# SOME ASPECTS OF LABORATORY TESTING AND COMPUTER MODELLING OF TWO-STAGE COMPRESSORS WITH INTER AND AFTER-COOLING

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SYNOPSIS Laboratory testing and computer simulation of a two-stage reciprocating air compressor with inter and after-cooling was undertaken with the purpose of studying the physical processes occurring in the compressor and obtaining information useful for design purposes. The testing includes measurements of instantaneous temperature and pressure in suction and delivery chambers and interconnecting pipes in both cylinders, instantaneous torque and average quantities, during the steady and transient state operation. Results are presented and the problems are discussed. The suitability of the micro-thermocouple for measurement of instantaneous temperatures, its dynamic behaviour and strength limitations, as well as mounting and calibration problems are considered. The measurement of pulsating pressure by means of piezoelectric and capacitive transducers in combined use are presented with specific reference to the effect of transducer location.

Results are reported of the computer modelling of the thermodynamic processes including the unsteady gas flow in the piping system with friction and heat transfer effects, and comparison made with experimental data. Some practical implications are discussed of the testing and computer modelling upon design.

## INTRODUCTION

1. The increasing use of reciprocating compressors and demands for improvement of compressor performance and reliability, have prompted both the development of laboratory testing techniques and computer simulation methods. These two approaches seem to complement each other; the simulation method, being considerably less expensive, appears particularly suitable for the prediction of the behaviour of new or redesigned compressors, but has to draw heavily upon experimental data relating to valve characteristics, fluid flow, heat transfer, etc.

2. Some experience of both laboratory measurements and computer modelling of single and two stage reciprocating compressors are reported here. The experiments involved the measurement of the instantaneous pressure and temperature in cylinder and suction and discharge chambers, and of the instantaneous torque. Also, some results from the model of the two stage compressor will be discussed.

## EXPERIMENTS

3. The oscillating character of the basic thermodynamic processes in reciprocating compressors imposes severe requirements upon the instrument characteristics, particularly when instantaneous variables are to be measured. Although the fundamental frequency of the oscillating process in a typical compressor is only of the order of several tens of Hz, sharp peaks appear on temperature and pressure-time profiles. Superimposed are high frequency pulsations associated with valve movement and pressure wave propagations, which demand good dynamic response of the instrument. Acceptable frequency response is needed, with small signal distortion. Furthermore, transducers must fit into the available space and the presence of the transducers should not disturb the flow. All these factors pose serious constraints upon instrument selection.

4. Various methods for temperature measurement in reciprocating compressors have been reported, but thin wire based transducers, such as resistance thermometers and thermocouples, seem to have the widest applicability. The limited space available for mounting, and the high frequency of temperature change dictate the use of temperature sensing devices of miniature dimensions. However they may not possess the mechanical strength and durability to endure the dynamic forces being exerted upon them. It is known that the resistance thermometer has better sensitivity than the thermocouple of the same wire diameter because the thermocouple head usually has a larger dimension, but the resistance thermometer wire must be longer to achieve the same sensitivity in order to eliminate the influence of the ends. For the measurements described here Cr-Al microthermocouples of various wire diameters below 100 microns were used; that with the wire thickness of 12.5 microns was selected as the optimum satisfying both dynamic response

and mechanical durability.

5. The temperature sensed by the wire and the real gas temperature are related by the energy equation which defines the instantaneous heat balance of the thermocouple head. Neglecting the radiation heat exchange between the thermocouple and the gas:-

$$(T_g - \beta \frac{w^2}{c_p} - T_w)\alpha D\pi dxdt = \rho c D^2 \frac{\pi}{4} \frac{\partial T_w}{\partial t} dxdt +$$

$$+ \sigma\epsilon (T_w^4 - T_s^4) D\pi dxdt + D^2 \frac{\pi}{4} \frac{\partial^2 T_w}{\partial x^2} dxdt \qquad (1)$$

The required gas temperature can be evaluated by the numerical solution of equation (1) if the heat transfer coefficient and the temperature distribution along the wire are known together with other coefficients appearing in the equation. Some solutions of the simplified forms of equation 1 are available in literature; in particular if the heat conduction term is neglected then equation 1 becomes an ordinary differential equation, (Ref. 1). With the assumption of a constant heat transfer coefficient and a knowledge of the temperature and geometry of the surrounding walls, the equation can be further simplified and solved for any given thermocouple. In special cases, when the changes of the gas temperature and the heat transfer coefficient are defined by a simple analytical function, an analytical solution can be obtained, but this can serve only for the theoretical analysis of the thermocouple response to a regular periodic input (18). The method based upon the use of two thermocouples of different dimensions which enable the elimination of the unknown heat transfer coefficients may be mentioned here also. However, often it suffices to consider the simplified form of equation 1 containing the heat transfer term only.

$$\tau \frac{dT_w}{dt} = T_g - T_w \qquad (2)$$

where $\tau = \rho c D/4\alpha$ represents the time constant of the thermocouple. The other effects represented by the other terms in equation 1 are accounted for through static errors, evaluated for each effect separately. The evaluation of the thermocouple time constant is not always straightforward due to the uncertainty in evaluation of the heat transfer coefficient. Standard relationships of the form $Nu = CRe^m$ have been used for evaluating the values of the constants with C and m being proposed for each particular geometry and physical condition, but the main problem seemed to be the accurate assessment of the effective gas velocity needed for the calculation of the Re number. Because of the uncertainty in evaluating the gas velocity, values between 2 and 10 m/s were tested in order to assess the influence of the gas velocity upon the thermocouple wire response. In this range the Reynolds number ranges in value from 0.68 to 3.4 for which, according to Hilpert, C and m should be assigned the values of 0.89 and 0.33 respectively. Because of the small value of the exponent m, the influence of the gas velocity is rather small. So, for 2 m/s the time constant has a value $\tau = 4.88$ ms, while for 10 m/s $\tau = 2.85$ ms. Periodic changes of pressure and temperature, causing periodic mechanical stresses in the thermocouple, require careful manufacture of it in order to achieve satisfactory durability. In addition difficulties are encountered when connecting the thermocouple wire with compensating leads of considerably larger diameter. Fig. 1 shows two of several variants of the thermocouple used in the experiments. The first achieved an average life time of about four working hours, while the second one, with connections between the thermocouple wire and the compensating leads outside the body, gave considerably better endurance.

6. In order to verify the thermocouple time constant and evaluate experimentally its frequency response, the dynamic calibration of the thermocouple was performed using the method suggested in ref. (17). A strong light source with a system of mirrors and lenses and an orifice, served as a radiation heat source. Between the light source and the thermocouple a rotating perforated disc was placed. The rotational speed of the disc was recorded firstly with a photo-cell and later with an inductive type transducer. The ratio of the thermocouple signal amplitudes at a certain disc speed to the steady signal gave directly the amplitude-frequency characteristics of the thermocouple, as shown in Fig. 2 together with typical signal records at 30.7 and 94 Hz. Experimental results, presented in the form of the logarithmic amplitude response, followed qualitatively the form of the first order characteristics so confirming the assumption implicit in equation 1. The intersection of the asymptotes corresponds to an amplitude fall of about 3 db, and this value gives the angular frequency of $\omega_o = 250$ rad/s (f=40 Hz) which corresponds to a time constant value of $\tau = 4$ ms. This result is in close agreement with the earlier calculated value of $\tau$ for a gas speed of about 6 m/s. The above analysis and the results obtained indicate that the thermocouples used can follow only about the first 3 to 6 harmonics of the temperature fluctuations in a typical reciprocating compressor, considering that at 1000 rev/min the basic frequency is 16.7 Hz. Higher harmonics would require the use of smaller diameter wire and would pose substantial practical problems.

7. Some indications of the behaviour of the thermocouple in unsteady conditions and the influence of various effects upon it can be obtained by analysing the thermocouple response to the gas temperature represented by a simple analytical function, when the simplified dynamic equation (2) can then be solved analytically. However, to predict the response to the specific temperature changes which occur in a compressor cylinder, it is necessary to take into account

various effects such as different cooling and heating times and rates. The gas temperature was therefore simulated to represent closely a typical temperature-time record in a compressor cylinder and equation (2) was solved numerically to yield the thermocouple temperature $T_W$. The gas temperature simulation was based upon the experimental results: expansion and compression processes were represented by polytropic curves obtained from pressure records and cylinder volume calculated from the kinematic relationships, while the suction and discharge processes were represented by linear temperature changes. With these assumptions, the actual thermodynamic process was highly simplified, but the temperature profile obtained served as a good qualitative representation of the actual process for the purpose of evaluating the thermocouple response to the periodic signal which is generated in a compressor cylinder. Results are given in Fig. 3, where the modelled gas temperature is shown together with the calculated thermocouple temperature for two values of the gas velocity, 2 and 10 m/s, and for two wires, 12.5 $\mu$ and 100 $\mu$ in diameter. The results show that the 12.5 $\mu$ diameter thermocouple follows the gas temperature closely and could be accepted for practical use, particularly at higher gas velocities. The largest discrepancy occurred at the peak temperature, where the thermocouples show a temperature 12°C lower than the actual gas temperature. These results confirm the earlier conclusion that the influence of the gas velocity is small, while the influence of the wire diameter is predominant. The thermocouple with wire diameter 100 $\mu$ showed a very poor response: the difference between the maximum and minimum temperature was only 25°C whereas the actual difference was more than 200°C. An experimental temperature-time record in the cylinder of the compressor used, showed reasonably good agreement with the modelled temperature as presented in Fig. 4.

8. For the assessment of the thermodynamic processes, valve functioning and compressor performance in general, the pressure record in the cylinder serves as a basic parameter. However, depending upon the dimensions and configuration of the fluid inlet and outlet systems and the valve characteristics, the pressure in the suction and discharge chamber may vary considerably, influencing the basic processes in the cylinder. Hence their measurement is of importance for analysis of the behaviour of valves and piping system. Of particular interest are the pressure waves in the inlet and outlet system generated by the valve movement, the dynamic effects of which may influence the compressor efficiency and may be a source of noise. High frequency and low amplitudes impose special requirements upon the measuring instruments, which differ from those for measuring the pressure in cylinders. In general, the pressure transducers suitable for measurements in reciprocating compressors should possess, in addition to the usual standard dynamic characteristics, some additional qualities such as high natural frequency, high sensitivity and small dimensions, and should preferably enable measurement to be made of both absolute and differential pressures. Basically, all pressure transducers could be classified in three groups: transducers with a diaphragm (inductive, capacitative or resistive), piezoelectric and hot wire or hot film transducers. Piezoelectric transducers have been frequently used for pressure measurements in I.C. engines, being suited for measurements of high frequency and high amplitude pressure pulsations. Their attractive feature is the high mechanical impedance resulting in a low energy consumption, as well as high natural frequency. This latter feature is of particular use in the present application where even fortieth and higher harmonics may contribute to the final form of the pressure diagram. Furthermore, because of the high inlet impedance there is no need for external feed back elements, enabling the manufacture of compact transducers of miniature dimensions. The main shortcoming of the piezoelectric transducers, is the high output impedance and the lack of response to a steady signal. The lack of response to a steady signal poses difficulties if the absolute pressure, e.g. the intermediate pressure in a two-stage compressor, is to be measured. This difficulty can be overcome by use of special adapters with fast-acting valves which permit rapid connection with the atmosphere or some other known reference pressure (13).

Diaphragm transducers have the widest application for measuring unsteady pressures, both absolute and differential, but their use in reciprocating compressors is somewhat limited by their large dimensions, which prevent mounting them directly at the place where the pressure is to be measured. External mounting with a connecting channel, with its own dynamic characteristics, may result in serious errors.

9. In the present work two types of pressure transducers have been used: the "Kistler" piezoelectric, and the "Disa" capacitive diaphragm-type transducer. Because of the small gas velocities and fairly uniform pressure distribution within the cylinder, the pressure measurements could be performed anywhere in the cylinder, but the transducer mounting is restricted by the compressor geometry. The cylinder head and the valve body usually serve best for this purpose. In spite of small compressor dimensions, a piezoelectric transducer was successfully mounted within the valve body as shown in Fig. 5, enabling direct contact of the transducer with the space concerned. However, the capacitive transducer was mounted externally, its diaphragm being connected with the cylinder via a connecting passage drilled through the valve body and the connection lead through the cylinder head.

This channel must affect the dynamic characteristics of the transducer. In order to evaluate the effect of the connecting passage, an approximate analysis was performed and the results compared with the measurements obtained by the piezoelectric transducer with no such passage. Assuming the pressure changes to be small, and regarding the system as being of the lumped parameter type (justified if the channel length is short compared with the wave length of the fluctuating pressure) the channel behaviour may be represented by a second order linear dynamic system, yielding the ratio of the measured and real pressure in the cylinder in the form of the standard relationship:

$$\frac{p}{p_c} = 1/\sqrt{\left[1 - \left(\frac{\omega}{\omega_o}\right)^2\right] + 4\left(\frac{\omega}{\omega_o}\right)^2} \qquad (3)$$

$$\omega_o = c/L\sqrt{\frac{V}{V_k} + \frac{1}{2}} \qquad (4)$$

$$\xi = \frac{4\pi\mu L}{\rho c V}\sqrt{\frac{V}{V_k} + \frac{1}{c}} \qquad (5)$$

The natural circular frequency $\omega_o$ and the damping factor $\xi$ for gases for the case when the channel volume, $V_k$, and the volume of the space below the diaphragm V (Fig. 6) are of the same order of the magnitude, are given by equations 4 and 5 respectively. Here c is the velocity of sound. From the above relationship it can be concluded that a large volume of connecting passage improves its dynamic characteristics, introducing smaller measuring error. However, the presence of the connecting passage necessarily increases the compressor clearance, which has several well known effects. In order not to increase excessively the compressor clearance, the pressure channel should be kept as small as possible, which is contrary to the earlier defined requirements for better dynamic characteristics. The pressure channel was selected to have the shortest possible length. Its maximum natural frequency was evaluated and the pressure measurements compared with those obtained by the piezoelectric transducer.

10. Pressure recording was performed with both piezoelectric and capacitive transducers in the cylinders, suction and discharge chambers and pipe systems of several types of reciprocating compressor. Fig. 8 shows the comparison of the pressure record in a first and second stage cylinder of a two-stage compressor, obtained by piezoelectric and capacitive transducers. To obtain at least an approximate level of the absolute pressure with the piezoelectric transducer, only the first record, obtained immediately after starting the transducer from rest (before the electric charge had drifted) was used. Results indicate noticable, although not excessive, differences between the two records. Some small differences could be expected due to the variation of the ambient and working conditions, since the measurements had not been taken simultaneously. However, even if the differences in the expansion and compression lines could be regarded as acceptable, the suction and discharge periods differ considerably. Both transducers record some pressure oscillations during these periods (generated by the valve movement) but the record obtained with the capacitive transducer shows in all cases considerably higher amplitudes. Since the basic frequency of the pressure oscillations is about 0.5 kHz, they should be recorded equally well by both transducers, considering that their frequency response span up to 20 kHz. Perhaps the excessive amplitudes of the pressure oscillations recorded by the capacitive transducer could be ascribed to the effect of the connecting passage. The natural frequency of the passage, calculated from equation 4 was about 800 Hz which is close to the fundamental frequency of the pulsating pressure, yielding near resonant conditions. With a value of damping factor $\xi = 0.03$, as calculated from equation 5, there may have been considerable amplitude amplification as well as phase shift. Substantial changes in the connecting passage dimensions were not possible considering the dimensions of the compressor and the transducer, and the records obtained with the piezoelectric transducer were taken as the more reliable. However, the capacitive transducer records served to determine the reference pressure, repairing this deficiency of the piezoelectric transducers.

11. In order to evaluate the compressor shaft power and gain some further insight into the dynamics of the moving components of the compressor, torque measurements were made. Several types of torque transducers were used including a self made strain gauge transducer with sliding copper rings for signal transmission. The most satisfactory results were obtained with the commercially available inductive contactless transducer "Vibrometer", which enabled the instantaneous torque to be measured with good accuracy. The recording of the instantaneous torque proved to be especially useful for the investigation of the transient dynamic behaviour of the compressor, in particular during the initial starting period, when the shaft torque reached a value several times higher than that at steady conditions. Some difficulties with respect to the signal repeatability were experienced initially, when variable speed transmission was used to vary the compressor speed at the constant speed of the driving electrical A.C. motor. It was found later that the flexible belt-type transmission gear variator "Flender" was the source of the torque oscillations, the frequency of which was different from that due to the compressor speed. Consequently the measured signal was gradually changing from one cycle to another. When the compressor was connected directly to the A.C. motor, the recorded signals were repeatable. A typical record of the torque variation as a function of

crank angle during a single cycle of a two-stage two-cylinder reciprocating compressor with cranks of the LP and HP cylinders displaced by $90^o$, is shown in Fig. 10 for the steady state operation. The integral under the curve yielded the average torque and a shaft power that differed only by 3% from the shaft power evaluated by measurement of the driving motor electrical power consumption and the available data for the motor efficiency.

## COMPRESSOR SIMULATION BY COMPUTER

12. During recent years research efforts have been directed towards the development of powerful and economical simulation methods which will predict compressor performance under working conditions. This approach could complement or replace much expensive laboratory testing. Encouraging results have been achieved, but it is evident that such simulations still have to rely upon data which can only be obtained experimentally. Simulation is achieved by mathematical modelling of the phases in the cyclical physical process: the thermodynamics of expansion and compression in the cylinders, the dynamics of valve motion and the flow through the valves, the gas flow and the associated wave motion in the suction and discharge piping systems, heat transfer with the surroundings, and processes in auxiliary equipment. It is notable, however, that the predictions of the pressure wave motion still pose the main difficulties; their simulation takes most of the computing time and is responsible for most of the uncertainties which appear in the predictions. Some of the results of the authors work along these lines will be reported soon elsewhere (21); here only a brief account will be presented. A series of computer models were developed to describe single and two-stage reciprocating compressors fitted with automatic reed valves. During the development, the results published in literature were abundantly used and some new contributions were made concerning the numerical scheme suitable for the computer calculations. The basic method used was that of Benson et al., but recently some tests employing other numerical methods by MacLaren et al, have given encouraging results.

13.Typical predictions of the pressure-crank angle diagram during one cycle with cylinders of a two stage air compressor with air inter - and after-cooling are presented in figures 11-14, together with experimental records obtained with the piezoelectric pressure transducers, described earlier. The presented results show an acceptable agreement on the whole although some discrepancies are noticable on the diagram portions that correspond to the suction and discharge periods, indicating certain deficiencies in the modelling of the valve behaviour, in particular in the case of the second stage cylinder. Fig. 13, 14, presenting the pressure pulsations in the discharge chamber of the first cylinder and in the suction chamber of the second cylinder,show qualitatively similar results, but even more pronounced discrepancies, clearly indicating some damping of the amplitudes of the predicted pressure variation. Although the applied numerical scheme for solving the flow equations based on the characteristic mesh method for the selected mesh size might have caused some smoothing of the pressure oscillations, the major source of uncertainties is inherent in the assumed valve characteristics which had been selected from data supplied either by the valve manufacturers or available in general literature. An extensive study would be required before these uncertainties, the deficiencies of the numerical methods and the assumptions in the mathematical equattion could be reduced. Nevertheless the predicted air flow was only 3.6% higher than that measured, while the predicted volumetric efficiency was 0.823, compared with the measured value of 0.80.

14.A mathematical model of the dynamics of the the compressor moving parts was also developed. The predicted torque is shown in Fig. 10 where it is compared with measured values. The comparison shows a satisfactory agreement despite the crudeness of the mathematical model, indicating that here too, the computer simulation may yield information useful for design and research purposes.

## CONCLUSIONS

15.Some experience of laboratory measurements and computer simulation of reciprocating compressors has been reported. The application of the microthermocouples for measuring the instant temperatures in cylinders and other elements of the compressors yielded satisfactory results, but also indicated the limitations of the use of wire-based thermometers for transient measurements in reciprocating compressors. For pressure measurement in small compressors the piezoelectric transducers appeared to be most suitable because their small and compact dimensions enable flush mounting. Some experience with the externally mounted capacitive transducers and the influence of the interconnecting channels upon the transducer's dynamic response has also been discussed. The computer simulation of the thermodynamic processes in the compressor and the dynamics of its moving parts showed satisfactory quality, but indicated that at present simulations have to be supported by experiments and that further improvements of both the mathematical models and the numerical methods of solution should be pursued.

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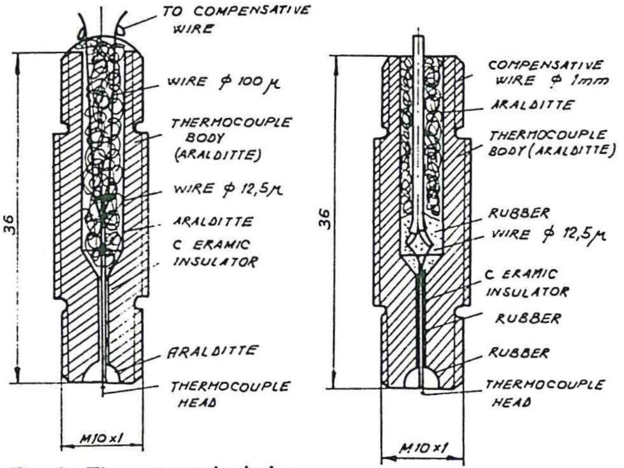


Fig. 1 Thermocouple design

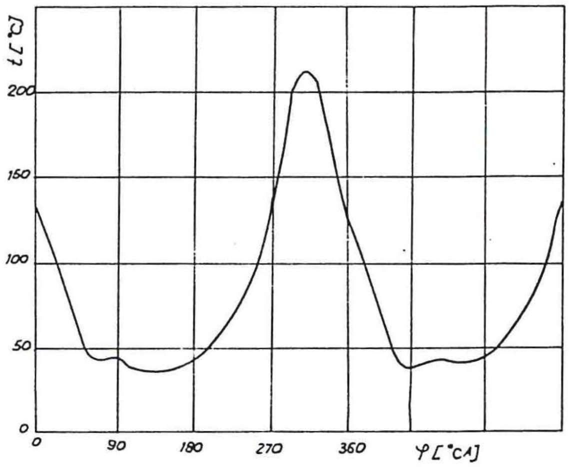


Fig. 4 Experimental temperature record in the cylinder

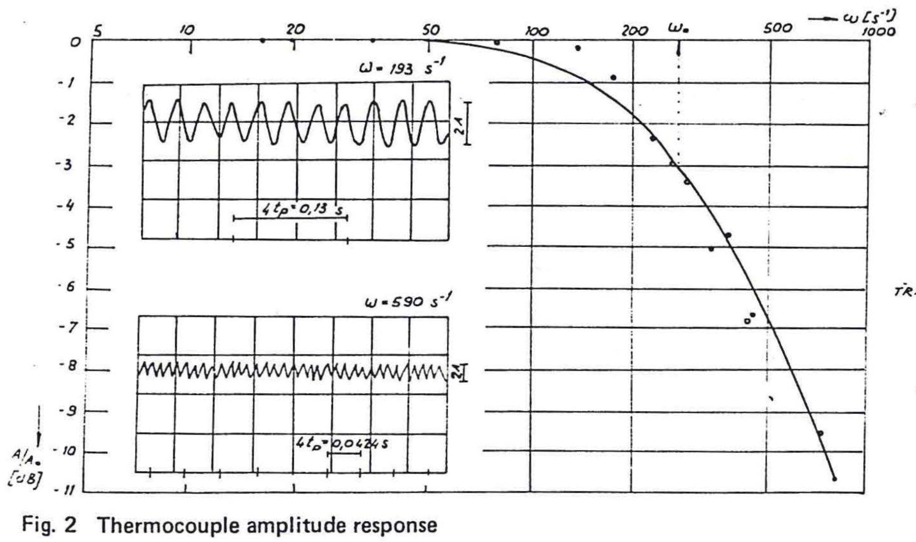


Fig. 2 Thermocouple amplitude response

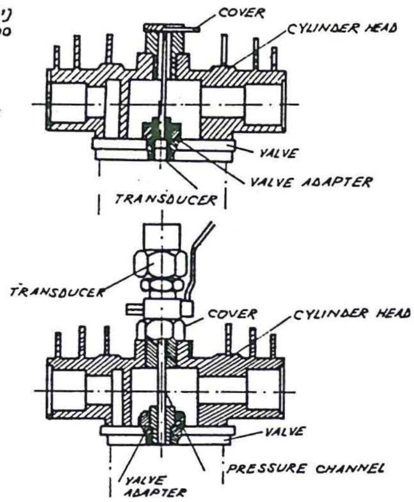


Fig. 5 Pressure transducer mounting

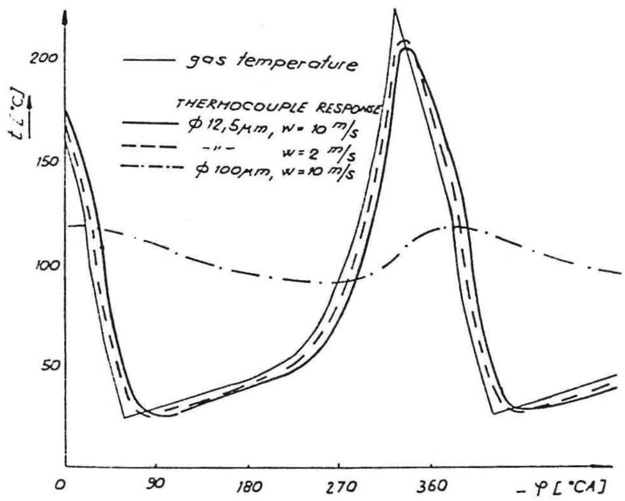


Fig. 3 Modelled gas temperature and thermocouple response

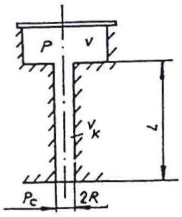


Fig. 6 Pressure channel

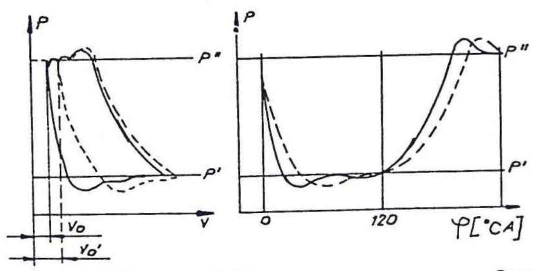


Fig. 7 Influence of clearance

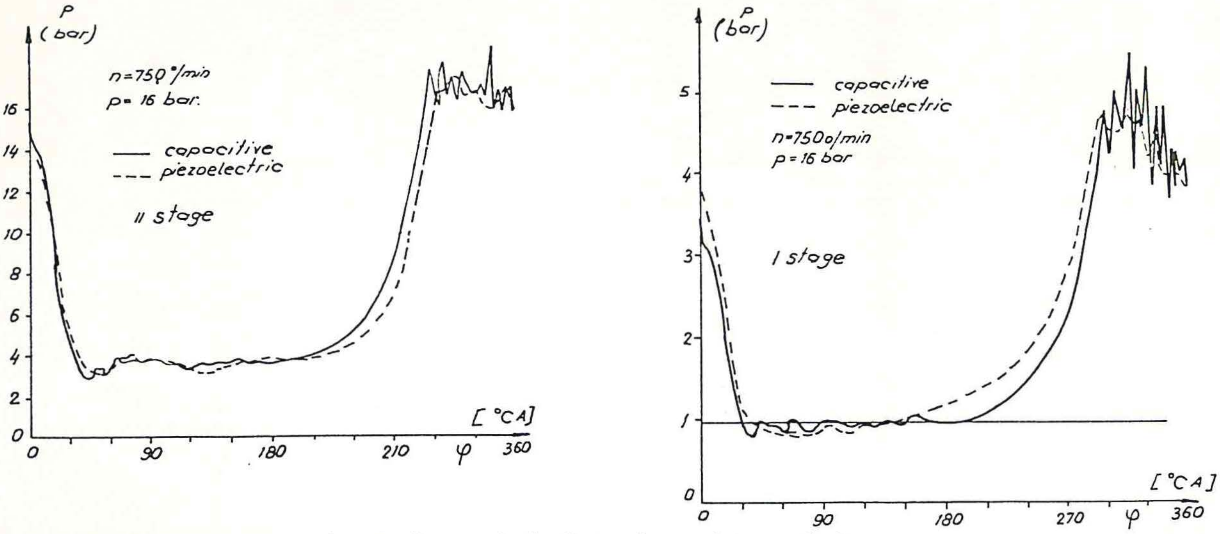


Fig. 8 Experimental pressure-crank angle diagrams in the first and second stage cylinders

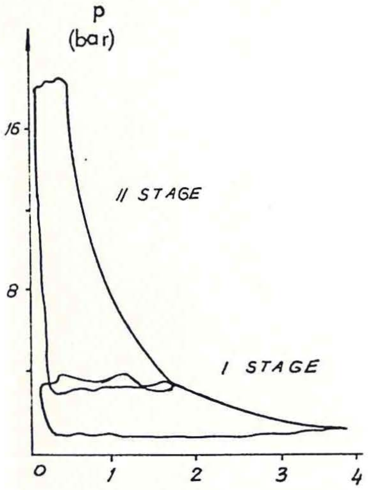


Fig. 9 Experimental pressure-volume diagrams in the first and second stage cylinders

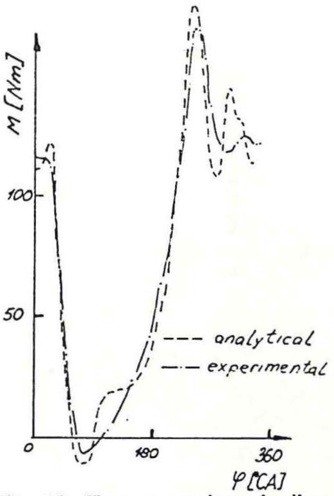


Fig. 10 Torque-crank angle diagrams

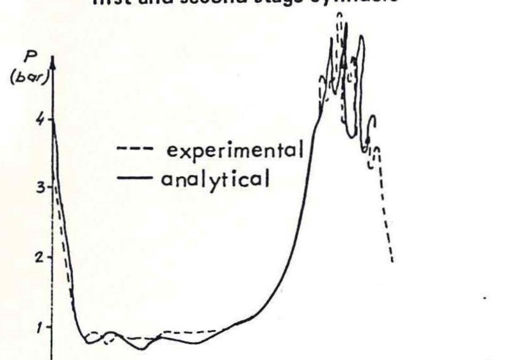


Fig. 11 Pressure-crank angle diagrams in the first stage cylinder

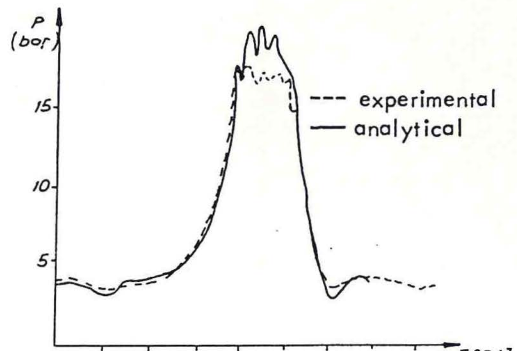


Fig. 12 Pressure-crank angle diagrams in the second stage cylinder

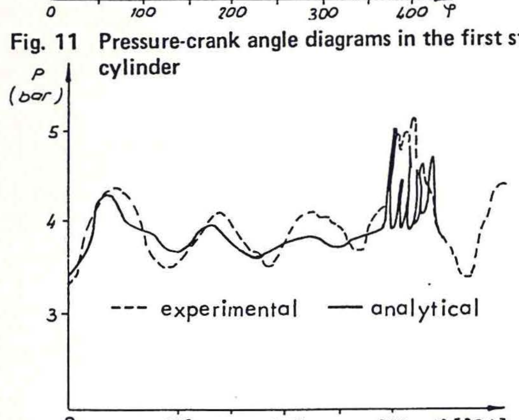


Fig. 13 Pressure-crank angle diagrams in the discharge chamber of the first stage

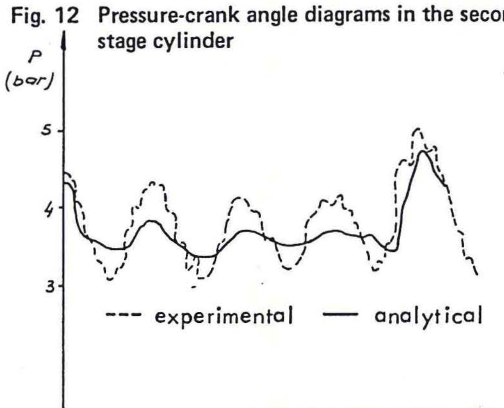


Fig. 14 Pressure-crank angle diagrams in the suction chamber of the second stage