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Computer Simulation of Valve Dynamics as an Aid to Design

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INTRODUCTION

The paper presents some results from a research work carried out at the Division of Refrigeration Engineering at the Norwegian Institute of Technology under the guidance of professor G. Lorentzen. The purpose of the project was primarily to develop a computer simulation model for calculating the valve behaviour under operating conditions, taking into consideration the pressure pulsations in suction and discharge. The computer model may be applied in many ways. Simulated test runs of a proposed valve design can be arranged in short time. It is also possible to use the model to develop simple design rules, so that optimal valve designs can be approached by simple calculation.

The work was primarily concentrated on ring-plate valves, but the results may be applied to other designs.

COMPUTER MODEL DEVELOPMENT

Experimental Works

The computer model has to be calibrated by comparing the calculated behaviour with measurements on real valves under normal operation. The major aim of the measurements is to find the relation between forces and valve reaction. The gas flow through the valve produces a force which determines the valve movement together with the spring forces and other minor effects (friction, oil sticking etc.). The gas force depends on the pressure difference across the valve and the flow conditions. It is therefore necessary to indicate the pressure oscillations on both sides together with the valve movement. It is desirable to be able to calibrate the measuring equipment during operation to eliminate the possible influence of temperature and oil content on the transducer characteristics.

An experimental discharge valve with the necessary electronic equipment was developed and installed in a single-cylinder compressor of the uniflow type, Fig. 1. The valve lift is indicated in three points around the pheriphery in order to discover any tilting. Capacitance transducers were used and the dimensions of the electrodes appear in Fig. 2. The transducers were calibrated statically before installation. In order to make it possible to check the calibration during operation, thin horizontal electrodes were installed at definite levels. This arrangement is also shown in Fig. 2 (3). When the valve plate passes the electrode, a sharp capacity signal peak is produced. When the signals from the two types of pick-ups are recorded together, it is possible to fix the necessary calibration level, as shown by an example in Fig. 3.

The pressure pulsations were indicated by means of piezoelectric transducers. Fig. 4 shows a cross section of the pick-up which was used to calibrate the pressure signal. A small plate (A) is balanced between the system pressure p_1 and the constant calibration pressure p_2. Whenever the pressure difference across the plate is turning, the plate starts to move from one seat to the other. Again the capacitive effect is used to indicate this motion. The delay due to inertia is small, as evidenced by the sample recording in Fig. 5.

During the experiments the compressor was working with air in an open system. The plant allowed continuous variation of compressor speed and pressures. The temperatures and static pressures throughout the system, together with the compressor pumping capacity and energy consumption, were determined by standard methods. The valve and pressure oscillations were recorded by an UV-recorder. The recordings were digitalized and fed into a computer. Process diagrams could be drawn to any desired scale by means of an automatic drawing machine.

Test series were run on two different valves. Fig. 6 demonstrates a typical result from the first one. The installed springs were too strong, choking the gas force, so that the valve could not open completely. It oscillated heavily, bouncing back on the valve seat several times during the opening period. The mechanical strain on the valve plate was increased, and such oscillations should be avoided.

When the valve was operated without springs in the second series, it behaved more favourably from a mechanical point of view, as can be seen in Fig. 7. However, the valve closing was delayed, and this caused blow-back and reduced volumetric efficiency. This was confirmed by the direct capacity measurements.

Computer Model

The simulation model requires mathematical expressions for the physical processes going on in the compressor. For the mathematical treatment the system was divided into three separate parts, mutually exchanging mass and energy. In the case of the discharge valve these are the cylinder, valve and discharge line system. The connections between the systems are visualized in the schematic block diagram of Fig. 8.

The cylinder process is described by the equation:

$$\frac{dp_c}{p_c} = \kappa \left(\frac{dm_c}{m_c} - \frac{dV_c}{V_c} \right) \qquad (1)$$

The cylinder volume V_c and its variation dV_c are known functions of the crank angle θ. The mass leaving the cylinder is calculated by the formula:

$$dm_c = - \frac{\alpha A_o}{h_o} \cdot h \cdot \sqrt{2\rho_c (p_c - p_d)} \cdot \frac{d\theta}{\omega} \qquad (2)$$

The flow coefficient α is known from static measurements on the test valve.

When the valve plate moves between the stops, the behaviour can be described by the simple dynamic equation, assuming one degree of freedom, which was confirmed by the experiments:

$$\frac{d^2h}{d\theta^2} = \frac{1}{m_v \cdot \omega^2} \cdot \Sigma F_v \qquad (3)$$

All the forces which could possibly affect the valve behaviour, have to be included and evaluated. The main forces are the following:

Gas force : $F_g = C_d \cdot A_v \cdot (p_c - p_d)$

Spring force: $F_s = F_{so} + C_s \cdot h$

The drag coefficient and spring characteristic are found by static measurements on the valve. Additionally the model includes the forces caused by damping, friction and gravity. The mathematical description of these forces is omitted here, but they include unknown "constants" which have to be found by the calibration of the model.

The effect of opening and closing delay due to oil sticktion at the boundaries is taken into account, and so is the reflection when the valve plate hits the seat or the stop.

The main purpose of the discharge line model is to yield the pressure p_d, needed in eq. 2. This pressure depends on the mass entering the discharge line through the valve and the pressure pulsations which are reflected from the open end. The discharge line process was calculated by a method of characteristics, described by Benson [1]. In the case of the test valve the gas was flowing directly into the discharge line and produced a known particle velocity, which is used as boundary condition at the valve end of the line. At the other end the pressure is assumed to be constant.

The result of the mathematical deductions is a system of simultaneous differential equations. These can be solved by stepwise integration, using the Kutta-Merson numeric method with adjustable integration steplength. The typical CPU-time needed for a complete simulation of two revolutions on the UNIVAC-1108 computer was 40-50 s. The computed process is presented as tables and drawings, allowing easy comparison with the measured results. The computer also calculates the mass delivered by the compressor and the valve energy losses during one revolution.

By the calibration good agreement between measured and calculated process was attained, both for valve behaviour and pressure oscillations. This can be seen from Figs. 9 and 10, giving some typical results. It was found that the gas and spring forces are the most important ones. The other influences may be neglected without any great error, at least for the valve type used in this project. The valve reflection at the seat was found to be very low ($C_{Refl} \approx 0.01$). At the stop the reflection depends on the actual geometry. When distance pins are used to reduce the maximum lift, a reflection coefficient in the order of 0.4 was found.

A similar model was developed for the suction valve system, applying the experience gained in analysing the discharge.

COMPUTER MODEL APPLICATION

Dynamic Design

As a result of the experiments the need for more adequate spring design rules was recognized. The simulations show that it is the balance between gas force and spring force which mainly determines the valve behaviour. The gas force variation is to some extent fixed in a specified compressor process, and the problem of choosing the appropriate springs remains. They should be sufficiently weak so that the gas force can hold the valve fully open. When the gas force decreases towards the end of the stroke, the springs should close the valve without oscillations and in time to avoid blow-back. That means that the valve should close exactly at pressure equalization between cylinder and suction or discharge.

Based on the experiences from computer model simulations a simplified equation for the valve closing process was designed. This could be solved to give a simple equation for calculating the optimal spring stiffness. The solution is based on the assumptions that no oil sticktion occurs, and that the springs have no initial force ($F_{SO} = 0$). The solution may be described by the following dimensionless formulaes:

$$\text{Discharge valve:} \quad B_* = 3.22 \cdot A_*^{0.53}$$

$$\text{Suction valve} \quad : \quad B_* = 2.16 \cdot A_*^{0.51} \qquad (4)$$

$$A_* = \frac{1}{2} \cdot C_d \cdot \left(\frac{A_{cyl} \cdot r}{\alpha A_o} \right)^2 \cdot \frac{A_v \cdot \rho_c}{m_v \cdot h_o}$$

$$B_* = \frac{\omega_s^2}{\omega^2} = \frac{C_s}{m_v \cdot \omega^2}$$

$$C_s = m_v \cdot \omega^2 \cdot B_* \qquad (5)$$

When the geometry and working conditions are given, together with the drag coefficient (C_d) and flow coefficient (α) of the valve, the dimensionless drag force A_* may be evaluated. The dimensionless spring force B_* and the optimal spring stiffness C_s can then be calculated from the equations given.

The design system was tested on the computer models. When the proposed springs were used, satisfactory valve behaviour was obtained, as evidenced by the simulation examples given in Figs. 11 and 12. For the suction valve delayed pressure equalization could occur and result in valve malfunctioning. The simulation in Fig. 12 shows that this problem is handled automatically. The design formulae were also used to calculate new springs for the discharge test valve, and good operation was obtained, Fig. 13.

The equations were based on the assumption of zero initial spring force. However, simulations indicate that initial spring force may be used without affecting the valve behaviour much. But then the spring constant must be adjusted, so that the maximum spring force at the valve stop is not changed. When the valve is operated not too far from the originally specified working conditions, good valve behaviour is maintained. Too low compressor speeds may lead to increased volumetric losses.

Other details may also be important to the dynamic behaviour of the valve. When the discharge or suction chamber volume is small, the pressure pulsations may affect the valve operation. This is demonstrated by the simulation run in Fig. 14. A sudden pressure decrease reaches the valve at the moment of closing, causing an extra valve opening and blow-back, with reduced volumetric efficiency (10 percent loss). This problem could be corrected by adjusting the discharge line length to omit sudden pressure increase or decrease at the dead center, Fig. 15. Consequently the pipe line length has to be considered in special cases. By the simulation shown in Fig. 15 an extra volumetric gain of about 3 percent was achieved as a result of the extra draining of the cylinder by low discharge pressure at the time of valve closure. The effect may be utilized intentionally both for the suction and discharge system to gain marginal volumetric improvement.

With the developed equations it should be possible to design the valves to function properly for a given compressor under specified working conditions. In special cases, where for instance the pipe-line pressure pulsations might affect the valve process, the computer model has to be used. However, the design formulae will be of benefit even here, giving optimal valve designs with less simulation runs, thereby saving expensive computer runtime.

REFERENCES

[1] Benson, R.S. and Üçer, A.S.: "An Approximate Solution for Non-steady Flows in Ducts with Friction". Int. Journal Mech. Science, vol. 13, 1971, pp. 819-824.

[2] Bredesen, A.M.: "Undersøkelse av ventilenes bevegelse i stempelkompressorer". ("Investigation of the valve behaviour in piston compressors"). Thesis for the technical licentiate degree at the Norwegian Institute of Technology, 1973, 319 pages in Norwegian.

LIST OF SYMBOLS

A_o	- maximum valve flow area	m^2
A_v	- valve plate front area	m^2
A_{cyl}	- cylinder cross section	m^2
A_*	- dimensionless gas force (defined in text)	-

B_*	- dimensionless spring force (defined in text)	-
C_d	- drag coefficient	-
C_s	- spring stiffness	N/m
F_v	- force acting on the valve	N
F_g	- gas force	N
F_s	- spring force	N
F_{so}	- initial spring force	N
h	- valve lift	m
h_o	- maximum valve lift	m
m_c	- mass content of cylinder	kg
m_v	- valve plate mass	kg
n	- compressor speed	rpm
p_c	- cylinder pressure	bar
p_d	- discharge line pressure	bar
r	- crank radius	m
V_c	- cylinder volume	m^3
α	- flow coefficient	-
θ	- crank angle	rad
κ	- adiabatic exponent	-
ρ_c	- density of cylinder gas	kg/m^3
ω	- compressor angular velocity	rad/s
ω_s	- resonance angular velocity of spring system	rad/s

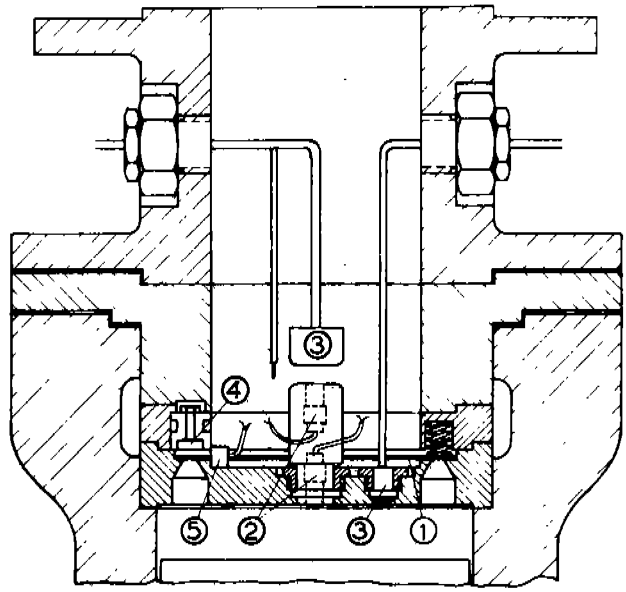

Fig. 1 Test valve with instrumentation

1 Valve plate
2 Piezoelectric pressure transducers
3 Pressure calibration pick-up
4 Capacitance valve displacement transducer
5 Valve displacement calibration pick-up

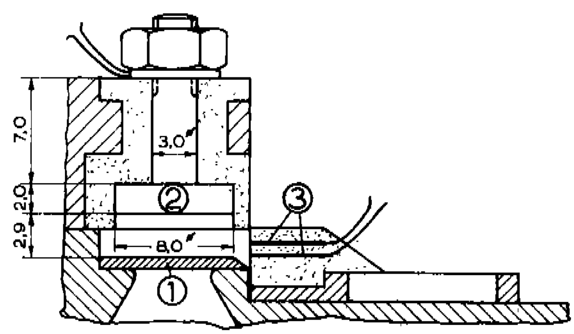

Fig. 2 Displacement transducer arrangement

1 Valve plate
2 Capacitance displacement transducer
3 Calibration electrodes

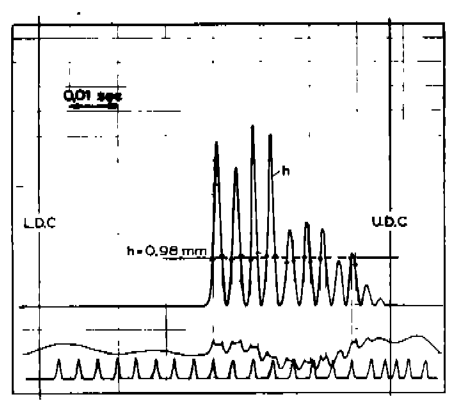

Fig. 3 Calibration of valve lift signal

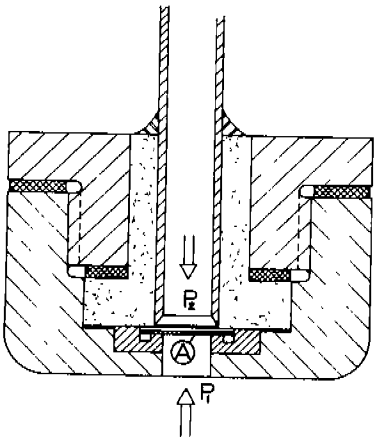

Fig. 4 Pressure calibration pick-up

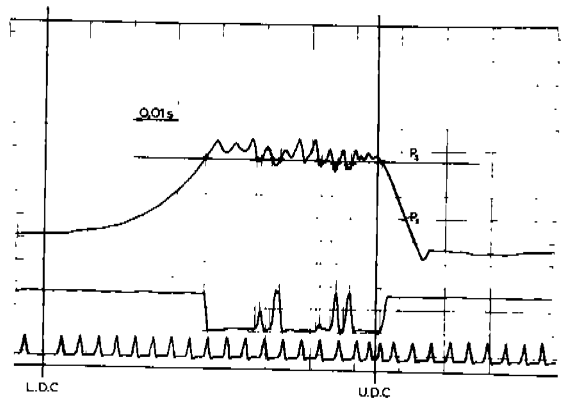

Fig. 5 Calibration of signal from the cylinder pressure transducer

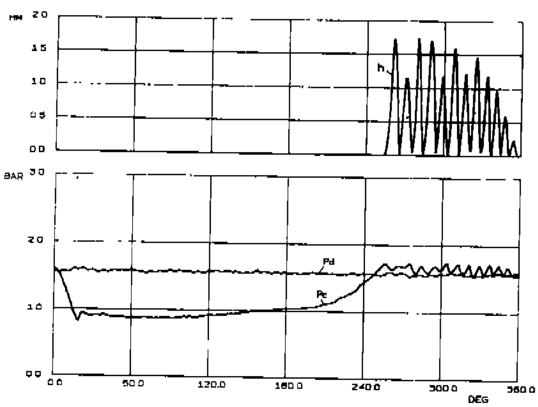

Fig. 6 Typical experimental result. Discharge valve operated with springs (F_{SO}=12 N, C_S=11400 N/m). Compressor speed 400 rpm.

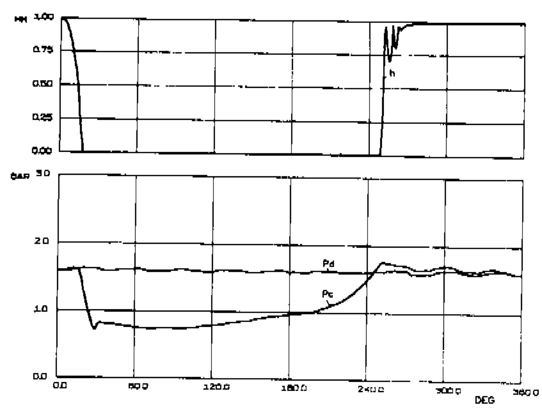

Fig. 7 Typical experimental result. Discharge valve operated without springs. Compressor speed 400 rpm.

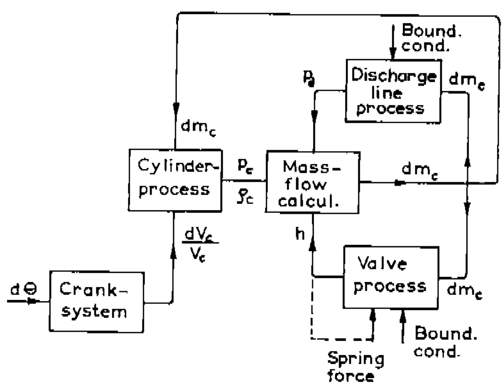

Fig. 8 Schematic block-diagram of discharge valve process

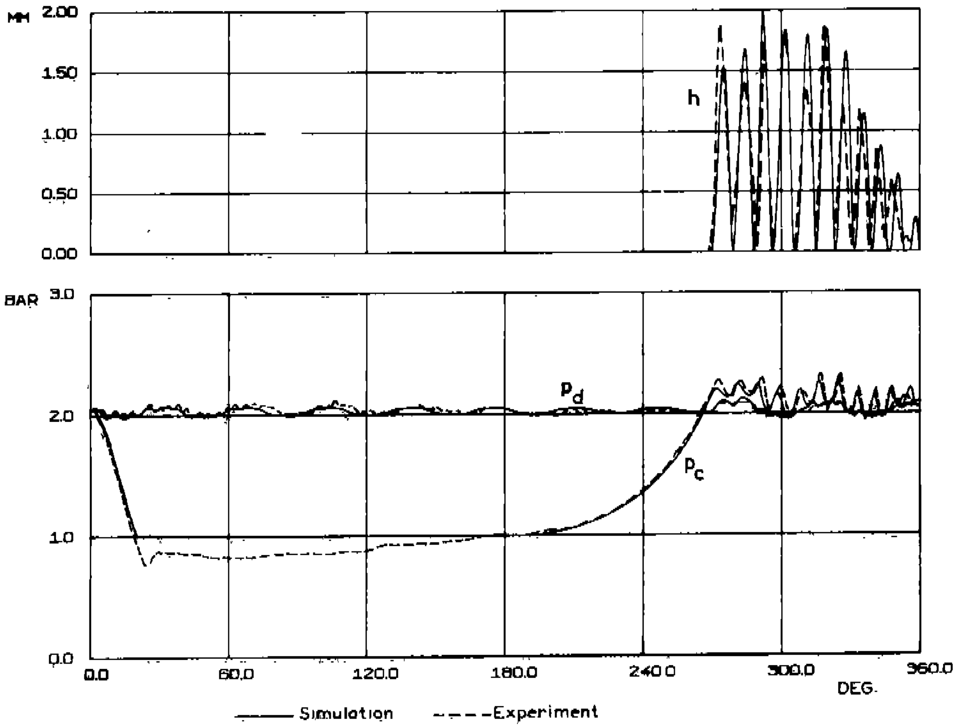

Fig. 9 Comparison of experiments and computer simulation for discharge valve operated with springs. Compressor speed 400 rpm.

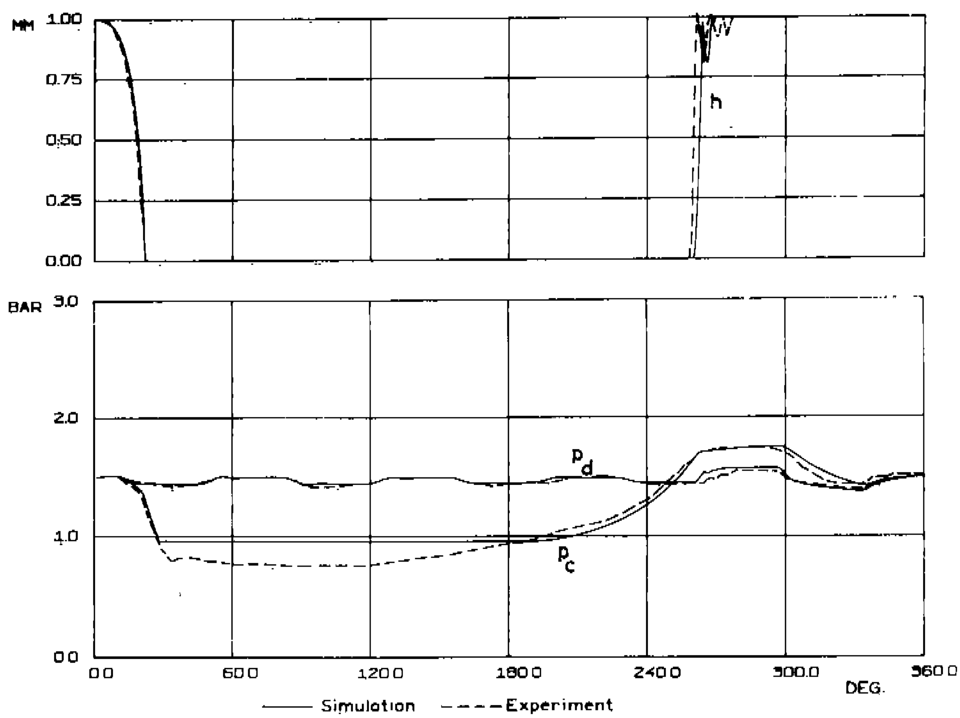

Fig. 10 Comparison of experiments and computer simulation for discharge valve operated without springs. Compressor speed 800 rpm.

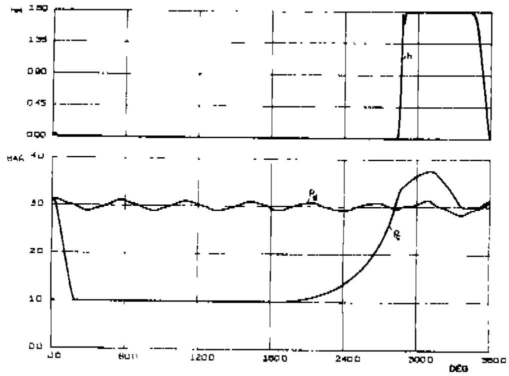

Fig. 11 Computer simulation of a discharge valve process. Valve springs designed according to eqs. 4 and 5. Compressor speed 600 rpm.

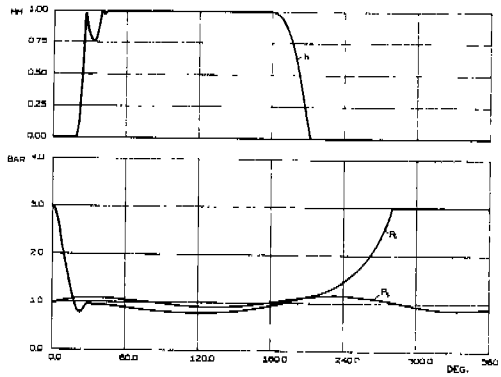

Fig. 12 Computer simulation of a suction valve process. Valve springs designed according to eqs. 4 and 5. Compressor speed 1450 rpm.

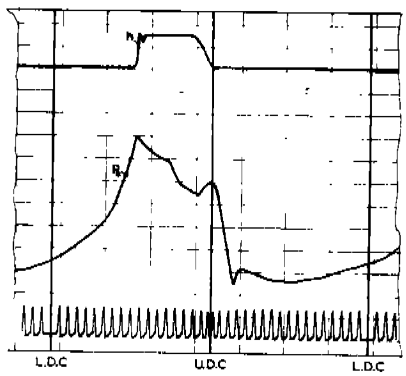

Fig. 13 Experiment recording of discharge test valve fitted with springs calculated by eqs. 4 and 5. C_s = 6000 N/m. Compressor speed 850 rpm.

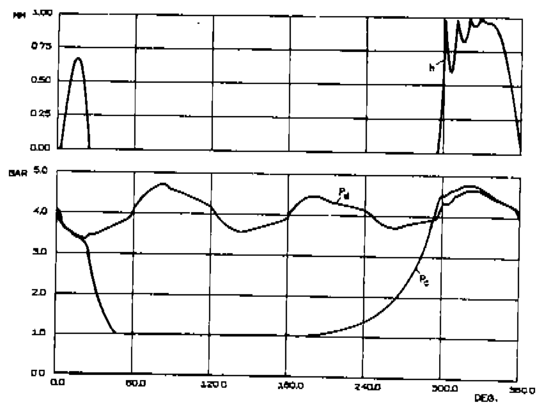

Fig. 14 Computer simulation demonstrating possible influence of discharge line pressure oscillations on the valve behaviour. Discharge line length 1.38 m. Comp. sp. 1450 rpm.

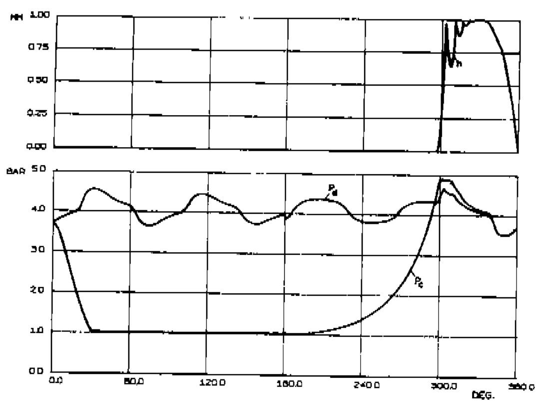

Fig. 15 Same as fig. 14, but the discharge line length has been adjusted to 0.92 m.