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Gas Pulsations in Screw Compressors - Part I: Determination of Port Flow and Interpretation of Periodic Volume Source

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GAS PULSATIONS IN TWIN SCREW COMPRESSORS -PART I: DETERMINATION OF PORT FLOW AND INTERPRETATION OF PERIODIC VOLUME SOURCE

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Abstract

Even in the absence of valves, flows through the suction and discharge ports of a screw compressor are oscillatory in nature. This paper utilizes the volume and port area curves based on rotor profiles and other geometric data for a typical twin screw compressor. The gas conditions and port flows are determined from thermodynamic analysis for a complete cycle of discharge and suction activity. This includes the conditions of under and over pressure. Typical Fourier spectra for port flow are presented.

Geometric Characteristics of a Twin Screw Compressor

The geometric property of a screw compressor can be represented by its volume curve, port area curve and seal line curves which are generated from the screw profiles and other fundamental geometric data. Figure 1 shows the fundamental procedures for obtaining these curves and Figures 2, 3, 4 & 5 show these curves for a typical 60 Hp twin screw compressor. The port area curve for a partial load is different from that at full load, due to the slide valve action which controls the load. The screw rotors of the example compressor are of the five male and six female lobe design and the wrap angle of the driving male rotor is 300 degrees.

Analysis of Periodic Port Flow

The following assumptions have been made to simulate the periodic flow through the ports:

(1) The fluid in each compression chamber is an ideal gas with constant specific heats C_p and C_v.
(2) No heat transfer occurs at the boundary of a compression chamber.
(3) Homogeneous distribution of the fluid properties.
(4) Quasi-steady change of the thermodynamic properties during the short time interval of a step.
(5) One dimensional isentropic flow through a port.
(6) The oil droplets do not mix well with gas in the chambers. The thermodynamic behaviors of the mixture can be well represented by the dry gas.
(7) The oil being injected into the chambers tends to accumulated on the outer boundary of the chambers due to centrifugal effect. Therefore most of the leakage paths are blocked by oil.

The chamber gas condition and the port flows are evaluated following stepwise the changes of the chamber volume and port areas which are obtained through the geometric analysis. By solving the simultaneous equations formulated based on the above assumptions, the gas state law, the mass conservation law, the first law of thermodynamics and the orifice flow consideration, we can calculate the gas condition in the chamber and flow through port for each step. Figures 6, 7, 8 and 9 show typical results of such an analysis for the example compressor mentioned above under the following conditions : Refrigerant gas R22 , male rotor rotation speed 59.2 cps , no blow hole and no leakage path, suction condition 83psia/55F and discharge pressure 241psia.

The total flow rate can be calculated by first multiplying the chamber mass flowing through the exhaust port by the rotation speed of the male rotor and the male rotor lobe numbers and then adding all flows from the chambers which are open to the exhaust port.

The pressure-volume diagrams under the three different loading conditions are shown in Figure 10. The area enclosed by each of these diagrams represents the theoretical work required to achieve compression per chamber. The power required to compress the gas is obtained by multiplying this work by the male rotor rotation speed and the male rotor lobe number.

The oscillatory flow at the exhaust port serves as an excitation to the discharge manifold where the pressure pulsation occurs. The shape of the volumetric flow at the discharge port as a function of time is critical in determining the gas pulsation in the discharge manifold and can be decomposed by Fourier expansion to obtain the magnitude of each harmonic component. Figures 11 and 12 show these pulsations in the time and frequency domains.

Table I lists typical results of the flow analysis of the compressor mentioned above under the specified inlet and outlet conditions of the compressor. The flow pulsation levels are the square root of the sum of the squares of the Fourier components at all harmonic frequencies.

Compressor load	100% load	75% load	50% load
Flow capacity (Kg/s)	1.404	1.048	0.820
Flow pulsation level (liter/s)	9.767	4.119	7.493

Table I: Typical flow performance of a screw compressor under various loads

Effects of Discharge Pressure and Leakage on Flow Pulsations

Besides the load effect under the same inlet/outlet pressure condition, we have studied the following:

(1) the effect of varying compressor outlet pressure on the flow pulsation of a twin screw compressor: By keeping the same condition of 83psia/55F at the compressor inlet, 100% load, and no leakage, the performance of the compressor as function of changing discharge pressure is shown in Table II and Figures 13, 14 and 15.

(2) the effect of various leakage levels defined by the average clearance at the seal lines: By keeping the same condition of 83psia/55F at inlet, 241psia at outlet and 100% load, the compressor flow pulsations as function of leakage are shown in Table III and Figures 16, 17 and 18.

From the observation of all the above conditions, we conclude that the most important factor which affect the flow pulsation is the pressure difference across the exhaust port at the start of the discharge process. This difference may be termed "overpressure" if positive and "underpressure" if negative. Generally speaking, the higher the overpressure level the higher the flow pulsation at the discharge port will be. A little underpressure will help reduce the flow pulsation. When the underpressure level is larger than a certain amount, it will have the similar effect as the overpressure condition. Also, at the beginning of the discharge process, there will be a small amount of reverse flow through the exhaust port whenever the underpressure condition occurs. It can be seen from Figure 12 that the pulsating flow level is the lowest at 75% load operation for the example compressor because its under/over pressure level is the lowest as can be seen from Figure 4 and 10. Similarly, the lowest pulsation occurs at a pressure a little lower than 310 psia which is the critical pressure of the chamber pressure where neither the overpressure nor the underpressure condition occurs(Figure 14 & 15). Also from Figures 16, 17 and 18, we see that the higher the overpressure magnitude, which is a function of clearance, the higher the flow pulsation will be.

Compressor outlet pressure	170 psia	241 psia	310 psia	350 psia
Flow capacity (Kg/s)	1.404	1.404	1.404	1.404
Gas pulsation level (liter/s)	16.618	9.749	4.308	4.648

Table II: Flow Pulsations at Various Discharge Pressures

Clearance	0.0 mm	0.03 mm	0.10 mm	0.30 mm
Flow capacity (Kg/s)	1.404	1.298	1.049	0.346
Gas pulsation level (liter/s)	9.750	8.812	6.618	3.329

Table III: Flow Pulsations for Various Leakage Clearances

References

(1) W. Soedel, "Introduction to Computer Simulations of Positive Displacement Type Compressors", Short Course Lecture Note, Ray W. Herrick Lab., Purdue University, 1972

(2) M. Fujiwara, etc., "Computer Modeling for Performance Analysis of Rotary Screw Compressor", 1984 International Compressor Engineering Conference at Purdue

(3) A. Fujwara and N. Sakurai, "Experimental Analysis of Screw Compressor Noise and Vibration", 1986 International Compressor Engineering Conference at Purdue

(4) S. Jonsson (SRM), "Performance Simulations of Twin Screw Compressors with Economizer," 1988 International Compressor Engineering Conference at Purdue

(5) P. J. Singh and G. C. Patel, "A Generalized Performance Computer Program for Oil Flooded Twin Screw Compressors", 1984 International Compressor Engineering Conference at Purdue

(6) Kwang-lu Koai and Werner Soedel, "Contribution to the Understanding of Flow Pulsation Levels and Performance of a Twin Screw Compressor Equipped with a Slide Valve and a Stopper for Capacity Control", 1990 International Compressor Engineering Conference at Purdue

Acknowledgement

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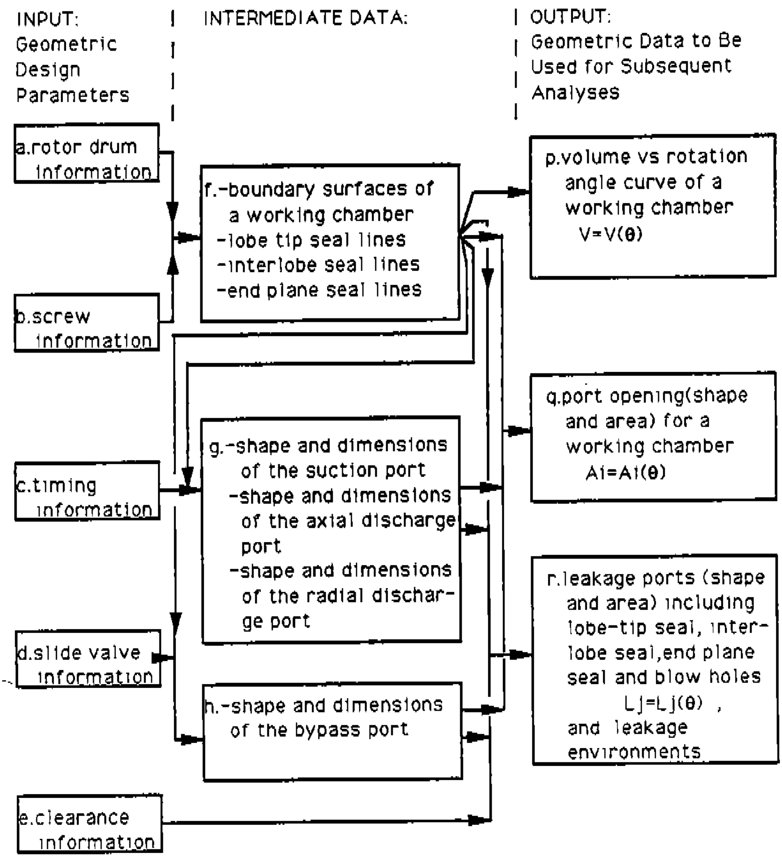

Figure 1: Geometric Investigation Procedure

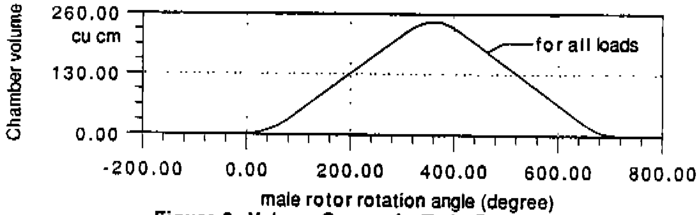

Figure 2 : Volume Curve of a Twin Screw Compressor

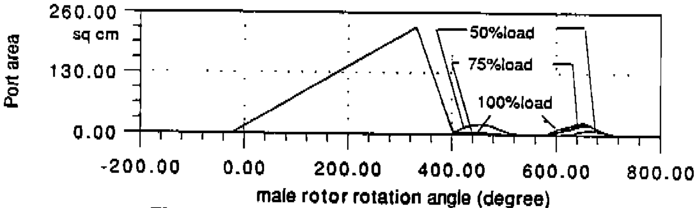

Figure 3 : Port Area Curve

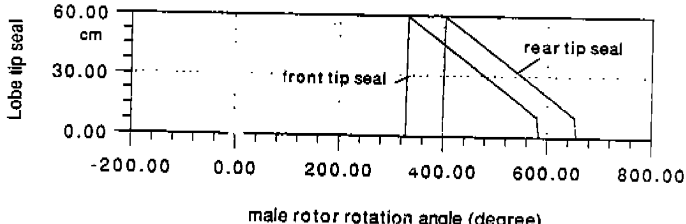

Figure 4 : Lobetip Seal Lines

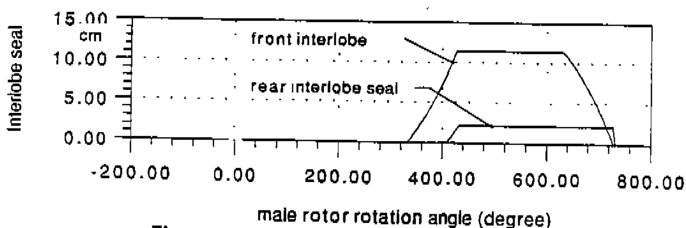

Figure 5 : Interlobe Seal Lines

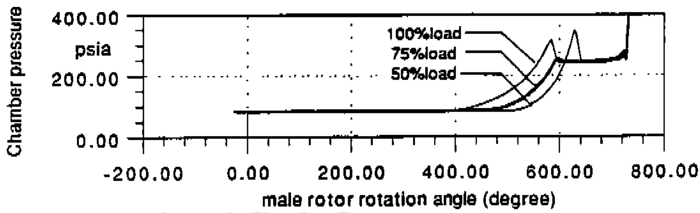

Figure 6 : Chamber Pressure

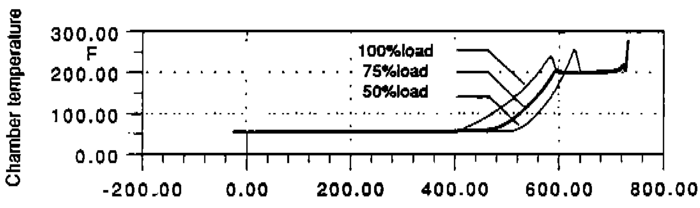

Figure 7 : Chamber Temperature

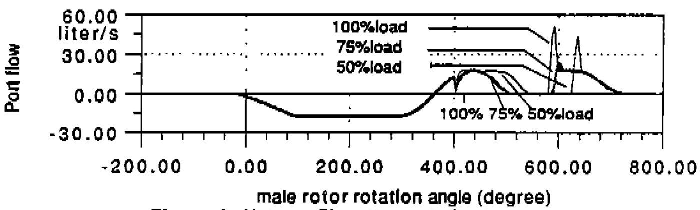

Figure 8 : Volume Flow through Port

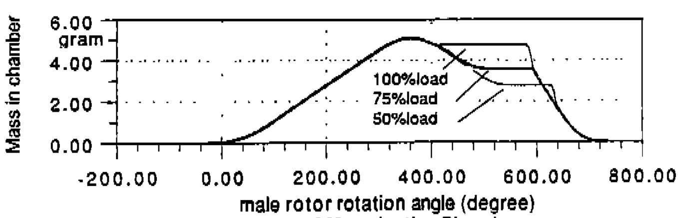

Figure 9 : Amount of Mass In the Chamber

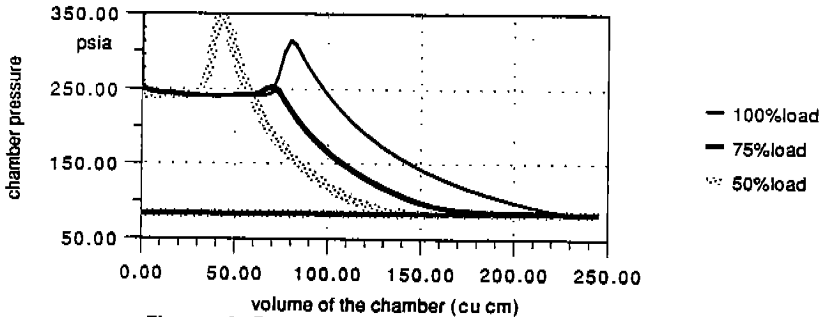

Figure 10: Pressure-volume Diagram at Various Loads(no leakage, constant inlet/outlet pressures)

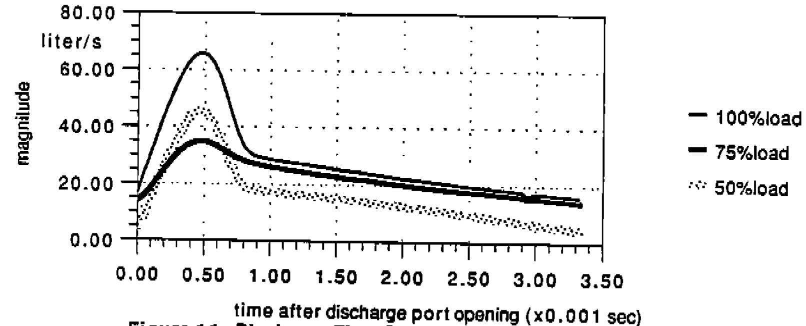

Figure 11: Discharge Flow Pulsation in One Period

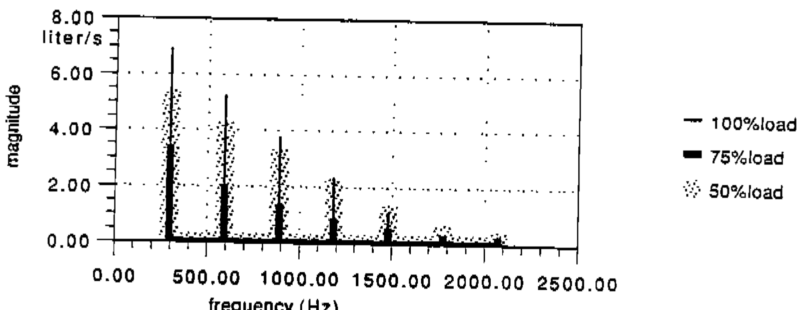

Figure 1 2 : Fourier Components of the Discharge Flow Pulsation

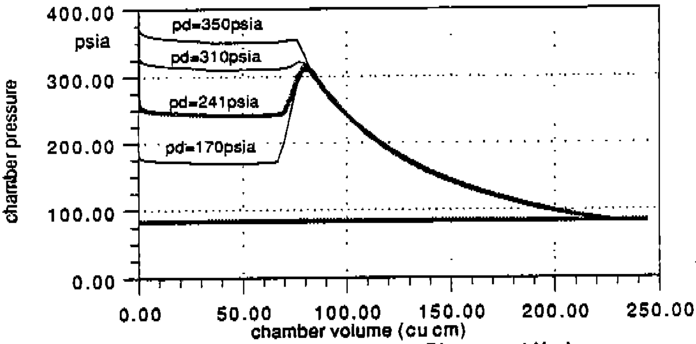

Figure 13: Pressure-volume Diagram at Various Outlet Pressure of the Compressor (full load)

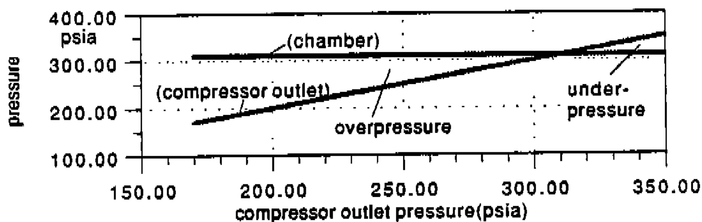

Figure 14: Over/under Pressure Condition at Various Outlet Pressure of the Compressor (full load)

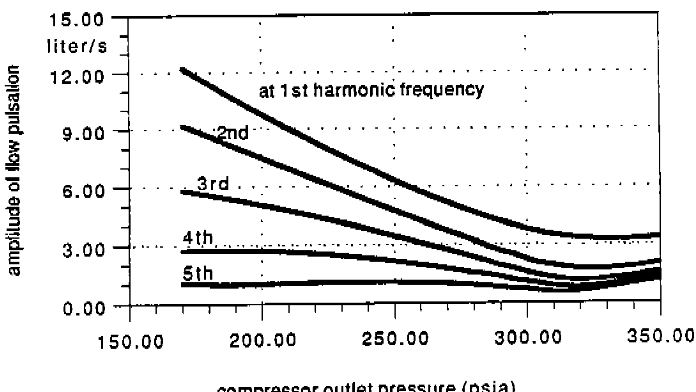

Figure 15: Flow Pulsation Amplitudes at Various Outlet Pressure of the Compressor (full load)

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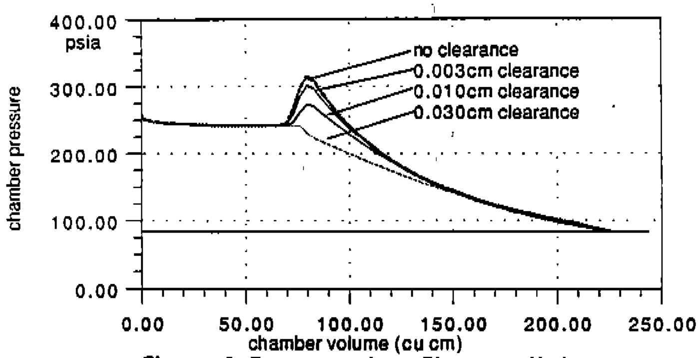

Figure 16: Pressure-volume Diagram at Various Leakage Clearances (full load)

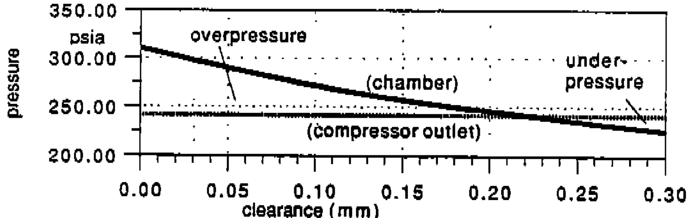

Figure 17: Over/under Pressure Conditions at Various Leakage Clearances (full load)

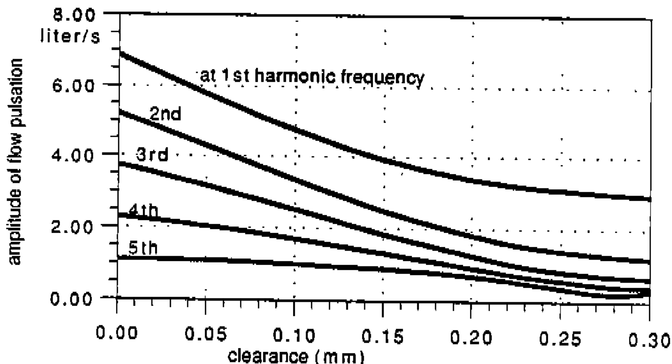

Figure 18: Flow Pulsation Amplitudes at Various Leakage Clearances (full load)