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AN INVESTIGATION OF THE COMPRESSOR SLUGGING PHENOMENON

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ABSTRACT

This paper presents the results of a preliminary experimental and analytical investigation of the slugging induced overpressures often encountered in refrigerant compressors. The experimental aspect of the project involved instrumenting a representative compressor, and then operating it under simulated slugging conditions. For the analytical study, two computer simulation programs were developed for the examination of the compressor processes under slugging-like conditions. Experimental and analytical results compare favorably as the trends and the qualitative features have been found to be similar.

INTRODUCTION

In typical refrigerant compressors, the maximum cylinder pressure limit of about 500 psi (3.43 MPa) is used for the design of structural components. But, this limit is exceeded considerably during the liquid slugging conditions, as the cylinder pressures could shoot up to very high values, in the range of about 700-3500 psi (4.8 - 24 MPa), consequently threatening the reliability of the compressor. The exact mechanism(s), effects, and other aspects of the liquid slugging phenomenon seem to be very complicated and not well understood.

An exhaustive review of literature on compressors (Hamilton 1974; Nieter and Singh 1984; Purdue Compressor Conference 1972-84; Soedel 1972) and closely related topics (Hammitt 1980; Singh 1980; Wylie and Streeter 1978) was conducted, but we could not find any paper that dealt with the slugging problem. Moreover, we utilized a computer information retrieval service to search the conference papers over 1973-1984 using the following key words, or any combination thereof: compressor, slugging, transient, and loading. Only 10 titles were found, but further examination of these papers revealed that they were not at all related to the slugging problem.

Since the literature is virtually nonexistent, we conducted a preliminary analytical and experimental investigation of the slugging problem with emphasis on the cylinder pressure overloading. Other aspects of the problem, such as how the liquid enters the cylinder and effects of oil and electric motor dynamics, were not considered. For the analytical study, we examined the cylinder compression process through two computer models. Predictions were compared with measured cylinder pressures obtained under slugging conditions induced in an experimental stand.

EXPERIMENTAL INVESTIGATION

Test Stand

A four-cylinder compressor was tested on an experimental slug test stand where slugging conditions were simulated by pumping liquid into the compressor. Only one out of the four cylinders was made operable and was directly connected to the suction and discharge pipings

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without any manifolding system. This cylinder was instrumented with a cylinder pressure (p_c) transducer, discharge plenum pressure (p_d) transducer, an accelerometer on the discharge head, and magnetic pick-up for the identification of crank angle θ through the location of the top dead center (TDC). Cycle voltage and current into the electric motor were also monitored with a voltmeter and an ampmeter, respectively.

The refrigerant (R-22) charge fed into the compressor was a combination of the pure liquid drawn from a reservoir and the cyclic refrigerant vapor at typical suction conditions. In this way, the amount of liquid and vapor in the refrigerant charge could be varied, between and including the total liquid and total vapor. Since there is no reliable method of measuring the mixture quality (x), which is the ratio of the vapor mass to total mass of the mixture, the state of the refrigerant fed was unknown.

It should be noted that an attempt was made to simulate the slugging condition as witnessed in the field by pumping as much cold liquid into an experimental compressor as possible. Even then, the slugging could not be induced in a deterministic manner. Our experimental tests have shown that the probability for the 'slugging' to occur at the compressor start-up increases as the quality decreases. In fact, we were unable to cause "slugging" unless the refrigerant charge was nearly liquid, i.e., $x \rightarrow 0$; and higher "slugging" pressures were evident with a larger amount of liquid charge.

The compressor was run at 1200 and 3600 rpm. The 'slugging' condition invariably occurred at the start up with a charge of the liquid refrigerant. Consequently the cylinder pressure at the start up peaked to a very high value. The peak cylinder pressures then dropped during subsequent cycles, and, after roughly a dozen cycles, the peak pressures were somewhat steady and approximately one tenth of the pressures measured at the start up. This condition was considered as "normal".

Results

Figure 1 shows a typical cylinder pressure (p_c) curve during the start up for 3600 rpm operation. We note that the curve is impulse-like during the first cycle with peak pressure to be 1200 psi (8.37 MPa). By the third cycle, the peak pressure drops substantially to about 685 psi (4.78 MPa) and after a dozen cycles to a steady value of 115 psi (0.79 MPa). After the first few cycles, the cylinder pressure profile looks "normal," as expected, without any impulse-like behavior. The only difference between 1200 rpm and 3600 rpm operation was that a secondary peak in p_c curve was noted in 1200 rpm data and not in 3600 rpm data.

Regarding the crank angle (θ) at which slugging-induced peak pressures were noticed, no consistency was evident as they ranged from 5° to 165° before the TDC. This may be due to the fact that the compressor does not come up to the full speed during one revolution, and, hence, there is uncertainty associated with the TDC marker during the first cycle, which is strongly influenced by the slugging.

The measurements of discharge plenum pressure (p_d) and motor current did not yield any significant information except that they followed the cylinder pressures. A better correlation was evident between the cylinder pressure and the structural acceleration at the discharge head, as peak-to-peak acceleration levels were found to be 250-280 g during "slugging" compared to 26-38 g during the "normal" operation. A summary of the typical experimental data is given in Table 1, where we find that during slugging the peak cylinder pressures are about 10 times and peak structural accelerations are about 7 to 10 times the values measured during the normal operation.

Even though the measured results may be different from those witnessed in the field, the qualitative behavior should be the same. Based on experimental study, we draw the following conclusions: (1) during slugging the cylinder pressure (p_c) profile is impulse-like with peak values of about 10 times those witnessed during the normal operation, and (2) the slugging condition and pressure overloading can be related to the quality of the refrigerant fed, which could not be measured.

Overall, it was difficult to draw any definite conclusions regarding the physical phenomenon that causes pressure overloading during "slugging". Thus, we had to conduct an analytical investigation of the compression process under slugging conditions.

ANALYTICAL INVESTIGATION

The polytropic process equation as given below has been used widely to simulate cylinder thermodynamics (Chlumsky 1965; Hamilton 1974; Nieter and Singh 1984; Soedel 1972).

$$p_c(t)\ [V_c^{\,n}(t)/m_c(t)]^n = \text{constant} = p_c(o)\ [V_c(o)/m_c(o)]^n \qquad (1)$$

Where V_c is the cylinder volume, m_c is the gas mass in cylinder, $t = \theta/\omega$ is the time, ω is the running speed in rad/s, and n is the polytropic index. This equation relates thermodynamic variables between two equilibrium states for a gas under the assumption that the process is quasistatic, i.e., internally reversible. The constant, n, is usually empirically determined. Even though n can assume any arbitrary value, for reciprocating compressors under normal operating conditions n lies within 1 and k where $k = c_p/c_v$ (Chlumsky 1965). Its value for "abnormal" operations, including slugging, is not known. We will examine it analytically.

We now study the cylinder compression or expansion process by considering a closed system, i.e., $m_c(t)$ = constant = m_c. Assuming that the refrigerant is an ideal gas of constant R:

$$p_c(t)\ V_c(t) = m_c R\ T_c(t) \qquad (2a)$$

$$\text{or} \qquad V_c(t)\ \dot{p}_c(t) + p_c(t)\ \dot{V}_c(t) = m_c R\ \dot{T}_c(t) \qquad (2b)$$

where superscript $\dot{}$ implies differentiation with respect to t. The application of the first law to the system under consideration yields the following equation:

$$\dot{Q}(t) = m_c\ c_v\ \dot{T}_c(t) + \dot{W}(t) = m_c\ c_v\ \dot{T}_c(t) + p_c(t)\ \dot{V}_c(t) \qquad (3)$$

where $\dot{Q}$ is the heat transfer rate (positive to the system) and $\dot{W}=p_c\dot{V}_c$ is the work transfer rate (positive from the system). Combining Equations 2 and 3 we get

$$V_c(t)\ \dot{p}_c(t) + [1 + (R/c_v) - (R/c_v)\dot{Q}(t)/(p_c(t)\dot{V}_c(t))]\ p_c(t)\ \dot{V}_c(t) = 0 \qquad (4)$$

In order to examine this equation, we differentiate Equation 1 with respect to t, assuming that n is a constant

$$V_c(t)\ \dot{p}_c(t) + n\ p_c(t)\ \dot{V}_c(t) = 0 \qquad (5)$$

Comparing Equations 4 and 5 we get

$$n = 1 + (R/c_v) - (R/c_v)\dot{Q}(t)/(p_c(t)\dot{V}_c(t)) \qquad (6)$$

Recognizing that $R = c_p - c_v$, we can express Equation 6 in terms of ratio H as

$$H = \dot{Q}(t)/\dot{W}(t) = \dot{Q}(t)/p_c(t)\dot{V}_c(t) = (k-n)/(k-1) \qquad (7)$$

Table 2 lists various values of n and H, and Figure 3 shows typical p-V plots for a closed system. We note that in order to obtain a very large increase or decrease in pressure, p_c, given a small change in volume, V_c, the polytropic index must assume very large or negative values. From Table 2 we know that such caes are possible only when heat transfer rates greatly exceed the work rates. This situation is obviously not encountered during the normal operation; however, it is worthwhile to pursue it for the slugging operation using a computer simulation model with variable n.

COMPUTER SIMULATION

Assumptions and Mathematical Models

Extensive information on the mathematical modeling of compressors, for normal operation, of course, is available (Hamilton 1974; Nieter and Singh 1984; Soedel 1972). For this study we simplified the models in order to reduce computational costs while still retaining sufficient accuracy. For instance, the suction and discharge systems are assumed to be anechoic lines (as pipes were directly connected to the valve plate in the experiment) at suction pressure ($\overline{p}_s$) and discharge pressure ($\overline{p}_d$), respectively. Some other assumptions are as follows: (1) uniform refrigerant properties throughout cylinder, (2) no leakage past piston, (3) constant running speed ω, (4) negligible oil and frictional effects, (5) negligible kinetic and potential energies associated with mass flux, (6) no heat exchange between cylinder and plenum gases, (7) one-dimensional flow, and (8) the cylinder gas energy to be entirely internal energy since average velocities are small.

In order to characterize the cylinder thermodynamic process under both normal and slugging operations, the polytropic process model with a variable n was chosen. The following two computer simulation models depending on the equation of state used were developed: Model I, an ideal gas model for the refrigerant vapor, and Model II, an empirical state model to describe refrigerant properties including the two-phase region.

MODEL I: IDEAL GAS/VARIABLE POLYTROPIC INDEX

We assume the refrigerant state to be a homogenous mixture described by the ideal equation of state (Equation 2). The thermodynamic process is considered to be polytropic (Equation 1) with an index n that can be chosen arbitrarily. Other mathematical models for the fluid flow, valve dynamics, and kinematics were the same as described by Nieter and Singh (1984).

Figure 3 shows cyclic cylinder pressure (p_c), suction pressure (p_s), discharge pressure (p_d), suction value displacement (y_s), and discharge valve displacement (y_d) curves for R-22 as an ideal gas with n = 1.10. For a closed system, this value of n corresponds to H = 0.5 as k = 1.2 for R-22. This implies that the heat transfer is out of the system as $\dot{Q} = -0.5(-\dot{W})$ where work is negative for the compressor according to our sign convention. Thus, Figure 3 illustrates typical time histories under normal operation. Note that the cylinder pressure profile is influenced near TDC by the discharge anechoic line.

Now we arbitrarily increase n by 10 times to simulate slugging. For a closed system, this value of n corresponds to H = -49 or $\dot{Q} = 49(-\dot{W})$; this implies substantial heat transfer into the system. For n = 11.0, we find abnormally high cylinder pressure (p_c) with very sharp, eratic fluctuations, as shown in Figure 4. Such characteristics were, in fact, seen in the experimental data measured during actual slugging conditions. The suction valve displacement for n = 11.0 (Figure 4) is not significantly different from the case of n = 1.10 (Figure 1). But, the discharge valve displacement (y_d) exhibits a large increase in its duration for n = 11.0 as it is open approximately twice the duration noted for n = 1.10. Also, there are some abrupt oscillations and bounces against the valve stop in Figure 4 for n = 11.00.

In Table 3, a comparison of some pertinent variables and indices corresponding to n = 1.1 and n = 11.0 is given. Drastic differences in several performance indices are evident. The volumetric efficiency has gone numerically above 100% for n = 11.0 from a nominal value with n = 1.10! The mass flow rate is more than doubled, but the associated actual work required increased by nearly a factor of five. The overwork at discharge due to the excessive cylinder pressure above the average discharge pressure increased by a factor of about 20, while the underwork at suction increased only by a factor of about 3.

MODEL II: REAL EQUATION OF STATE/VARIABLE POLYTROPIC INDEX

Refrigerant Properties

For this model, empirical equations of state as described by Chan and Haseldon (1981) were used to compute R-22 properties in both superheated vapor and two-phase regions. Computational algorithms were written to solve for the properties iteratively. Our property computations for the two-phase mixture were found to be in excellent agreement with published data (third reference) as the errors were less than 0.1%. For the superheated vapor phase, errors of less than 1% were found for all states except at higher pressures and temperatures. For instance, at p = 260 psia, T = 859.69°R, the specific volume was in error by 5.45%. Thus, the refrigerant property computations for the superheated vapor phase were excellent at moderate pressures and temperatures and still reasonably good at high pressures and temperatures. But, computational problems arise at extremely high pressures and temperatures which are witnessed during "slugging". When such pressures are approached, numerical computations in several refrigerant property subroutines become too large for computer storage, especially when the rate of pressure change is computed. The numerical simulation is then terminated. Nevertheless, we can still study the cause(s) leading to unusually high cylinder pressures because we know slugging conditions are beginning.

Computer Simulation Results

For the thermodynamic process, the polytropic model with variable n similar to Model I was again used. Additionally, we could vary the initial quality (x) of the refrigerant mixture fed to the compressor.

Figure 5 shows predicted time histories corresponding to n = 1.1 and x = 0.2 at 3453 rpm. As the initial quality x at n = 1.1 is decreased, the peak cylinder pressure increases as shown in Table 4. Figure 6 shows typical results for a further reduction in the initial quality x to 0.1. The numerical computation is, however, terminated when p_c is 650 psi (4.48 MPa). In all of the studies corresponding to n = 1.1, which is the normal condition as the heat is being transferred out of the system and is small, the computed quality x was found to decrease during the compression process. Table 4 shows that $x \rightarrow 0$ in two cases associated with high pressures.

In order to examine the effect of the heat transfer rate, Q, we vary n arbitrarily while holding the initial quality x = 0.2. We note from Table 4 that the peak pressures are much smaller than the ones observed for n = 1.1. This is primarily due to the fact that the quality x increases for higher n because of the substantial heat transfer into the system. Thus, this simulation demonstrates that the peak pressures are not only related to the initial quality but also to the heat transfer process during the compression.

CONCLUDING REMARKS

We believe that the study presented here is original and of significance, as a comparison of the computer simulation results with those obtained experimentally indicates some striking similarities. The analytical models presented here show that high cylinder pressures would be obtained when one of the following conditions is satisfied: the polytropic index assumes high values, given that the refrigerant could be described by the ideal gas equation, or the initial refrigerant quality fed to the compressor is very low, given a nominal value of the polytropic index. Even though the computer simulation models are simple, they predict trends and qualitative features very well. It should also be noted that both simulation models predict lower pressure peaks than those found in the experimental data. This is partly due to the fact that a homogeneous mixture was assumed. Also, Model II stopped computing because of the numerical difficulties associated with higher pressures. Nevertheless, both models clearly point out the strong influence of the heat transfer process and the initial charge quality on the cylinder compression process.

In order to investigate this problem more in depth, we have developed two additional mathematical models: Model III, an energy equation/two-phase refrigerant model, and Model IV, a model in which the refrigerant is assumed to behave as a separated two-phase fluid composed of an ideal gas and an incompressible liquid (Prater et al.; 1984). Two fluid configurations were considered in Model IV: one in which the liquid slug was located on the piston and a second in which the liquid slug was located at the cylinder head, near the valves. These models will be reported later in a separate paper. All four simulation models have been used to perform parametric design studies in order to find a set of design requirements that could lower the magnitudes of slugging induced pressures.

REFERENCES

Chan, C.Y. and Haselden, G.G. 1981, "Computer-based refrigerant thermodynamic properties. Part 1: basic equations," International Journal of Refrigeration, Vol. 4, #1.

Chlumsky, V. 1965. Reciprocating and rotary compressors, SNTL, Prague.

DuPont. Publications T-22, C-30, X-88F, and X-88B1, E.I. DuPont de Nemours & Co. (Inc.). "FREON" Products Division, 1400 N. Meecham Road, Schaumburg, Illinois, 60195.

Hamilton, J.F. 1974. "Extensions of mathematical modeling of positive displacement type compressors." Purdue University.

Hammitt, F.G. 1980. Cavitation and multiphase flow phenomena, McGraw-Hill.

Nieter, J.J. and Singh, R. 1984. "A computer simulation study of compressor tuning phenomena." Journal of Sound and Vibration, 97(3), 475-488.

Prater, Jr., G. Nieter, J.J. and Singh, R. 1984. "Simulation of cylinder pressure overloading caused by the slugging conditions." The Ohio State University, report submitted to Copeland Corporation.

Proceedings of Purdue Compressor Technology Conferences, 1972, '74, '76, '78, '80, '82, and '84.

Singh, R. 1980. "Modeling of fluid transients in machines." Shock and Vibration Digest, 12, (6 and 7).

Soedel, W. 1972. "Introduction to computer simulation of positive displacement type compressors." Purdue University.

Wylie, E.B. and Streeter, V.L. 1978. Fluid-transients, McGraw-Hill.

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TABLE 1
Typical Measured Compressor Data

Measured Quantity	During Slugging		During Normal Operation	
	1200 rpm	3600 rpm	1200 rpm	3600 rpm
peak cylinder pressure p_c	1135 psi (7.82 MPa)	1200 psi (8.37 MPa)	105 psi (0.72 MPa)	115 psi (0.79 MPa)
crank angle θ corresponding to peak pressure p_c	113°	173°	148°	159°
peak to peak corresponding to peak cylinder pressure p_c	250g	280 g	26 g	38 g acceleration

θ measured from the bottom dead center (BDC)

TABLE 2
Polytropic Index n and Ratio $H = \dot{Q}/\dot{W}$ for an Ideal gas in a Closed System

n	$-\infty$	-1	0	$0 < n < 1$	1	$1 < n < k$	k	$2k-1$	$n >> 2k-1$
H	∞	$\frac{k+1}{k-1}$	$\frac{k}{k-1}$	$\frac{k}{k-1} > H > 1$	1	$1 > H > 0$	0	-1	$H << -1$

TABLE 3
Predictions by Model I at 3453 rpm: Normal Condition (n=1.1) vs. Slugging Condition (n=11.0)

Computed Variable	n = 1.1	n = 11.0
peak p_c	428 psi (2.95 MPa)	823 psi (5.74 MPa)
θ corresponding to peak p_c	178°	110°
mass flowrate through suction valve per cycle	1.5×10^{-3} lbm (0.68×10^{-3} kg)	2.9×10^{-3} lbm (1.32×10^{-3} kg)
volumetric efficiency	77.7%	153%
Ideal work/cycle	236.9in-lb_f(26.78 N-m)	478.9in-lb_f(54.13 N-m)
actual work cycle	271.5in-lb_f(30.69 N-m)	1072.9in-lb_f(121.27N-m)
under work/ideal work	3.8%	2.8%
over work/ideal work	9%	52.7%
Torque required	43.2 in-lb_f(4.88 N-m)	1072.9 in-lb_f (121.27N-m)

TABLE 4
A Summary of Predictions by Model II

Polytropic index, n	Initial Quality x(%)	peak (p_c) (psia)	Quality x Corresponding to peak p_c(%)
0.5	0.20	360.0	→ 0
1.1	0.20	553.1	0.125
1.1	0.15	640.1	→ 0
1.1	0.10	650.0	→ 0
5.0	0.20	415.0	0.80
7.0	0.20	400.0	0.85
7.0	0.10	570.0	0.70

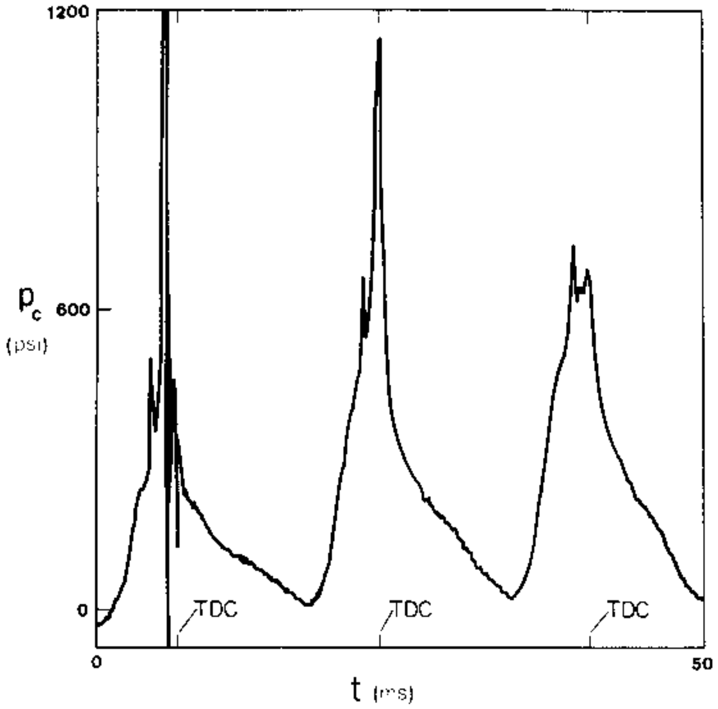

Figure 1. Measured cylinder pressure (p_c) time history during slugging. The compressor was started at $t = 0$

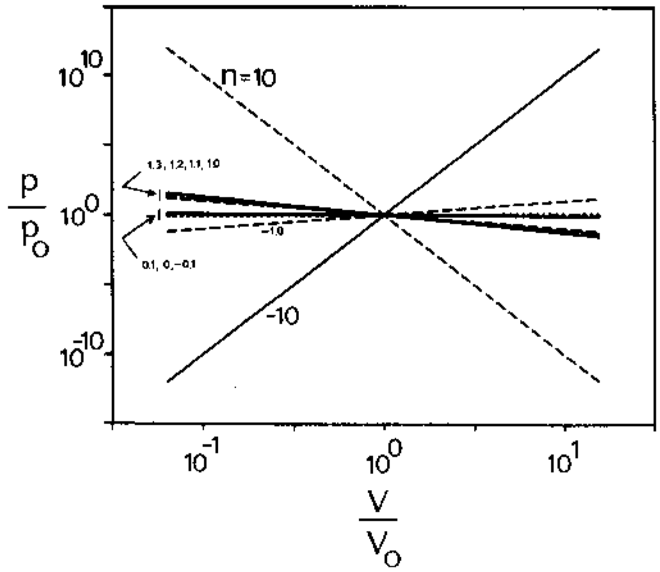

Figure 2. Pressure-volume relationship for an ideal gas in a closed system for various values of n. Here p_o and V_o refer to the reference or initial conditions; these are taken to be equal to the suction line conditions in a positive displacement compressor

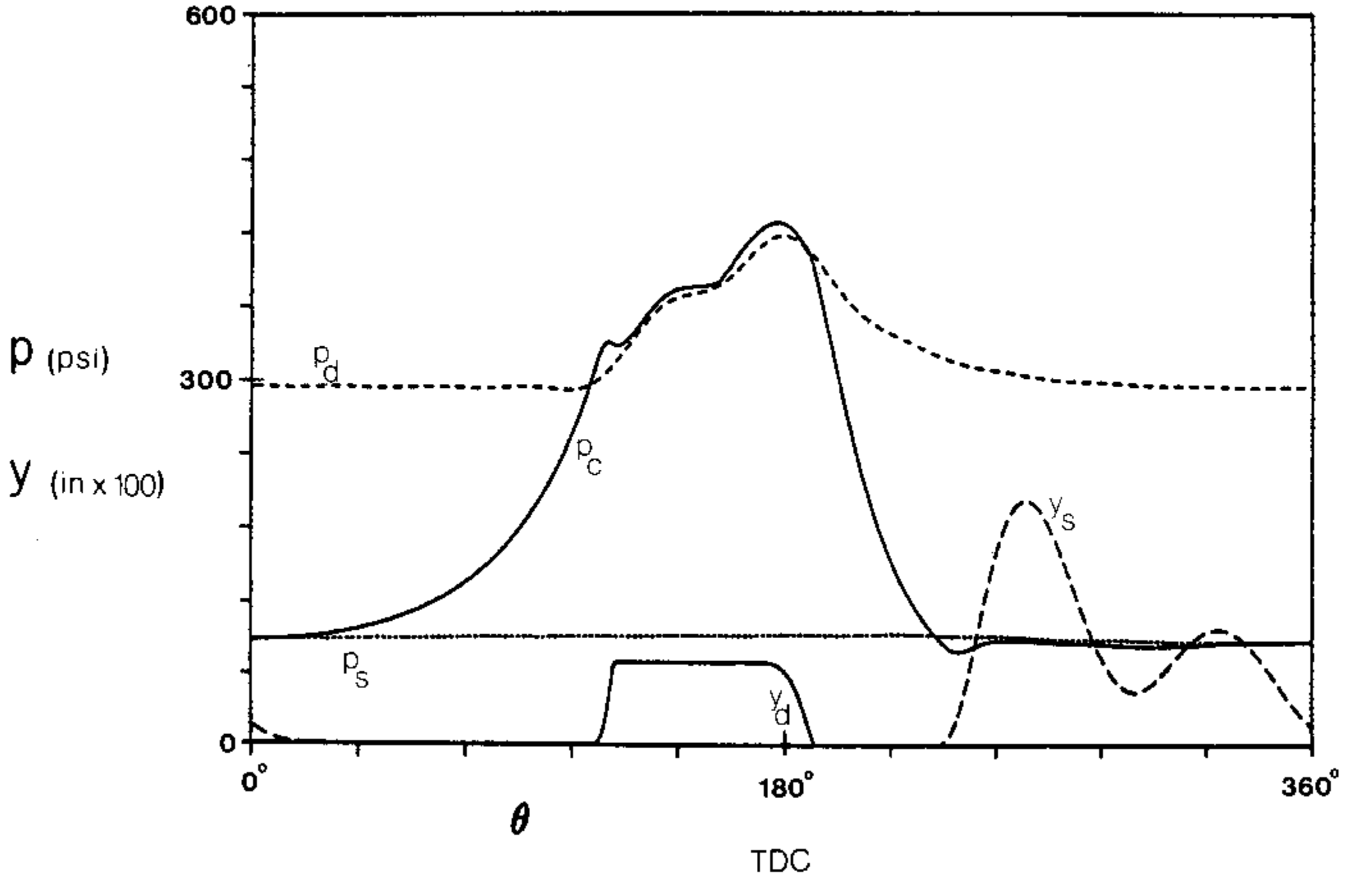

Figure 3. Cyclic predictions by Model I during the normal operation ($n = 1.1$) at 3453 rpm. Here, p_c = cylinder pressure, p_d = discharge plenum pressure, p_s = suction plenum pressure, y_d = discharge valve displacement, and θ = crank angle in deg from the bottom dead center

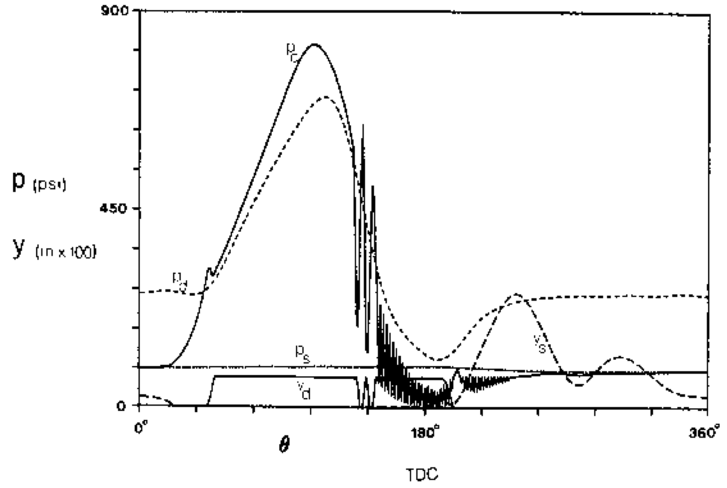

Figure 4. Cyclic predictions by Model I during the slugging condition (n = 11.0) at 3453 rpm

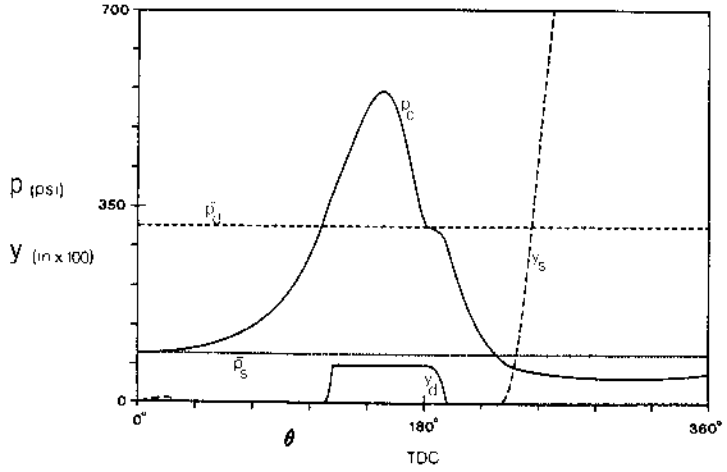

Figure 5. Cyclic predictions by Model II for n = 1.1 and initial quality x = 0.2. Here, p_s = suction line pressure as p_d = discharge line pressure

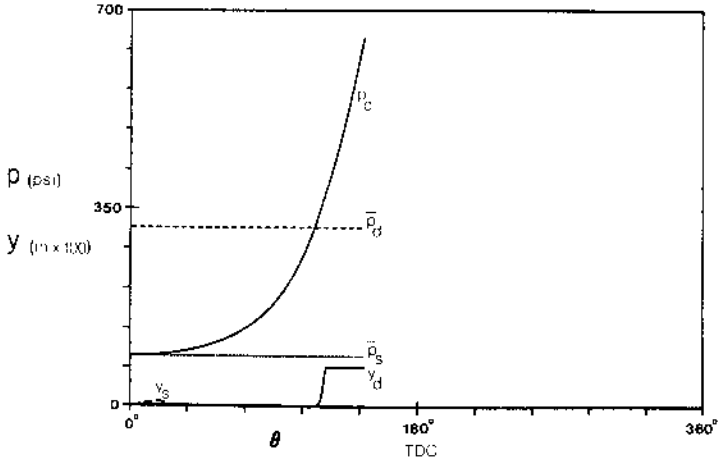

Figure 6. Cyclic predictions by Model II for n = 1.1 and initial quality x = 0.1. Note that the simulation was terminated due to the numerical difficulties