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ABSTRACT

In typical refrigerant compressors, peak cylinder pressures under liquid slugging conditions are as high as 10 times the peak pressures during the normal operation. In this paper we model this phenomenon by considering the compression process with a separated gas and liquid media in the cylinder. Two cases of this model for various liquid fractions are simulated. Even though time histories corresponding to both cases are slightly different, both models yield very high peak pressures, similar to those measured, when the compressor is "pumping" mostly liquid. Various aspects of the problem are discussed here.

NOMENCLATURE

A	flow area
b	slider-crank length
C	coefficient of discharge
g_c	gravitational constant
k	ratio of isentropic specific heat capacities
$\dot{m}$	mass flow rate
n	polytropic index
p	pressure
t	time

V volume

x refrigerant quality (on mass basis)

y valve displacement

α liquid fraction (on volume basis)

θ crank angle

ρ density

Subscripts

0 reference statement (mean suction line condition)

a clearance volume

c cylinder

d discharge

g gas

l liquid

s suction

INTRODUCTION

During liquid slugging conditions, cylinder pressures could shoot up to very high values, 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 Proceedings 1972-84; Soedel 1972) and on fluid transients in general (Hammitt 1980; Singh 1980; Wylie and Streeter 1978) was conducted, but we could not find any paper that dealt with the slugging problem.

PREVIOUS STUDY AND SCOPE OF CURRENT WORK

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 the following two mathematical models [Singh et al, 86]. Both of these models assumed a homogenous refrigerant mixture.

(I) The refrigerant state of the mixture was assumed to be described by the ideal equation of state. The thermodynamic process was considered to be polytropic. We varied the polytropic index (n) to simulate the slugging behavior.

(II) For this model, empirical equations of state as described by Chan and Haseldon (1981) were use to compute R-22 properties in both superheated vapor and two-phase regions. Computational algorithms were written to solve for the properties iteratively. 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.

The above mentioned analytical models show that high cylinder pressures would be obtained when one of the following conditions is satisfied: the polytropic index n assumes very high values, given that the refrigerant could be described by the ideal gas equation, or the initial refrigerant quality x fed to the compressor is very low, given a nominal value of the polytropic index. Even though these 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 this paper we advance the previous work by considering the case where the refrigerant is assumed to behave as a separated two-phase fluid composed of an ideal gas and an incompressible liquid. Two fluid configurations are considered: one in which the liquid slug is located on the piston and a second in which the liquid slug is located at the cylinder head near the valves. Thus we will be examining the two-phase compression process in a reciprocating compressor. The overall problem can be defined as follows: since the compressor is a gas compression device and not a liquid pump, how will it respond to pure liquid or a combination of gas and liquid entering it?

EXPERIMENT

A compressor was tested on an experimental slug test stand where slugging conditions were simulated by pumping liquid into the compressor. The cylinder was directly connected to the suction and discharge pipings 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 the 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., $x \to 0$; and higher "slugging" pressures were evident with a larger amount of liquid charge.

MEASURED RESULTS

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".

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.

Following the first main peak, we also note in Figure 1 a rapid drop in pressure to virtually zero pressure. This "negative impulse" like profile is what one would expect from cavitation type phenomenon.

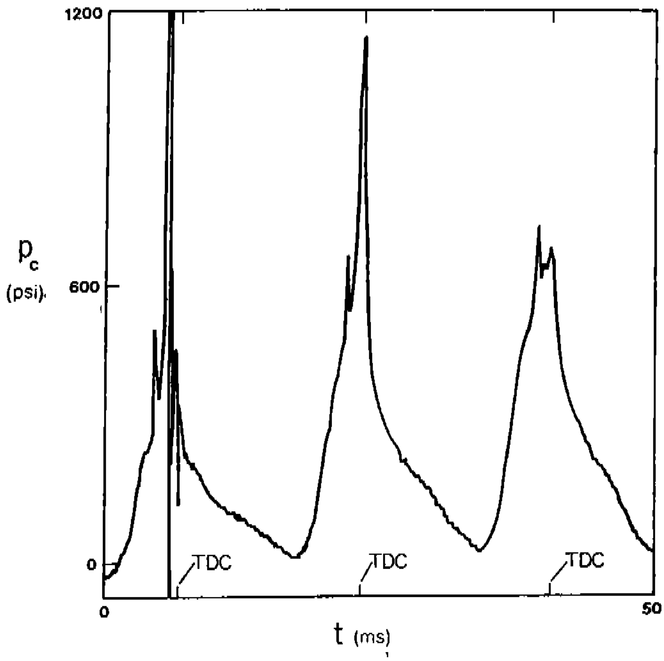


Figure 1. Measured cylinder pressure (p_c) time history during slugging. The compressor was started at t = 0.

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, (2) a negative impulse like profile follows the first positive impulse peak, and (3) 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.

MATHEMATICAL SIMULATION MODELS

Fluid Configurations

We assume that the cylinder working fluid consists of a separated two phase media. The refrigerant liquid slug of volume V_l is considered to be incompressible. The refrigerant portion of the fluid of volume V_g, is modeled as an ideal gas. In any given situation, the total volume is given as follows; see Nomenclature for the identification of symbols

$$V_l + V_g = V_a + (\pi D^2 b/2) \qquad \text{(1-a)}$$

Now, define the liquid fraction α on volume basis as

$$\alpha = V_l/(V_l + V_g) = V_l/(V_a + \pi D^2 b/2) \qquad \text{(1-b)}$$

The following two different fluid configurations are considered.

Case 1: The liquid slug is located on the piston as shown in Figure 2(a); in this case the vapor leaves the cylinder first when the discharge valve opens and the liquid is pumped through the valve. Case 2: the liquid slug is located at the valve plate and the vapor is in between the liquid slug and piston, as shown in Figure 2(b); in this case the liquid is discharged first when the discharge valve opens. We assume that in both cases no phase changes and mixing occur during operation.

Models for Vapor Portion

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 infinite volumes, i.e. p_s = constant and p_d = constant. Some other assumptions are as follows: (1) uniform refrigerant properties throughout vapor volume, (2) no leakage past piston, (3) constant running speed , (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. For details on the mathematical models used here, see paper by Nieter and Singh (1984).

Model for Liquid Portion: Case 1

For this model, the compressor acts as a pure vapor compression machine with reduced swept volume till all of the vapor is purged from the cylinder. Then, the compressor starts to behave like a pump as the liquid is being discharged through the valve. Since the liquid is essentially incompressible, the mass flow rate through the discharge valve is

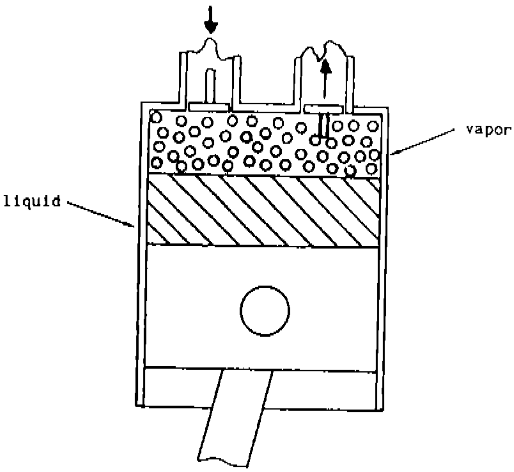
$$\dot{m}_{d1}(\theta) = \rho_1 \dot{V}_c(\theta) \tag{2}$$

We model the discharge valve as a hydraulic orifice

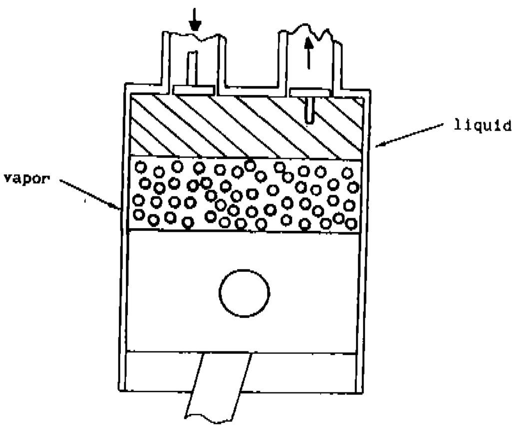
$$\dot{m}_{d1}(\theta) = C_d\, A_d\, [2\, g_c(p_c(\theta) - p_d)/\rho_1]^{\frac{1}{2}} \tag{3}$$

We note that the liquid portion affects essentially the cylinder pressure $p_c(t)$ computation. A backward finite difference approximation is used for Equation (2) and (3) along with a fourth order Runge-Kutta solver for valve dynamics. The computer simulation was found to be very sensitive to the step size ($\Delta\theta$). By trial and error, $\Delta\theta = 0.1°$ was deemed to be suitable during the compression stroke.

When the discharge valve closes at the top dead center (TDC), we find that the liquid still fills the clearance volume V_a. And, when the suction valve opens we assume that only vapor enters the cylinder. Consequently, during the next cycle the liquid fraction is very small and given by $\alpha = V_a/[V_a + \pi D^2 b/2]$.



a Working Fluid Configuration for Case 1 Simulations.



b Working Fluid Configuration for Case 2 Simulations.

Figure 2. Working fluid configuration. (a) Case 1 simulation. (b) Case 2 simulation.

Model for Liquid Portion: Case 2

During this case the liquid is pumped as soon as the discharge valve opens. The cylinder pressure $p_c(\ )$ is computed by solving the following equations simultaneously.

$$\dot{m}_{dl}(\theta) = C_d\ A_d[2\ g_c(p_c(\theta) - p_d)/\rho_l]^{\frac{1}{2}} \tag{4}$$

$$p_c(\theta) = (1/\rho_g(\theta))^n = p_c(0)(1/\rho_g(0))^n \tag{5}$$

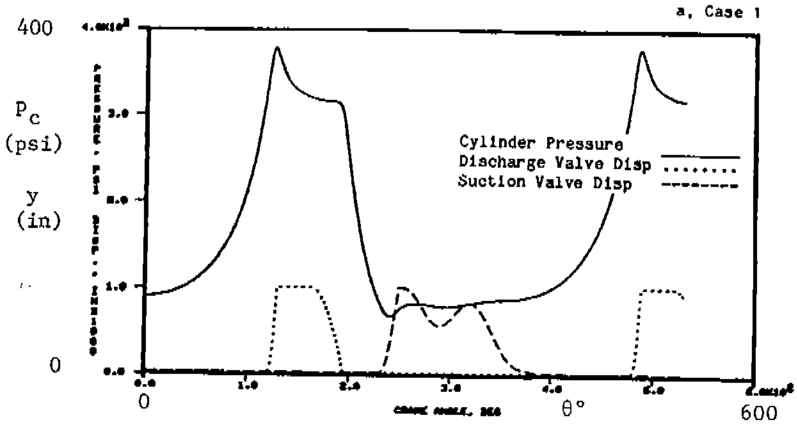
Here ρ_g is the density of the vapor fraction as the vapor is being compressed in between the liquid slug flow and the piston motion. The Newton-Ralpson method is used to solve Equations (4) and (5), like Case 1 model, a step size of $\Delta\theta = 0.1°$ was used. After the liquid is pumped out of the cylinder, the simulation reverts back to the vapor models. In this case, no liquid remains in the cylinder and we assume that no further liquid enters the cylinder. Hence, the compressor behaves like a normal vapor compression machine during the second cycle.

RESULTS AND DISCUSSION

Figure 3 shows the Case 1 and Case 2 cylinder pressure plots for a liquid fraction $\alpha = 0.0$ (pure vapor). As expected, the pressure and valve motion curves for both cases are identical. The maximum cylinder pressures are approximately 380 psia which occur at crank angles of 120 degrees.

If we increase the liquid fraction α to 0.10, the simulations yield the plots of Figure 4. The Case 1 and Case 2 results are now radically different. For Case 1 the pressure curve looks similar in shape to that of a pure vapor ($\alpha = 0$) until the discharge valve opens and the vapor portion of the working fluid is purged. At this point, however, liquid begins to be pumped from the cylinder, and because of its incompressibility, the cylinder pressure increases instantaneously. The piston velocity decreases as the compression process continues, and the cylinder pressure correspondingly decreases until, immediately after the top dead center, it reaches a value on the order of 10^{-4} psia. The suction valve then opens, the cylinder pressure increases, and the remainder of the cycle continues on much as before. The abrupt drop to an extremely low pressure that occurs after the top dead center, due to the liquid left in clearance volume, provides conditions that could cause cavitation. Such rapid changes of state can lead to very high pressure transients. In fact, several of the experimental plots show a secondary pressure peak preceded by an inordinately low pressure. Although our Case 1 simulation program does not model the cavitation phenomenon and the resulting transients, its results do show that it is a possibility.

In Case 2 simulations, the liquid is pumped as soon as the discharge valve opens, and thus the maximum pressure is reached earlier than in Case 1. After all the liquid is eliminated from the cylinder, there is an instantaneous pressure drop. The



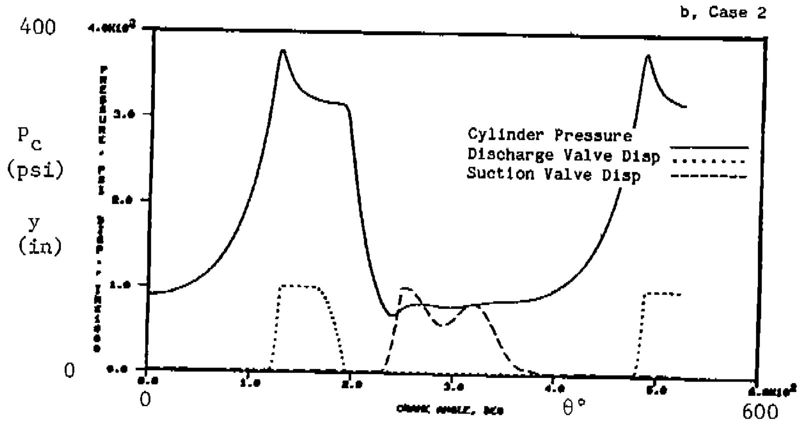


Figure 3. Cylinder pressure and valve displacements time history for a liquid fraction of $\alpha = 0.0$.

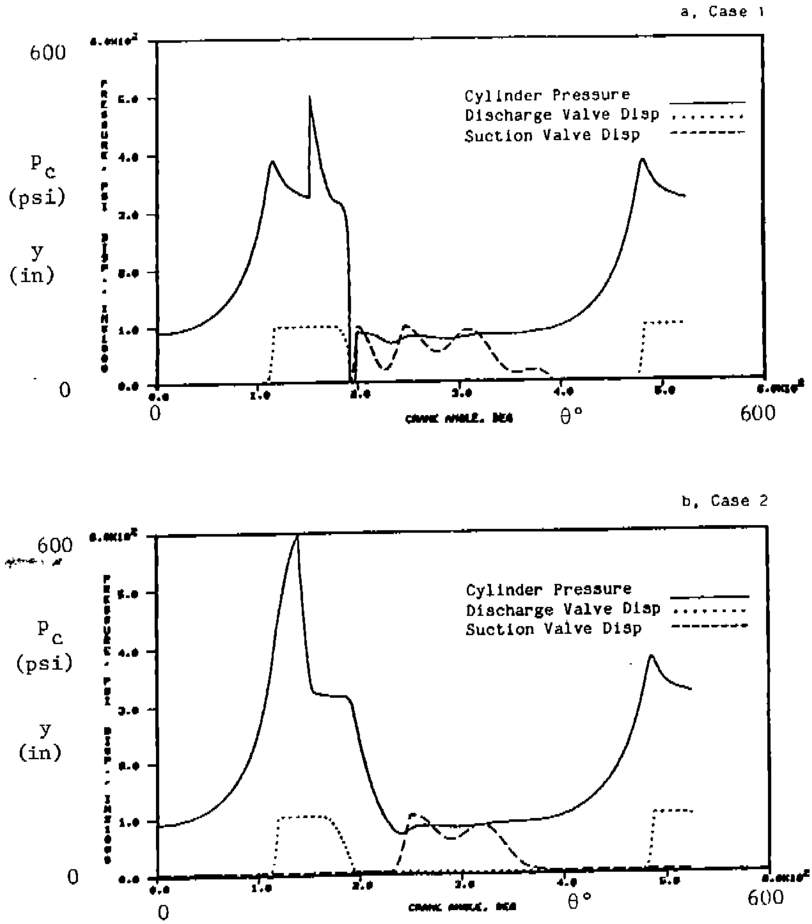


Figure 4. Cylinder pressure and valve displacements time history for a liquid fraction of $\alpha = 0.1$.

remainder of the process then looks exactly like the corresponding portion of the pure vapor simulations. Since no liquid remains in the clearance volume at TDC, a cavitation-like condition does not develop.

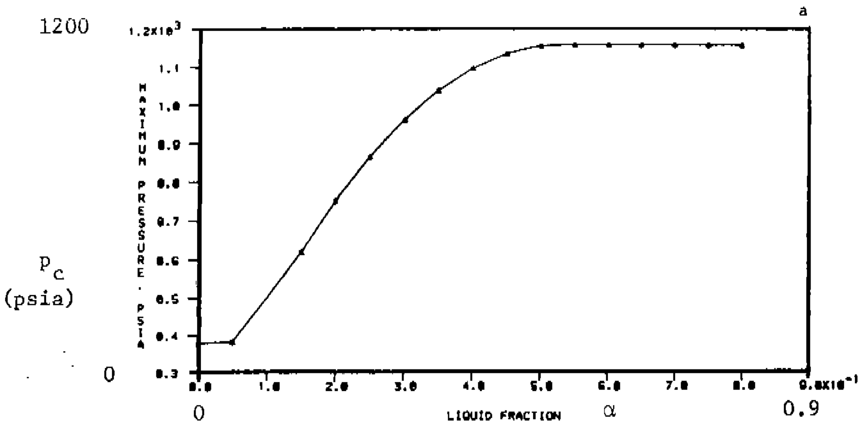
Several major differences may be noted when comparing Figure 3 with Figure 4. For both Case 1 and Case 2, the maximum cylinder pressures for the two phase working fluids are greater than those expected from a pure vapor working fluid. In addition, the discharge valve opens earlier and remains open longer. If we increase the liquid fraction α, the maximum cylinder pressure increases, as shown in Figure 5.

Figure 5 shows that for both Case 1 and Case 2, the maximum cylinder pressure increases with increasing liquid fraction until it reaches a maximum of approximately 1150 psia. This value is nominally three times greater than the corresponding pure vapor maximum. Because the Case 1 simulation assumes that no vapor is present in the cylinder when liquid is being discharged, this limiting maximum is reached at a lower liquid fraction than in Case 2. In addition, the large pressure increases do not begin in Case 1 until the liquid volume exceeds the clearance volume (α= .05). In Case 2 they begin immediately.

Finally, in Fig. 6 we see that the equivalent polytropic index n continually increases with increasing liquid fraction. Furthermore, the index goes to positive infinity as the liquid fraction approaches 1.0. Such a polytropic index implies a process involving an incompressible fluid, and is consistent with the assumptions made in the homogenous fluid mixture mathematical models by Singh et al. (1986).

In comparing the separated fluid results of this section with the experimental results, one notes several striking similarities. Most obvious is the location and shape of the cylinder pressure peaks. For example, in Fig. 1 we see a small initial peak, a much larger primary peak, a rapid pressure drop, and then, following quickly, another peak. Such behavior is very similar to the Case 1 simulation results. In addition, the presence of a third pressure peak after a pressure drop is consistent with the behavior expected from a cavitation type phenomenon.

In concluding this discussion a few comments on the consequences of our assumption of constant discharge pressures are in order. During the operation of an actual system, discharge pressures tend to increase as flow through the valve occurs. These increases are due for the most part to downstream flow impedances. Since a specified discharge flow rate requires a fixed pressure differential across the valve, the increased discharge pressures increase the cylinder pressure as well. Therefore, our assumption of a fixed discharge pressure is conservative, and leads to lower cylinder pressures than we might otherwise expect. To increase the accuracy of the simulations, one would have to model the discharge valve and the downstream



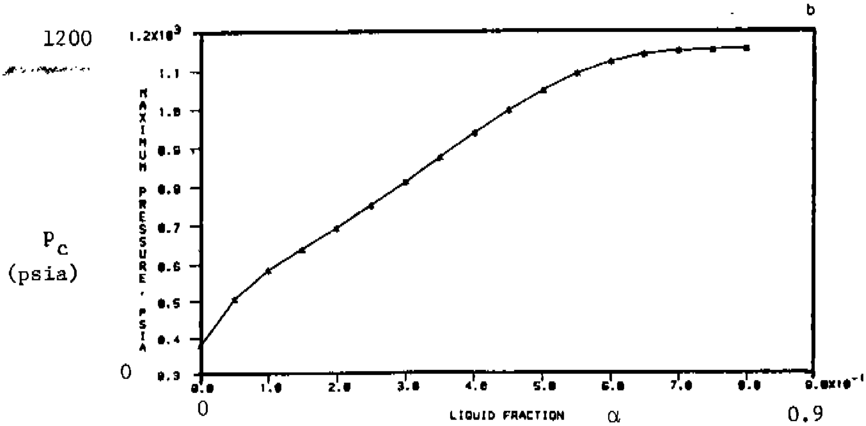
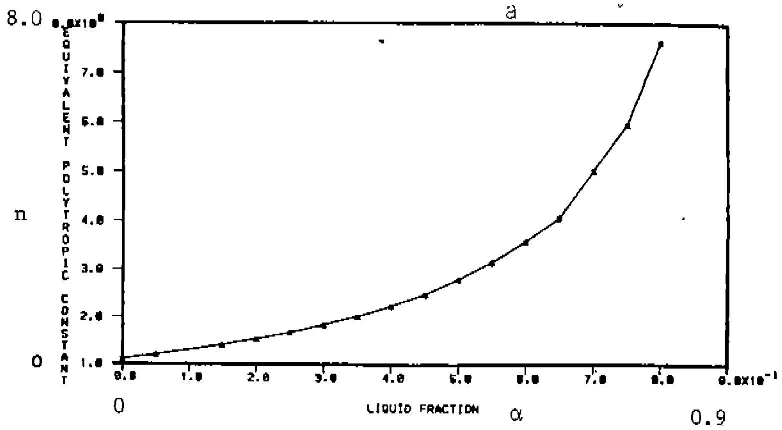


Figure 5. Peak cylinder pressure versus liquid fraction α . (a) Case 1 (b) Case 2.



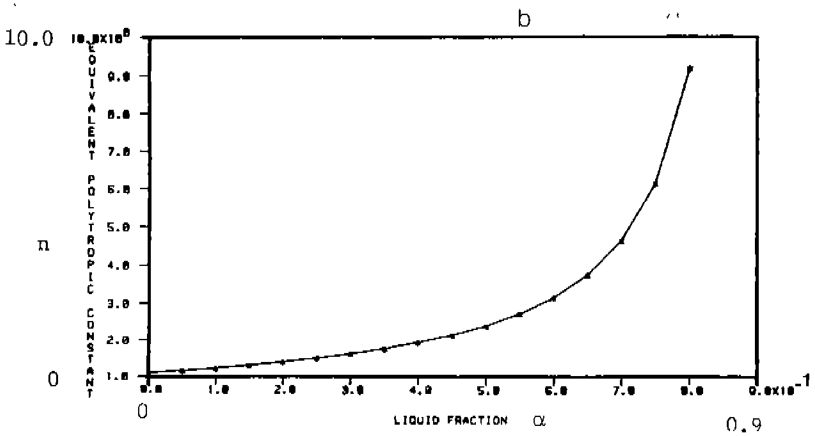


Figure 6. Computed polytropic index n versus liquid fraction α . (a) Case 1 (b) Case 1.

flow passages as a sequence of orifices in series and parallel and then determine an effective flow impedance.

CONCLUDING REMARKS

The most important result of the mathematical model presented here is that increasing percentages of liquid in a separate two-phase fluid result in increased maximum cylinder pressures. Furthermore, when the liquid portion of the fluid rests on the piston, conditions are created that are conducive to the formation of cavitation pressure transients. Both of these effects have been witnessed in measured data. An increasing liquid fraction also results in: (i) smaller maximum pressure crank angles (with a limiting value of $\theta = 90°$), (ii) earlier discharge valve opening angles and longer discharge valve open intervals, and (iii) higher equivalent polytropic constants (ranging from $n = k$ at $\alpha = 0.0$ to $n \to \infty$ as $\alpha \to 1.0$).

Further analytical and experimental work in this area is in progress.

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TABLE I.
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 accel. corresponding to peak cylinder pressure p_c	250g	280g	26g	38g

measured from the bottom dead center (BDC)