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Contributions 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

K. Koai
Purdue University

W. Soedel
Purdue University

CONTRIBUTIONS 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

by
Kwang-lu Koai and Werner Soedel
Ray W. Herrick Laboratories
Mechanical Engineering School
Purdue University
West Lafayette, Indiana 47907 U.S.A.

Abstract

This paper discusses from a theoretical viewpoint the flow pulsation of a twin screw compressor and how it is related to performance. The compressor is equipped with a stopper in addition to a slide valve for controlling its capacity and the compression ratio. The volume ratio(V_i), the flow capacity, the power consumption rate and the flow pulsation level are determined based on a geometrical characterization in terms of volume versus shaft angle as input and a thermodynamic analysis of a screw compressor for no leakage. Typical results are shown in a plot form for various operating conditions defined by the stopper stroke, the bypass port opening area, the driving rotor speed and the compressor discharge pressure.

Introduction

Traditionally the capacity of a twin screw compressor is controlled by a slide valve, which is a wedge shaped solid block constituting a part of the rotor housing. Activated by a control mechanism, the slide valve can move in the direction parallel to the axes of the rotors. Figure (1) shows the shapes of a typical slide valve and a rotor housing. At full load, the front end of the slide valve is in contact with its facing plane on the stationary part of the rotor housing and the medium in the compression chamber is totally enclosed after the suction port closes. The shape of the rear end of the slide valve is designed to allow the discharge of the compressed gas in the radial direction at a certain V_i ratio at full load. At partial loads, the slide valve is pulled a distance toward the exhaust end of the compressor and a bypass port is created in front of the valve, which leads the medium in the compression chamber to flow back to the suction side of the compressor. Also the radial discharge port on the slide valve moves by the same distance as the stroke of the slide valve. Both the flow capacity and the compression ratio change following the slide valve movement. Figure 2 shows the slide valve, the bypass port and the radial discharge port viewed from the compressor top.

A recent development in the compressor industry is to introduce a second slide valve or stopper in front of the original slide valve for better control of the screw compressor(reference 1). With the combined action of two valves, the control of the compressor is diversified. There is some interest in how screw compressors of this type perform at various operating conditions. This paper attempts to contribute to the understanding by way of a theoretical simulation and to extend the discussion by including flow pulsation level predictions.

Geometric Characterization and Thermodynamic Analysis

The volume curve and seal line curves of a screw compressor can be generated based on its screw profile and fundamental geometric data. If the dimensions and the strokes of the slide valve and the stopper are known, the port area curve can be determined too. The thermodynamic analysis of a screw compressor is conducted following stepwise the evolution of the compression chamber. By solving the simultaneous equations based on ideal gas and isentropic flow assumptions, the conservation laws and assuming no leakage, the gas conditions and the port flows can be calculated for each step. Figures 6, 7 and 8 and show the port area curves, the volume curves, the internal pressures, the port flows and the masses of the medium in the compression chamber of a typical 60 hp screw compressor at various valve positions as designated in Figures 3(b), 4(b) and 5(b). Other operating conditions are: R22 refrigerant as the medium, driving rotor speed 59. 2 cycle per second, and compressor inlet/outlet conditions of 83.2psia/55°F and 241psia/180°F. Figure 9, 10 and 11 show the corresponding pressure-volume diagrams. These diagrams and the calculated flow conditions determine the compressor performances. i.e. the effective volume ratio, the flow capacity, the power consumption rate and the flow pulsation level can be evaluated. For an ideal single stage refrigeration cycle, these performances represent: the chamber pressure when the discharge port opens, or equivalently the "overpressure" or "underpressure" level, the heat capacity, the inverse of the coefficient of performance(COP), and the pressure pulsations respectively under the operating conditions specified. Figure 12 shows such performances of the example compressor at different combinations of the stopper location and the spacing between the slide valve and the stopper. Other operating conditions are the same as before. Figures 13 and 14 are the same as Figure 12, except that the performances at different rotating speeds or compressor outlet pressures are examined.

Flow Pulsation and Performance

(1) With the introduction of a stopper in addition to the conventional slide valve, the control of a twin screw compressor can be achieved with more flexibility. The flow pulsation and the performance of the twin screw compressor vary as functions of the stopper stroke and the bypass port opening.

(2) The effective volume ratio(V_{ie}), which is the volume ratio with a density correction for the chamber volume at the closing of the suction port, is subject to the geometric limitations of the screw compressor design. As can be seen from Figures 12(a), 13(a) and 14(a):

A. The V_{ie} value can not exceed the limit set by the locations of the ending line of the suction port and the starting line of the axial discharge port on the rotor housing. This limit can occur only with the bypass port fully closed. Point A's in the figures correspond to the situation that the radial discharge port on the slide valve coincides with the axial discharge port. The radial discharge port disappears as the slide valve is pushed further toward the discharge side of the compressor. If the stopper and the slide valve together are pulled beyond point A away from the discharge side, the V_{ie} value drops gradually.

B. If the bypass port opens, the highest V_{ie} point occurs when the radial discharge port on the slide valve coincides with the axial discharge port too(points B,C&D). The V_{ie} value drops gradually when the slide valve is pulled or pushed away from this highest point. Maximum V_{ie} values at all partial loads are approximately the same because they are mainly controlled by the length of the slide valve. They are lower than the maximum V_{ie} for full load condition because the bypass port is usually designed to close at a certain time lag after the suction port closes.

(3) The flow rate of the medium is determined by the density of the inlet gas and the timing for the compre-ssion process to start. At full load and assuming no leakage, the flow rate is constant because the compression process starts when the suction port closes(no bypass port). For partial loads, the flow rate decreases as the slide valve is pulled toward the compressor discharge end and the chamber is not closed to the suction gas until the bypass port closes. At the same stopper location, the larger the bypass port opening is, the less the flow rate will be.

(4) The power consumption rate, defined as the power required for transporting a unit mass flow through the compressor, is determined by dividing the work obtained from the P-V diagram by the mass flow per chamber. As can be seen from Figures (9),(10) and (11), the higher the over/under pressure level (the chamber pressure at the start of discharging relative to the compressor outlet pressure) is or the earlier the compression process starts, the more work will be required for each chamber. If the compression process starts at approximately the same time, the power consumption rate will be roughly proportional to the ratio of the over/under pressure level to the mass flow rate.

(5) The gas pulsation level is roughly proportional to the over/under pressure level and inversely proportional to the increasing rate of the discharge port area at the start of the discharge process.

(6) By comparing Figures 12,13 and 14, we see that:

A. The V_{ie} curves are almost the same because they represent the geometric properties of the screw compressor that do not change. The density correction factors are approximately the same for different stroke arrangements with bypass port opening.

B. The flow rate plots are the same if the compressor geometry, the suction gas density and the rotation speed are kept the same. Its value changes proportionally with the rotation speed.

C. The over/under pressure behavior depends on the compressor inlet and outlet conditions. The averaged gas pulsation level changes consistently with the over/under pressure level.

(7)In response to a load demand change of the refrigeration system, there are in general two ways for the compressor to catch up: by changing the mass flow rate or by changing the compression ratio, or both. The control system of the screw compressor may be designed by following the above general trends to properly move the stopper and the slide valve or change the rotation speed of the driving rotor to cope with the system load call, such that the optimal values of the equivalent volume ratio, the power consumption rate(or equivalently COP) and the gas pulsation can be obtained.

References

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(2)Lars Sjöholm, "VARIABLE VOLUME-RATIO AND CAPACITY CONTROL IN TWIN SCREW COMPRESSORS" Proceedings of the 1986 International Compressor Engineering Conference at Purdue pp494-508.

(3)Kwang-lu Koai and Werner Soedel, "GAS PULSATION IN TWIN SCREW COMPRESSORS - PART I : DETERMINATION OF PORT FLOW AND INTERPRETATION OF PERIODIC VOLUME SOURCE " Proceedings of the 1990 International Compressor Engineering Conference at Purdue(to appear).

Acknowledgement

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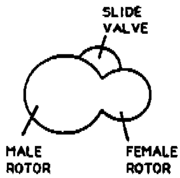


Figure 1: Cross sectional View of a Slide Valve and a Rotor Housing

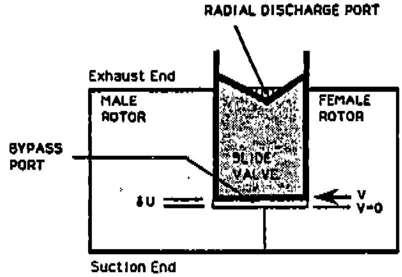


Figure 2: Top View of the Interior of a Screw Compressor

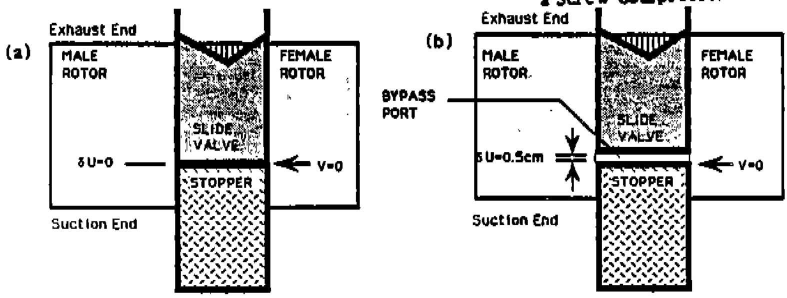


Figure 3: Stopper Located at the Reference Position
(a) No Bypass Opening
(b) 0.5 cm Bypass Port Opening

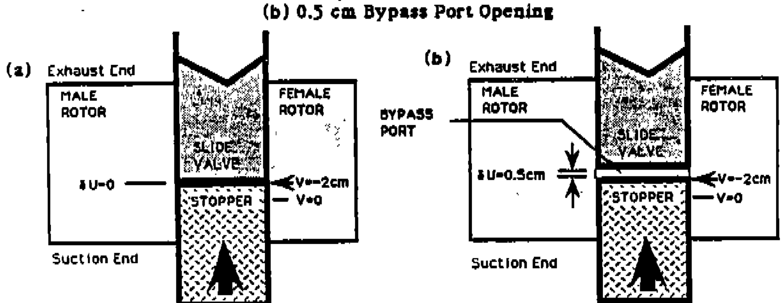


Figure 4: Stopper Pushed 2 cm with respect to the Reference Position toward the Exhaust End
(a) No Bypass Opening
(b) 0.5 cm Bypass Port Opening

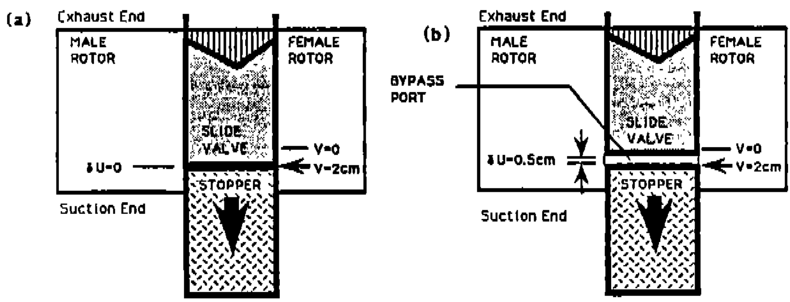


Figure 5: Stopper Pulled 2 cm with respect to the Reference Position away from the Exhaust End
(a) No Bypass Opening
(b) 0.5 cm Bypass Port Opening

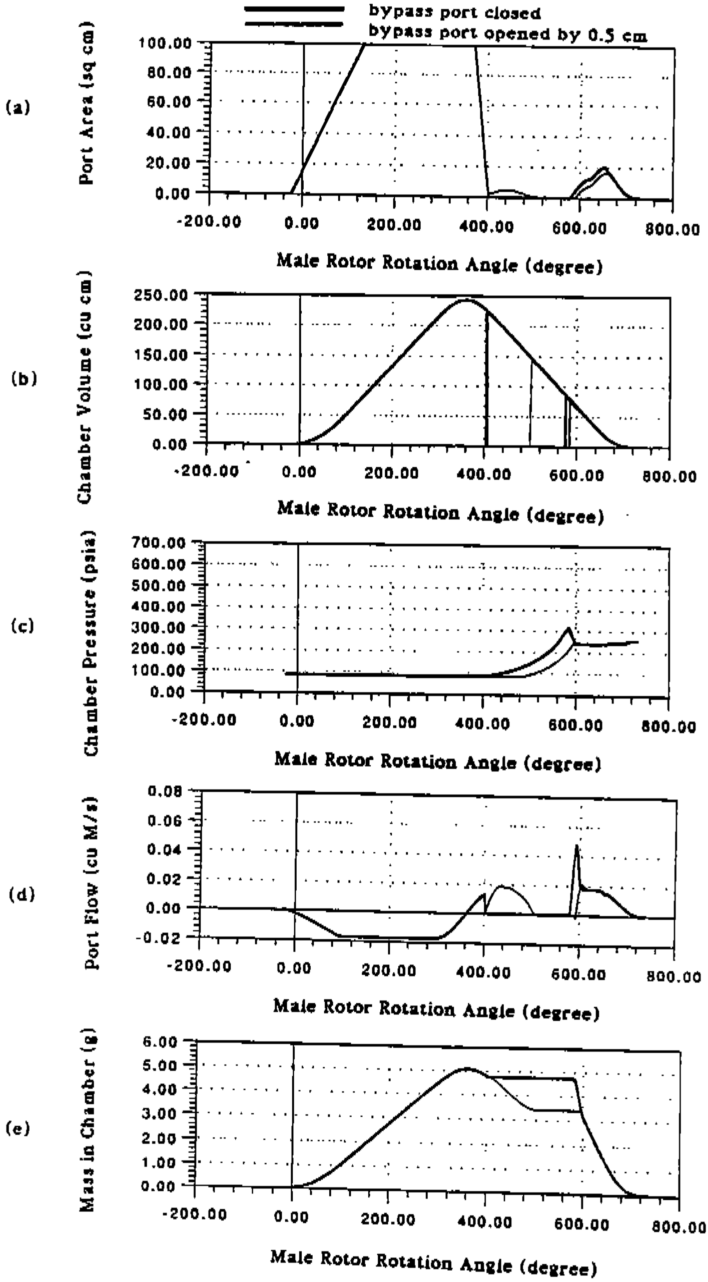


Figure 6: Chamber Conditions with Stopper Located at the Reference Position
(a) Port Area (sq cm)
(b) Chamber Volume (cu cm)
(c) Chamber Pressure (psia)
(d) Port Flow (cu M/s)
(e) Mass in Chamber (g)

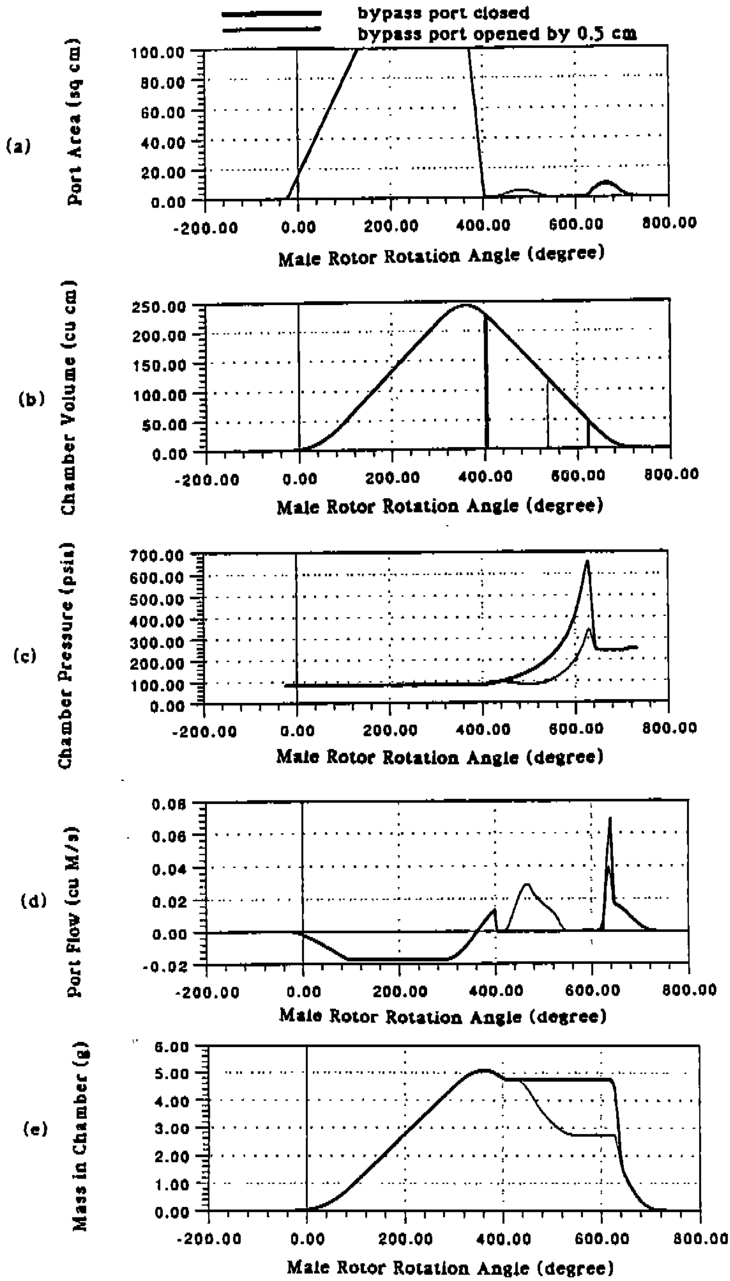


Figure 7: Chamber Conditions with Stopper Pushed 2 cm toward the Exhaust End
(a) Port Area (sq cm)
(b) Chamber Volume (cu cm)
(c) Chamber Pressure (psia)
(d) Port Flow (cu M/s)
(e) Mass in Chamber (g)

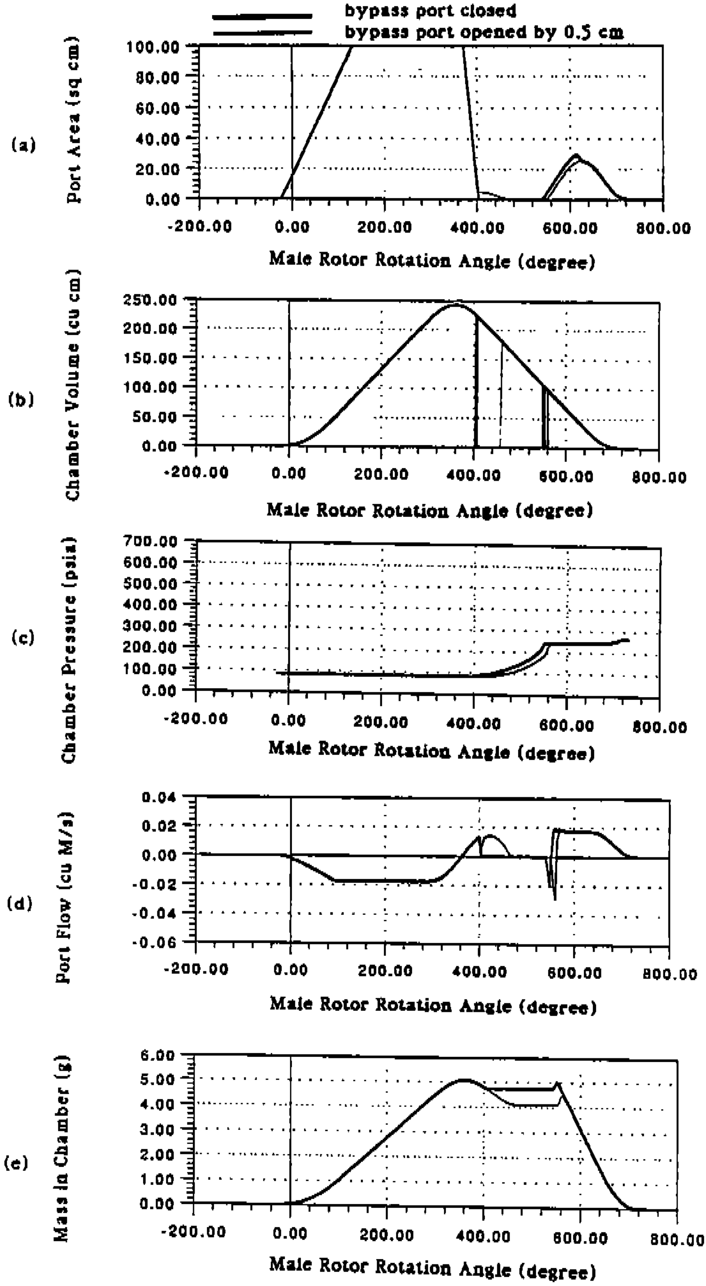


Figure 8: Chamber Conditions with Stopper Pulled 2 cm away from the Exhaust End
(a) Port Area (sq cm)
(b) Chamber Volume (cu cm)
(c) Chamber Pressure (psia)
(d) Port Flow (cu M/s)
(e) Mass in Chamber (g)

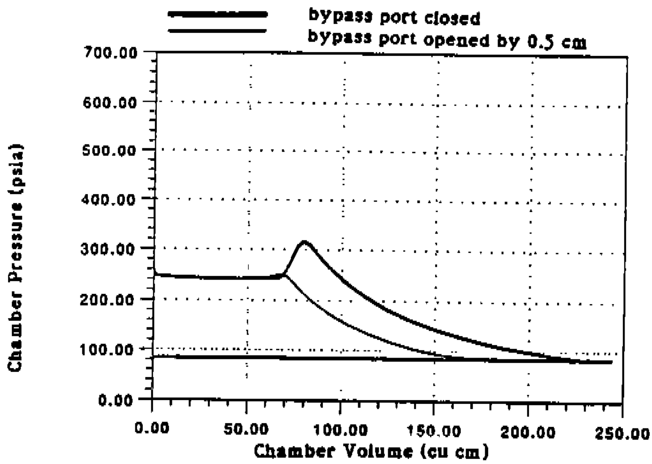


Figure (9) Pressure Volume Diagram with Stopper Located at the Reference Position

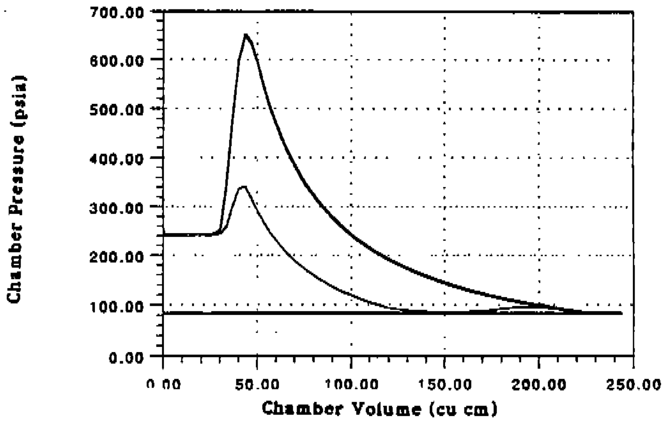


Figure (10) Pressure Volume Diagram with Stopper Pushed 2 cm toward the Exhaust End

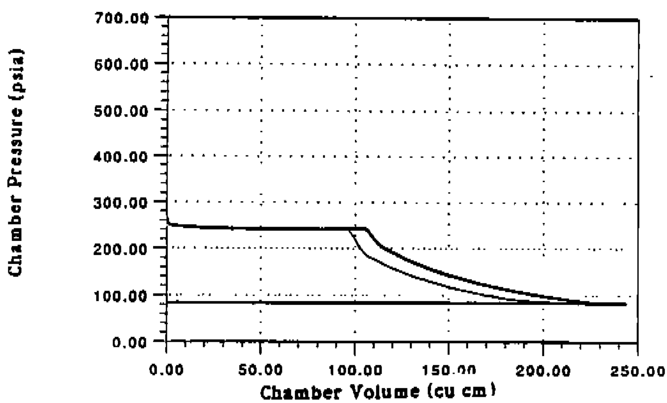
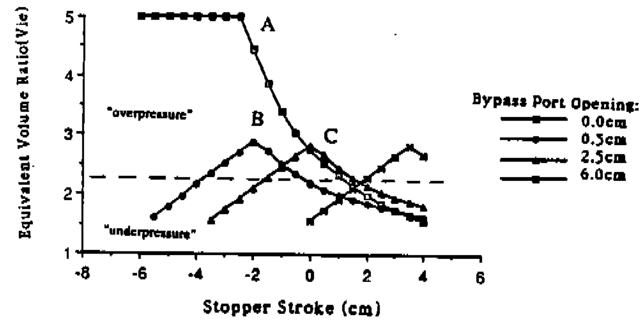
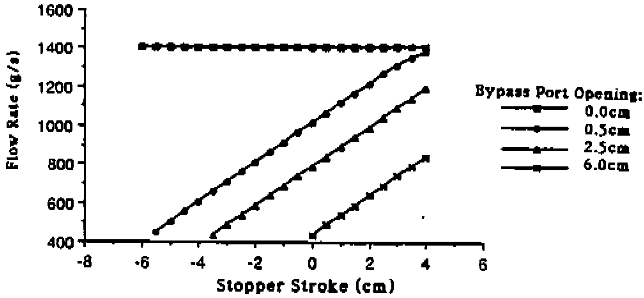
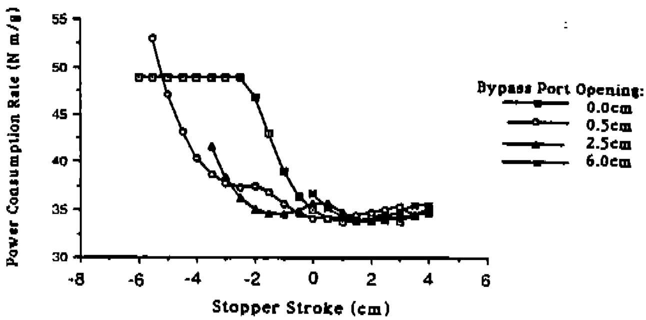


Figure (11) Pressure Volume Diagram with Stopper Pulled 2 cm away from the Exhaust End







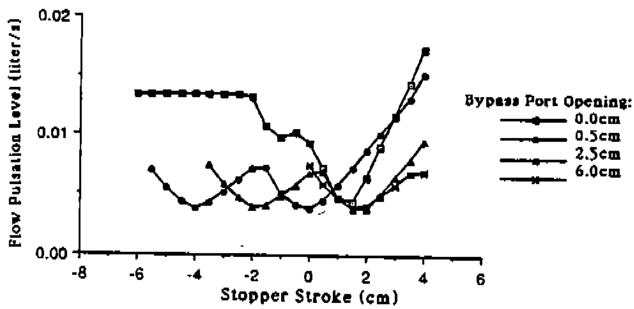


Figure 12: Performances of a Typical 60 HP Screw Compressor at 241 Psia Compressor Outlet Pressure and 59.2 CPS Speed

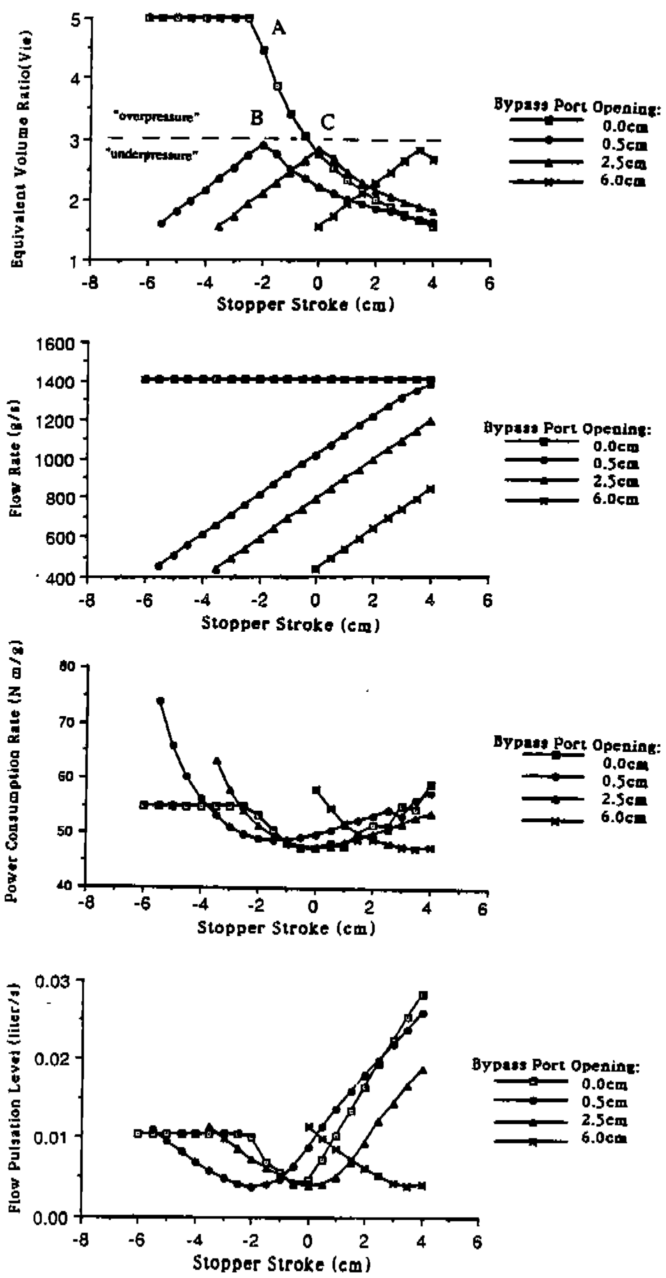


Figure 13: Performances of a Typical 60 HP Screw Compressor at 350 Psia Compressor Outlet Pressure and 59.2 CPS Speed

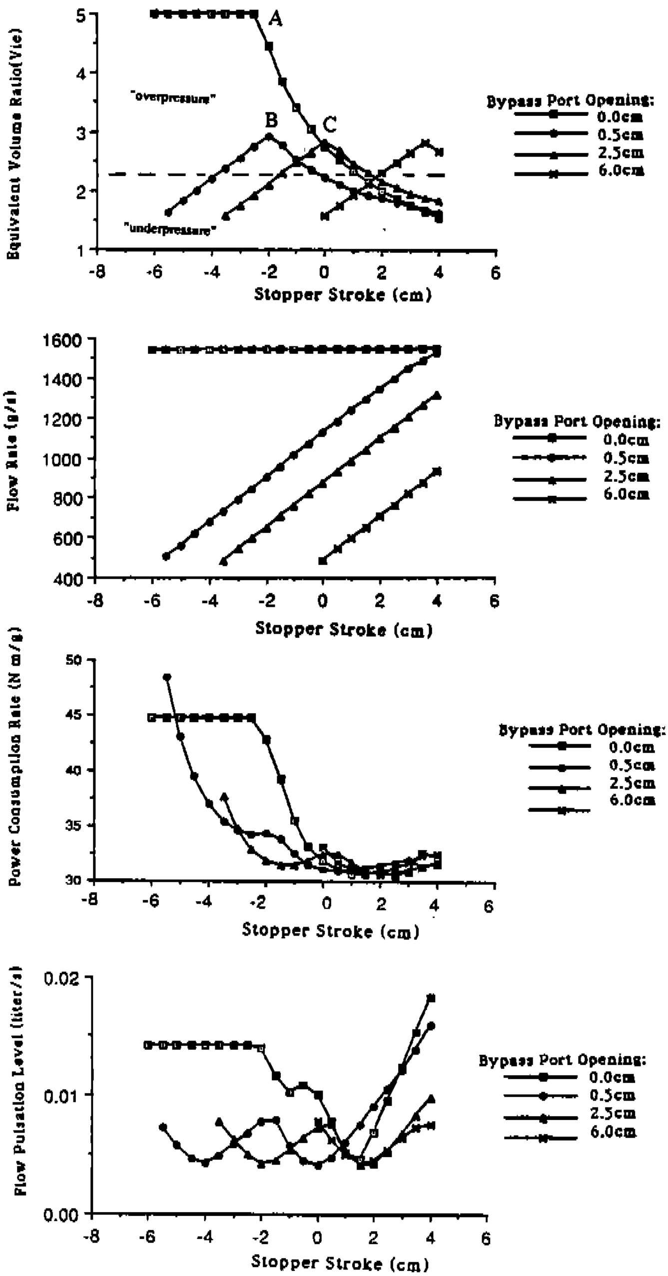


Figure 14: Performances of a Typical 60 HP Screw Compressor at 241 Psia Compressor Outlet Pressure and 65.0 CPS Speed