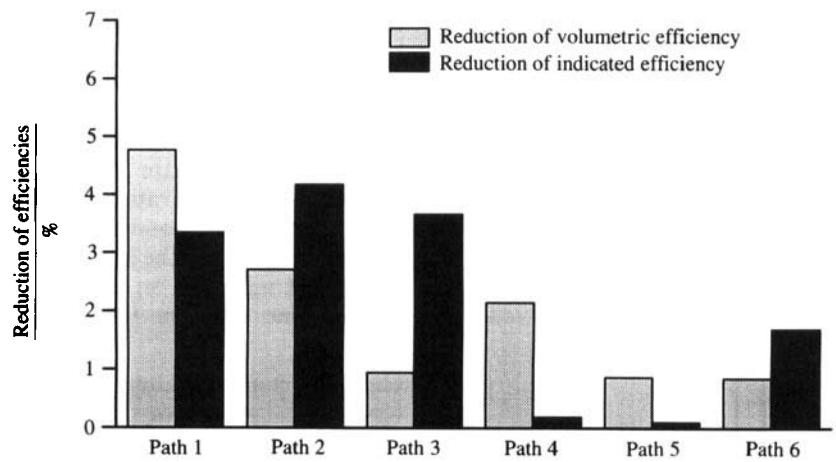

Fig. 7 Reduced efficiencies: leakage compared with zero leakage path by path (rotational speed: 3500 r/min; evaporating temperature: 253.15 K)

Table 3 Leaking flowrates through each leakage path (rotational speed: 3500 r/min; evaporating temperature: 253.15 K)

	Leaking outflow rate kg/min	Leakage inflow rate kg/min	Net leaking outflow rate kg/min
Path 1	5.60	0.27	5.33
Path 2	16.72	13.76	2.96
Path 3	13.03	11.99	1.04
Path 4	2.31	0.00	2.31
Path 5	0.94	0.00	0.94
Path 6	3.85	2.84	1.01

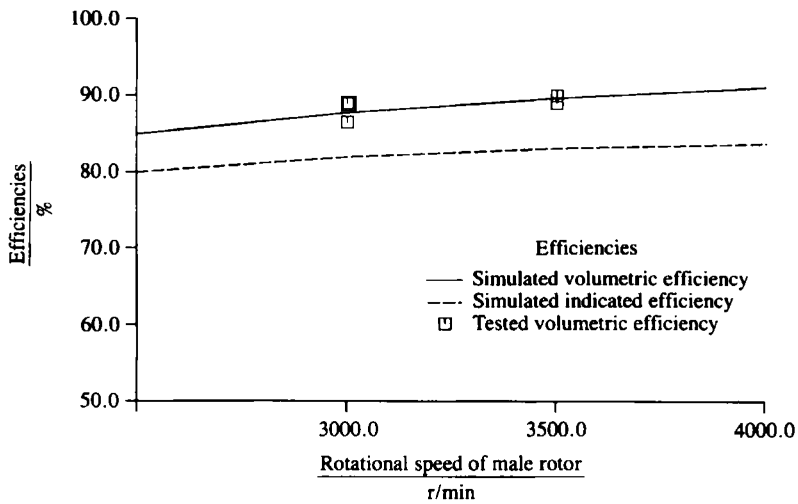

Fig. 8 Predicted and tested efficiencies versus rotational speed of male rotor (evaporating temperature: 253.15 K)

loss as a function of speed compared with the volumetric efficiency as a function of speed. The authors will discuss this in a separate paper.

5 APPLICATION OF THE RESULTS OF THE LEAKAGE ANALYSIS

Each leakage path has its own unique influence on the volumetric and isentropic indicated efficiencies. A big reduction in volumetric efficiency is not necessarily accompanied by a big reduction in isentropic indicated efficiency. From the point of view of saving energy it is more important to obtain a high isentropic indicated efficiency than to obtain a high volumetric efficiency. Obtaining a higher volumetric efficiency means that a smaller machine can be used for the same capacity and that the cost of the compressor is likely to be reduced.

To improve the performance of a compressor by reducing the leakages through the different leakage paths, optimizing the geometric parameters of a profile must be the most valuable method, since path 3 (the cusp blow hole), path 1 (the contact line) and path 4 (the compression start blow hole) depend strongly on the profile parameters chosen by the designer. Paths 3 and 1 have a particularly strong influence on the isentropic indicated efficiency. With regard to reducing the leakages through path 2 (the rotor tip sealing line), path 5 (the rotor end clearance at the section end) and path 6 (the rotor end clearance at the discharge end), compressor designers should give consideration to an analysis of the thermal and dynamic distortions of the rotors and housing so as to choose optimal clearances for paths 2, 5 and 6. The control of manufacturing quality is critical to this endeavour.

A method used to optimize the parameters of a profile has been presented by the authors (**2**), as has a procedure to obtain the optimum clearance distribution between the rotors (**5**). It has been used to reduce the leakage through path 1, which has a considerable influence on both the volumetric and isentropic indicated efficiencies.

The results of the leakage analysis show that the cusp blow hole has a very important influence on the isentropic indicated efficiency. Consequently, the improvement of a profile should concentrate on reducing the cusp blow hole area, although its influence on the volumetric efficiency is very small. The designer must determine his or her priorities; that is which is more important, compressor energy efficiency or compressor size? Usually the reduction of the cusp blow hole area will result in a little longer contact line length, but the total leakage area across the contact line can be reduced by the method described in reference (**5**). Reducing the cusp blow hole area may result in increasing the compression start blow hole area, with a consequent, perhaps slight, net reduction in volumetric efficiency, but with no significant influence on the isentropic indicated efficiency.

For the compressor specified in this paper, path 2 (the rotor tip sealing line) has the most important influence on the isentropic indicated efficiency (see Figs 5 and 7). The method presented by the authors in reference (**2**) can be used to optimize the rotor parameters of a compressor, such as the wrap angle and the length–diameter ratio, so as to change the leakage table and reduce the leaking rates through path 2 significantly.

Making use of the above ideas and the optimization method presented in reference (**12**), an improved SRM D profile shown in Fig. 9 has been developed by the authors' profile generation program, which can generate profiles under the definitions of the SRM symmetrical circular, A and D profiles. It is presented as an example used to show how the profile parameters and rotor parameters, etc., influence the leakage pattern and thus the volumetric and isentropic indicated efficiencies. The authors wish to suggest that the designers of twin screw

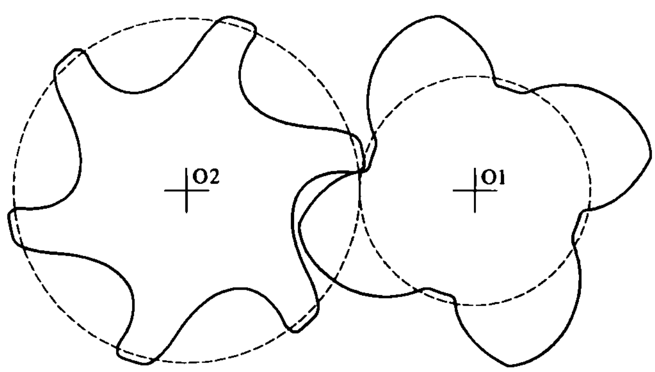

Fig. 9 A proposed improved profile

compressors may design higher quality compressors by carrying out a similar leakage analysis and following a similar optimization procedure to that described in the example given here. However, when improving a profile, in addition to considering thermodynamic performance, manufacturing cost [as discussed in references (**1**) and (**5**)] must also be considered.

The objective of the example presented here is to minimize the reduction of the indicated efficiency due to path 3 (the cusp blow hole) shown in Fig. 5. A procedure that can be used to reduce the cusp blow hole effectively but does not reduce the addendum of the female rotor is presented in detail in reference (**12**) by the authors. The same method is used to reduce the blow hole of the profile shown here in Fig. 9. Figure 10 shows for this compressor the reduced volumetric and isentopic indicated efficiencies due to the effects of the different leakage paths. The proposed machine has almost the same theoretical capacity as the existing machine described in the last section. Both machines have the same rotor diameter, the same wrap angle of the male rotor, the same ratio of rotor length to diameter and equal bore diameters. The predicted results in Fig. 10 are for the same running conditions as in Fig. 5 and the clearances of the different parts of the machines are exactly the same. Compared with the existing machine, the isentropic indicated efficiency has improved by 1.95 per cent and the volumetric efficiency has improved by 0.57 per cent. Thus in this case the change in the leakage pattern does improve the isentropic indicated efficiency; 1.83 of the 1.95 per cent improvement in the indicated efficiency is obtained by changing the leaking outflow and leaking inflow rates through the cusp blow hole (path 3). Comparing Fig. 10 with Fig. 5 shows that the reduced indicated efficiency due to the cusp blow hole in the proposed machine has become the third in importance after the tip sealing lines (path 2) and the contact line (path 1). The reduced volumetric efficiency due to path 3 in the proposed machine becomes the most unimportant of these effects. The other efficiency reductions in Fig. 10 remain almost unchanged compared with those shown in Fig. 5, and the leaking rates through each leakage path except the cusp blow hole (path 3) also remain almost unchanged (see Table 4).

If the method presented in reference (**5**) is used to obtain the optimum clearance distribution between the male and female rotor and the optimization method presented in reference (**2**) is used to reduce the rotor tip sealing line lengths, the losses in the volumetric and indicated efficiencies due to paths 1 and 2 can be reduced. Repeated applications of the procedure described here can further improve the performance of the compressor.

6 DISCUSSION AND SUGGESTIONS FOR FURTHER WORK

The same value of the product c_1c_2, effectively a single leakage coefficient, is used in the model for *all* leakage paths, despite the fact that the temperature and pressure effects under load tend to increase some clearances and reduce others. Table 5 attempts to summarize the position.

The presence of oil reduces all clearances and leakages, as intended. Under load the compressor is hotter

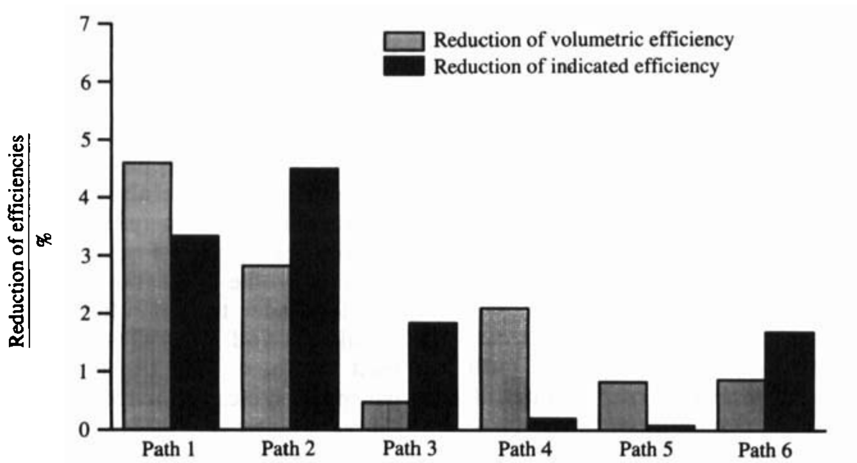

Fig. 10 Reduced efficiencies for a new proposed compressor (rotational speed: 3000 r/min; evaporating temperature: 253.15 K)

Table 4 Leaking flowrates through each leakage path for a new proposed compressor (rotational speed: 3000 r/min; evaporating temperature: 253.15 K)

	Leaking outflow rate kg/min	Leakage inflow rate kg/min	Net leaking outflow rate kg/min
Path 1	5.16	0.27	4.89
Path 2	16.81	13.81	3.00
Path 3	6.25	5.74	0.51
Path 4	2.23	0.00	2.23
Path 5	0.89	0.00	0.89
Path 6	3.55	2.62	0.93

Table 5 Influence of operational load on clearance (+, an increase; −, a decrease; 0, no change)

Path	Influence on clearance		
	Oil	Pressure	Temperature
1. Contact line	−	+	−
2. Tips	−	−/+ (outer/inner)	−
3. Cusp blow holes	−	0	0
4. Compression start BH	−	0	0
5. Suction end face	−	0	−
6. Discharge end face	−	0	−

inside (the rotors) than outside (the casing). Consequently, all clearances (other than paths 3 and 4) will be reduced due to the temperature effects of load, but the relative magnitude of this effect from one path to the next is unknown to the authors at present.

The influence of gas pressure under load is more complex since it tends to separate the rotors, giving a tip-to-bore clearance which is increased on some regions of the bore surfaces and decreased on others. The orientations of the forces separating the rotors are roughly NE on one and NW on the other; hence the net effect is difficult to predict but an increase seems most probable. At the contact line the clearance must increase.

Measuring the mass flowrates through each leakage path one at a time or simultaneously as the compressor operates seems at first sight to be an impossible task, except that it is not in the creed of the engineer to deem any task to be impossible, and certainly not after a first examination. Rigs could be constructed from compressor parts in static arrangements to permit the measurement of each leakage mass flowrate individually for a range of combinations of male rotor angle and pressure difference, but the fluid behaviour in a running compressor will almost certainly differ considerably from the behaviour in static rigs, especially for oil-injected machines in which the behaviour of the oil, its motion and distribution within the cavity must be affected enormously by the rotation of the rotors.

However, building static rigs in an attempt to reproduce the leakage behaviour in dry compressors operating normally may very well produce useful results. To represent the effect of oil, an oil blockage coefficient, guessed or inferred from measurements of macrobehaviour, could be introduced. A number of reasonably complex rigs would be required to be designed and built and fully instrumented; a significant investment of resources and time would be required. As a consequence, the authors believe that their approach of modelling, combined with 'organized trial and error' to determine values of the product c_1c_2 which give good agreement with measurements of macrobehaviour determined from readily available standard compressor tests, is a viable economic alternative to constructing a number of special-purpose rigs and will produce results much more quickly. The authors believe that the work presented here is valid in itself and, at the very least, is valid as an exploratory investigation. Furthermore, it even has a reasonable prospect of being sufficient in itself to achieve the immediate objectives: namely the construction of an accurate user-friendly computerized thermo-fluid model of a twin screw compressor which models leakage flow sufficiently accurately to make it a useful tool for research, development and design purposes.

Measuring the pressure–volume diagram in this machine, though more complicated than in a reciprocating compressor, presents no insurmountable difficulties and is a task planned by the authors.

7 CONCLUSIONS

1. A computer program has been developed to analyse the leakages of a refrigeration helical screw compressor. The program can calculate the leaking outflow rate, the leaking inflow rate and the net leaking outflow rate through each leakage path. The effect on the volumetric and isentropic indicated efficiencies of the machine due to the leakages can be calculated. The accuracy of the predictions for each path individually is not known at present, but it seems probable that the model is overestimating the leakage through paths 5 and 6 and underestimating it through paths 1 and 2. Special-purpose static test rigs designed to reproduce the conditions in an oil-free compressor need to be built to investigate the leakage paths one by one. However, comparison with test data for a complete compressor shows the macropredictions of the authors' computer model to be good.
2. The leakage analysis for the existing profile and the existing compressor has been completed, and the leakage patterns for different running conditions, including different rotational speeds, have been obtained. Each leakage path has an influence on compressor efficiencies different from that of any other path. A path, apparently very important to volumetric efficiency such as cusp blow hole, in fact has very little influence on volumetric efficiency, but has a very considerable influence on isentropic indicated efficiency. From the point of view of saving energy it is more important to obtain a high isentropic indicated efficiency than to obtain a high volumetric efficiency. Volumetric efficiency has an important influence on compressor size: the higher the volumetric efficiency the smaller the compressor is required to be.
3. Making use of the leakage analysis to determine the influence of the three most important leakage paths, that is the contact line (path 1), the rotor tip sealing line (path 2) and the cusp blow hole (path 3), shows clearly which of these paths results in the biggest reduction of the indicated or volumetric efficiencies for the existing machine. As an example to show how to use the leakage table to improve the performance of a compressor, an improved profile is proposed to reduce the influence of path 3, which gains a higher isentropic indicated efficiency. The change in the leakage pattern achieves a considerable improvement in isentropic indicated efficiency, although the improvement in volumetric efficiency is not large.

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