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COMPUTER SIMULATION OF THE WORKING PROCESS IN THE CYLINDER OF A RECIPROCATING COMPRESSOR WITH PIPING SYSTEM

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INTRODUCTION

The non-stationary flow of gas through a piping, which results from the operation of a reciprocating compressor, is manifested both by mechanical effects upon the piping and automatic valves and by an effect upon the working process of the compressor. This non-stationary flow through a piping is very often called "gas pulsation".

Greatest attention is usually paid to the mechanical effects of pulsations,because these are often a cause of failures of piping systems as well as of automatic valves. Wherever breakdowns and stoppages of production occur due to pulsations, corrective measures are absolutely necessary and no manufacturer or user of compressor equipment will hesitate to carry them out as quickly as possible.

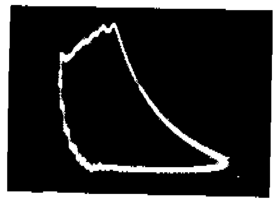
What is rather less known, is the connection between the gas pulsations in the pipings of reciprocating compressors and the energy necessary for the compression of the gas. There are cases where the compressor input is a few per cent higher due to pulsations, than necessary. The financial evaluation of these losses represents a considerable amount after a number of years. As, however, the compressor works reliably otherwise, the user normally does not take any measure for the reduction of pulsation level and often does even not realise the value of the energy losses.

An example from the practice will show the value of such losses. In Fig. 1 is a pressure-volume diagram of the working process of a compressor whose discharge section is affected very distinctly by gas pulsations (a). After a modification of the piping by addition of a perforated pipe as a damping element, the shape of the pressure-volume diagram is improved substantially (b). The compressor input has thus been reduced by 5 per cent.

Fig. 2 shows p-v diagrams deformed by gas pulsations in the discharge piping.

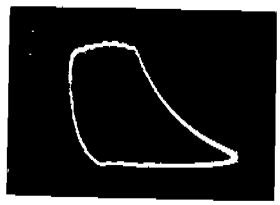
From the point of view of mechanical effects upon the compressor equipment, the gas pulsations are always considered harmful. However, their effect upon the working process of the compressor is usually not evaluated so distinctly. Now and than it is even possible to hear an opinion that the compressor output can be raised by pulsations. An extensive experimental research could prove objectively the effect of pressure pulsations upon the working process of a compressor. It need not be stressed, though, that this way not only requires much time but is also rather costly.

a)



p-v diagram
affected by pulsations in discharge pipe

b)



p-v diagram
discharge pipe with damper

FIGURE 1

Another, much faster and also much cheaper way of obtaining at least a general idea of the effect of pulsations upon the working process of the compressor, is the model technique with the aid of numerical computing methods. Thanks to computers, this technique is getting more and more popular.

ANALYTIC MODEL

The piping systems of reciprocating compressors are usually of various shapes and therefore it is not possible to create an universal model which would meet the connection of a reciprocating compressor with any piping system. If connections are sought between the working process of a compressor and the gas pulsations in a piping, this should always be done only for a certain compressor and the corresponding piping system.

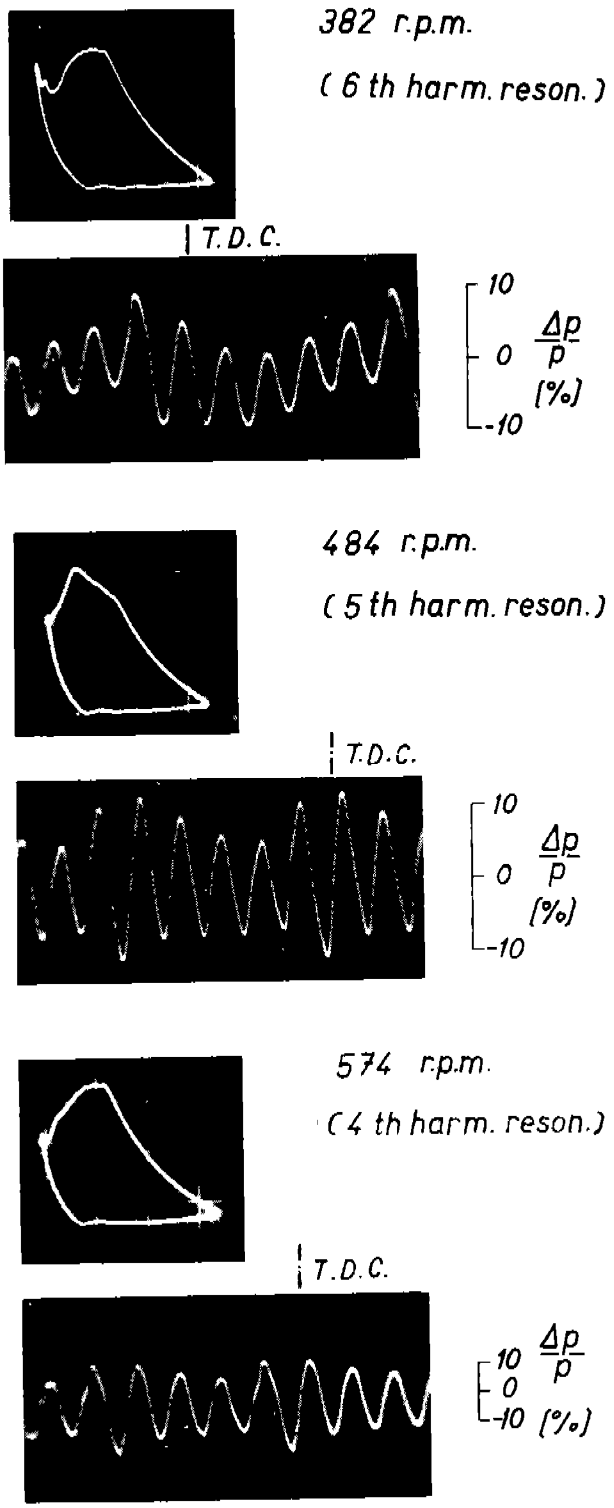


FIGURE 2

p-v diagram and gas pulsations in discharge valve chamber

Air: suction pressure 1 kp cm^{-2}
discharge pressure 3 kp cm^{-2}

Notation

a	Acoustic velocity
g	Gravitational acceleration
h	Stroke of piston
l	Length of pipe
p	Pressure in pipe
$p1$	Pressure in valve chamber
p_o	Mean pressure in pipe
r	Crank arm
t	Time
v	Velocity of gas
v_p	Piston speed
x	Coordinate of length
F_p	Area of cross section of cylinder
F_v	Flow area of valve
L	Indicated work of one cycle
Q	Volume of gas
R	Gas constant
T_s	Temperature in suction
T_d	Temperature in discharge
V	Volume of working space of cylinder
ε	Expansion coefficient
$\varkappa$	Adiabatic coefficient
λ	Length of wave
$\lambda 1$	Length of wave whose frequency equals the compressor speed
ϱ_o	Specific mass of gas at mean pressure
ϕ	Flow coefficient of valve
ω	Angular velocity
Q_s	Gas volume at suction pressure p_{os}

The selection of a suitable model is, however, facilitated by the usual practice of including dampers both in the suction piping and in the discharge piping. If a damper of suitable dimensions is chosen, a higher level of the gas pulsations will be limited practically to the piping between the working area of the compressor and the damper, while in the damper itself and in the piping behind it the gas pulsations will already be so low that their influence upon the working process in the cylinder can be neglected. Thus it will not be a great error to consider the pressure in the damper as constant and to limit the solution of the problem of the effect of gas pulsations upon the working process to the piping from the compressor to the damper.

The above considerations lead to very simple model. Fig. 3 . It is a single-acting compressor to whose suction and discharge a piping of a constant cross section is attached. The free ends of the suction and

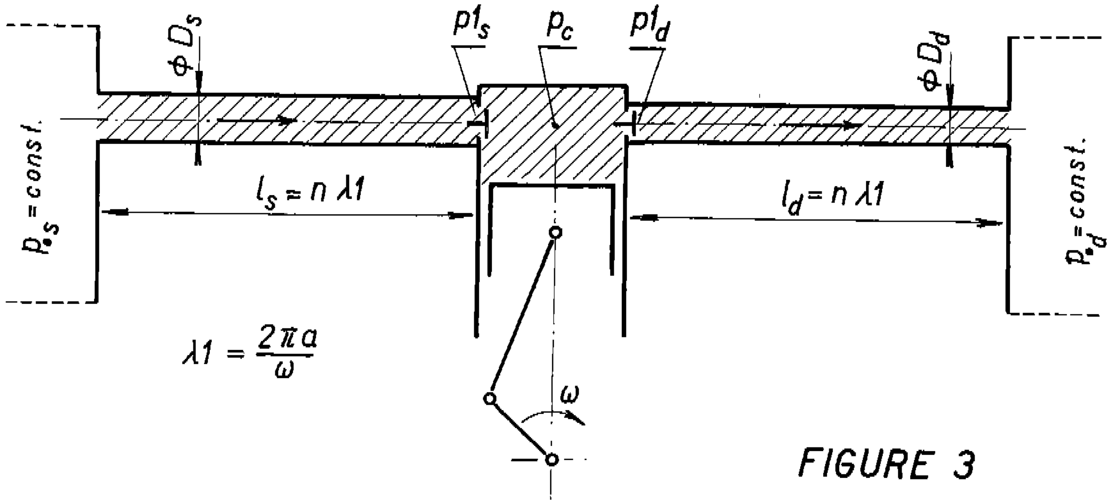


FIGURE 3

discharge pipings open into spaces with constant pressures. On such idealized model it is possible to show the effect of pulsations upon the working process of the compressor by changing the length of the suction or discharge piping from $l = 0$ to $l = 0{\cdot}25\lambda 1$, where $\lambda 1$ is the length of a wawe whose frequency equals the compressor speed. A gradual change of the length within the above range will result in appearance of resonances with the individual harmonic components of the suction or discharge impulses including the basic speed frequency.

The model, obtained so far, can be called a " mechanical model " because it still represents a machine, even though a considerably simplified one. For the determination of an analytic model corresponding to this mechanical model it is necessary to know the relations by which the studied processes are governed. These relations are generally known and their application does not cause any difficulties as long as they are used separately. However, difficulties are encountered, if it is necessary, as in our case, to study the relations simultaneously. A solution in closed form is not known yet and thus one of the possibilities of attaining some results is the numerical computation. The use of a computer is then absolutely necessary.

Analytic modelling has its specific advantages when compare with an experiment. If only a certain phenomenon is followed, the work can be made easier and other influences can be simplified. The absolute values of the results obtained by computing will most probably differ from experimental results, but the functional interdependences, which are important above all, remain unchanged.

Working space of compressor

Adiabatic compression and expansion are presumed in the working space. The influence of heat convection through the walls and the influence of leakages are not considered. To a pressure change in the working space applies the formula

$$dp = -\varkappa p \frac{dV}{V}$$

where

$$dV = v_p F_p \, dt - dQ$$

$$v_p = r\omega\left(\sin\omega t \pm \frac{r}{2l_r} \sin 2\omega t\right)$$

l_r - length of piston rod

Valves

The operation of the valves can also be idealized during the study of the influence of pulsations upon the working process in the cylinder. It is assumed that the valve will fully open, as soon as in the direction of the flow the gas pressure before the valve is higher than behind it, and that it will close fully at once if the situation is reverse. The properties of the flow will be expressed by coefficient which be considered as a constant. The gas quantity flowing through the suction valve per a time unit will then be

$$\frac{dQ}{dt} = \phi \, \varepsilon \, F_v \sqrt{2gRT_s} \sqrt{\frac{p1_s - p_c}{p_c}}$$

and that flowing through the discharge valve

$$\frac{dQ}{dt} = \phi \, \varepsilon \, F_v \sqrt{2gRT_d} \sqrt{\frac{p_c - p1_d}{p1_d}}$$

Piping

A non-stationary flow through a piping is usually expressed by relations

$$\varrho_0 \frac{\partial v}{\partial t} + \frac{\partial p}{\partial x} = - \varrho_0 z\, v$$

$$\varrho_0 a^2 \frac{\partial v}{\partial x} + \frac{\partial p}{\partial t} = 0$$

The member $-\varrho_0 z\, v$ expresses the losses which are linearly dependent on the velocity of the gas. The above relations are known as " telegraphic equation ".Here,too, it is possible to simplify the solution. In a straight piping with a constant cross section the losses occur mostly on its open end during discharge into an open space or during suction from it. Thus no great mistake will be made if inside the piping the non-stationary flow is expressed by a wave equation (where losses are neglected) and if at the end of the piping all losses are expressed summarily by Z . This coefficient indicates what part of the total gas velocity in the case of a wave proceeding to the open end of the piping will be lost after its rebound.

If reliable dependences between the course of the working process in the compressor cylinder and the gas pulsations in the attached piping are to be obtained, it is necessary to solve the preceding relations simultaneously. In connection with the solution of partial differential equations it is necessary to describe also their boundary and initial conditions.

One of the boundary conditions is the flow of gas through a valve. It is not, however, a constant function, because it depends both on the pressure in the working space and on the pressure in the valve chamber and is thus affected by the gas pulsations. Another boundary condition is the assumed constant pressure on the open end of the piping.

The initial conditions cannot be determined beforehand. They must be chosen and the calculation is to be repeated until there are no substantial differences between the values at the beginning and at the end of the computing cycle.

The method by which the non-stationary flow through the piping will be solved is decisive for the selection of the calculation method which will have to be used for the whole system of equations. The method of characteristics had proved very useful in the preceding works (1,2) and was, therefore, also used here.The differential equations of the pressure change in the working space and the equation of the flow through the valve were adapted to it and have been converted to difference equations. The whole equation system was then included in the program for the computer and was adapted for computation by the " step by step " method.

COMPUTATION RESULTS

Numerical computation is always based on concrete values, therefore it was also necessary in this case to specify the dimensions of the compressor and of the piping. The data, included in the computation, correspond to the ČKD 2 SK 240 B compressor type. It is a compressor which has been used in a number of experiments and the values from the measurement can thus be used for the verification of the computation.

Data included in computation :

Compressor :

piston diameter:	0.24 m
stroke :	0.15 m
speed :	480 r.p.m.

dead space 5 per cent of swept volume

Valves :

flow area of suction or discharge valves: 0.0044 sq.m.

flow coefficient ϕ : 0.5

Piping :

diameter of suction piping : 0.15 m
diameter of discharge piping: 0.10 m

Variants of computation :

1) Compression ratio: 4	a)	Course of suction at ideal discharge (Fig.4)
	b)	Course of discharge at ideal suction (Fig.6)
2) Compression ratio: 2		Course of discharge at ideal suction (Fig.5)

In all the three variants the length of the piping changed from $l = 0$ to $l = 0{\cdot}25\lambda 1$ gradually by $\Delta l = \frac{\lambda 1}{180}$ .Compressed gas: air.

The target of the computation program was determination of the course of pressure in the working space (p_c) and of the corresponding pulsation in the attached piping in dependence on time. The values were calculated for one work cycle with a step corresponding to the rotation of the crankshaft by 2 degrees. Air pulsations were specified by values of pressure and flow per time unit at the beginning of the piping and by values of flow per time unit at the end of the piping.

From these basic values were derived: the indicated work of one work cycle

$$L = \sum_{0}^{\frac{2\pi}{\omega}} p_c F_p v_p \Delta t$$

and the total delivered quantity of air by one work cycle

$$Q_s = \sum_{0}^{\frac{2\pi}{\omega}} \frac{dQ_s}{dt} \Delta t$$

The print of the computation results of the basic program was not suitable for a dis - tinct elaboration of the influence of the piping length and thus also of pulsations upon the operation of the compressor. A further program was therefore derived, in which only the values L and Q were printed identically with the original program, and then the discharge efficiency

$$\eta = \frac{Q_s}{F_p\, h}$$

and the maximum and minimum pressure amplitudes of the gas pulsations at the beginning of the piping (in the valve chamber), which occured during the computed cycle.

A further modification of the results made it possible to determine also the dependence of the indicated work on the length of the piping in the form of the proportional values:

relative work of one cycle

$$\frac{L - L_o}{L_o}$$

and

relativ specific work of one cycle

$$\frac{L/Q_s - L_o/Q_{os}}{L_o/Q_{os}}$$

where L_o is the indikated work and Q_{os} the total delivered quantity of air by one work cycle without influence of pulsations. The graphs of the results are shown in Figs. 4 to 6 .

CONCLUSION

The analytic modelling based on numerical computation methods is always limited to actual cases. Neverthless, if the results are modified adequately, they can be given a wider validity than that corresponding to the computed problem.

If gas pulsations are considered from the point of view of economy, they always mean a loss in the discharge. A slight increase of the discharge efficiency caused by suction from the dead space by the inertia of the flowing gas, has no practical sense. A distinct increase of losses is observed wherever there is a higher pulsation level due to the resonance of the gas column. The first three harmonic components of the discharge pulse are manifested most.

A comparison of the discharge with a compression ratio of 2 and a compression ratio of 4 shows that the influence of pulsations is manifested to a relatively greater degree in the case of the lower compression ratio.

The situation in the suction seems to be more favourable. By a suitable tuning of the gas column in the suction piping it is possible to increase the discharge efficiency. In our case this increase was approx. 7 per cent. Literature contained an example where an increase of the output by as much as 14 per cent was reached by way of experiment in a compressor with double-acting first stage (3).

The computation indicates, identically with the said experiment, that the maximum discharge efficiency is obtained at a resonance with the 2nd harmonic component of the suction pulse. Fig. 4 shows that the final phase of the suction is of decisive importance for the compressor output.

From the point of view of economy, however, it is instructive that losses grow faster than the discharge efficiency. Therefore both the ratio

$$\frac{L - L_o}{L_o} \quad \text{and} \quad \frac{L/Q_s - L_o/Q_{os}}{L_o/Q_{os}}$$

reach higher values at a resonance than at a low pulsation level. It is natural, because a higher pulsation level brings along increased losses which can be covered only by increased compression work. However much the ratio between the compression work and the delivered quantity of gas may be affected by the character of the losses, in no case should compressed gas be obtained more advantageously by means of pulsations than without them.

It would be premature to make a definite conclusion regarding the influence of pulsations upon the working process of a compressor. The submitted paper should be therefore considered only as a prognosis of this influence, meant to render the final solution easier.

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WORKING PROCESS AND GAS PULSATION IN SUCTION PIPE

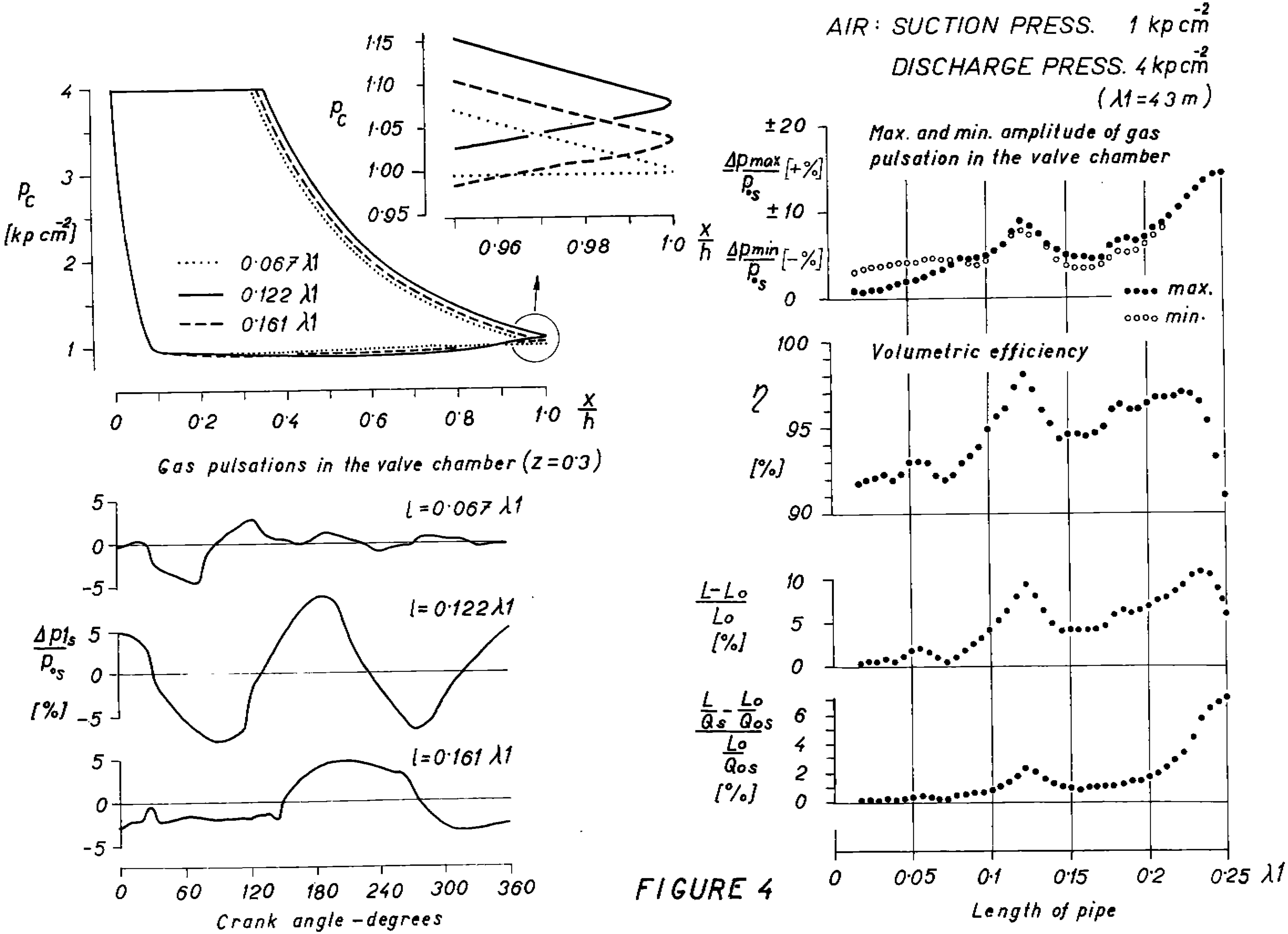

FIGURE 4

WORKING PROCESS AND GAS PULSATION IN DISCHARGE PIPE

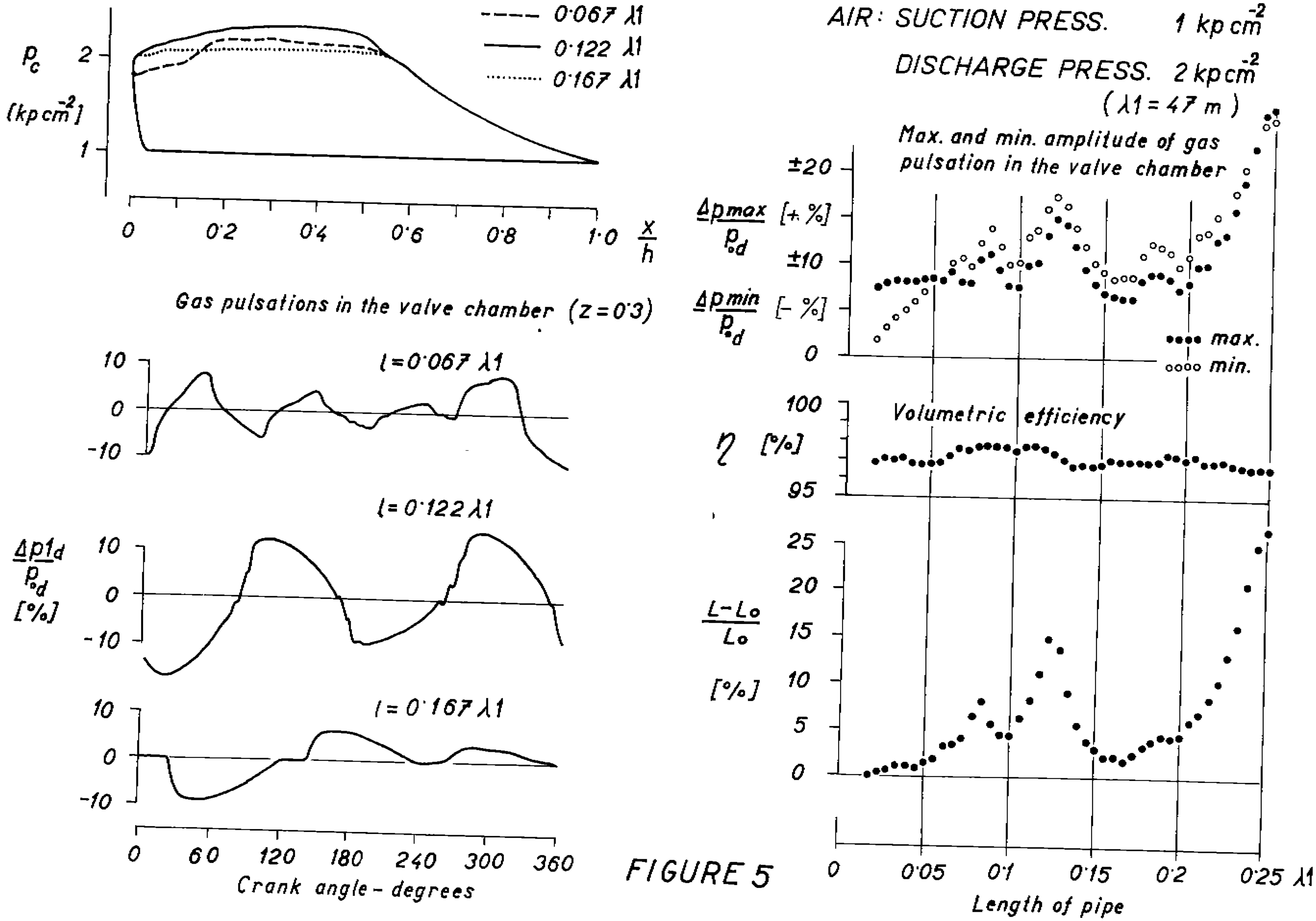

FIGURE 5

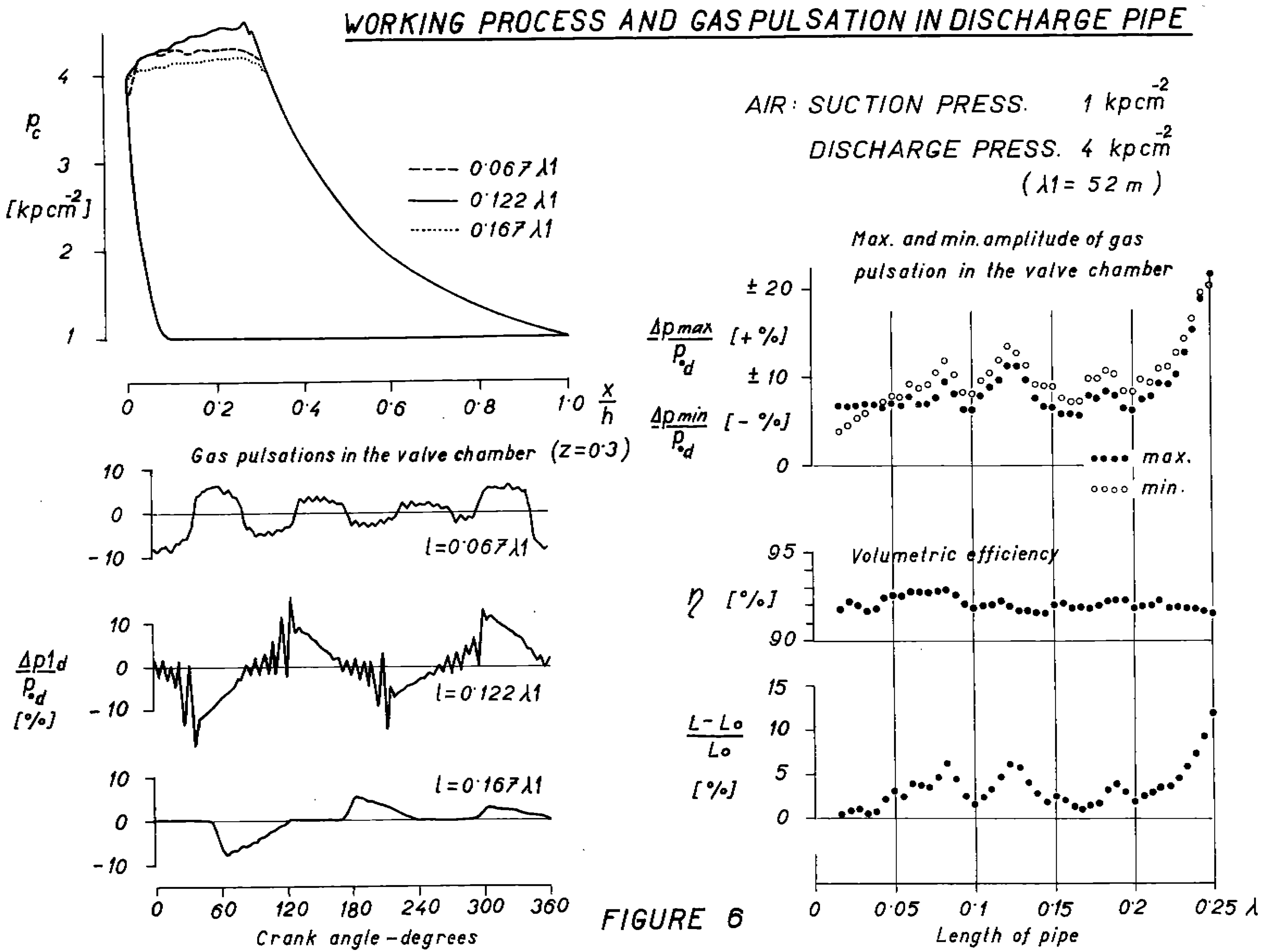
WORKING PROCESS AND GAS PULSATION IN DISCHARGE PIPE
AIR: SUCTION PRESS. 1 kpcm⁻²
DISCHARGE PRESS. 4 kpcm⁻²
(λ1 = 52 m)
p_c [kpcm⁻²]
4
3
2
1
---- 0·067 λ1
—— 0·122 λ1
······ 0·167 λ1
0
0·2
0·4
0·6
0·8
1·0
x/h
Gas pulsations in the valve chamber (z=0·3)
10
0
-10
l=0·067 λ1
l=0·122 λ1
l=0·167 λ1
Δp1d/p₀d [%]
0
60
120
180
240
300
360
Crank angle - degrees
Max. and min. amplitude of gas pulsation in the valve chamber
± 20
± 10
0
Δp max/p₀d [+%]
Δp min/p₀d [-%]
•••• max.
oooo min.
Volumetric efficiency
η [%]
95
90
(L-Lo)/Lo [%]
15
10
5
0
0
0·05
0·1
0·15
0·2
0·25 λ1
Length of pipe

FIGURE 6