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A Comparison of Computer Simulation Techniques of Gas Flow in Multiple Single-Stage and Two-Stage Reciprocating Compressor Systems

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A COMPARISON OF COMPUTER SIMULATION TECHNIQUES OF GAS FLOW IN MULTIPLE SINGLE-STAGE AND TWO-STAGE RECIPROCATING COMPRESSOR SYSTEMS

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ABSTRACT

A brief review of the two general computer programs for simulating the flow in single and double stage reciprocating compressor systems is presented. The main differences in the computation schemes used are discussed. The computer time and input requirements, and output capability of the two programs are compared. The merits and disadvantages of the two simulation techniques are reported. Both computer programs are tested at various conditions to predict the pressure pulsations at several locations along a multiple reciprocating compressor system with a receiver, and a two-stage compressor system. Mass flow, volumetric efficiency, and indicated power predictions are compared with the known experimental results.

INTRODUCTION

A reciprocating compressor generates unsteady flow in its associated piping, due to the intermittent delivery and suction through its inlet and discharge valves. It has been established that the pressure fluctuations effect the compressor performance and valve behavior. A complete simulation of compressors with their suction and delivery pipes is therefore necessary to predict the compressor performance at different conditions. Compressor kinematics and thermodynamics, valve dynamics, gas exchange, and unsteady flow in the pipes should be solved simultaneously in this case. Problem generally becomes a solution of unsteady flow in pipes with the compressor being one of the boundary conditions of the problem. Various methods have been used for the solution of unsteady flow in compressor piping systems by assuming that pressure fluctuations are small, and linearizing the governing equations (1-4). It has been shown by Benson and Üçer (5,6), that a general simulation program for compressor systems may be written, utilizing the method of characteristics in the solution of non-linear, hyperbolic partial differential equations which describe the flow in pipes. The gas flow was assumed to be homentropic but the effect of friction was included in this simulation. In the recent years solution methods other than method of characteristics were also employed in the simulation of single-stage reciprocating compressors (7). The original computer program of Benson and Üçer was rewritten by Veryeri (8). The program was then extended to two-stage compressor systems. The assumption of homentropic flow in the compressor pipe systems imposes the condition of constant entropy and no heat transfer. There has been an extensive amount of research work done on the solution of unsteady gas flow in I.C Engine manifolds considering the longitudinal entropy gradients, heat transfer, and friction (9-11). A new compressor system simulation program was developed based on the experience gained from engine simulation work of Benson (12). The application of this program was first made to a simple single-stage compressor system (13). It was then extended to multiple-single-stage and two-stage compressor systems with receivers (14). This paper gives a comparison of these two compressor simulation programs applied to single-stage and two-stage systems.

COMPUTER PROGRAMS

For convenience, we will refer to the two simulation programs under consideration as:

Program I — Flow is assumed to be homentropic in pipes but frictional effects are included. Heat transfer is not considered from any part of the system.

Program II — Friction, heat transfer and longitudinal entropy gradients are included to the gas dynamic model of flow in pipes. Heat transfer from cylinder and receiver are considered.

The differences and similarities of the two simulation programs are going to be discussed in this section. Program I uses continuity, momentum, and homentropic assumption in the modelling of unsteady one dimensional gas flow in the pipes; whereas Program II uses first law of thermodynamics with continuity and momentum equations. Both programs assume the gas to be perfect, and use method of characteristics for the solution. Numerical technique used for the solution is mesh method. The time step is found from the stability of the solution using Courant, Frederics, Lewy criterion (15) in both programs. Program I uses appropriately defined Riemann variables as dependent variables of the problem (16). The dependent variables, used to solve the unsteady flow in Program II, are Riemann variables and entropy. Entropy of the gas is represented by the speed of sound after isentropic expansion from known pressure and temperature to the reference pressure (17). Both programs utilize non-dimensional variables. In transforming the variables from characteristic directions to mesh points linear interpolations are used. Empirical relations are utilized to calculate heat transfer coefficient and friction factor in Program II (13). Program I uses an empirical method for the adjustment of entropy level in the discharge pipe systems (5).

Compressor cylinder is simulated assuming no wave action in it. Generalized energy equation with the continuity equation are used in the mathematical modelling in both programs. Flow through the valves is assumed to be one-dimensional and the actual flow is reconstructed by introducing the effective flow area obtained from blowing tests. Drag coefficient of the valve is assumed to be constant. It is assumed that the flow from pipe to the cylinder through the valve is isentropic. For inflow to the pipe, it is assumed that the flow from cylinder to the valve throat is isentropic, whereas the flow from throat to the pipe is adiabatic in both programs. A pipe terminating with a converging nozzle is also treated in the same way. An iterative procedure, in which properties are corrected at the pipe ends is used for calculating inflow to the pipe in Program II (10). An iterative technique is utilized in both programs, for the simultaneous solution of valve dynamics, gas exchange, compressor kinematics, and flow at the end of pipes. Program I checks the convergence of instantaneous cylinder pressure and temperature, whereas Program II iterates on instantaneous pressure and mass in the cylinder. Valves are assumed to be one degree of freedom systems in both programs, and their equation of motion is solved numerically. Valve plate is assumed to be displaced parallel to its seat in both programs. No stiction is considered. However viscous drag, initial force and viscous damping are included.

Pipe ends, open to large volume of air at constant uniform properties, are treated as open ends using one-dimensional energy and continuity equations. For inflow to the pipe Program II uses an iterative procedure, similar to the one used for valves, in correcting the properties at the pipe ends.

Receivers are simulated, assuming no wave action and uniform properties in them at any instant of time. Pipe ends connected to the receivers are treated as open ends. Instantaneous properties in receiver are calculated using an iterative procedure in both programs (6,14).

At pipe junctions, pressure is assumed to be the same in all branches at any instant of time in both programs (11). For Program I the above mentioned pressure condition is necessary and sufficient. For Program II condition of equal entropy at every branch is also necessary. For the calculation of heat transfer from compressor cylinder in Program II, Annand's (18) equation is used to determine the heat transfer coefficient. Heat is assumed to be transferred by natural convection from the receiver, which has a constant wall temperature (14).

Both simulation programs were written, so that simulation of large, complex multi compressor receiver systems is only restricted by the available memory of the computer. The organization of the two programs are given in Fig. 1a and Fig. 1b. Figures show the general layout of the programs as block diagrams. Programs are made up of large number of subroutines of two types. Primary subroutines are used to simulate the physical phenomena at the components of the compressor system; such as pipes, compressor cylinders, receivers, junctions etc. The secondary subroutines deal with the solutions of complex equations. This organization makes it possible to change the pulse generator (reciprocating compressor in this case) without much effort into other types of positive displacement machinery such as sliding vane compressors and Roots blowers.

INPUT AND OUTPUT

Most of the data necessary to initiate the simulation is common to both programs. Program II requires more data for the calculation compared with Program I, and it reveals more information as output. Data common to both programs are: Compressor system parameters which reproduce the system to be simulated in the computer memory; pipe lengths and diameters; atmospheric pressure and temperature; initial pressure in pipes; throat area of nozzles; receiver volume and initial pressure in receiver; compressor cylinder bore; connecting rod length; crank radius; clearance volume; valve effective flow areas; viscous damping factor for valves; spring stiffnesses; drag coefficients of valve plates; crank angles for starting and stopping the calculation; crank angle which starts cylinder calculations when both valves closed; data for output organization. Program II requires the following additional data: Initial temperature of gas in pipes and receivers; pipe wall temperatures; compressor cylinder and receiver wall temperatures. The number of components of the compressor systems to be simulated are read at the entry to the program with a number of system parameters. Remaining compressor system parameters and the data necessary to simulate the components are read at the first entry to the primary subroutines.

Output may be organized as it is required by the use of data in both programs. Program I gives pressure at required crank angle steps and locations along the pipes. Valve displacements, pressure in the receivers and compressor cylinders may also be obtained at the required crank angle steps. Integrated mass flow rate, volumetric efficiency, and indicated power of each compressor in the system are also given. In addition to the output given by Program I, Program II gives temperature of the gas at required locations along the pipes and temperature in compressor and receiver, at the requested crank angle steps.

Table 1a. Results of System I

Test Specification			Percent Deviation From Experiment			
			Mass Flow Rate		Indicated Power	
Nominal Speed(rpm)	Phase Angle	Receiver Volume(m^3)	Program I	Program II	Program I	Program II
700	in phase	4.96×10^{-3}	8.8	3.4	16.6	9.7
700	180^{0} apart	1.10×10^{-3}	5.9	4.4	13.6	5.45
400	90^{0} apart	2.36×10^{-3}	3.5	2.8	14.3	7.0
400	in phase	2.36×10^{-3}	3.4	2.4	13.7	6.9
700	180^{0} apart	2.36×10^{-3}	4.7	0.3	10.3	4.1
400	180^{0} apart	4.96×10^{-3}	4.5	3.1	13.3	9.8

Table 1b. Results of System II

Test Specification		Percent Deviation From Experiment			
		Mass Flow Rate		Indicated Power	
Nominal Upstream Pressure(bar gauge)	Nominal Speed(rpm)	Program I	Program II	Program I	Program II
9.0	500	9.9	5.9	28.9	16.3
9.0	700	11.5	7.6	33.0	11.4
5.0	500	7.9	4.7	18.0	8.0
5.0	700	9.7	6.6	21.4	10.2
7.0	600	9.5	7.3	23.4	16.2
7.0	600	9.7	5.4	22.3	8.3

DISCUSSION OF RESULTS

Two different systems were simulated using programs I and II, and the results were compared with the experimental ones. For convenience we will refer to systems as:

System I — Two single-stage compressors are feeding into a small receiver. The delivery side of the system terminates with a converging nozzle operating at sonic conditions (Fig.2).

System II — A two-stage compressor is feeding into a large receiver (Fig.3).

Details of the compressors and the experimental setup of System I are given in reference (6). The two- stage compressor system is described in full detail in reference (14).

Phase angle between compressors, compressor speed, and receiver volume were the parameters changed during the tests performed by System I. Delivery pressure and compressor speed were altered during the tests performed by System II. Test conditions are tabulated in Tables 1a and 1b. Pressure predictions using Program I and II are compared with the experimental results of Systems I and II in Figures 2 and 3 respectively.

Two System I results, at completely different conditions will be discussed. Due to the 90 degrees phase angle between the compressors shown in Figure 2b the wave form produced is more complex compared to the results of the test shown in Figure 2a. The comparison between experimental pressure diagrams and predicted pressure diagrams show that both programs are succesfull in predicting the amplitude and frequency of pressure waves. However, it can be said that Program II is marginally better in computing the complex wave forms (Fig.2b). Program I utilizes the entropy adjustment in the delivery pipe system with the pipe length correction for heat transfer (6). Figure 3 shows the results obtained from System II at two different compressor speeds and delivery pressures. Second stage, of the two-stage compressor, leads first stage by 80 degrees. It is clearly seen that Program I completely fails to predict the pressure diagram at the interstage pipe. Although entropy level adjustment and pipe length correction were applied to the interstage pipe, Program I is unable to reproduce the physical phenomena. This is due to the importance of heat transfer which is not considered in the model utilized by Program I. The error in predicting the mean pressure in the interstage pipe also influences the pressure diagrams in the first and second stages.

A comparison of predicted and measured mass flow rates and indicated powers is given in Table 1. In Table 1a, total mass flow rate through system I is used in calculating percent deviation from measured value. Percent deviations of indicated powers are based on the total power consumption of the two compressors. Deviations of predicted volumetric efficiencies from measured values are not listed in Table 1. This is because they are equal to mass flow deviations since compressor capacity is reduced to the same suction state in finding experimental and predicted volumetric efficiencies. Both simulation programs predicted higher values of mass flow and indicated power almost in all cases. For system I mass flow predictions were made only slightly better by Program II. However, Program II calculated an avarage deviation of indicated power from experimental value of 7 % compared to 14 % of Program I. Program I was unable to predict the indicated power of the two-stage compressor system as expected. It must be noted that Program II also made an avarage of 11.7 % error in computing indicated power of System II. This might be due to the assumptions in the modelling of heat transfer from pipes and compressor cylinder. It is clearly seen that Program II gives better agreement with experimental results. This is due to the various energy dissipating factors considered in the simulation.

The two computer programs were also compared on the basis of execution time. Simulations were made on an I.B.M. 370/145 computer. The execution time depends on the fineness of the grid on the time distance plane and the complexity of the system to be simulated. Total number of meshes used for System I and II were 30 and 22 respectively. For fixed number of meshes a decrease in compressor speed increases the computing time due to the decrease of time step. Program I used 87 seconds to compute one crank revolution of System I while the compressors were running at 700 rpm. Program II used, on the average, six times more execution time to simulate System I. 140 seconds were used by Program II to compute one crank revolution of System II at 700 rpm. Program I simulated the same system approximately one thirds of the time used by Program II. One last point may be noted here; the difference between the execution times of Program I and II increased as the simulated system became more complex.

COMMENTS AND CONCLUSIONS

From the comparison of simulations made by the two programs, it is concluded that both programs are capable of predicting the pressure diagrams along single-stage compressor systems satisfactorily. Although Program II is slightly better than Program I in pressure diagram prediction, Program I may be preferred because it is six times faster than Program II. Simulation of two-stage reciprocating system can only be done by Program II. Program II has a superiority over Program I in predicting mass flow, volumetric efficiencey, and indicated power. However, as it is slower than Program I, program selection for simulation depends on the available computing time. Due to the organization of the programs, simulation of other positive displacement compressor systems may be made without much diffuculty. The compressor boundary condition subroutine of existing programs may be replaced by suitable mathematical models of the positive displacement compressors to be simulated.

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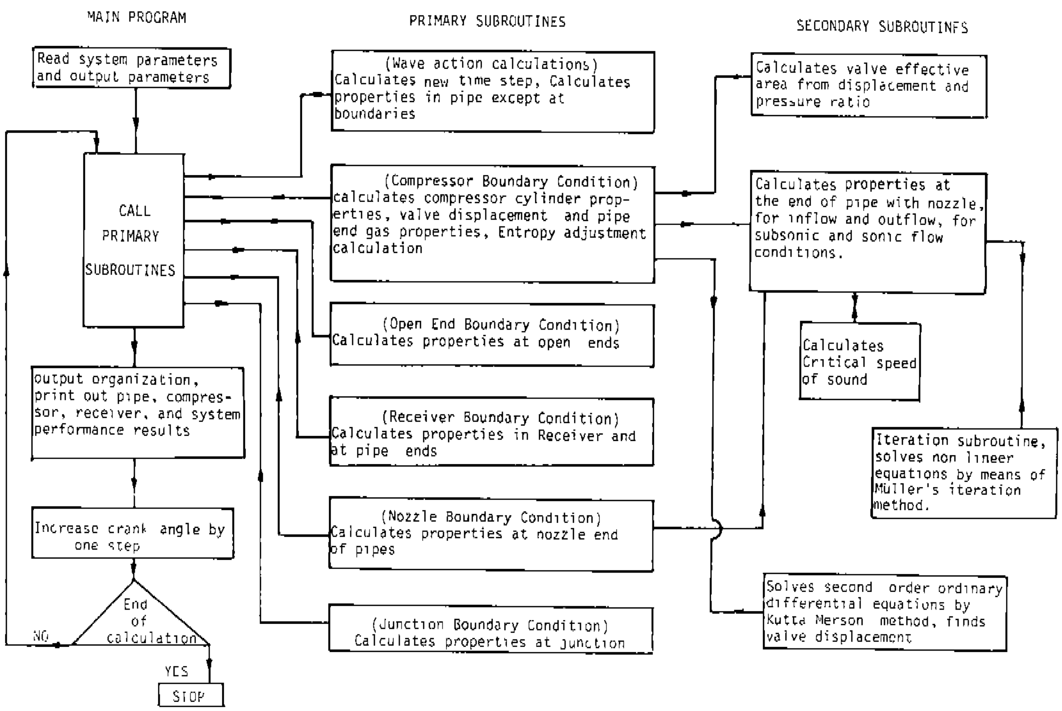

Fig. 1a. Block Diagram of Program I

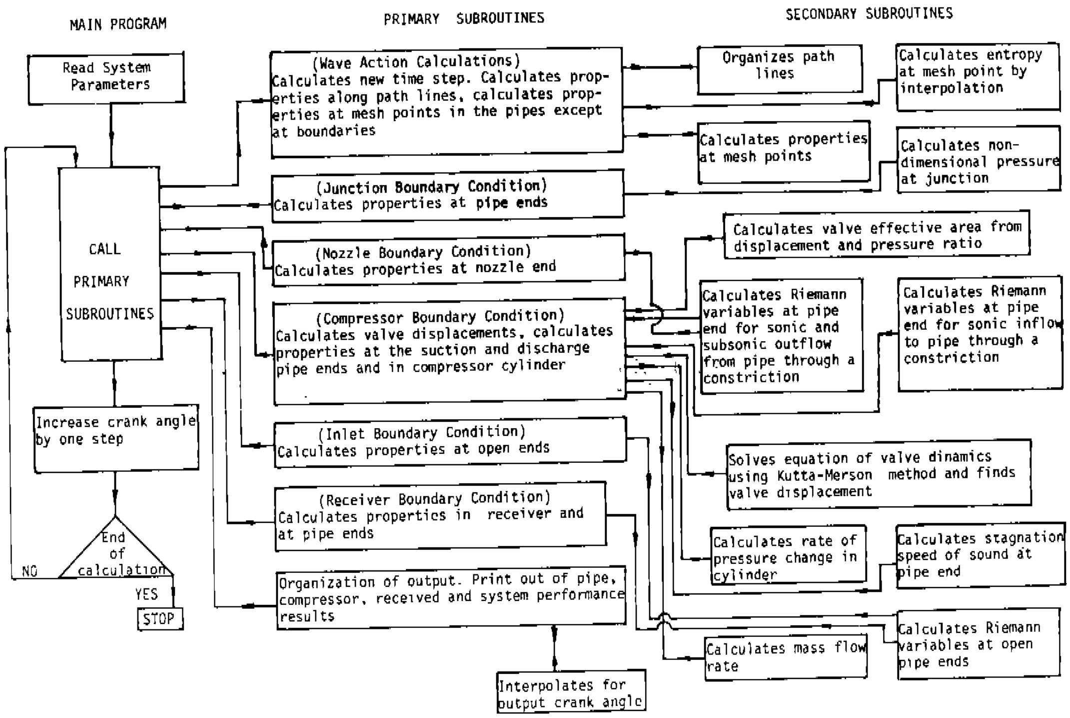

Fig. 1b. Block Diagram of Program II

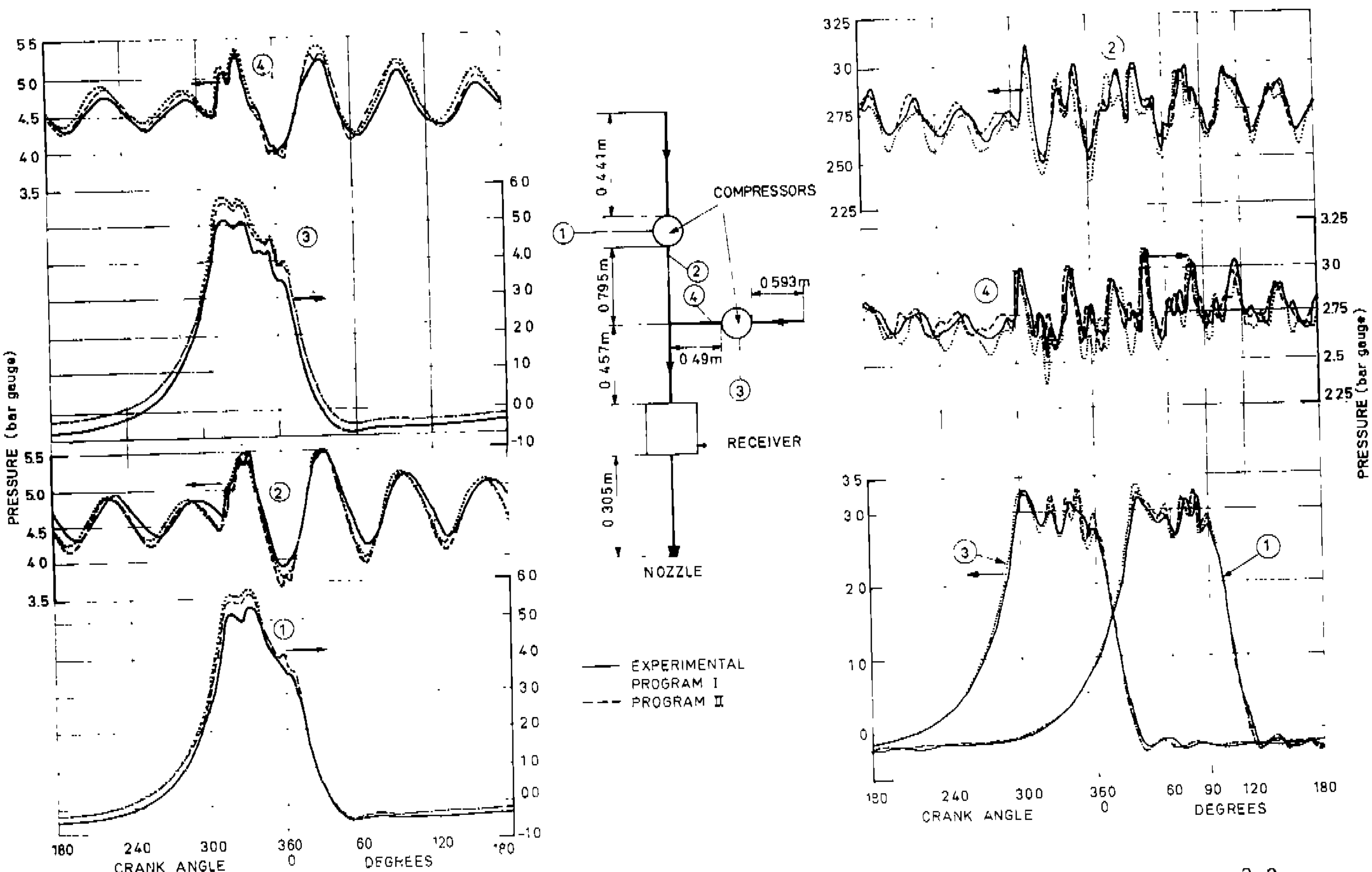

Fig.2a. 700 rpm, receiver volume $4.96\times10^{-3}m^3$

Fig.2b. 400 rpm, receiver volume $2.36\times10^{-3}m^3$

Fig.2. Comparison of Experimental and Theoretical Results (System I)

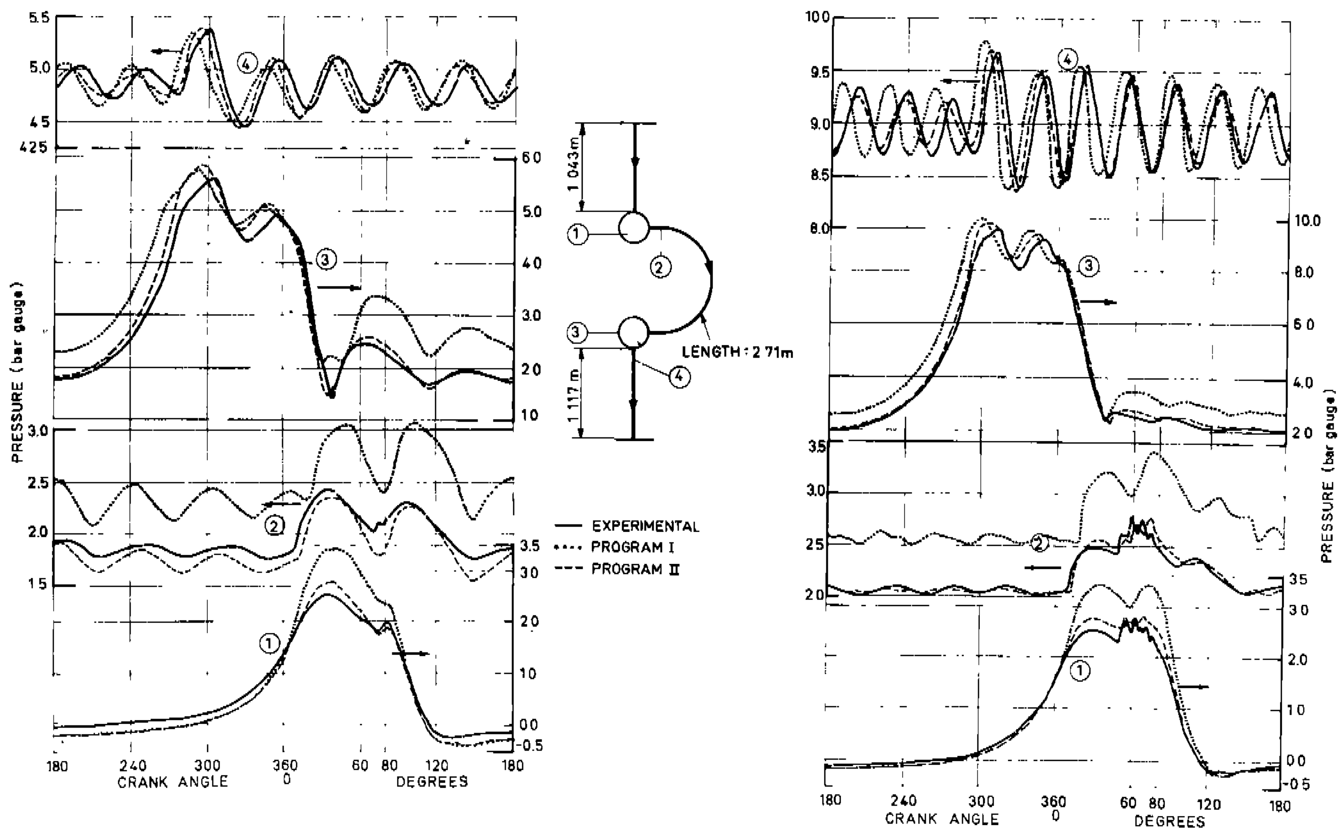

Fig.3a. 700 rpm, delivery pressure 5 bar gauge. Fig.3b. 500 rpm, delivery pressure 9 bar gauge.

Fig.3. Comparison of Experimental and Theoretical Results (System II)