

Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

2012

Incorporating 3D Suction or Discharge Plenum Geometry into a 1D Compressor Simulation Program to Calculate Compressor Pulsations

Jack Sauls
jsauls@trane.com

Keith Novak

Follow this and additional works at: <http://docs.lib.purdue.edu/icec>

Sauls, Jack and Novak, Keith, "Incorporating 3D Suction or Discharge Plenum Geometry into a 1D Compressor Simulation Program to Calculate Compressor Pulsations" (2012). *International Compressor Engineering Conference*. Paper 2043.
<http://docs.lib.purdue.edu/icec/2043>

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at <https://engineering.purdue.edu/Herrick/Events/orderlit.html>

Incorporating 3D Suction or Discharge Plenum Geometry into a 1D Compressor Simulation Program to Calculate Compressor Pulsations

Jack SAULS^{1*}, Keith NOVAK²

Ingersoll Rand - Trane
Compressor Development and Technology
3600 Pammel Creek Road; La Crosse, WI 54601 USA

¹Email: jsauls@trane.com, Phone: 608-787-2517

²Email: keith.novak@trane.com, Phone: 608-787-4105

*Corresponding Author

ABSTRACT

A 3D calculated impedance transfer matrix method has been developed and demonstrated to provide accurate and efficient prediction of pressure pulsations in positive displacement compressors. The method is easily incorporated into 1D compressor simulation models, where a single impedance matrix, normalized to density and speed of sound, can be used to model all compressor running speeds and operating conditions.

The general transfer matrix or “4 pole” method is documented by Bilal, et al (2010) where the hybrid approach used here is also introduced. More recent developments of this method are presented by Novak and Sauls (2012). This paper describes how the model has been integrated into a family of 1D thermodynamic simulation programs used for design and analysis of reciprocating, scroll and screw compressors. A review of how the model compares to test results for a twin screw refrigeration compressor is provided. In addition, we show one example of the type of design study that can be carried out using the 1D simulation with the hybrid model.

1. INTRODUCTION

So called one-dimensional thermodynamic simulation models are often used in the design and analysis of positive displacement compressors. It is at once an advantage and a weakness of these tools that they model the compressors using simplified representations of geometry, and tribological, fluid- and thermodynamic processes. Models have generally improved over the years and are often supplemented with empirical factors that render them more accurate, at least for compressors that are appropriately similar to those used to provide data for the models. However, the simplifications ultimately limit the ability of the programs to represent reality; this is a particular problem when it comes to the issue of pressure pulsation in the static flowpath components upstream and downstream of the primary compression elements – the piston/cylinder/valve assembly in a reciprocating compressor, fixed and orbiting involutes in the scroll and the meshing rotor pair of a screw compressor.

The first issue is that the problem is dynamic. Compression process analyses in virtually all contemporary thermodynamic simulations are accomplished by solving appropriate differential equations as initial value problems with time (or a surrogate such as crank angle along with a prescribed running speed) as the independent variable. To do this, rather more detail is provided for the geometry of the compression mechanism than for other components modeled, but the geometry representations are, nonetheless, 1D or perhaps “quasi-2D.”

Several pulsation analysis procedure options and how they might be used in design and analysis of screw compressors are reviewed by Koai and Soedel (1990). Here, the basic principles of the so-called 4 pole method are explained. The methods require determination of a transfer function relating mass flow rate and pressure; there are various ways to compute the required matrix of coefficients. These range from a lumped parameter model where the

flowpath is considered a collection of volumes and interconnecting restrictions (or “necks”) to use of a finite element model of the actual three-dimensional geometry to compute the required coefficients.

A method based on 3D models of the static flowpath elements was presented by Bilal, et al (2010). In that report, the authors approximated the actual geometry of the suction side flowpath using an assembly of interconnected tubes and volumes. This approach has the advantage of simplicity since the transfer functions required for the calculations had been derived for the pipe and volume elements. It was determined early on, however, that the complex geometries found in many positive displacement compressors did not lend themselves to being modeled as a collection of simple, elemental shapes. Considering the complexity of the geometry and the observations on the various methods provided by Koai and Soedel, this is not too surprising.

As a result of the experience and observations from the literature, we elected to build a simulation procedure using the 4 pole method with transfer function data computed from finite element analysis of actual flow path geometries. This paper is one of two describing continuing development of the “hybrid approach” to pulsation modeling and its application to positive displacement refrigeration compressors. Section 2 of this paper outlines development of the modeling approach, focusing on its inclusion in compressor thermodynamic simulations. Specific application in a screw compressor simulation is reviewed in Section 3, where we offer comparisons to experimental results. Section 4 illustrates application of the modeling approach by way of a design study looking at the effect of screw compressor discharge porting. Concluding remarks are offered in Section 5.

2. REVIEW OF THE MODELING APPROACH

The 4 pole method is well documented; cf. Singh and Soedel (1979) and Soedel (2006). Development and validation of a hybrid approach to pulsation modeling is described work by Bilal, et al (2010) and Novak and Sauls (2012). In the latter, a companion to this paper, examples are provided for a single cylinder reciprocating compressor. The method itself, however, is not limited to a particular type of compressor and once it had been developed and validated for the reciprocating case, it was applied in simulations for both screw and scroll compressors.

Figure 1 shows a schematic of the set of thermodynamic simulations we use for positive displacement compressor design and analysis. There are three separate programs, one for each compressor type. The compressors share many component types such as bearings and motors. There are generic models to compute loads, losses, etc. for these components and these models are available in the same form to each of the programs. This is the level at which the hybrid pulsation model exists. The module was written in a way as to not include any compressor-specific information or analysis methods. In addition, as shown in the bottom row of the chart, there are fluid flow, thermodynamic and cycle analyses available to the component models or the main programs as required.

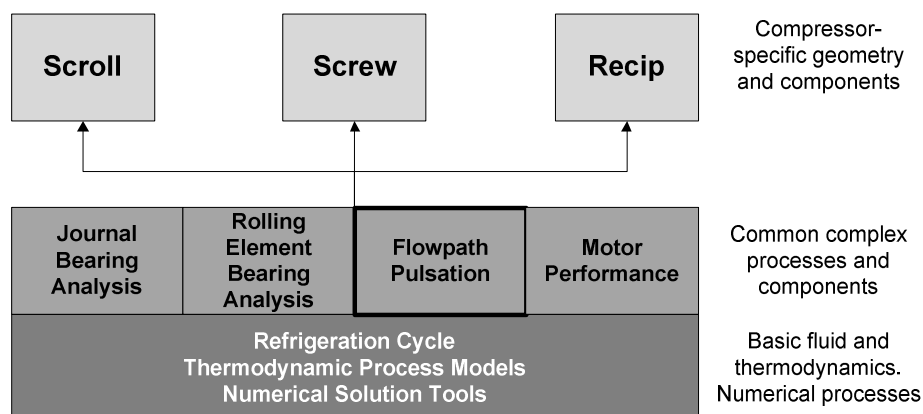


Figure 1: Positive displacement compressor thermodynamic simulation tools

The pulsation module accepts input from the simulation program that describes the variation in mass flow rate with time at one face of the flowpath for which pulsation is to be computed. Current values of average fluid properties at

the inlet are also provided along with flags controlling some calculation and output options. The flowpath's impedance matrix data, which was computed using a finite element analysis as described in Novak and Sauls (2012), is stored in a file whose name is passed to the pulsation module.

Pressures at the flowpath inlet, flowpath exit and at user specified locations within the calculation domain are returned to the simulation program. The actual process is iterative: the pulsation is driven by the time variation of the mass flow entering the domain and the variation in flow rate is affected by the pressure pulsation at the interface with the compression element. This is simply illustrated in Figure 2. In this case, only one pulsation – that at the compressor discharge – is shown. However, the model can be used for any connection to the compression elements where the mass flow rate varies with time and where the connected flowpath has been modeled to generate the impedance matrix data. The iterative process starts with the pressure (1) set to a uniform value. Executing the calculations for the compressor (2) results in a mass flow rate at the discharge port (3) which varies with time. This information is used in the pulsation calculations (4) to update the pressure characteristic (1). The process is repeated until convergence, signaled by a minimal pressure change between iterations, is reached.

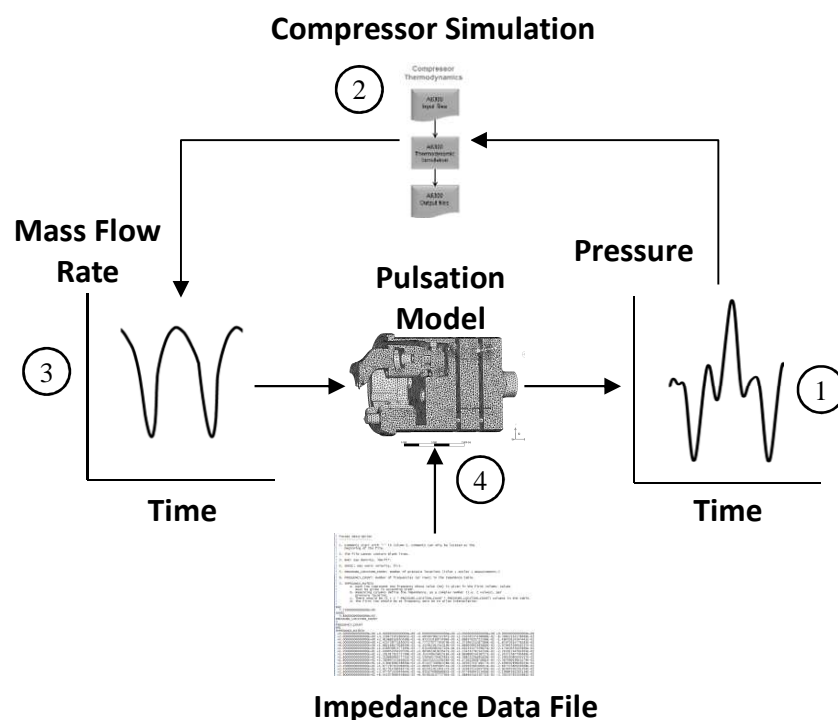


Figure 2: Iterative solution process for analyses with pulsation

Experience to date with using the pulsation module for analysis screw compressor discharge pressure pulsation has been positive. There have been some issues with convergence although the approach reported by Zhou, et al (2001) has made the analyses more robust.

Because the impedance matrix coefficients can be normalized to the density and speed of sound of the fluid entering the domain, the 1D calculation can be run at any condition, thus allowing us to derive the benefit of the simple simulation approach of rapid analysis of many operating conditions, speeds and/or design options. This does not, of course, extend to the modeling of the flowpath elements for which the pulsation is calculated. As has been noted, we found it necessary to model the actual 3D flowpath geometry and use FE analyses to determine the impedance matrix. Nonetheless, the hybrid approach offers a reasonable balance between modeling effort, execution speed and accuracy of results.

Once the thermodynamic simulation program with the pulsation model included was verified, we built a model of an existing screw compressor for which discharge pressure pulsation data was available. Results of the comparison between the model and these tests are provided in the next section.

3. TEST AND ANALYSIS FOR A REFRIGERATION SCREW COMPRESSOR

In order to validate the model's ability to represent actual compressor performance, we ran simulations of a compressor for which extensive test data was available. The compressor was designed for operation with refrigerant R134a and has a displacement of 525 m³/hr at its rated speed. Both the axial and radial discharge ports were designed for a 3.1 built-in volume ratio. The compressor was tested at speeds over a range of about 40% to 110% of nominal. Several dynamic pressure transducers were installed in the discharge flowpath to measure the discharge pressure pulsation characteristics.

Figure 3 shows the comparison between tested and calculated pressure pulsation in the discharge plenum of the compressor for three operating speeds as indicated. The pressure and time scales are the same for each of the speeds. All comparisons presented here are for operation at 3.9:1 pressure ratio.

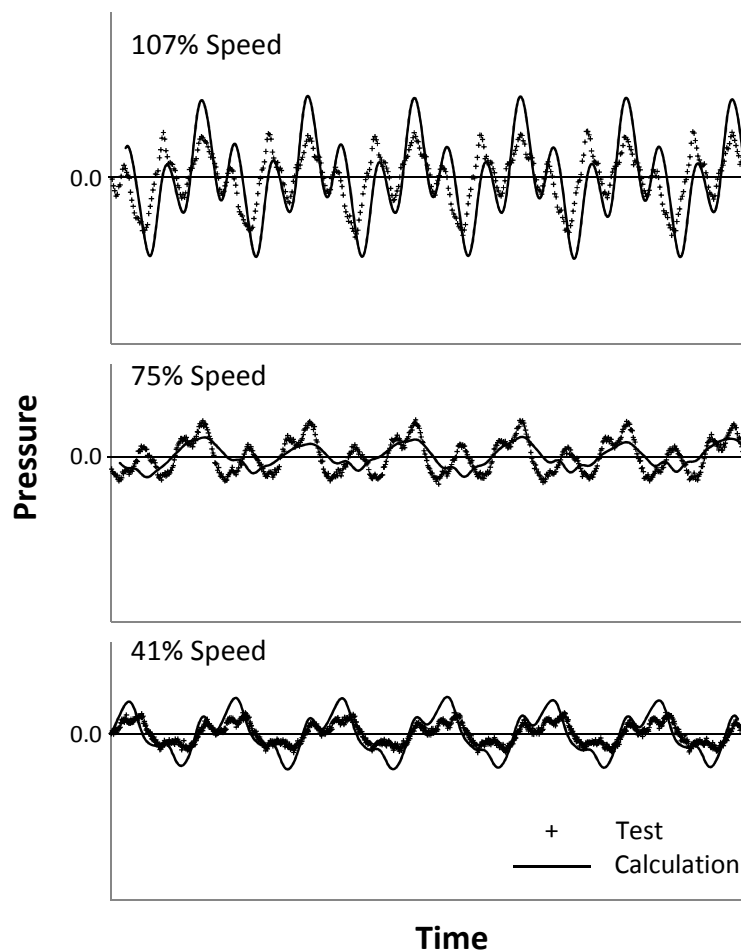


Figure 3: Screw compressor discharge pressure pulsation – comparison of calculated and measured results

The agreement between test and calculations is fair, although not as good as was seen in the case of the reciprocating compressor reported by Novak and Sauls (2012). In addition to the first order effect that the model is a relatively simple numerical representation of a complex dynamic process, there are factors at play specific to this particular

comparison. Most notably, the test data was acquired during a speed sweep. Since the compressor was not actually set to the standards of “steady state” operation used when performance data is acquired, there is a degree of uncertainty as to details of the actual operating condition.

Sonic velocity can have a significant effect on the calculated results, especially if the compressor’s fundamental operational frequency or its harmonics are near a resonance of the flowpath. This effect could be studied easily with the hybrid model coupled with the thermodynamic simulation. One of the features of the hybrid approach is that the impedance matrix is essentially normalized to the speed of sound and density. This allows for the matrix to be corrected to the actual operating conditions of the analysis. Hence, it is a simple matter to run the analyses over a range of operating conditions to vary the temperature entering the discharge plenum. This was done for the 75% speed case by calculating the pulsation over a range of compressor inlet temperatures while maintaining the same inlet and discharge pressures.

The analysis that generated the results shown in Figure 3 was run at a compressor inlet temperature 8.3°C above the saturation temperature of the refrigerant. A series of runs were subsequently made for different levels of superheat, varying from 11.1°C down to 0°C. Computed pulsation characteristics were noticeably affected; the case for 0°C superheat applied to the analysis at 75% speed is shown in Figure 4 where the time scale is magnified to show only a single cycle at the rotor lobe passing frequency.

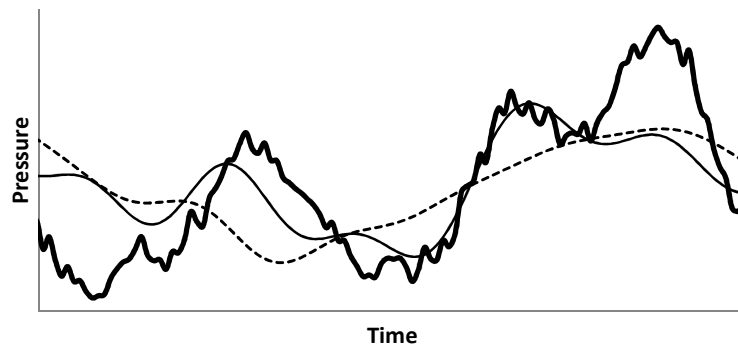


Figure 4: Example of the effect of temperature on computed pulsation characteristics

The heavier weight solid line shows measured data. The dashed line is the original computed pulsation as shown in Figure 3. The lighter weight solid line are results at the reduced operating temperature. The sonic velocity at the point entering the discharge flowpath in this case is 142.7 m/sec compared to 147.3 m/sec of for the original case.

Repeating the temperature sensitivity study at the 107% showed similar sensitivity in that there were some marked differences due to the temperature of the analysis; in that case, the original analysis as shown in Figure 3 provided the best agreement with data. At 41% speed, however, the temperature level had almost no effect at all on the calculated results.

Our broad conclusion based on this comparison is that the model is capable of fairly representing reality. As a result, we feel it should be useful in design optimization studies and for quantifying pulsation and resulting performance effects as affected by variations in operating conditions such as pressure ratio and running speed. An example of such a design study is given in Section 4.

4. SCREW COMPRESSOR DESIGN STUDY

One benefit of the transfer function approach is that it can be embedded in the thermodynamic simulation which models the compressor with an extensive set of relatively simple component and fluid flow models. Such simulations have been used by many commercial and academic organizations to successfully identify good quality designs. They are fast and allow designs to be defined with relatively few inputs. The method we have chosen for pulsation calculations allows for these benefits, with one exception – the exploration of the effect of flowpath

geometry itself. The hybrid method requires a detailed model of the three-dimensional geometry and building this cannot be incorporated into the process.

Nonetheless, there are many studies that can be carried out where the flowpath geometry is not a design variable. In this section, we present the results of a screw compressor study where the sizes of the two discharge ports, the radial port in the main housing and the axial port in the discharge housing, are varied independently. The port size is indicated by the volume ratio associated with the start of porting for the port in question. The original design has both the axial and radial ports sized for setting a 3.1 built-in volume ratio.

In this study, the two ports were varied over a range of volume ratios from 2.9 to 3.6 in increments of 0.1, resulting in 64 combinations of axial and radial port size. Eight of these had the ports at the same size, with radial/axial volume ratios of 2.9/2.9, 3.0/3.0, etc. For the rest of the cases, the ports were each associated with different volume ratios: 2.9/3.6, 2.9/3.5, ..., 3.0/3.6, ..., etc. All of the calculations were carried out at 107% running speed at a pressure ratio of 3.9:1 with 8.3°C superheat at the compressor inlet.

Each case was run first without pulsation then again with the pulsation model active. Finally, after examination of the results from these two analyses, we input a specific pressure pulsation characteristic to demonstrate what could be achieved IF the characteristic could be achieved with proper selection of design parameters. Results are shown in Figure 5 and discussed in the remainder of this section.

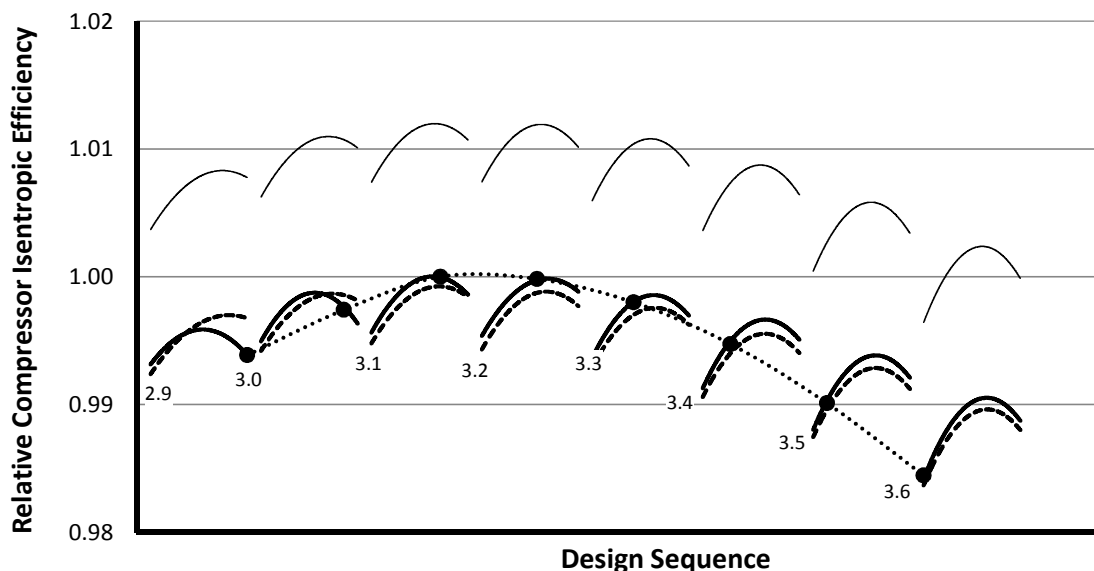


Figure 5: Calculated effects of compressor volume ratio options on overall performance

There is a lot of information in Figure 5. First, note that all of the efficiencies are normalized to the highest value calculated for the case where the pulsation model was used in the analysis. There are eight sets of curves, each representing a fixed radial port geometry, identified by the volume ratio associated with that particular port – the volume ratios are labeled under the curves. Each curve is a line through eight points, each of these representing a different axial port. The far left side of each curve is for an axial port volume ratio of 3.6; the right hand end is for the 2.9 volume ratio case. The data are plotted against case number – a sequence running from 1 to 64.

The heavier weight solid lines are results for computations including pulsation; the dashed lines are for cases without pulsation included in the analysis. The dotted line through the dark symbols is the locus of points where the radial and axial ports are at the same volume ratio. Finally, the lighter weight lines at higher efficiency show the results for the imposed “improved” pulsation which is the same for each case.

Comparing the cases run with and without the pulsation model activated reveals some interesting characteristics. First, while the efficiency levels are very slightly higher, there is very little effect of pulsation on the characteristics

themselves, except maybe for the case of 2.9 radial port volume ratio. The highest efficiency is with the 3.1/3.1 port combination in both cases and the overall effects of volume ratio variation are essentially the same. This was a bit surprising considering the fact the pulsation levels varied considerably with configuration. Figure 6 shows the overall magnitude of the pulsation defined as maximum pressure – minimum pressure evaluated in the time domain; the results are normalized to the highest pulsation level which occurs with the 2.9/2.9 configuration.

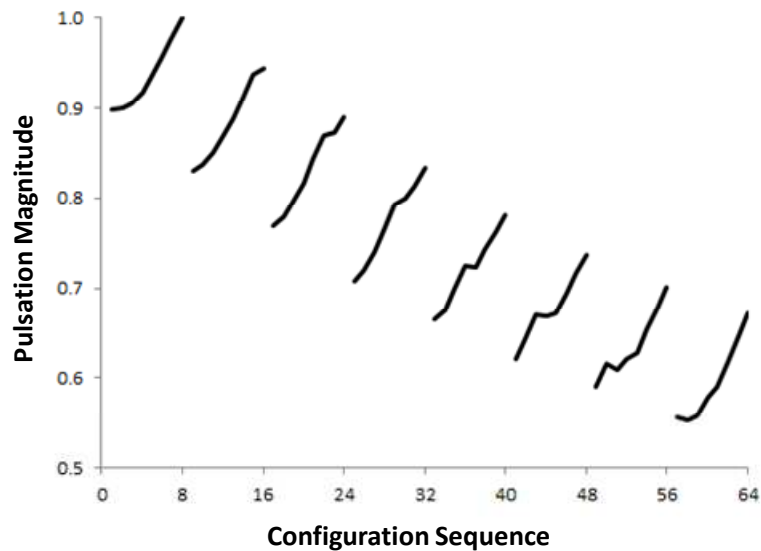


Figure 6: Magnitude of discharge pressure pulsation, normalized to maximum value computed

Efficiency seems relatively independent of pulsation magnitude as defined. Rather, volume ratio effects are dominant. The peak efficiency overall occurs with both ports at a volume ratio of 3.1. For other radial volume ratios, the best efficiency is found with axial ports at a different volume ratio. In fact, each curve representing a fixed radial port volume ratio reaches its peak efficiency at an axial port volume ratio of 3.1.

Figure 5 shows use that including pulsation results in a small efficiency increase. This is because the interaction between the time varying discharge pressure and pressure in the rotor interlobe space exhausting into the discharge flowpath results in a small reduction in overpressure losses. Details of this effect are highlighted in Figure 7.

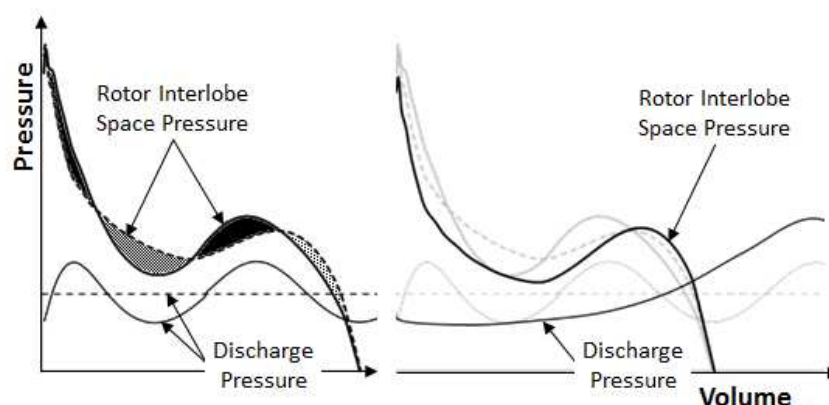


Figure 7: Screw rotor discharge process shown in a pressure-volume chart (indicator diagram)

The figure shows pressures plotted against the volume of the screw rotor pair interlobe space, the familiar pressure-volume (PV) or “indicator” diagram. The area under the rotor pocket pressure curves represent work done on the fluid, so differences in multiple curves shown in the chart can be quantified in terms of differences in power

required to drive the compressor. The chart on the left side of the figure compares the cases for radial and axial port volume ratios of 3.1/3.1, the highest efficiency case. The dashed lines are for the case with no pulsation; solid lines are results from calculation with pulsation. The differences due to pulsation are highlighted by the shaded areas. The lighter areas are where there is less power associated with the case with pulsation, the darker areas are where there is more power used than in the no pulsation case. The net difference in area is a small benefit in favor of the case with pulsation; the amount of the difference is exactly proportional to the difference in efficiency.

On the right side of the curve, we show an “optimized” pulsation. In fact, this is simply an arbitrary input created to force a more favorable discharge process and to quantify, more or less, the potential for pressure pulsation control to provide a performance benefit. The results of imposing this particular pressure pulsation are illustrated in Figure 5 (the lighter weight solid lines at higher efficiency levels) where there is a calculated performance improvement of just over 1%.

This last exercise simply illustrates the potential for tailoring pulsation to provide a benefit. Whether such other benefit can be achieved in practice requires a design study that includes evaluation of alternative discharge flowpath geometries and variation in other compressor design features.

5. SUMMARY AND CONCLUSIONS

The hybrid approach as we have developed and applied it for pulsation calculations for our positive displacement refrigerant compressors has proven to compare quite favorably with experimental results from a reciprocating compressor. The agreement with test data from a screw compressor was not quite as good, perhaps a result of the four pole method’s limitations in simulating effects at higher frequencies and to the uncertainty in the actual discharge temperature coupled with the computed results’ sensitivity to this factor. All-in-all though, we consider the approach to be well suited for compressor design and application studies.

The method has been implemented in a stand-alone MATLAB model and as a module that can be connected to any of our three positive displacement compressor thermodynamic simulation programs. Thus, we can take advantage of the method’s accuracy in an environment that allows for rapid calculation of the effects of operating and design parameters when the flowpath for which pulsations are computed is defined. Changing the design of these flowpath elements does, however, require a more time consuming process of modeling and execution of the finite element analysis.

The simple design study carried out for an R134a screw compressor produced the somewhat surprising result that discharge plenum pulsation (for the compressor as designed) seemed to have little effect on overall isentropic efficiency. This is in spite of the fact that the level of pulsation in the different configurations varied by a factor of nearly 2:1. Built-in volume ratio effects were the primary factor in determining the efficiency variation characteristics which generally follow the pattern explained by the simplest of first-principles analysis as presented by Sauls (1982).

Further work with the models for efficiency improvement is planned. In addition, there is the question of how variation in pulsation might affect overall compressor noise levels. This was beyond the scope of the present work, but the procedures developed can likely provide some general insights. We will be looking into this area as well.

REFERENCES

- Bilal, N., Adams, D., Novak, K., Sauls, J., 2010, A Hybrid Approach of Calculating Gas Pulsations in the Suction Manifold of a Reciprocating Compressor; *Proceedings of the 2010 International Engineering Conference at Purdue*, Purdue University.
- Koai, K., Soedel, W., 1990, Gas Pulsations in Screw Compressors - Part II: Dynamics of Discharge System and Its Interaction With Port Flow, *Proceedings of the 1990 International Compressor Engineering Conference at Purdue*, Purdue University.
- Novak, K., Sauls, J., 2012, Comparing 3D Calculated Impedance Transfer Matrix Pressure Pulsations to CFD and Operating Compressor Pressure Measurements; *Proceedings of the 2012 International Engineering Conference at Purdue*, Purdue University.
- Sauls, J., 1982, Performance Characteristics of Positive Displacement Compressors, *Proceedings of the 1982 International Compressor Engineering Conference at Purdue*, Purdue University.
- Singh, R., Soedel, W., 1979, Mathematical Modeling of Multicylinder Compressor Discharge System Interactions, *Journal of Sound and Vibration*, vol. 63, no. 1: p. 125-143.
- Soedel, W., 2006, *Sound and Vibrations of Positive Displacement Compressors*, CRC Press.
- Zhou, W., Kim, J., Soedel, W., 2001, New Iterative Scheme in Computer Simulation of Positive Displacement Compressors Considering the Effects of Gas Pulsations; *Transactions of the ASME*, Vol. 123, pp. 282-288.