The Influence of Leakage on the Performance of Refrigerant Screw Compressors

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Zusammenfassung

In diesem Bericht wird der Einfluß von Leckspalten auf die Verdichterleistung beschrieben. Diese Arbeit wurde zwischen 1989 und 1990 während der Entwicklung eines 164 mm, R-22 Schraubenverdichters durchgeführt. Verfahren zur Bestimmung der kritischen Spalte beim Stillstand als auch während des Betriebs werden beschrieben. Versuchsergebnisse über den Einfluß des Spaltes und ein Vergleich zwischen Berechnung und Messungen werden ebenfalls zur Verfügung gestellt.

Summary

This report describes a study of the effect of clearances on compressor performance, carried out in 1989 - 1990 during development of a 164 mm, R-22 screw compressor. Procedures for determining critical clearances at compressor assembly and models for estimating clearances in a running compressor are described. Test results showing the effect of individual clearances are given and a comparison with analytical results is provided.

1. Introduction

In designing a screw compressor for production, it is important to know the impact of clearance levels and tolerances on compressor performance. More effort is required to achieve low clearances and small clearance variations where leakage has the greatest effect. Clearances that have less impact on performance do not need the same degree of effort in their control. There has been considerable work published on the effect of clearances on performance, both before and since the work reported here. Rinder /1/, Peveling /2/ and Buthmann /3/ reported on leakage flow models and calculation of clearances considering such factors as injected oil, operating pressures and temperatures, and specific features of the leakage gaps. More recently, Kauder and Gödde /4/, Kauder and Rau /5/ and Fleming, et al /6/ have reported on the issues of clearance analysis, leakage computations and the effects of various leak paths on performance.

Information from /1/, /2/ and /3/ was incorporated into a comprehensive thermodynamic model during work on semi-hermetic, oil injected, R-22 compressors for water-cooled water chillers. This model was then used to design a 164 mm compressor. This was our first production compressor for air-cooled chiller applications to use the rack-generated rotor profile /7/. After the first compressors were built, a test program was begun to compare actual performance to design expectations and to determine the effects of design variables such as clearance.

For these tests, it was necessary to know the clearances for each compressor configuration. This required knowing both the clearances of the assembled compressor and how the operating pressures and temperatures would change these clearances. The methods developed to determine the clearances in order to properly interpret the test results are described in section 2. Results of experimental and analytical studies of a 164 mm compressor designed for R-22, air-cooled water chiller applications follow in sections 3. and 4.

2. Assembled Clearances

A computer assisted clearance analysis and build documentation procedure was written for screw compressors with slide valve unloaders. The purposes of this program are to provide a detailed picture of assembled compressor clearances (both cold, as assembled and as running under load) and to generate a complete build summary of each tested compressor.

The method used involves linking data from inspections of critical parts to a master build log, written using Microsoft EXCEL™. The program coded into the build log uses the input data to compute clearances, generate clearance distribution charts and format a multi-page build summary. The program will compute rotor-to-housing and rotor-to-slide valve radial clearances along with rotor-to-housing axial clearance and rotor intermesh clearance at the assembled center distance. The following effects are included in the clearance analyses:

- Bearing clearance, rotating eccentricity and stationary eccentricity
- Rotor and bearing deflection due to gas loading
- Bearing clearance applied in the average load direction or rotors orbiting in the bearings
- Slide valve rotation due to key and key slot width differences
- Slide valve located away from rotors (typical running condition) or towards rotors
- Dowel pin eccentricity calculations for clearance adjustments.

Use of the assembly log during an experimental program offers the following advantages:

- Analysis of effects of alternative part selections on clearances
- Identification of reliability or performance threatening clearance situations before running the compressor
- Rapid analysis of effects of reworking parts to improve reliability or performance
- Complete documentation of selected build geometry

The clearance analysis starts with coordinate measuring machine (CMM) measurements of the three major compressor parts: the main rotor housing (which includes the inlet end bearings), the discharge bearing housing, the slide valve unloader. Rotor shaft extension diameters and runouts are also measured. In addition, the rotor pair is measured manually to determine profile diameters and runouts, rotor pair intermesh clearances and backlash. Finally, mounted bearing under-roller diameter and data on other features is taken during assembly of the compressor. All of this data is transferred to a set of database files that make up the assembly log program.

When a compressor is assembled, the serial numbers for the parts selected for the build are entered into a data form along with the measurements obtained during the build. Data related to the effects of operating pressures and temperatures can also be included. This form is read by the assembly log spreadsheet, which then gets data for each part from the databases in separate spreadsheet files. When all of the data is loaded into the main assembly log, the built-in calculations generate tables of clearances, clearance charts and data on critical assembled dimensions arranged on separate pages that can be printed to provide a complete build documentation. The assembly log also allows numerous measurements not required for the clearance calculations to be entered for the purpose of documentation only.

Figure 1 shows a schematic of the compressor with assembly log measurement points and the general coordinate system. The inlet end of the compressor is to the left. Points U and T define the axis along which the center of the male rotor bore is assumed to lie for the CMM inspection.

This axis can be defined arbitrarily as the assembly log uses coordinate transformations for the clearance calculations. Point U is the measured center of the inlet end bearing and T is the theoretical center of the housing bore at the discharge end. Points W and V define the measurement axis for the female rotor. G and F are the measured centers of the male and female discharge end bearings, respectively. Planes labeled N, P, R and S are locations at which CMM data is taken to define the bores.

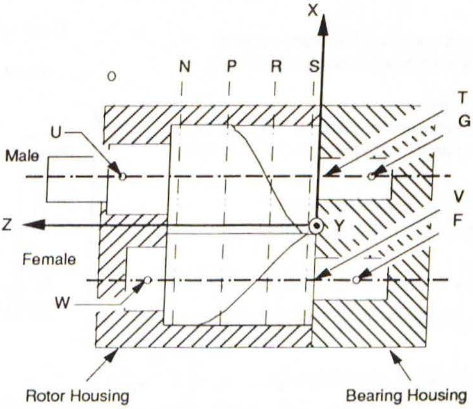


Bild 1 / Figure 1

Points U and G on the male rotor and W and F on the female define the centerlines of the rotor shafts. The effect of bearing clearance is added to these positions to locate the shaft in the assembled compressor. In analyzing the running compressor, bearing deflection is added. To compute the clearances between the rotor and housings, the measured rotor profile diameters are distributed along the axis of the rotor shaft (U' G' for the male...see Figure 2...and W' F' for the female). At each measurement plane (N, P, R or S) points on the bores are defined by measuring their x,y coordinates in a local coordinate system whose origin is on the U-T or W-V axes. This is coordinate system **L** (for Local) in Figure 2.

Coordinate system **Q** in Figure 2 is in the same axial (z) plane as **L**, but with its origin on the rotor axis. The rotor axis differs from the housing measurement axis for the following reasons: [1] the center of the discharge end bearing G is different than the discharge end of the measurement axis T due to offset of the bearing housing relative to the rotor housing both by design and because of tolerances; [2] at each end, the rotor axis shifts in the bearing clearance. The direction of the shift is assumed to be in the direction of the average gas load; [3] if an analysis of a running compressor is requested, the shaft ends in the bearings shift due to deflection of the bearings and at each measurement plane the axis moves, again in the direction of the gas load, due to bending of the shaft and bearings.

The computation of the clearance between the rotor and housing bores is carried out for each point measured on the housing bores. Typically, we measure six points on each bore at each measurement plane for a total of 24 (x,y) coordinates in the male bore and 24 for the female. At each measurement plane, the x,y coordinates of the housing bore are transformed into the **Q** coordinate system. The clearance is then computed as the difference in the distance from **Q** to the transformed coordinates x,y and the radius of the rotor at the measurement plane.

The transformations and clearance calculations applied to a point on the male rotor housing are defined in equations (1). Coordinates x' and y' are the CMM points measured in the **L** coordinate system, represented in the **Q** coordinate system. D' is the measured rotor profile diameter at the inspection location plus the rotor eccentricity. D' thus defines the envelope within which the rotor rotates. The origins of both the **L** and **Q** systems are defined relative to a global coordinate system *(X,Y)* whose origin in turn is set by measurements of the dowel pin holes in the main rotor housing.

L Local housing coordinate system
Q Local rotor axis coordinate system

Subscripts
k - measurement plane (k = N, P, R or S)
i - measurement point (i=1,2, . . ., npts)

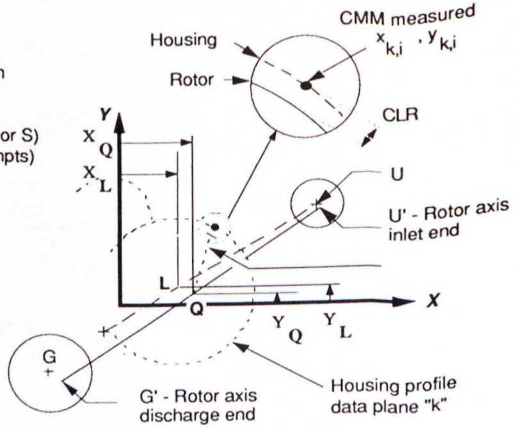
Equations 1
Radial Clearance Calculations

$$x'_{k,i} = x_{k,i} + X_{L\,k} - X_{Q\,k}$$

$$y'_{k,i} = y_{k,i} + Y_{L\,k} - Y_{Q\,k}$$

$$D_k' = D_k + \text{eccentricity}_k$$

$$CLR_{k,i} = \sqrt{x'^{\,2}_{k,i} + y'^{\,2}_{k,i}} - \frac{D'_k}{2}$$



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The calculation is carried out for each point on the housing for which CMM data is available. The computed clearances are stored in a table in the assembly log spreadsheet and are also used to generate plots of clearance at each measurement plane.

Calculation of clearances between the rotors and the slide valve unloader are carried out in a manner similar to that used for the housing. However, there are additional considerations for locating the surface of the slide valve before the clearances can be evaluated. Figure 3 shows the slide valve geometry and coordinate system. Again, **L** denotes the local measurement axis for CMM data and the surface coordinates x,y are relative to these local systems. Figure 4 shows the other elements of the slide valve and slide valve bore geometry that are required for the clearance analyses. The assembly analysis includes the effects of slide valve rotation, horizontal shift and vertical shift. For the vertical shift, it is assumed that the slide valve is down (towards the rotors) if the compressor is not running, but is pushed away

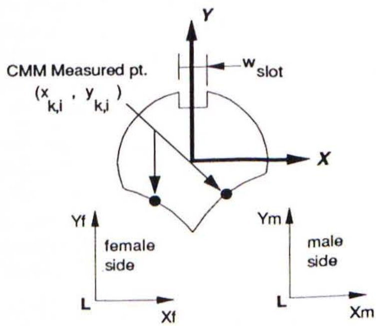
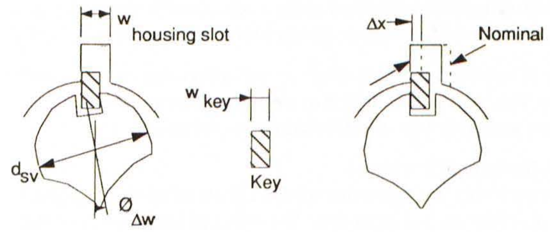


Bild 3 / Figure 3

from the rotors and rotated towards the male rotor if the compressor is under load. The amount of rotation or shift is computed using the measured values for the slide valve diameter, the housing bore diameter, the width of the slide valve key and the widths of the slots in the housing and the unloader. Given the location and orientation of the slide valve, the computation of clearances between the rotor and slide valve surfaces is carried out as it was for the housing. Measured coordinates x,y are transformed into the rotor axis system Q and the clearance is computed as the difference in the distance from the origin to x', y' and the local radius of the rotor (plus eccentricity).



Rotation from Differences in Key and Key Slot Widths

Offset Housing Slot Location

Bild 4 / Figure 4

Figure 5 shows the chart of clearances at planes R and S (nearest the discharge end of the compressor) for build 1.7 of the 164 mm compressor tested during this project. This chart shows clearances for the compressor as assembled, with no thermal or bending effects included. The innermost arcs represent the rotors and the data represents the locations of the housing bores and slide valve surfaces, with the actual values magnified. The short lines pointing away from the outer circular grid lines represent the direction of the average gas loads at the discharge end of the rotors. The effect of the slide valve rotation is evident in this chart: the clearance is relatively large over the top of the female rotor and smaller at the cusp; just the opposite is true on the male rotor side.

In addition to radial clearance calculations, the assembly log computes an average rotor intermesh clearance. Clearances measured for each rotor pair are stored in one of the database spreadsheets. The assembly log program retrieves the

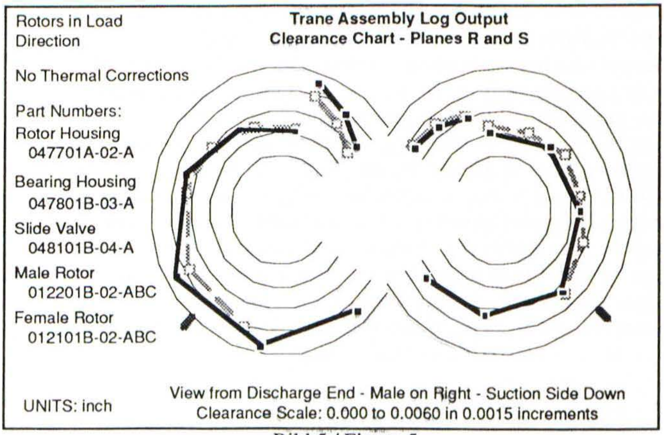


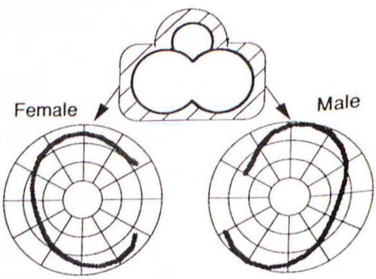
Bild 5 / Figure 5

clearance data for the rotor pair selected then adjusts the clearance to the center distance computed for the housings and bearings. The change in average intermesh clearance with center distance is computed using a rotor profile design program, which calculates the rotor pair clearances from either design profile clearances or using CMM measured profiles.

The final clearance of significance is that between the rotors and the discharge bearing housing face. This clearance is built in to the compressor at assembly and the value measured is recorded in the assembly log...no calculations are necessary.

3. Running Clearances

The assembly log approximates the effects of operating pressures and temperatures by using built-in models and input data. The effect of temperature on the radial clearances comes from finite element modelling of the rotor housing. Figure 6 shows the change in clearance around the rotor bores computed for a discharge temperature of 100°C. This information has been converted into equations describing the change in clearance as a function of angular location around the rotor bore. For each housing measurement location, the clearance change due to thermal distortion is subtracted from the clearance computed for the cold housing. The correction applied is a function of the housing temperature, which varies from the inlet end to the discharge end. The assembly analysis allows a scaling of the clearance shift to be applied as a function of the distance along the measurement axis to account for this change in temperature.



Polar graph radial scale :: 1 unit = 10 µm

Bild 6 / Figure 6

The measured rotor intermesh clearance is adjusted for the effects of temperature and pressure loads using data from the computer programs used for design of the compressor and rotors. Gas loads on the rotors and at the bearings are computed using the compressor thermodynamic model. This data is used by a shaft deflection and bearing selection program to compute the deflections due to rotor bending and elastic deformation in the bearings. These data are input into the assembly analysis. The program uses the deflections to compute true running center distances and then calculates the intermesh clearance, correcting the measured clearances using the specified change in clearance with center distance. A final correction to the intermesh clearance is made to account for the effect of temperature on the rotors at constant center distance. This correction is determined using the screw rotor design program, which can determine clearances between rotors at any center distance and which includes a model for the change in rotor profile due to the effect of operating temperatures. Figure 7 shows results computed for two rotor temperature levels - cold and in a compressor running at 52°C discharge temperature. An analysis carried out for the rotors at constant temperature, but with center distance reduced by 25 µm is also illustrated. These calculations result in the following data for the 164 mm rotor pair tested for this study:

Change in average mesh clearance, temperature effect	=	-3,0 µm / °C
Change in average mesh clearance, center distance effect	=	0,7 µm / µm

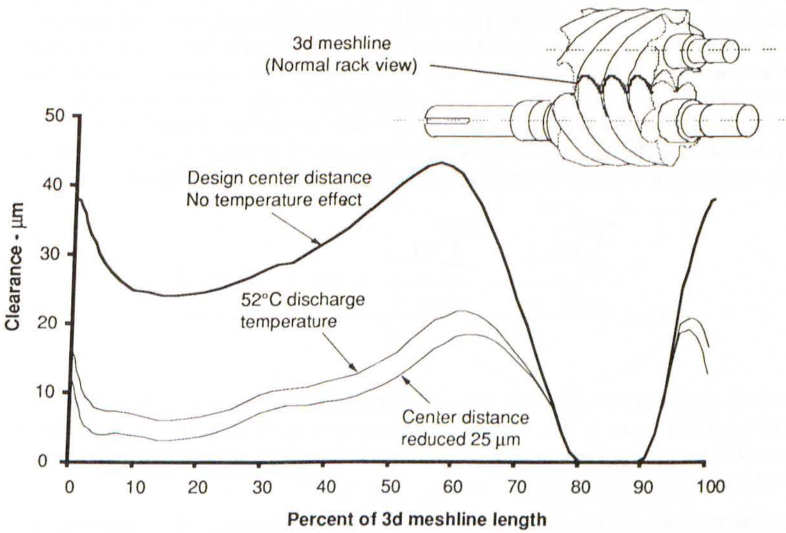


Bild 7 / Figure 7

The clearance calculation methods described in section 3 were used to determine the actual clearances for the 164 mm, R-22 compressor tested to measure the clearance effect on performance. The experimental program is described in section 4.

4. Experimental Program

A 164 mm R-22 compressor was used to test the effects of critical clearances on performance. During this program, the radial clearances between the rotors and the housing bores and between rotors and slide valve were varied in several ways. In addition, the effects of discharge end clearance and intermesh clearances were studied.

The first test was used as a baseline and no special clearances were set...the compressor was run using standard parts; clearances were determined using the assembly analysis described in section 3. Clearances between the rotors and housing and between the rotors and the slide valve were then varied. First, material was removed from the slide valve surfaces to add approximately 127 µm of radial clearance. The rotors and housing bores were unchanged. The next test was to increase the discharge end clearance (both rotors) by 40 µm. Then, the rotors were ground to a smaller diameter to increase the average clearance. Following this test, a set of offset dowel pins was constructed to shift the rotors towards the slide valve. The average clearance (difference in size of the rotors and housing bores) was not changed. Finally, the center distance was varied to test the effect of rotor intermesh clearance.

During the tests of the effect of radial clearance, the changes in clearance were complex. The clearance between the rotor and housing bore was different than between the rotor and slide valve and the clearances varied along the length of the rotor housing and circumferentially around the rotors. Tests showed that the compressor was, as would be expected, more sensitive to clearances between the rotor and the slide valve as this is the high pressure region. The clearance between the rotors and housing bores in the vicinity of the slide valve and especially near the discharge end also have a larger effect. A "representative clearance", Clr_r, was devised for the tests to allow a single clearance value to be assigned to each configuration. The representative clearance is defined as the average of [1] all clearances measured between the rotors and the slide valve and [2] the clearances between the male and female rotors and the housing measured at the discharge end (measurement plane S in Figure 1) at the top of the housing; this location is shown in Figure 8B. Calculation of Clr_r is illustrated below:

$$Clr_r = \frac{\dfrac{\sum_{n=1}^{npts\_m} clr_{sv,m}}{npts\_m} + \dfrac{\sum_{n=1}^{npts\_f} clr_{sv,f}}{npts\_f} + clr\_t_m + clr\_t_f}{4}$$

The various compressor builds where the radial clearances are varied and test results for each are documented in Figure 8. Test results for a pressure ratio of 4:1 at 21 and 27 bar discharge pressure (P2) are shown. The compressor was run in R-22 at 3480 RPM with a male rotor tip speed of 29.8 m/sec.

The representative clearance for the compressor with slide valve unloader is shown to be a good measure of the clearance effect. The radial clearance effect is to cause an 0.5% loss in efficiency for each 10 µm increase in clearance. Furthermore, the tests show that the changes in radial clearance have very little affect on the delivery or volumetric efficiency of the compressor.

It is also useful to compare pairs of tests. Builds 1.35 and 1.4 have the same clearances between the rotors and the main housing both as measured by the average clearance and the clr_t value. However, the average clearance between the rotors and slide valve was increased by approximately 130 µm for build 1.4, resulting in a 3.5% loss in efficiency.

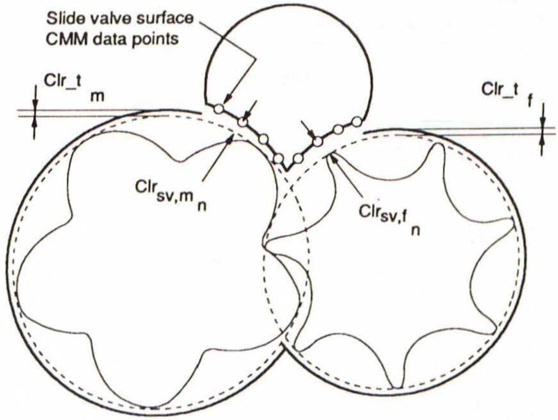
For build 1.6, the rotors were ground to a smaller diameter and the high clearance slide valve used for builds 1.4 and 1.5 was replaced. As a result, the overall average and slide valve clearances increased relative to build 1.35, although only a small amount...the representative clearance was 10 µm greater. Compared to build 1.35, the performance was unchanged for 21 bar discharge pressure and down 1% for the higher P2.

Build 1.7 uses the same parts as 1.6. However, eeccentric dowel pins were used to raise the discharge end of the rotors to reduce the slide valve clearance and the clr_t values while keeping the same average clearance. This change resulted in a 1% improvement in overall efficiency.

Changes in two other critical clearances were made during the test program. After tests of build 1.4, build 1.5 was made by increasing the discharge end clearance between the rotors and bearing housing by 35 µm. The center distance between the rotors was increased by 41 µm after tests of build 1.7 to make build 1.8, the last configuration tested.

Table 1 gives the results of the end clearance tests, comparing builds 1.4 and 1.5. Table 2 compares builds 1.7 and 1.8 to show the center distance effect. The data shown is for operation

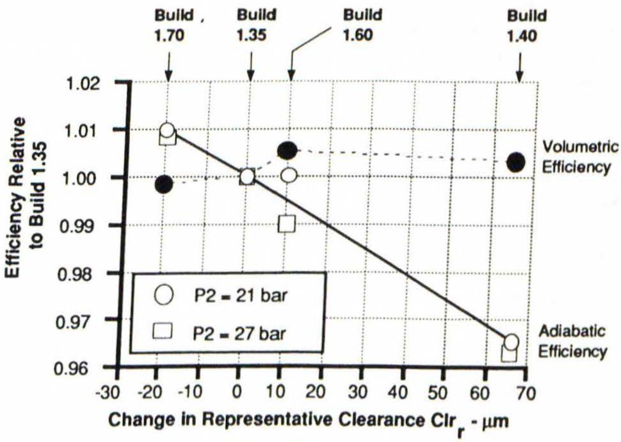
(A)



(B)

Build	Rotor Housing to Rotor: Overall Average – Male	Overall Average – Female	Clr_t at Plane S – Male	Clr_t at Plane S – Female	Slide Valve to Rotor: Average of all CMM data – Male	Average of all CMM data – Female	Clr_r
1.70	74	94	38	38	119	114	77
1.35	51	81	51	64	130	145	97
1.60	. 79	99	76	76	142	135	107
1.40	51	81	51	64	264	270	162

(C)



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(A) Definition der Spaltes — Clearance Definitions
(B) Spalte der gepruften Konfiguration — Test Configuration Clearances (µm)
(C) Gemessener Einfluß des Radialspalts — Measured Effect of Radial Clearance

at 21 bar discharge pressure at a pressure ratio of 4:1 and a male rotor tip speed of 29.8 m/sec. The change in discharge end clearance is known directly as it is determined when the compressor is built. The rotor mesh clearances, however, must be computed from the known center distance and cold rotor clearances using a model for the effect of operating pressures and temperatures. The rotor set tested had an average mesh clearance of 63,0 µm when inspected at a center distance 50 µm above nominal. The rotor profile design program was run with modified male and female profiles to give the same clearance pattern as that measured. The model was then run with the modified profiles, but at increased center distance and elevated rotor temperatures computed using the thermodynamic simulation and a bearing and shaft analysis program. This analysis was done for both builds 1.7 and 1.8...the clearances in Table 2 are the average clearances (total leakage area / total length of the 3D contact line) computed by the rotor profile design program for each of these builds.

Table 1
Discharge End Clearance Effect

Build	Discharge End Clearance - µm	Efficiencies* Adiabatic	Volumetric
1.4	36,5	0.680	0.851
1.5	71,5	0.671	0.850

Table 2
Rotor Mesh Clearance Effect

Build	Center Distance - mm	Mesh Clearance - µm	Efficiencies* Adiabatic	Volumetric
1.7	127.85	12,7	0.689	0.844
1.8	127,87	48,3	0.679	0.828

* Includes effect of inlet gas-cooled motor, oil separator and piping losses

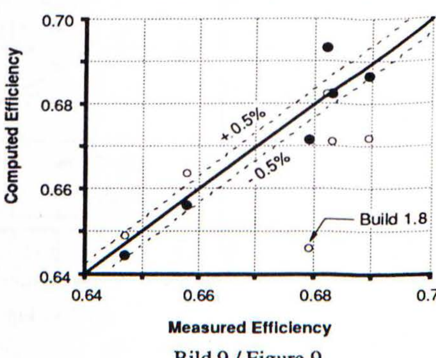
The tests show that both of these clearances have an affect on efficiency. For discharge end clearance, the loss in adiabatic efficiency is 0.4% for each 10 µm increase with almost all of the change coming from increased power...volumetric efficiency is not affected by the change. For the rotor clearances, the loss is the same - 0.4% for each 10 µm increase, with this change coming from loss in volumetric efficiency.

5. Comparison with Computation

Figure 9 shows computed and measured efficiencies compared. The open symbols are for leakage computations using a constant flow coefficient, α, of 1.0 for radial and intermesh leakages, where

$$\alpha = \frac{\text{Actual mass flow rate}}{\text{Isentropic flow rate}}$$

Agreement with the data is fair, except for build 1.8. Here, the computed loss in capacity due to the increased mesh clearance was too high. Peveling, in



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reference /3/, shows that the leakage flow coefficients can be less than 1.0 and will vary according to the leak path geometry, the actual clearance and the Reynold's number of the leakage flow. Variations in flow coefficient as described in this reference (Figure 7.9 for intermesh clearances and Figure 7.10 for radial clearances) were used in a new calculation. For build 1.6, coefficients were determined that provided good agreement with the data. Then, computations were carried out allowing the coefficients to vary for the other builds. This comparison is plotted in Figure 9 with the solid symbols. Using the variable coefficients gave much better agreement with data. The average error for constant α is 1.8% but is only 0.7% for the variable α case. However, the model does not predict all of the cases well, probably because the leakage flow rate, leak path geometry and clearance are varying throughout the compression cycle. A single clearance, single coefficient model is not sufficient to describe such a complex situation.

5. Conclusions

The computerized assembly analysis procedure, which allows computation of all critical clearances in any compressor, has proven to be a valuable tool. For the clearance study reported here, the program allowed an accurate documentation of all test compressors, leading to a clearer understanding of the test results. Using the spreadsheet for the assembly program allowed for easy use and input since data forms could be created. Output is easily formated and tables and graphs are available. The original program did prove to be difficult to maintain as the calculations were distributed throughout the spreadsheet and it was impossible to get a complete program listing. Newer versions of the EXCEL™ spreadsheet have been used to create a simpler version now in use and we continue to document our tests using the assembly log.

Data for the 164 mm compressor with the rack-generated rotor profile showed generally expected trends for the radial and intermesh clearances. The sensitivity to end clearance was greater than expected. However, the effect of end clearance as reported here has been duplicated in other tests of the 164 mm compressor, a 142 mm version using the same rotor profile and a 127 mm compressor with a newer generation of the rack-generated rotor profiles. This work led to methods for design, tolerancing and manufacturing of parts to minimize the rotor pair intermesh clearance and the clearances between the rotors and slide valve. In addition, assembly procedures were developed to insure setting of the proper discharge end clearances for good performance and reliability.

Analytical studies coupled with the experimental program show that reasonably good agreement with data can be expected if actual running clearances are known. Even a perfect thermodynamic model cannot produce proper results if the geometry of the compressor is not accurately described. Adding the modified flow coefficient model gave even better results. However, even with this improvement, the model is still calibrated to test results. Since this work, the program has been improved by allowing more complete definitions of clearances. Radial clearances can vary throughout the compression and variations in intermesh clearance from the inlet end to the discharge end can now be modeled. Plans for future work include adding a calculated flow coefficient based on these experiments and on the work reported in /3/.

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