OIL STICTION IN AUTOMATIC COMPRESSOR VALVES

A thesis presented for the Degree of

Doctor of Philosophy

by

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SUMMARY

This thesis describes experimental and theoretical investigations into oil stiction effects in automatic compressor valves.

The experimental programme involved the testing of disc valves in a compressor in which the working fluid could be varied and the oil circulation controlled. The main purpose of the experiment was to measure, in as direct a manner as possible, the time lag in valve opening due to the presence of oil between the valve disc and its seat. The design of the apparatus also allowed the pressure difference across the valve, required to open it, to be measured. These tests were carried out under various conditions of back pressure and rate of change of pressure.

The experimental programme also involved the measurement of the stiction force in a test rig in which the parameters of seat geometry, oil viscosity, and rate of application of lift on the valve disc could be controlled. The stages involved in the rupture of the oil film are described and coefficients obtained from the analysis of these test results are utilised in computer programs which simulate the stiction process in a working compressor.

The theoretical model, for the simulation of the phenomena during the initial opening phase of the valve, is based on the Navier-Stokes Equations using thin film approximations. The theoretical model is programmed for use in a digital computer. Subroutines in the programs take account of surface tension and cavitation in the oil film.

The model is not completely analytical. As stated above, it/

it utilises a number of experimentally determined coefficients, modified to suit conditions in a working compressor.

Both the effect of rate of change of pressure across the valve and flexure of the valve are shown by the mathematical model to be significant. Trends indicated by variation of these parameters and others, such as seat geometry and oil viscosity, are mirrored by experimental results.

CHAPTER 1

GENERAL INTRODUCTION

AND REVIEW OF PREVIOUS WORK ON OIL STICTION

1.1 Oil Stiction

In the past decade, the increased use of the digital computer has accelerated the amount of research being carried out on the simulation and design of automatic reciprocating compressors. As theoretical models of compressors have been gradually improved and corrected, some discrepancies have been found between theoretical predictions and experimental results. Many of the unknown coefficients have either been estimated or ignored as being not important. However, it has become apparent that the more coefficients that are known, the better can be the prediction of the compressor performance.

One particular discrepancy that has been observed for some time concerns the times at which the valves open and close. It has been deduced that this discrepancy or "time lag", in practice, is due to oil being trapped between the valve reed and its seat or stop and so causes a sticking effect, thus delaying the opening of the valve. This oil is necessary in reciprocating compressors for lubrication purposes and some of it becomes entrained with the compressed gas and tends to be deposited on the valve seat.

It has been surmised in the past that this sticking effect is due to the viscous action of any entrained oil between the valve reed and its seat, and has gradually become known through (mis)usage as oil stiction.

This project was initiated at the behest of Danfoss A/S, compressor manufacturers of Nordborg, Denmark. They were concerned about the inability of their compressor simulation programs to give adequate comparison with experimental results during the opening phases of their compressor valves. They hoped that if a proper analysis/

analysis was made of the processes involved, and a better understanding obtained, a subroutine could perhaps be implemented in their programs to take account of stiction.

1.2 Aims of the Investigation

An investigation has been made of the phenomenon which is known in reciprocating compressor design as oil stiction. The project set out to develop a mathematical model which would predict the stiction force in the equation of motion of the valve. An experimental programme was also initiated which, it was hoped, would corroborate the results predicted by the theoretical model.

The two programmes were carried out simultaneously. This meant that the experimental programme was influenced by trends observed in the theoretical predictions and also that results obtained experimentally were incorporated in the mathematical model.

Because of this, it has sometimes been necessary to cross-reference sections. This practice, however, has been kept to a minimum.

The theory which was developed is suitable for either suction or discharge valves. However, to avoid duplication and clumsiness, all valves referred to will be discharge valves unless otherwise stated.

1.3 Previous Work

Although more intensive research has been carried out into the operation of the valves than into any other part or function of a reciprocating compressor, very little investigation has been done on the effect of oil stiction on the valves.

As/

As long ago as 1955, Lorentzen (13) remarked that, in some cases, the adherence of the valve plate to its seat had a marked influence. Again, in 1963, Lorentzen (14) made mention of the fact that even at slow nominal speeds there was some effect of oil making the valve plate stick to its seat, thus giving a pressure peak at opening.

Later, in 1965, Czaplinski (6) in a review of work done in valve dynamics observed that, for optimum values, the valve should open and shut without bounce and without sticking to the seat or guard.

However, it appears that nobody attempted to measure the sticking force in even a simple manner, although most investigators were aware of its existence and had an idea, perhaps, of its effect.

MacLaren and Kerr (15) suggested that it might be possible to determine a coefficient experimentally with the valve assembly outside the compressor, or deduce it by comparison between the predicted results and accurate experimental results. They followed up this theme (16) by concluding that discrepancies between experimental and theoretical results at the opening of the valve were, after checking other possibilities, attributable to the sticking of the valve to its seat. However, they apparently found this sticking effect to be intermittent and unable, therefore, to be readily accounted for in an analysis.

Wambsganss (21) did go a step further towards the appreciation of the phenomenon. He suggested that a delay in opening of the valve could arise due to the cohesive forces in the oil present between the valve seat and the valve reed in its closed position. This sticking force would have to be overcome by the increased/

increased pressure drop across the valve before it could open. Like MacLaren and Kerr, he also thought that the effect could be investigated through a comparison of theoretical and experimental results using various values for the sticking force, since "a theoretical treatment would be quite involved".

In 1967, Najork (19) attempted to take account of the effect of oil between the valve and its seat by using a "stiction number". This stiction number was deduced experimentally from the pressure differential at the instant of opening and the parameters of the valve. Although he had, at the time, very little experimental data on which to base his number, this appears to have been the first attempt to take account of the stiction force holding the valve onto its seat.

Brown and Lough (4) showed the time lag, exhibited by valve discs sitting on flat seats when subjected to a sudden change of pressure across the valve. The response was compared with that of a quartz piezo-electric pressure transducer. Even although the time lags were very small, they thought that, if the work was extended to include rates of pressure rise across the valve of the same order as those encountered in the field, the technique could be used to investigate, in a more comprehensive manner, the behaviour of automatic valves in a dynamic situation. Using this method, the parameters could be controlled more effectively than in a working compressor. (It was this technique which interested Danfoss A/S and hence prompted the present investigation).

In 1974, Giacomelli and Giorgetti (7) undertook an experimental investigation into oil stiction in ring valves. They based their tests on the stop rather than on the valve seat since, in/

in their case, they considered the time lag in closing was of more importance than the time lag in opening. Their apparatus was a simple gravity pull arrangement. A pneumatic device held a simulated valve plate against a stop end the measurement taken was the time lag between the support being removed and the simulated valve ring leaving the stop. The valve plate took longer to fall because it had first of all to overcome the oil force between itself and the stop. Different weights could be attached to the valve plate to compare the time lag under various forces. They then concluded that the oil stiction time lag was the same immaterial of the compressor speed, since there was no rate of pressure change across the stop unlike the seat. Recommendations to overcome the time lag included: the reduction of the oil quantity to a bare minimum; the increase of the spring force to return the valve to its seat; and the drilling of small holes in the stop to reduce the surface area.

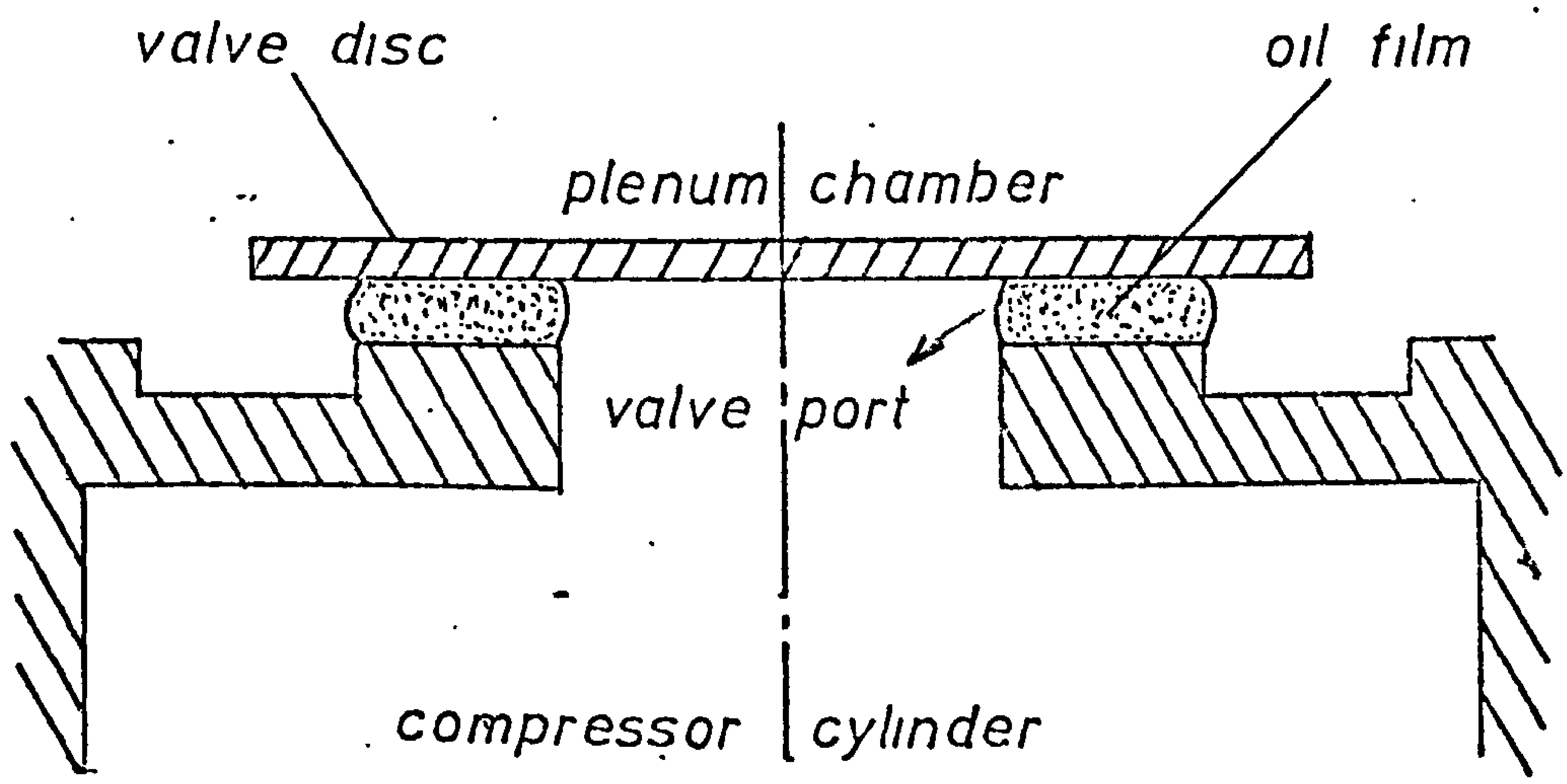
Brown et al (5) published a paper which attempted to show theoretically the processes involved in the oil stiction of valves. An experimental programme was also carried out in which figures for the time lag and oil force were obtained under normal working conditions in a compressor. These results corroborated trends predicted by the theoretical model

CHAPTER 2

DEVELOPMENT OF THEORETICAL MODEL

2.1 Initial Approach

The first mathematical model considered was a rigid, flat, circular disc separated by a thin film from a flat valve seat as shown in Fig. 2.1.1.



DIAGRAMMATIC REPRESENTATION OF VALVE

FIG. 2.1.1

The stiction force was assumed to be due to the viscous flow of the thin oil film between the valve disc and its seat.

The initial assumptions made were:

1) The oil film remained continuous.

2) No slip took place at the metal-oil interface.

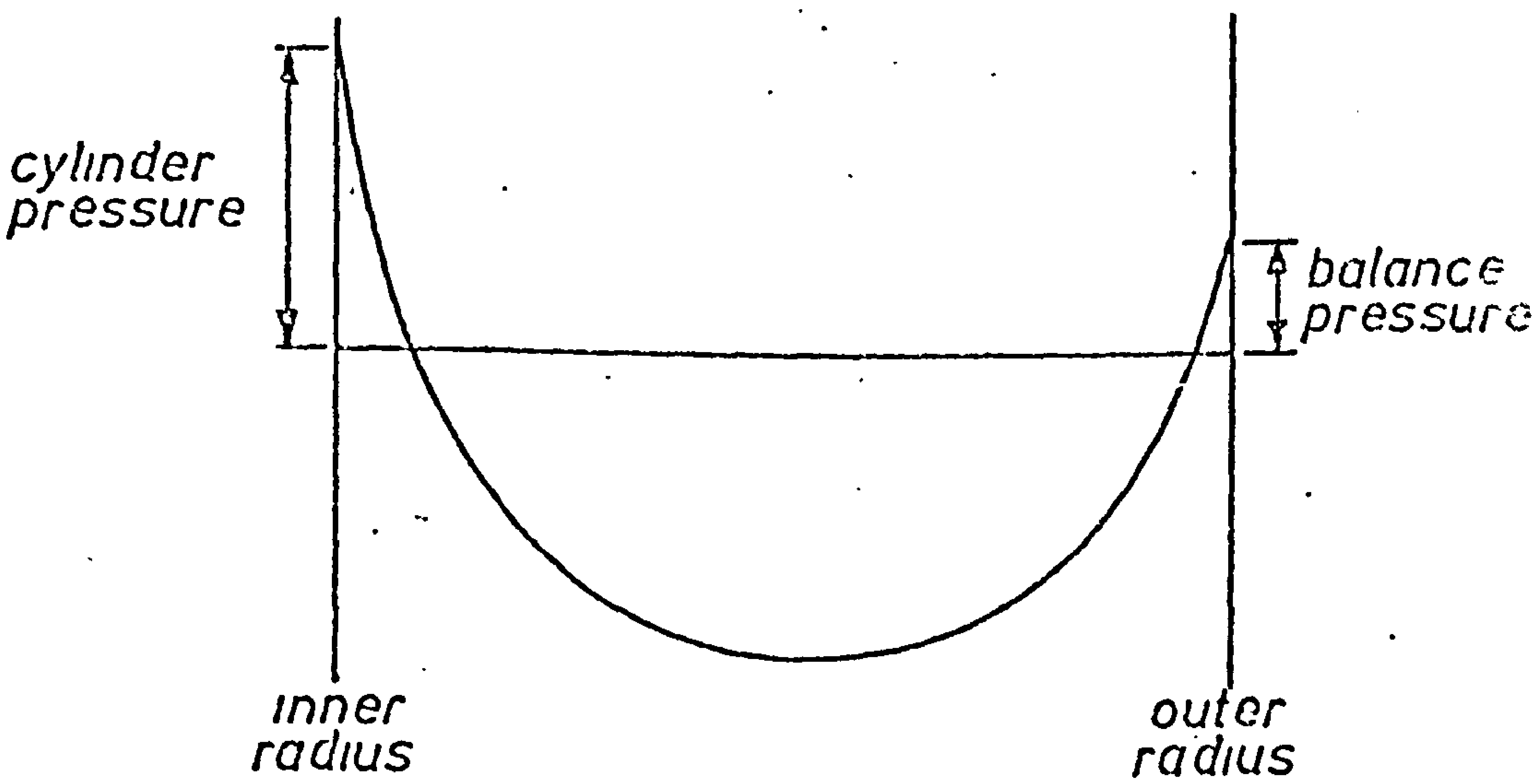
3) The viscosity remained constant.

The valve considered was a discharge valve. The pressure underneath was, therefore, the varying cylinder pressure and the pressure above was the steady plenum chamber pressure. It was only the steady plenum chamber pressure that held the disc on its seat/

seat when there was a smaller pressure in the cylinder. No retaining spring force was considered.

When the pressure in the cylinder exceeded the balance pressure, the valve was prevented from moving by the stiction force of the oil. The stiction force was assumed to be the tension in the oil, which in turn was caused by the pressure difference across the valve.

As the pressure difference across the valve increased, the tensile force in the oil increased and oil flowed radially. The pressure profile was computed and is of the form shown below in Fig. 2.1.2. This meant that the oil flowed radially inwards from both the inside and outside edges of the valve seat.



PRESSURE DISTRIBUTION IN OIL FILM ACROSS VALVE SEAT.

FIG. 2.1.2.

When the oil began to flow radially, the disc was allowed to move axially - this, in turn, caused an increased radial flow of the oil film. The film was considered to be ruptured when the inner/

inner radius of the oil film became equal to the outer radius.

2.2 Construction of the Mathematical Model

Because the oil film between the two surfaces was very thin, thin film approximations were made to the Navier-Stokes Equations which were then written in the form

$$\frac{dp}{dr} = \mu . \frac{d^2u}{dZ^2} \;;\; \frac{dp}{d\theta} = 0 \;;\; \frac{dp}{dZ} = 0 \;;$$

$$\frac{du}{dr} + \frac{u}{r} + \frac{d\omega}{dZ} = 0$$

where u = radial velocity

ω = axial velocity

These equations were used by Hays and Feiten (12) when they developed a mathematical expression to predict the growth of cavitation bubbles in thin films. Their experimental technique was to pull two circular, optically flat discs apart when there was a thin film of oil between them. They could then observe, through the optical flat, the growth of the cavitation bubbles. The above equations were developed to determine the value of the oil pressure at any radius of the oil film.

Their theoretical model was similar to the one described in the previous section, but was somewhat simpler due to the fact that it had only one free boundary. The presence of a hole in the bottom disc complicates the process considerably. It was found to be unfeasible to construct a complete mathematical model for the case where two free boundaries occurred. The above expressions were, therefore, applied as finite difference equations, the technique for which is described in Appendix B. It will be shown later that this approach allowed the model to be more flexible and allowance/

allowance could be made for varying oil film thickness caused by non-flatness of the seat and valve reed distortion.

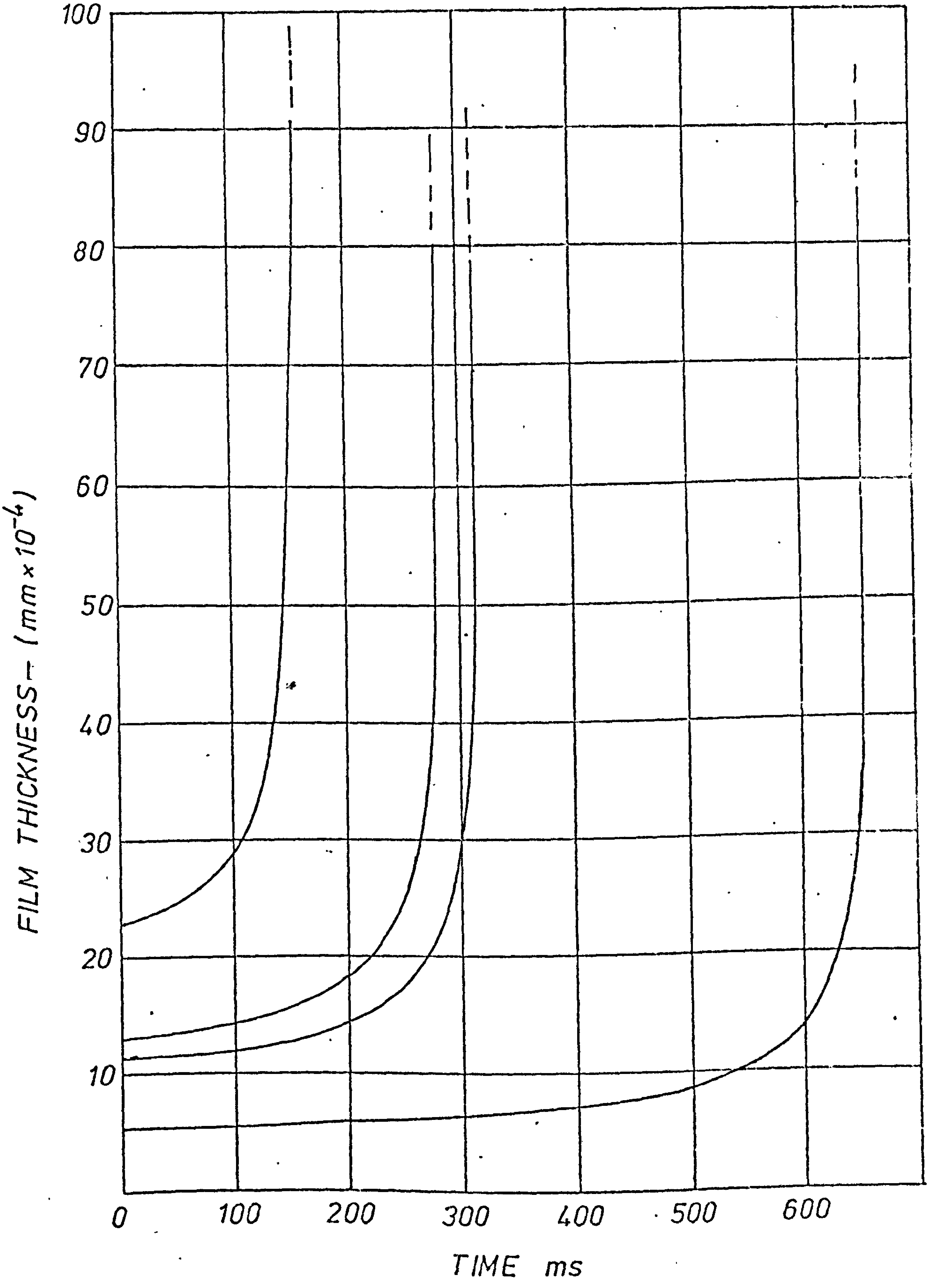
Using only the aforementioned equations and assumptions, it was possible to formulate a simple computer program which gave a crude approximation to the stiction process. This simple program unfortunately indicated time lags which were far greater than those obtained in practice and it became apparent that some other factors would have to be incorporated.

Because this crude model predicted large negative pressures in the oil film, the effect of cavitation in the oil film had to be investigated and it was partly for this reason that the force pull-off apparatus, described in Chapter 4, was designed.

2.3 Determination of Oil Film Thickness

When the program was run, it was found that one of the unknown parameters, the oil thickness, had a large bearing on the computed time lag obtained. Fig. 2.3.1 shows this dependance. By increasing the film thickness, the time for the film to rupture was considerably decreased. The times obtained were very much longer than the times measured in practice but the results did indicate the necessity of knowing the film thickness. The reasons for the excess time lag (errors of the order of 100 times) will be explained later in section 2.8.

It was not possible to measure experimentally the oil film thickness as no displacement transducer could detect the difference between a valve disc sitting on a dry seat and a disc sitting on an oily seat. An attempt was made to calculate the thickness by determining the minimum thickness that the surface tension at the oil/gas/



EFFECT OF INITIAL FILM THICKNESS ON RATE OF RUPTURE OF CLAVUS 27 OIL FILM (COMPUTED)

FIG. 2.3.1

oil/gas interface could withstand.

This method is explained in Appendix B.6.1.

It did give an idea of approximately what the oil film thickness would be, but eventually this method was superseded because of reed flexibility

It was unrealistic to assume that the valve disc was so rigid that it would not deflect into the port due to the pressure difference across the disc in its closed position. It was more reasonable to assume that the valve disc would deflect in proportion to the pressure difference across it and would actually be touching the seat on its inside edge as shown in Fig. 2.3.2.

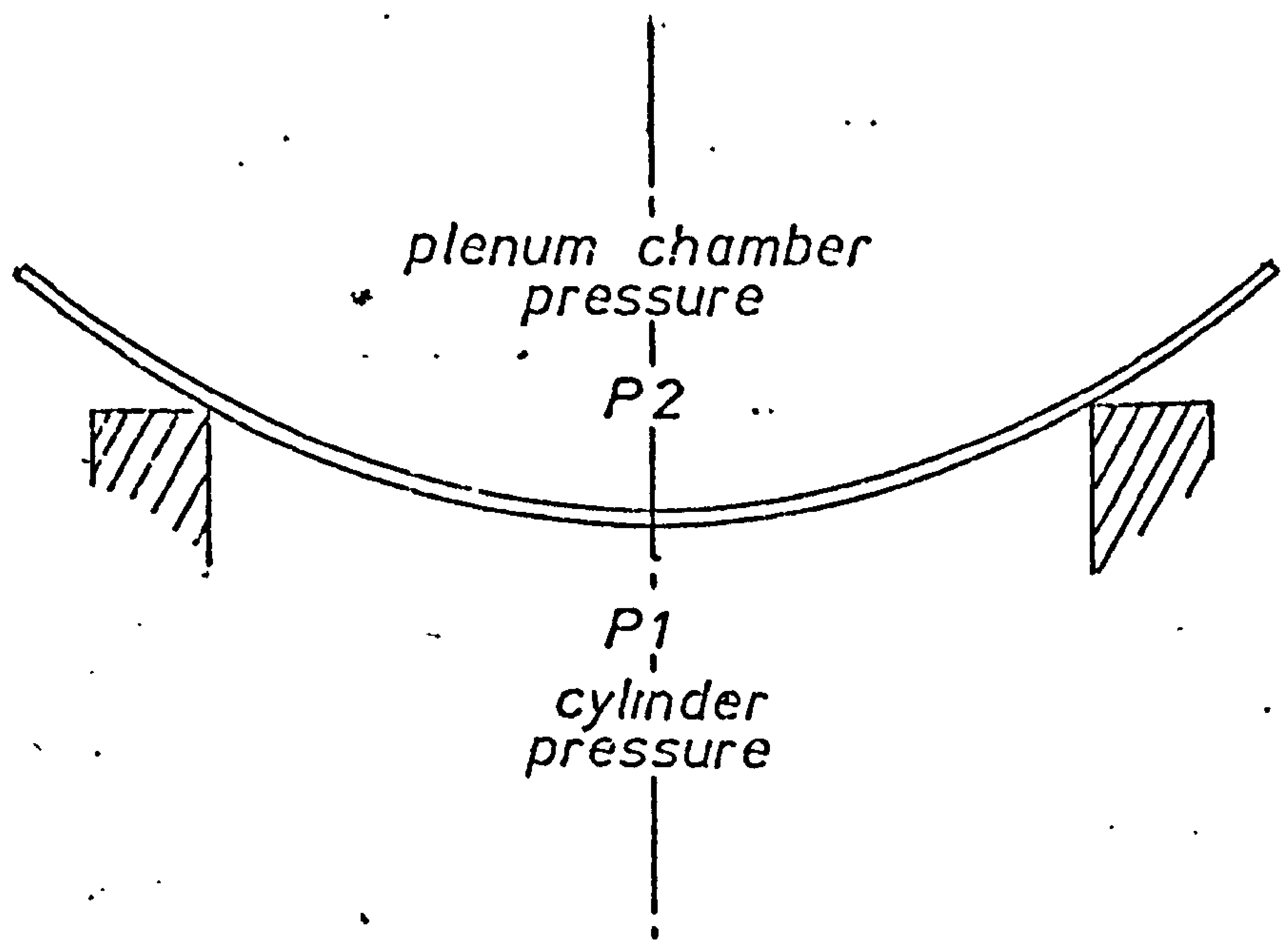


FIG. 2.3.2

(A modified program was used to check that, when closing, the valve disc did, in fact, touch the inside edge of the seat before the piston reached bottom dead centre and was not supported by a film of oil at the inside edge).

The/

The resulting wedge shape will be taken up by any available oil. As the pressure difference across the valve reduces, the valve will tend to lift and straighten out simultaneously, forcing oil out at the outer edge. Because of the pressure profile across the valve seat, the valve will tend to lift before pressure balance is reached. This is explained in section 2.4. At the balance pressure, the original wedge shape of oil will be rectangular in section and the oil thickness will be known.

This approach obviated the need to calculate an initial film thickness. The determination of the valve reed slope due to the pressure difference across it is fairly complex and is explained in full in Appendix B.6.2.

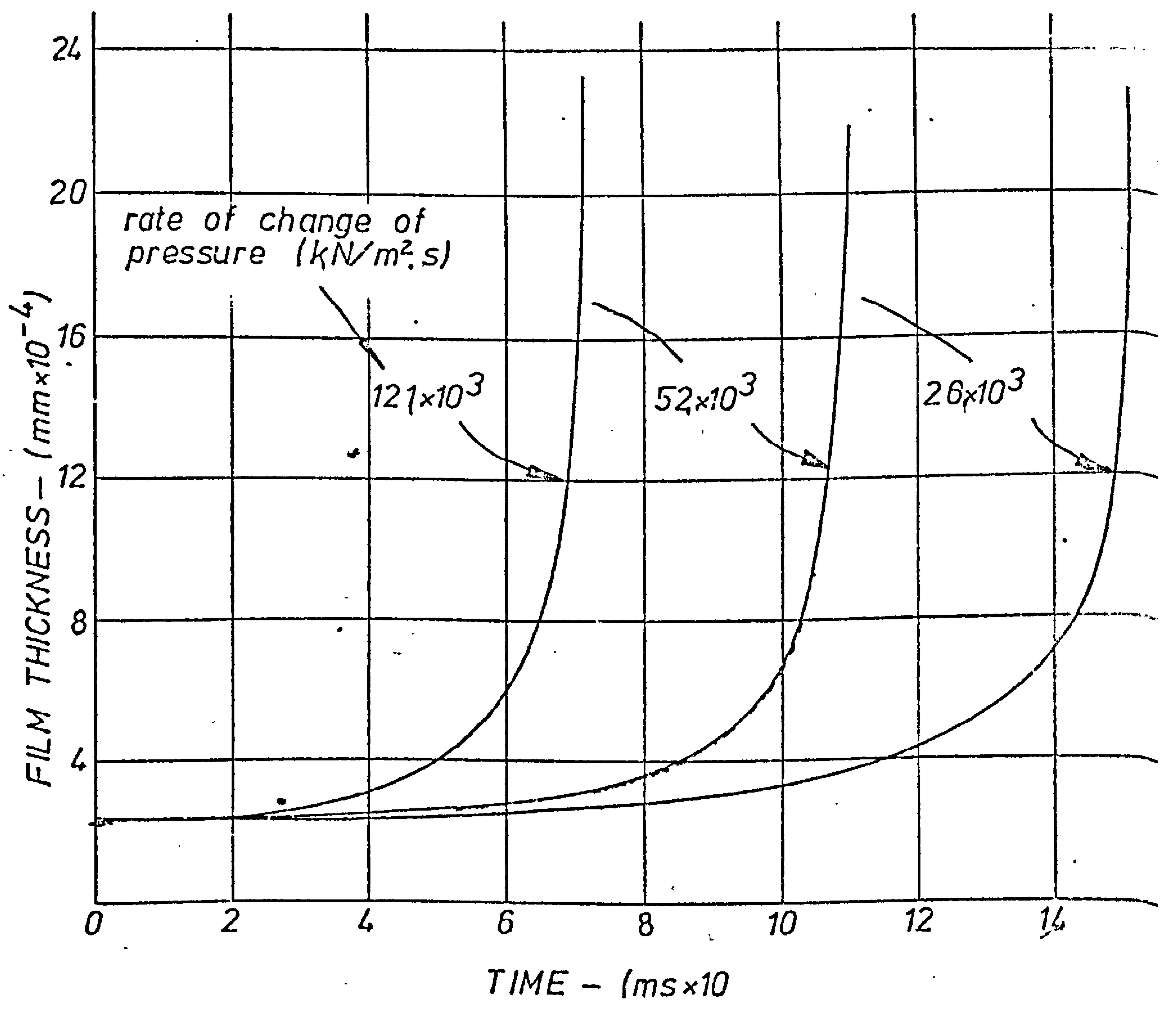
2.4 Effect of Rate of Change of Pressure

A uniform rate of change of pressure was first assumed for the opening sequence of the valve. Because the time lag was only a few milliseconds, the difference between the rate of change of pressure at the balance and the rate of change of pressure when the valve opened was thought to be small. This meant that the pressure below the valve was simply increased by a fixed amount, each computing step depending on the time interval chosen. Fig. 2.4.1 shows the dependence of the time lag on the rate of change of pressure.

This arrangement worked fairly well when the valve disc was considered to be rigid, since there was no valve motion until after pressure balance.

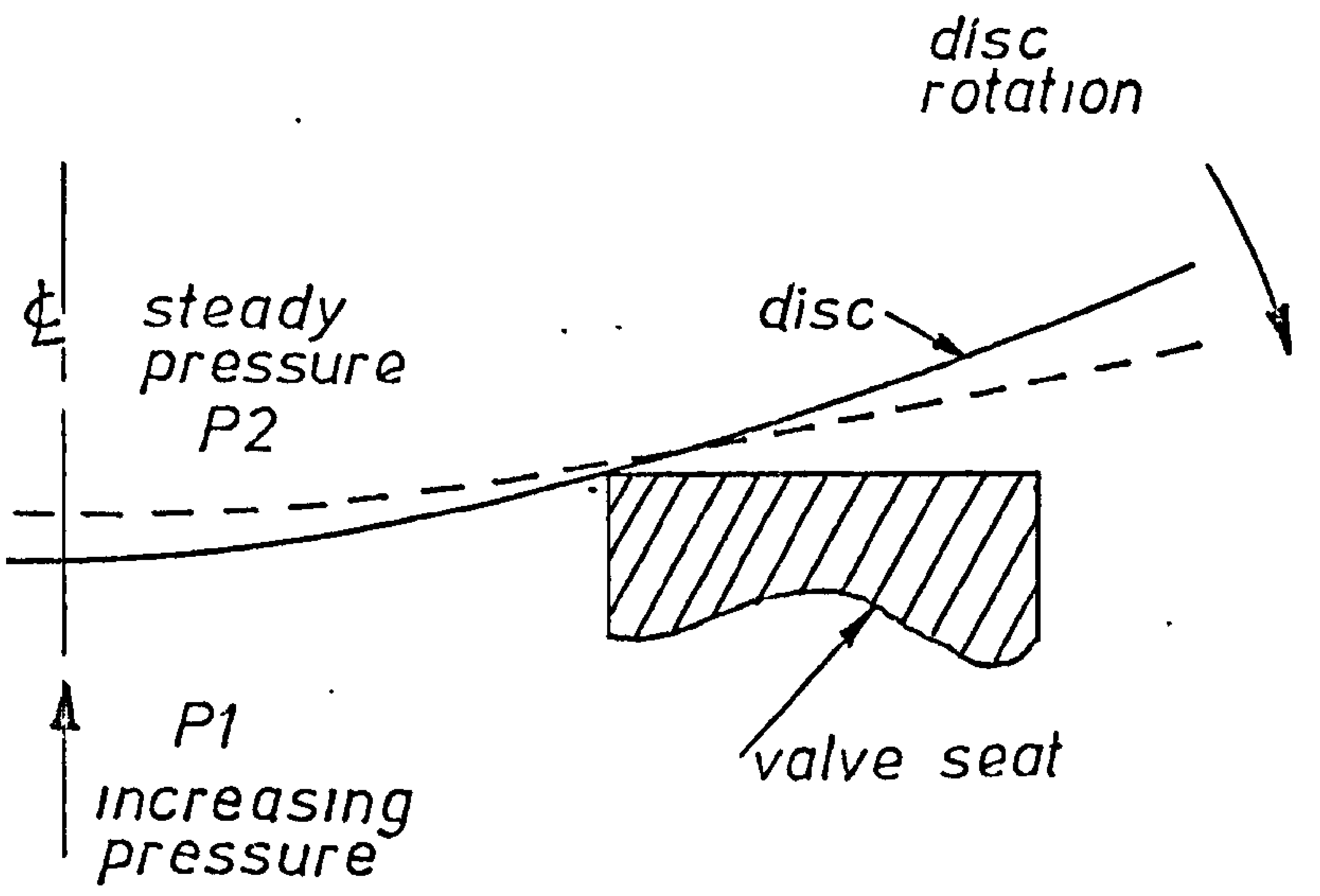
When the valve flexibility was introduced, the method of assuming uniform pressure change became inadequate. The reason for this was that the rate of valve rotation was dependent on the rate of change of pressure.

Consider/

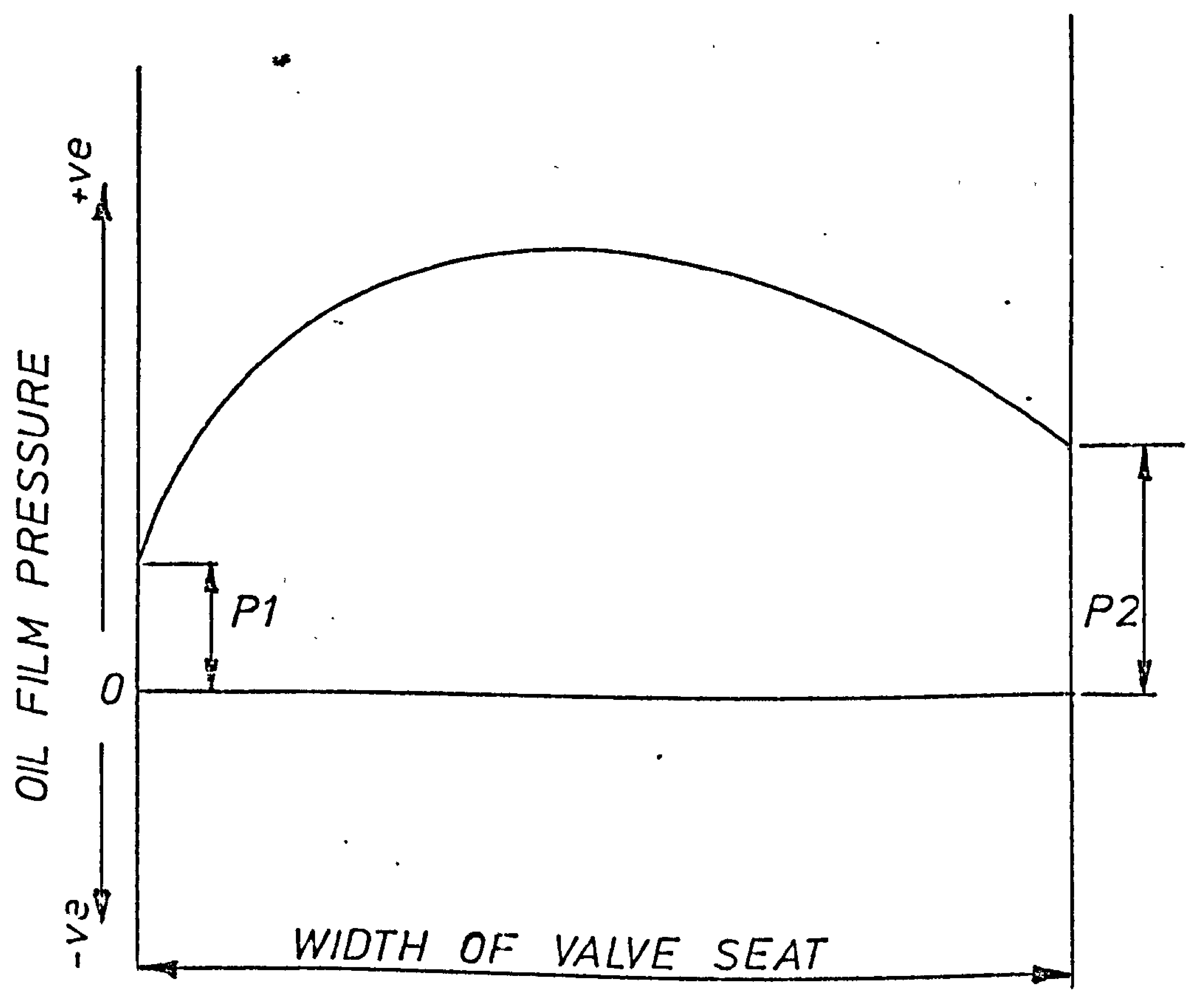


EFFECT ON TIME LAG OF VARIATION OF UNIFORM RATE OF CHANGE OF PRESSURE (COMPUTED)

FIG. 2.4.1



(a) CHANGE OF VALVE DISC SLOPE WITH PRESSURE DIFFERENCE



(b) OIL PRESSURE DISTRIBUTION ACROSS SEAT WIDTH

FIG. 2.4.2

Consider Fig. 2.4.2(a).

Because the valve slope is dependent upon the pressure difference across the valve, the rate of valve rotation is dependent upon the rate of change of pressure. The faster the valve is rotating, the greater will be the velocity of the oil forced out at the outer edge. A high rate of oil flow at the outer edge will require a pressure profile across the valve seat as shown in Fig. 2.4.2(b). This positive pressure profile will cause the valve to lift before the balance pressure. Therefore, if the rate of change of pressure is low, the radial oil velocity will be low and hence, the positive pressure profile will be smaller. The valve will thus not begin to lift until later.

Although the rate of pressure change can be assumed to be linear over the opening sequence of the valve after pressure equalisation, it cannot be assumed to be linear over the complete compression curve. A linear approximation is therefore unsuitable and a true value of the compression curve must be used. The method employed for determining this is explained in Appendix B.7.

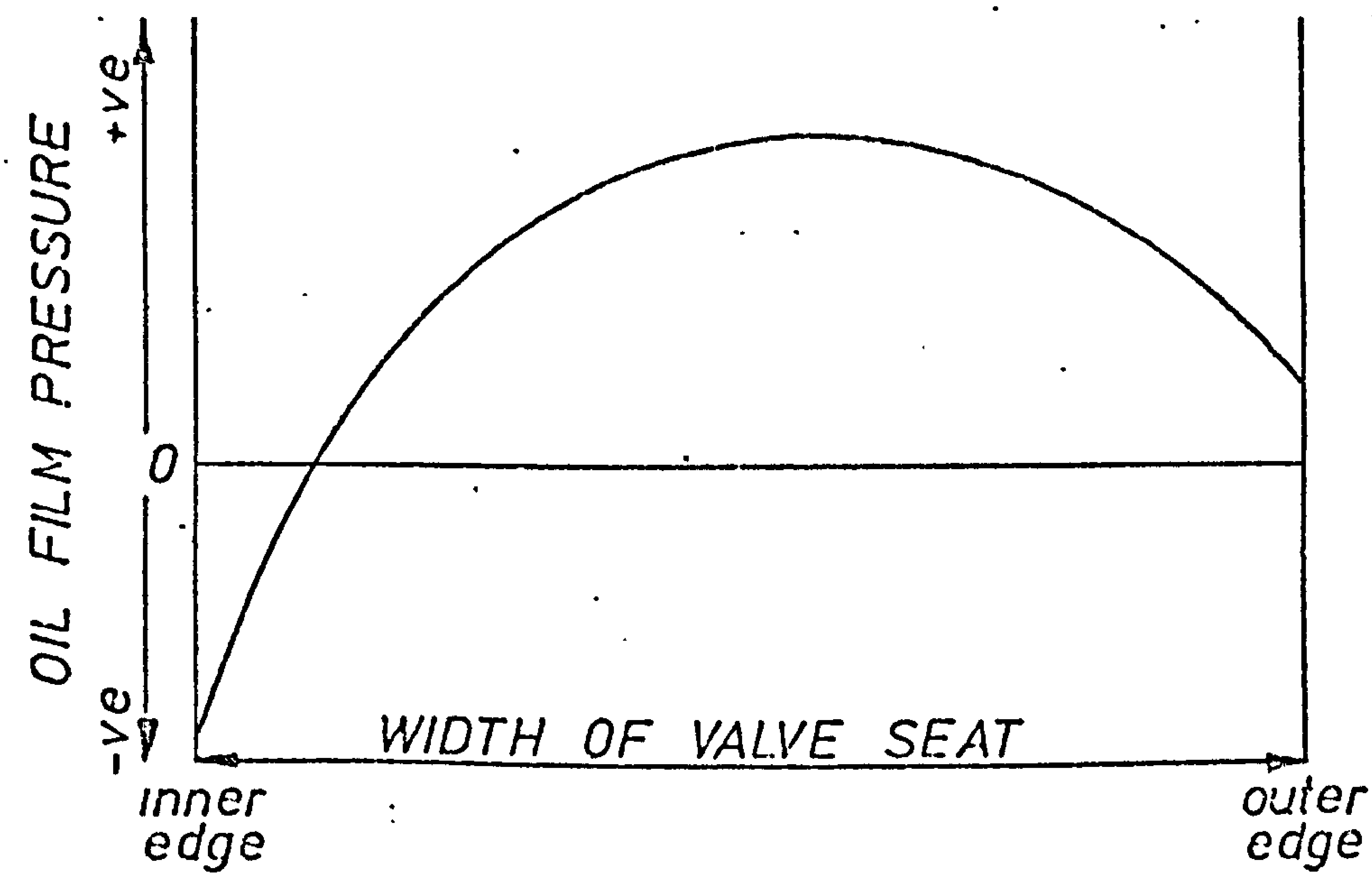
This method had to be used because of the dependence of the time lag on the pressure at which the valve first began to move.

2.5 "Gull-Wing" Effect

When the piston is at bottom dead centre, the pressure difference across the valve is a maximum and the valve disc is of the previously described deflected shape. As the pressure difference decreases, the valve disc tends to lift and at the same time straighten out. This gives an effect similar to a flapping motion and has been referred to during the course of this investigation as the/

the "gull-wing" effect.

This motion was incorporated into the model and a pressure profile of the form shown in Fig. 2.5.1 was produced immediately after the disc began to lift.



OIL PRESSURE DISTRIBUTION ACROSS SEAT WIDTH

FIG. 2.5.1

This pressure profile was very interesting. One observation was the fact that the disc began to lift at the inner edge of the seat well before the balance pressure. Another was that, because of the pressure profile, oil was being forced out at the outside edge. Most significant, however, was the very large pressure difference across the oil film close to the inner edge. This meant that if the valve opening at the inside edge was very small - as it would be when the valve just began to move - a "wire drawing" effect would be produced as gas was sucked past this edge.

This effect will be discussed more fully in "Discussion".

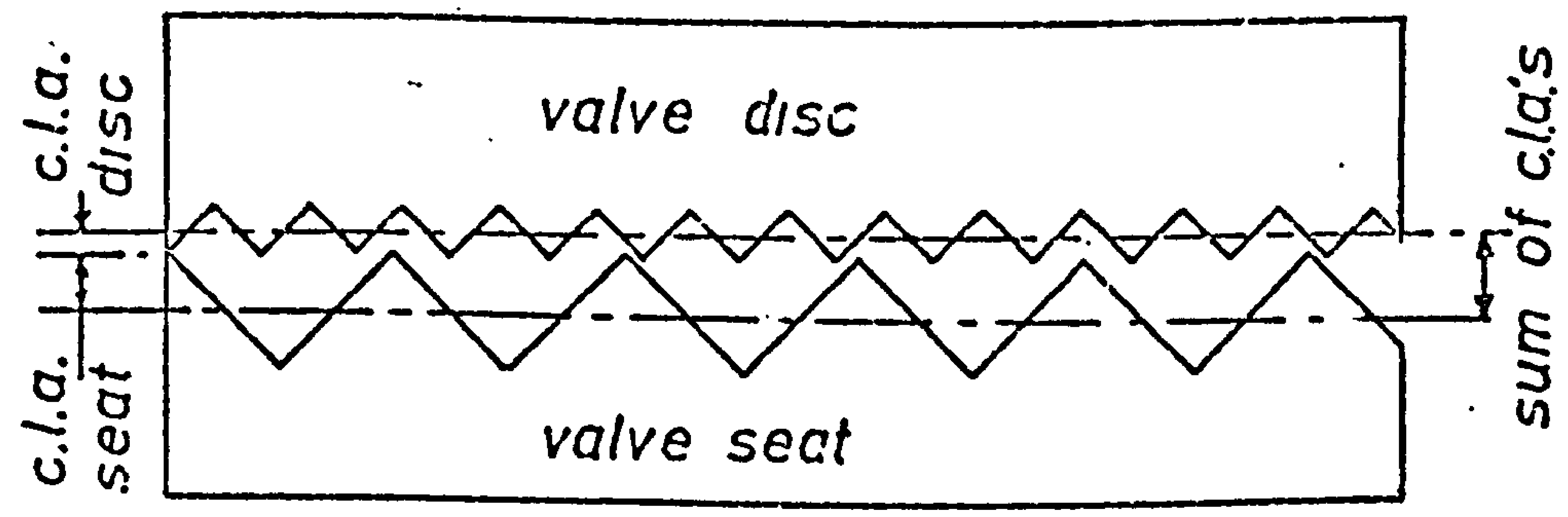
2.6/

2.6 Surface Finish of Valve and Seat

The oil film between the valve and seat was very thin. It was therefore greatly influenced by the surface finishes of the valve disc and its seat. Because the time lag was found to be affected by the oil thickness, it meant that a bad surface finish would give an effectively greater thickness which would result in a reduced time lag.

A term for surface finish was therefore introduced into the model. The CLA values of the valve disc and seat were measured and the mean value taken. Instead of considering the valve disc as touching the inside edge of the seat at the point of maximum deflection, a small gap equal to this mean value of surface finish was assumed to separate them.

This idea can perhaps be better appreciated if the surface finishes are considered to be of a saw-tooth form as shown in Fig. 2.6.1.



DIAGRAMMATIC INTERPRETATION OF EFFECTIVE FILM THICKNESS

FIG. 2.6.1

Admittedly/

Admittedly this method is not ideal as it is not possible to know just how the peaks will interlink. The gap at the edge could conceivably be the sum of the two peak-to-trough distances or else the difference of the two. The mean, however, will give an approximation to the true value.

2.7 Effect of Surface Tension

It was felt that because of the extreme thinness of the oil film, the effect of surface tension would be appreciable. Massey (17) mentions that the forces due to surface tension become comparable with other forces when solid boundaries of a liquid surface are close together. Tests on CLAVUS 27 (see Appendix C.1) showed that the direct force, due to surface tension, holding the disc on its seat was negligible when compared with the total stiction force measured in either the compressor apparatus or direct force-pull apparatus.

The surface tension is important, however, at the inside edge where the film thickness is equal to the mean value of the surface finishes and so is very small. The surface tension initially prevents the flow of oil at the inner edge and this will be shown later to lead to greater tensions being generated in the film.

2.8 Cavitation of Oil

Theoretically, a liquid will cavitate when the pressure is reduced below its vapour pressure. The cavitation phenomenon is well known and is thought by many workers (2 ,8 ,23) to be due to the growth of microscopic bubble nuclei, which are widely supposed to/

to exist in all liquids. If liquids did not cavitate, very large tensions would be generated in them before they ruptured. In fact, under certain conditions, hydrostatic tensions of about 300 bar have been measured (1). This, however, does not occur under normal conditions.

In practice, all organic liquids dissolve air and other gases and these are released when the liquid pressure drops below ambient. Hayward et al (9,10,11) have done extensive work to find the amount of gas dissolved in these liquids. The pressure at which these gases are evolved is much higher than the vapour pressure of the liquid and the gas evolution prevents the generation of all but the most modest of tensions.

As shown in Fig. 2.1.2, large negative pressures were seen to be generated in the oil film when the simple model was run on the computer. These negative pressures were generated in order to achieve a force balance caused by the pressure differential across the valve. It became apparent that the effect of gas evolution would have to be incorporated into the model, as it was unrealistic to allow such large tensions to be developed, which result in increased time lags.

This gas evolution helped to explain why the time lags shown in Fig. 2.3.1 were so large. The program was unable to consider the effect of gas evolution and its consequent destruction of the film.

2.8.1 Evolution Constant

The rate at which dissolved gas is evolved from a liquid is a complicated process dependent on a number of factors, e.g. surface tension, viscosity, percentage of dissolved gas in solution, rate of change of pressure, as/

as well as the absolute pressure of the liquid. This process is difficult to approach analytically, and is of more concern to the physical scientist rather than to the engineer. Some method had, therefore, to be devised to account of the fact that this evolution of gas broke up the oil film and accelerated the rate at which the valve left its seat.

Young and Fannin (24) investigated the vapour evolution rates from oil-refrigerant mixtures following expansion through a nozzle. Previously, no experimental work appeared to have been published on the rate of refrigerant evolution following a rapid reduction in pressure. Although their results were difficult to incorporate into the present project, they did show the large evolution rates possible over just a few milliseconds.

The values of evolution rate for this particular project were, therefore, based on experimental observations as will be explained more fully in Chapter 4. The problem was overcome by introducing a term which accounted for the increased rate at which the valve reed left its seat due to the increased volume of the film resulting from the evolved gas.

The evolution constant takes no account of whether the gas is initially dissolved in the oil, as in the case of an oil-refrigerant mixture, or secreted in microscopic or submicroscopic interstices of the valve reed or valve seat surfaces. The criterion was that the method employed simulated fairly accurately the results obtained in practice.

2.8.2 Time Delay Constant

When the valve disc is forced from its seat with a high velocity/

velocity, large tensions can be generated in the oil film before it ruptures. These large tensions produce negative pressures in the film much greater than the pressure at which gas is evolved. This, however, can be explained by the fact that the evolution of gas does not occur instantaneously but is time dependent, due to the fact that the submicroscopic bubbles referred to in the previous section take a certain time to reach a critical dimension after the pressure in the liquid is reduced. A "time delay constant" was, therefore, introduced into the computer program to delay the initiation of gas evolution.

Like the evolution constant, the time delay constant was deduced from experimental observations which are explained in detail in Chapter 4.

CHAPTER 3

COMPRESSOR TEST APPARATUS

3.1 Introduction

In order to conduct experiments in a manner as near to normal working conditions as possible, a comprehensive series of tests was carried out on a modified compressor. The principle of operation depends on the stiction-free performance of a gravity controlled valve disc being taken as a standard, against which the performance of a test valve disc can be compared.

Balanced-disc transducers, based on the "Farnborough" engine indicator, have been used for some time by Lorentzen and others in refrigerating compressor research. Lough and Brown noticed inaccuracies in these transducers when they were developing them for their own research programme. Differences were observed between the outputs of these transducers and the output of a Quartz Crystal transducer when recording pressure-crank angle diagrams from a reciprocating compressor. These differences were not obvious when the machine was first started up, but became apparent after a few minutes. The errors were also more noticeable if the width of the seats in the balanced-disc transducer were particularly wide compared to the port size. When the balanced-disc transducer was dismantled, a thin film of oil was found to have gathered on the transducer seat. Brown and Lough deduced that this film of oil caused the balanced-disc to adhere to its seat.

If the seat width was made very narrow, with respect to the port size, any small errors were eliminated. For this reason, in the tests associated with the compressor apparatus, a transducer with very narrow seats was used as a standard against which the performance of a test valve disc could be compared.

3.2 Principle of Operation

The balanced disc transducer consisted of a lightweight disc which could "float" between two seats as shown diagrammatically in Fig. 3.2.1.

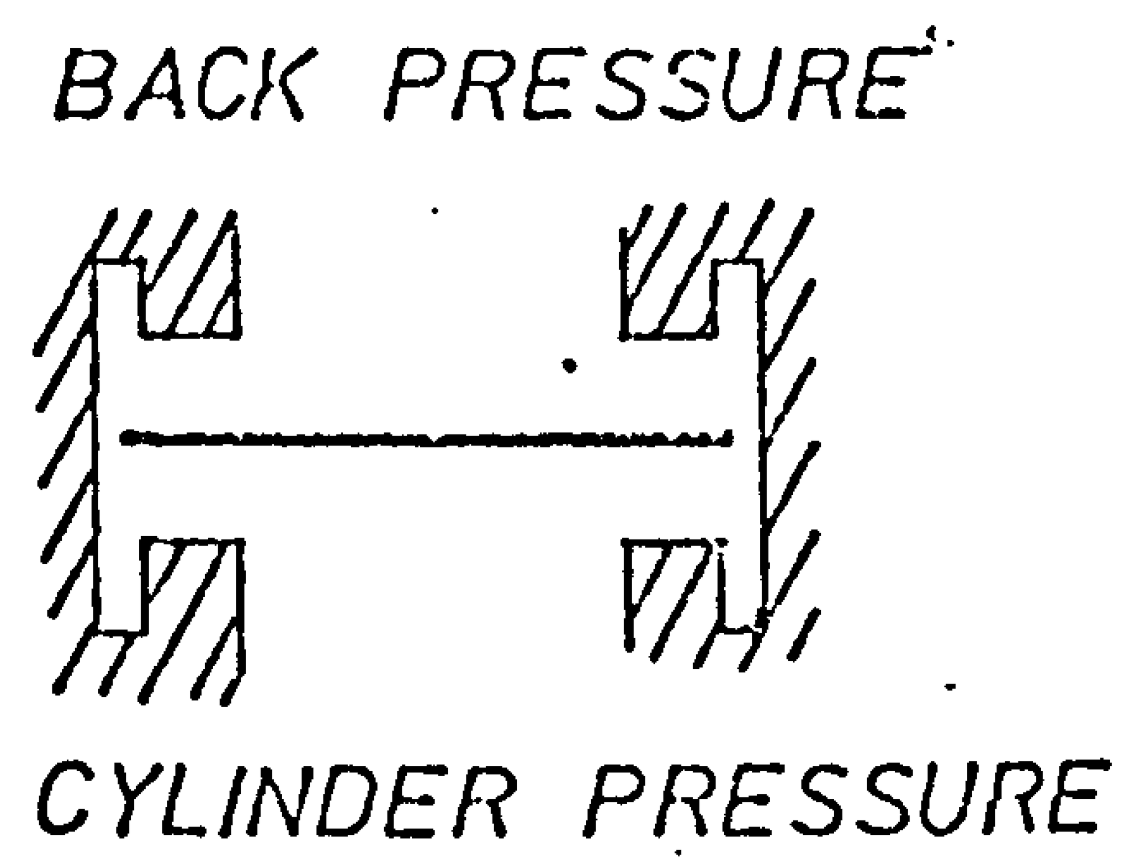


FIG. 3.2.1

The pressure on top of the disc was held steady. The disc could only move to the top seat when the pressure underneath equaled, and then exceeded, the pressure above. Similarly, the disc could only move back to the bottom seat when the pressure below fell to the level of the constant pressure above. Because of the small mass of the disc, gravitational load was negligible, as also was the inertia at the instant of opening (see Appendix C.2 and C.3).

If this balanced-disc transducer was placed in the head of a reciprocating compressor and the underside of the disc exposed to the fluctuating cylinder pressure, the disc would move up and down once every cycle.

When/

When oil was present on the seat, the disc would not move at the balance pressure. The pressure had to increase past the balance point to overcome the cohesive action of the oil between the disc and its seat. Because a finite time had to elapse to allow the pressure below the disc to increase, the disc exhibited a finite time lag in its motion after the balance pressure had been reached. This time lag was measured by comparing the response of a test disc sitting on a seat of a known profile with the response of a disc sitting on a sharp-edged seat, which Lough and Brown had previously shown to exhibit negligible time lag.

3.3 Compressor Modification

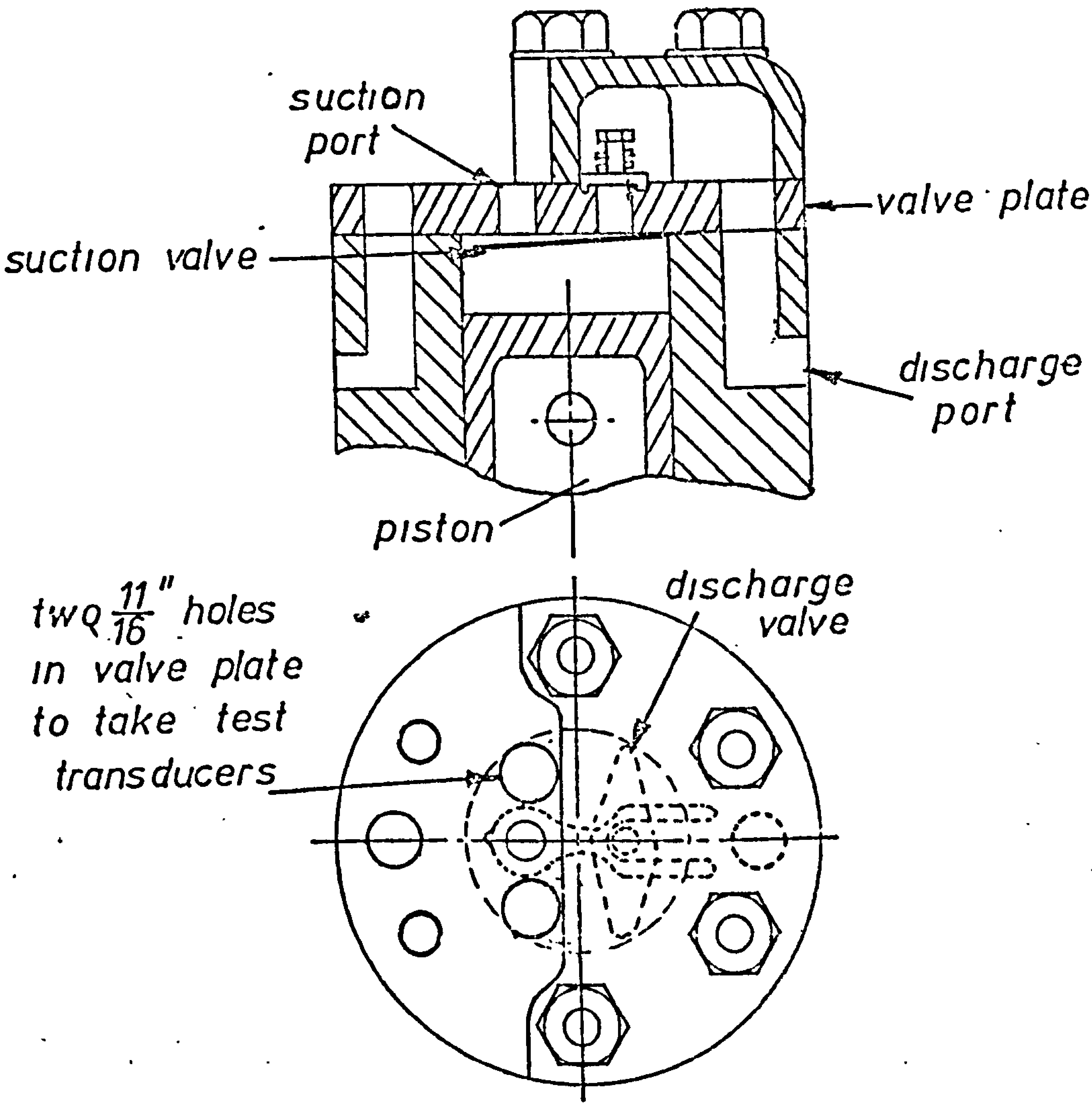
The compressor head was modified by boring two holes in the valve plate as shown in Fig. 3.3.1. In these holes were placed the two balanced-disc transducers. The complete assembly of one of the transducers in the modified head is shown in Fig. 3.3.2. Apart from the modification to the valve plate, a completely new head had to be made to accommodate the two protruding displacement probes.

3.3.1 Balanced Disc Transducer (Seats and Disc)

A detailed view of one set of seats is shown in Fig. 3.3.3. The balanced disc transducer consisted basically of a thin disc lying between two seats. The two seats were held in position by the body of the transducer.

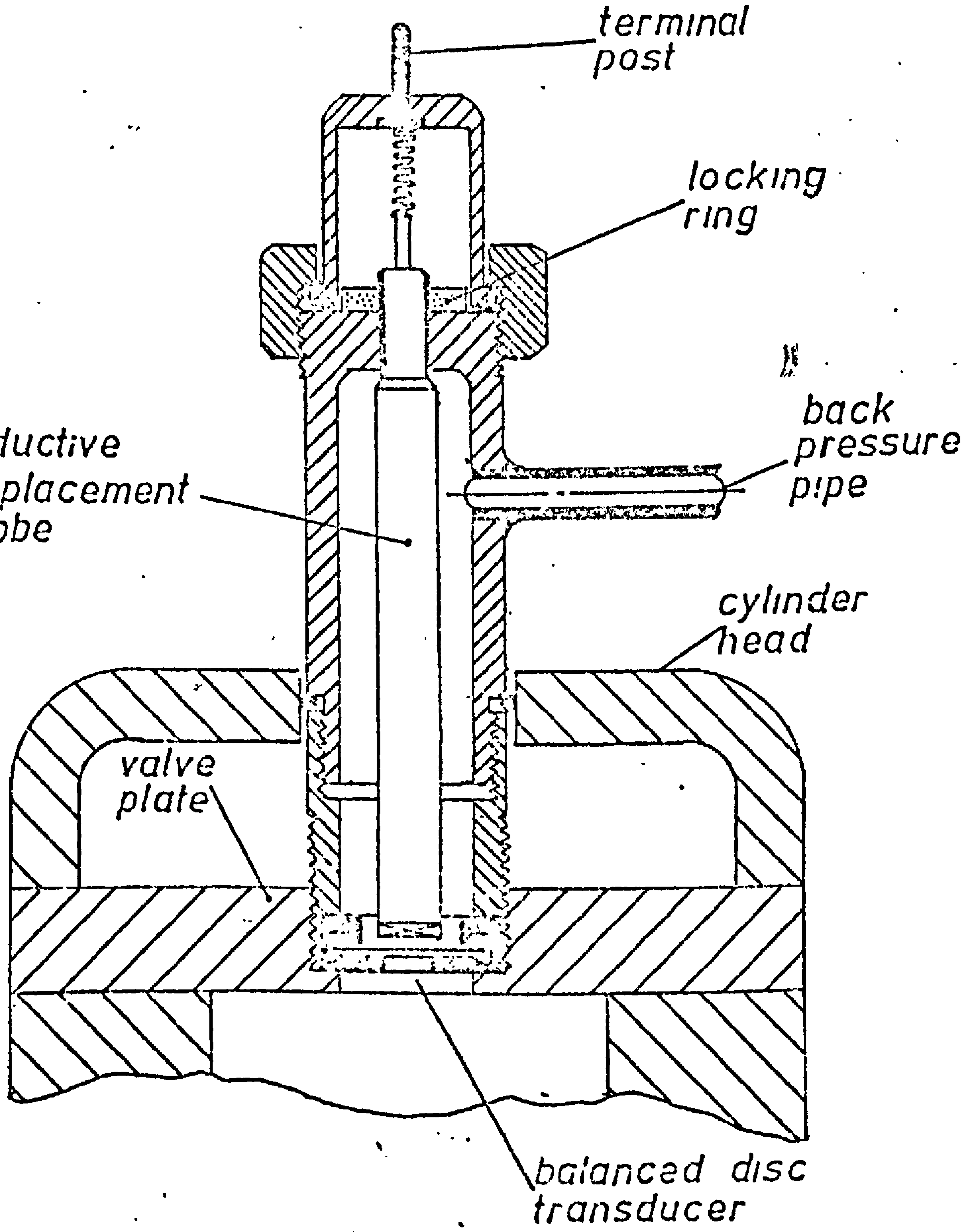
The disc moved up and down between the lower "stiction" seat and the upper retaining seat, depending on the pressure in the compressor cylinder.

The two seats were made of brass. The lower one could be made/



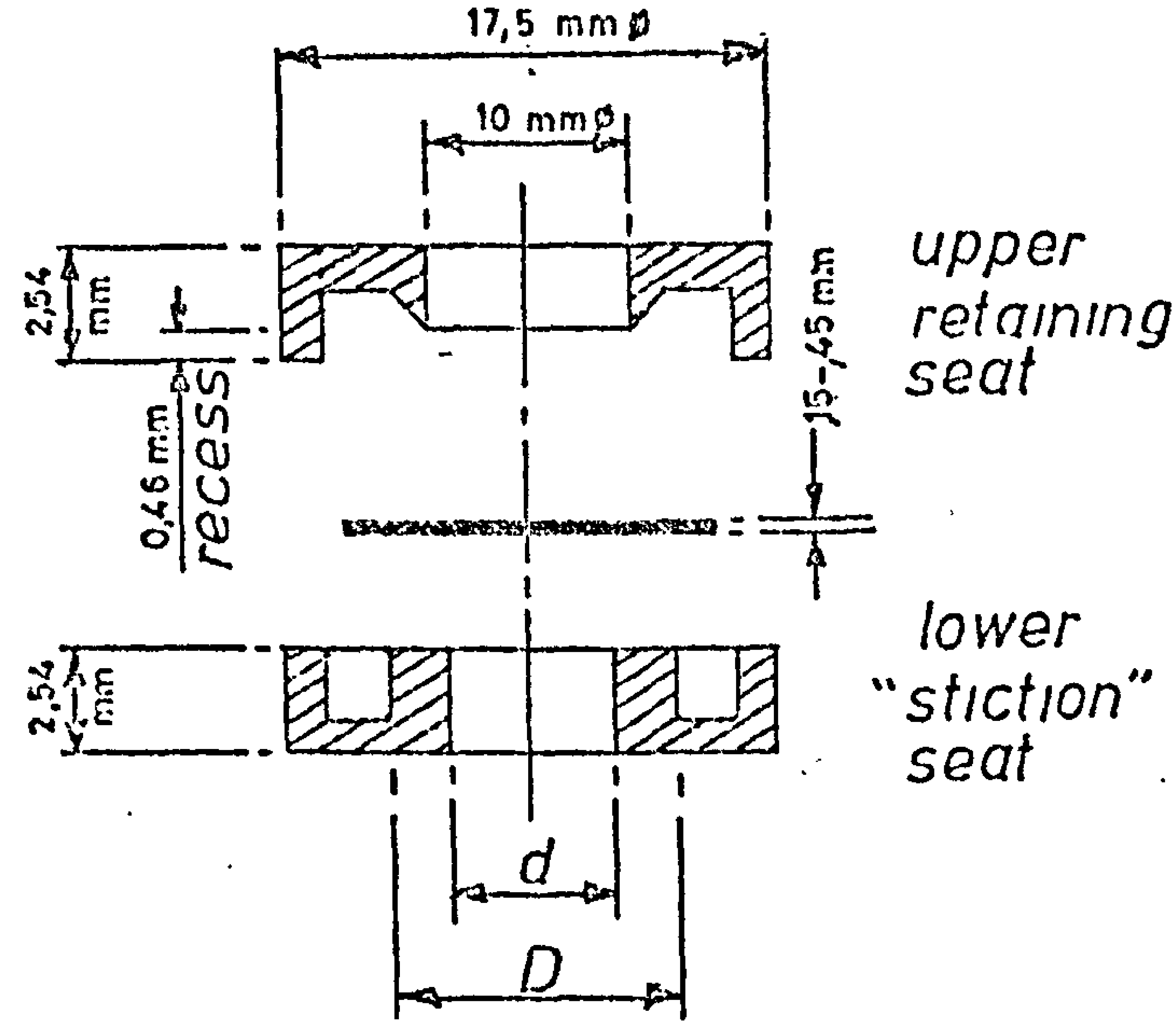
CYLINDER AND CYLINDER HEAD OF AIR COMPRESSOR SHOWING MODIFICATION TO VALVE PLATE TO ACCOMMODATE PRESSURE TRANSDUCERS

FIG 3.3.1



EXPERIMENTAL TRANSDUCER

FIG. 3.3.2



seat number	outside diameter D (mm)	inside diameter d (mm)
1	13,34	2,54
2	12,07	3,81
3	10,80	5,08
4	9,53	6,35
5	13,34	10,67
6	12,07	9,02
7	10,80	7,21
8	9,53	5,16

FIG. 3.3.3

made of any desired port size and width of seat. As can be seen, the surface of the seat is made flush with the outside wall. This allowed the surface to be lapped to a good surface finish.

The valve seat could be made to any required outside and inside diameters and a complete list of all seat profiles is also given in Fig. 3.3.3. The purpose of making the test seats to the dimensions shown was to separate, if possible, the effects of surface tension and surface area.

Seats numbered 1 - 4 had the sum of the circumferences of the inside and outside diameters the same. The reasoning behind this was to eliminate any possible direct surface tension force at the liquid/gas interface due to the extreme thinness of the film. This force was found later to be negligible compared with the other forces involved (see Appendix C.1). Seats numbered 5 - 8 had the same seat surface area. This was a first approach to investigate the cavitation/evolution phenomenon. It was thought that if gas was evolved at a particular pressure, then this pressure would spread and be the same over the whole area of seat. The force holding the disc on its seat (pressure x area) would, therefore, be the same for each seat. Although the initial philosophy was found to be slightly naive, or perhaps misplaced, the end result of having eight seats of known profiles enabled a more than adequate amount of experimental data to be compared, eventually with theoretical results.

The upper retaining seat was not flat but was sharp-edged. In this case, however, the seat was recessed from the seat wall. This recess allowed the disc to move freely between the two seats. The port size of the retaining seat was not important but was made sufficiently large enough to allow the inductive displacement probe to/

to pass through.

The two seats could easily be inverted to allow the steady pressure to be applied to either the upstream or downstream side of the transducer. This enabled the transducer to act as either a "suction" or a "discharge" valve.

If a disc to be tested was thicker than the recess in the retaining seat would allow, spacers could be inserted between the two seats to separate them and so give more clearance for the disc. The reason for using discs of varying thicknesses was to study the effect of valve flexibility.

3.3.2 Balanced-Disc Transducer (Body)

The body of the transducer was made in two parts for ease of assembly.

The bottom part simply held the two seats tightly together.

The function of the upper part was two fold:

1. It had a tapped hole in the top to allow the displacement probe to be held at any desired position. This position was then secured by means of a locking nut. (The reason for the position of the displacement probe being held accurately will be explained in section 3.6.2).

2. It had an inlet pipe to allow a steady or "back" pressure to be applied to the top of the disc. (Because the displacement probe was rectangular in section and the upper seat port was round, there was no air restriction involved).

An 'O' ring round the outside of the body prevented leakage from the plenum chamber into the atmosphere.

To prevent any leakage to atmosphere past the screwed part of/

of the displacement probe, a sealing cap was added as shown in Fig. 3.3.2. The cap was screwed down tightly to make sure the rubber ring made a good seal, and the electrical circuit was completed by means of a spring connecting the two terminal posts.

3.3.3 Experimental Configuration

In all tests, two such transducers were used.

1. A transducer fitted with sharp-edged seats (top and bottom).
2. A transducer fitted with a flat seat of the required profile on the bottom and a sharp-edged seat on top.

Even although the mass and inertia of the discs were negligible, two similar discs, i.e. of the same thickness, were always compared with each other.

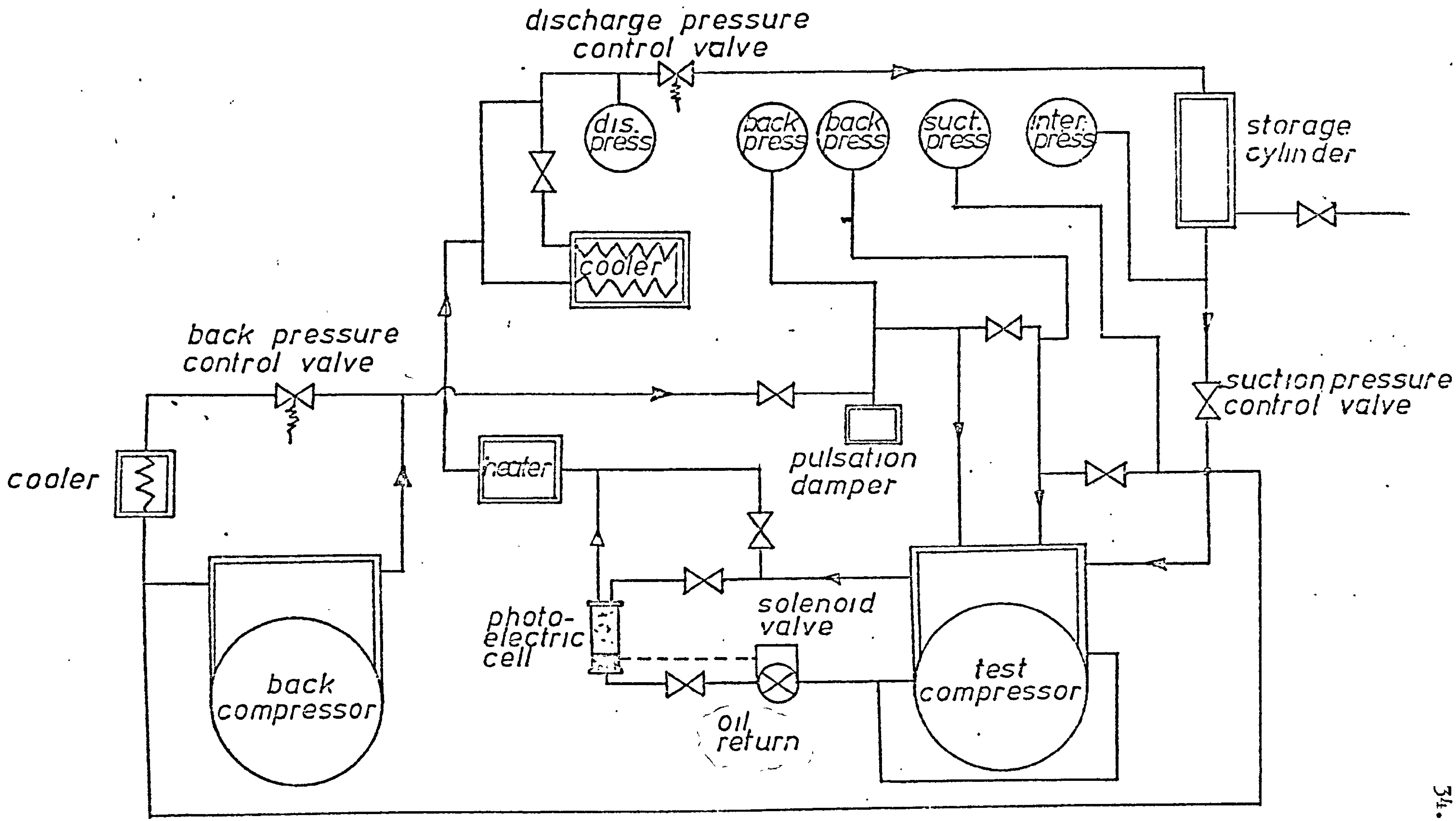
3.4 Compressor Test Circuit

The compressor test circuit is shown diagrammatically in Fig. 3.4.1. The development of the complete circuit was gradual and was built upon as the investigation progressed.

3.4.1 Compressor Configurations

Two compressors were used: one, the test compressor, with the head modification as described in section 3.3; the other, to provide the constant "back pressure" to the valve discs. Both compressors were single cylinder, single acting and driven through vee-belts by variable speed electric motors.

The two compressor circuits were interconnected. This closed circuit arrangement was required in order that Refrigerant 12 could be used as the working fluid, as well as air. Any other arrangement, e.g. external gas supply, would have resulted in a considerable/



CIRCUIT DIAGRAM OF COMPRESSOR TEST APPARATUS

FIG. 3.4.1

considerable loss of refrigerant. It was also found, during the development of the apparatus, that if an external air supply was used for the back pressure, a large build-up of gas resulted in the circuit. This build-up was caused by small leakages past the valve discs.

3.4.2 Pressure Control

Constant pressure expansion valves were used to control the discharge pressure of the "test" compressor and to control the gas flow round the "back" compressor circuit.

Due to the fact that both compressors were drawing gas from the same source, i.e. large storage cylinder in the case of R12, and the atmosphere in the case of air, any changes in the discharge pressure of the back compressor affected the suction condition. This change in suction condition thus affected the discharge pressure of the "test" compressor. This problem was obviated by installing an expansion valve just before the suction side of the test compressor.

In order to prevent any pressure pulsations on top of the discs, a pulsation damper was fitted immediately before them. Also, by fitting a shut-off valve between the damper and the "back" compressor, the back pressure could be further controlled.

It was found from experience that the liberal use of shut-off valves throughout the circuit was advantageous in case ancillary equipment had to be added or changed, particularly to prevent any unnecessary loss of refrigerant.

3.4.3 Pressure Measurement

The pressures at all required points were measured by the use/

use of standard pressure gauges. These gauges had been previously tested over their working range to ensure their accuracy and were checked thereafter at regular intervals.

It can be seen from the diagrammatic sketch of the compressor circuit that two back pressure gauges were fitted. Both of these were necessary to measure the separate pressures which could be applied to each valve disc. The application of the different pressures was accomplished by means of a shut-off valve, between the two discs, which prevented an increase of pressure on top of one disc. By being able to apply different pressures to each disc, it was possible to measure not only the time taken for one valve disc to leave its seat, but also the pressure difference required across the valve for it to leave its seat. This will be explained more fully in section 3.7.2.

3.4.4 <u>Oil Control</u>

Because the floating discs did not act as true valves in that there was negligible flow past them, a copious supply of oil had to be made available in the sump of the compressor. This resulted in a large flow of oil through the actual discharge valve.

In order to control this extra flow and to prevent it from travelling through the circuit and hence, blocking the valves and pressure gauges, an oil separator was fitted immediately after the discharge side of the compressor. Oil from the separator then returned to the sump of the compressor.

However, if this oil return was kept open continuously, the discharge pressure would fall to the suction pressure. To overcome this, a solenoid valve and photo-electric cell were fitted.

The/

The solenoid valve was actuated by a photo-electric cell. The photo-electric cell was operated by a copper ball, floating on the oil inside the glass separator, shutting off the light as the oil level rose. This opened the valve for a short time (approximately 1 second) until the oil level fell again, when it was then closed. This short opening time also prevented the discharge pressure from falling. This system thus ensured that a constant oil level was maintained in the sump of the compressor.

3.5 Temperature Measurement

It was necessary to measure the temperature of the oil underneath the discs because of the dependence of the oil viscosity on its temperature.

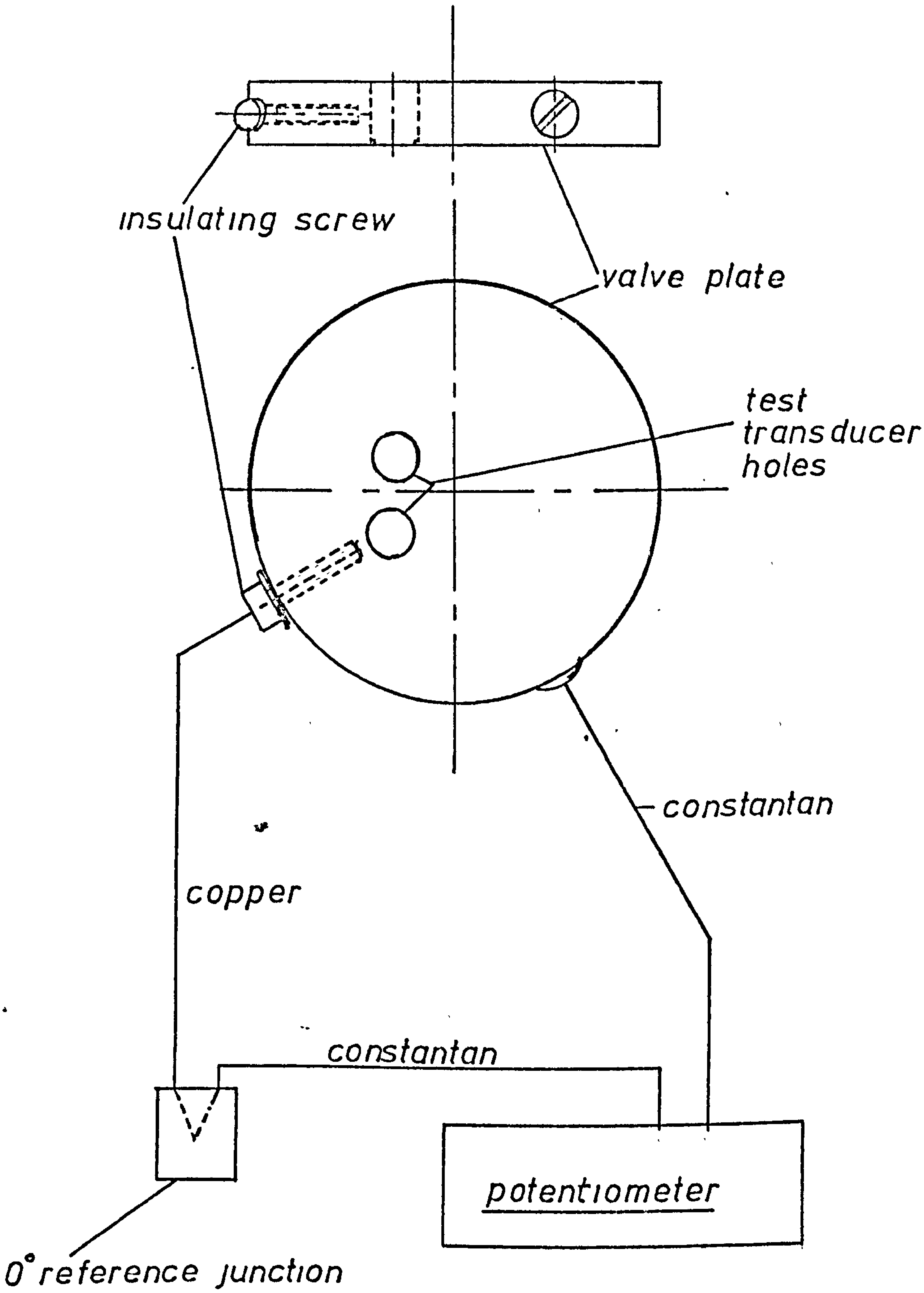
The method employed was to drill a hole radially in the side of the valve plate as near as possible to the balanced disc transducer without breaking through. A copper-constantan thermocouple was then inserted. The arrangement of the thermocouple circuit is shown in Fig. 3.5.1.

The reference junction was placed in a flask of shaved ice. The voltage was measured using a Cambridge potentiometer and the equivalent temperature determined with reference to tables in Weber (22).

3.6 Instrumentation

3.6.1 Introduction

In recent years, technological advances have simplified the construction and operation of measuring instruments and at the same/



DIAGRAMMATIC CONFIGURATION OF THERMOCOUPLE

FIG. 3.5.1

same time increased their accuracy and reliability. Instruments such as force, pressure, or displacement transducers now offer such a wide choice in make and mode of operation, as well as accuracy and price, that the need for manufacturing "home made" devices to suit a particular function is virtually unnecessary. Therefore, although detailed knowledge of the design of every type is not a prerequisite, a working knowledge is necessary in order that the most suitable type is chosen for a particular function. Instruments must not be treated simply as "black boxes".

3.6.2 Inductive Displacement Transducers

The displacements of the floating discs were detected using two inductive displacement transducers manufactured by Southern Instruments and the signals displayed on a 2-channel oscilloscope. The manner in which the transducers were held has already been described in section 3.3.2.

The transducers were displacement modulated, meaning that their outputs were proportional to disc displacement. However, it was found when calibrating them (see Appendix A.5) that this response was not linear, especially very close to the transducer. And when the transducer was placed at a distance relatively far from the disc, the output signal was not sufficiently large enough to be easily read on the oscilloscope.

Care had therefore to be taken when positoning the transducers to ensure that they were close enough to the disc to obtain a strong signal but sufficiently far enough away to keep the response reasonably linear.

3.6.3 F/M System

Standard oscillators and F/M discriminators were used in conjunction with the two inductive displacement transducers.

The oscillator was tuned to a set centre frequency and any voltage produced by the transducer due to a change in inductance caused a variation in frequency about this centre frequency.

The discriminator then detected this change in frequency and converted it to a d-c voltage and simultaneously amplified the signal in order that it could be read on a cathode-ray oscilloscope.

3.6.4 Oscilloscope

The oscilloscope needed to have two channels to allow the two signals to be displayed simultaneously. In order to hold the signals steady on the screen, an external trigger was also required so that times were measured from a set datum.

Since the time lag of the disc leaving its seat was very small compared to one revolution of the compressor, a time delay sweep was required in the oscilloscope so that the desired portion of the cycle could be expanded.

A Tektronix 7010 Storage Oscilloscope was used in the tests. Although other oscilloscopes available had the necessary functions required above, this particular type could retain one particular sweep and facilitated inspection of the oscillograms.

This meant that it could also be used in other experiments which required a "one-shot" operation.

3.6.5 Triggering Device

A small screw was fitted to the drive pulley of the compressor. As this screw passed an electromagnetic pick-up, it generated an electrical/

electrical signal which was used to trigger the oscilloscope. Because of the time delay mechanism available in the 'scope, the position of the trigger on the pulley was purely arbitrary. However, it is common practice to position the trigger at the "top dead centre" of the piston stroke. This position was also found to be convenient when determining the pressure wave in the cylinder (see section 3.7.4).

3.7 Compressor Apparatus Test Procedure

Both compressors were started up and allowed to run for approximately half-an-hour to allow them to reach steady conditions of pressure, temperature and oil flow. This time also allowed the electronic equipment to reach steady conditions. The speed of the back compressor was purely arbitrary so long as it could generate sufficient pressure on top of the transducer discs. The speed of the test compressor was varied from test to test in order to give different rates of change of pressure for a required back pressure. It was, therefore, run at the speed for the next required test during the half-an-hour pre-run. The reason for this was that the running speed influenced the temperature of the oil. Because the viscosity of the oil is dependent on its temperature, care had to be taken to ensure that the oil had reached a steady temperature before readings were taken.

The test seats were always thoroughly cleaned with a degreasing agent before each run commenced. Because of this, no stiction effect was observed immediately the compressor was started. However, this effect became apparent after only a few minutes when oil was deposited on the seats.

3.7.1 Time Lag Measurement

The back pressure on top of the discs was initially set low - usually atmospheric pressure if possible - and the time lag noted using the displacement signals on the oscilloscope screen. (Depending on the amount of fluid on the system, it was often only possible for the back pressure to be as low as 50 kN/m^2). The back pressure was then increased by regular steps and the time lag noted for each case.

The time lag measurement was obtained by taking the distance on the oscilloscope screen from the point where one disc left the sharp-edged seat to the point where the other disc left the test seat. A diagrammatic sketch of a typical oscillogram is shown in Fig. 3.7.1.

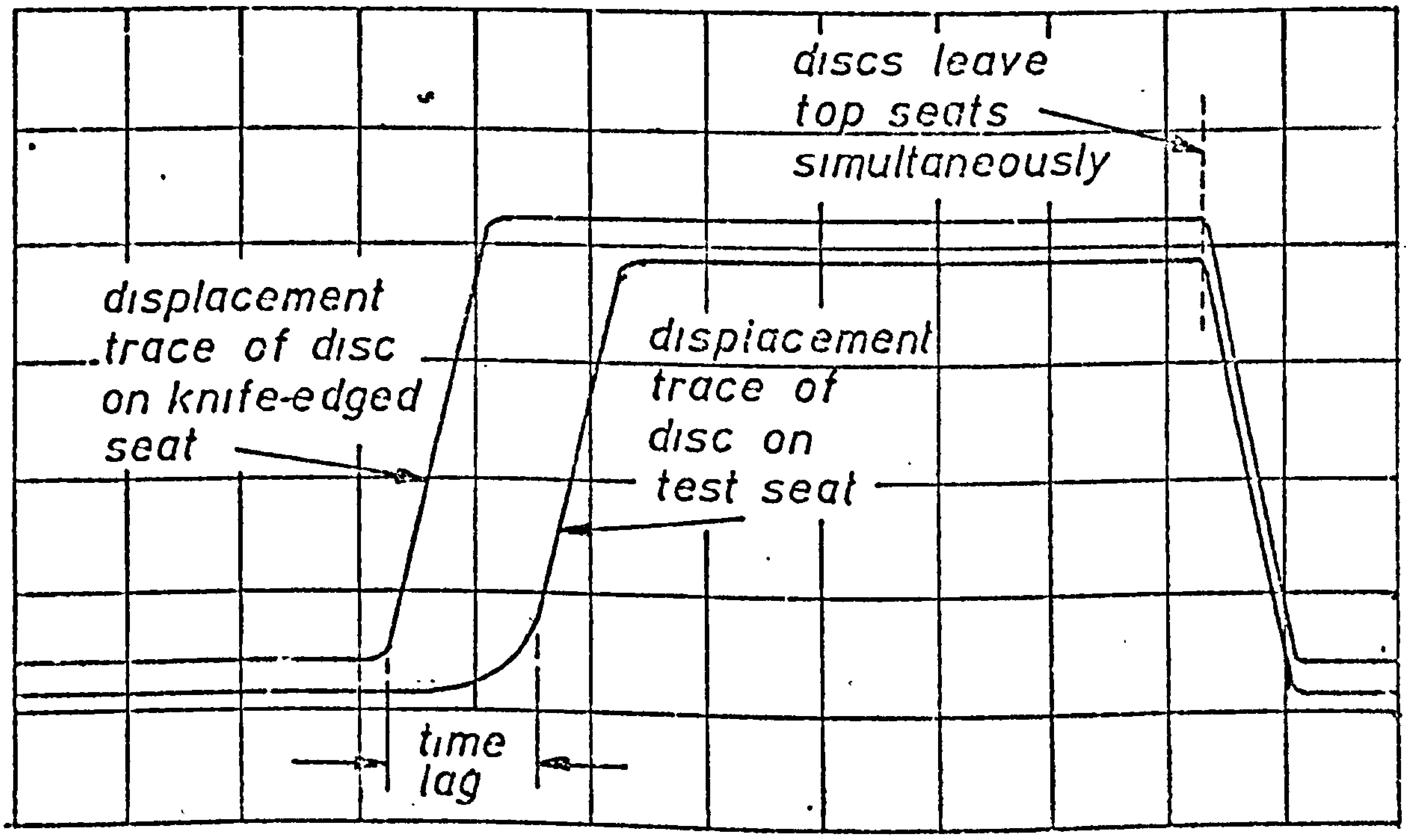


FIG. 3.7.1

The actual time lag was this distance multiplied by the time scale of the oscilloscope. The time scale could be expanded if/

if a more accurate measurement of the time lag was required.

The rate at which one disc left the test seat was not as sharp as the rate at which the other disc left the sharp-edged seat. The reduced initial rate was thought to be due to the viscous action of oil and so the disc was therefore still in contact with the oil film. Because of this, the time lag was measured to the point where the disc began to move at a rapid rate as shown above.

In general, this time lag remained quite constant, although small shifts or variations were sometimes observed. These were attributed to two possible causes. If only the test disc signal moved, this was thought to be due to incompletion of the film. However, if both signals shifted simultaneously (as was the more usual case), these small variations were (deduced to be) caused by belt slip. If the belt slipped, the speed of the compressor was affected and so the time to reach the balance pressure and also the time lag was affected. This would cause small shifts of the oscillograms.

3.7.2 Pressure Difference Measurement

The balance pressure of the discs could easily be found as this was the back pressure applied to the discs. The time lag could also easily be determined using the previously described method. However, the pressure difference across the test disc at opening, could not easily be determined. An attempt was made to measure the pressure difference, using the pressure v crank angle trace obtained as explained in section 3.7.4. Fig. 3.7.2 is referred to

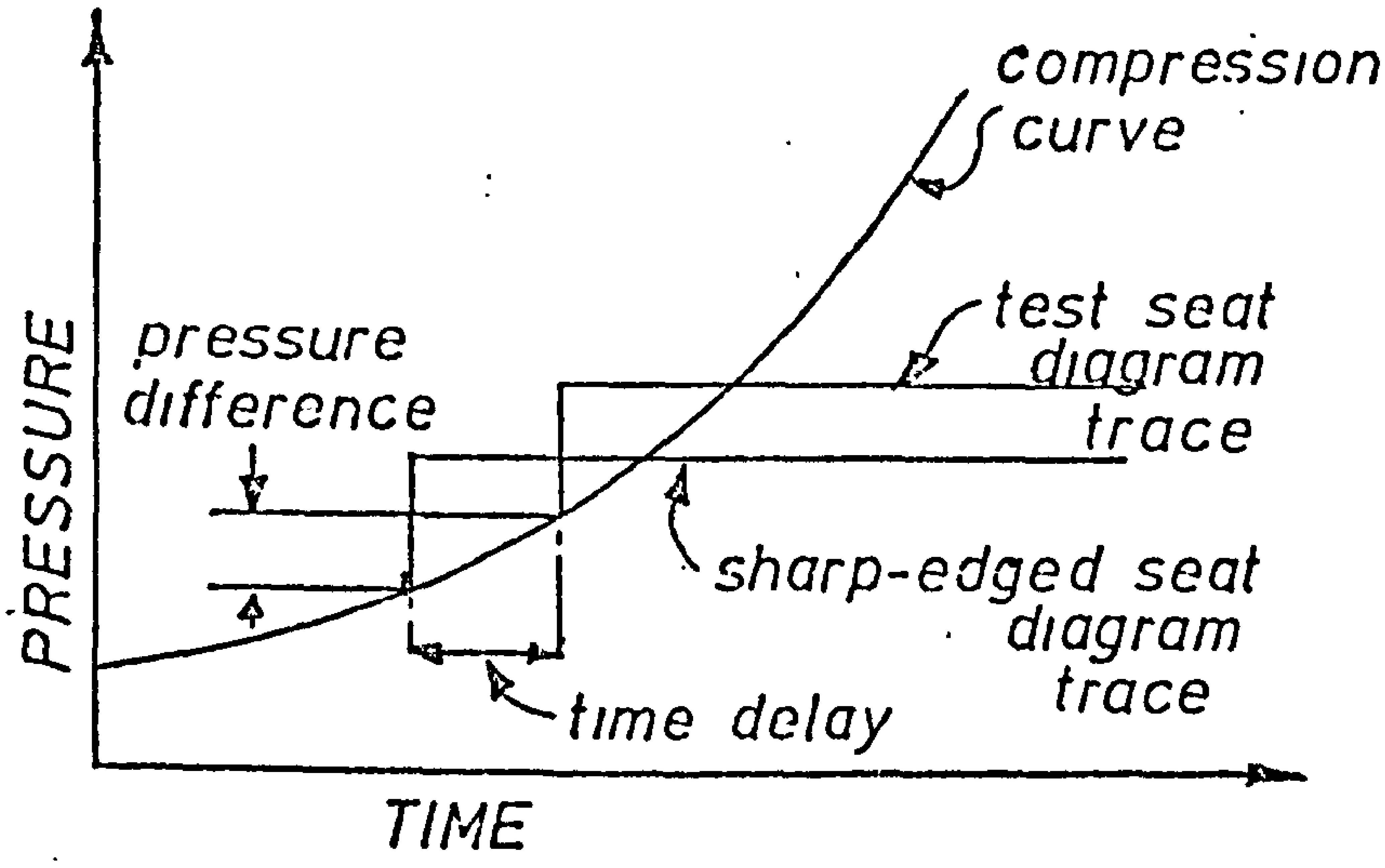


FIG. 3.7.2

If the exact pressure curve is known and the balance pressure and corresponding time lag are known, it should then be possible to measure the pressure difference as shown. This method was not found to be wholly successful. Its main fault was that it depended on: either producing a pressure curve for each test; or that the compressor speed and other conditions were exactly the same for a test run as they were when the pressure v crank angle trace was produced. The first way was impractical because of the time involved and the second was impossible to achieve because of errors caused by the aforementioned belt slip and gas losses past the piston rings which could not be guaranteed to remain constant. Because this attempt at determining the pressure difference did indicate some interesting trends, it seemed worthwhile to modify the apparatus to try to obtain a more accurate value for the pressure difference necessary to overcome stiction.

For/

For this reason, the method of applying different back pressures to each disc was devised. By applying different pressures to each disc until the displacement signals coincided, it was possible to observe directly from the pressure gauges the pressure difference required to lift the "stiction disc" from its seat.

Because the time lag and the pressure difference across the disc were both known, it was quite straightforward to calculate the mean rate of change of pressure across the disc after the balance pressure was reached, and to observe how the time lag was affected by the rate of change of pressure.

3.7.3 Temperature Measurement

The temperature in the head of the compressor was noted throughout the run of each test. However, this temperature varied little for each particular speed.

3.7.4 Determination of Pressure v Crank Angle Trace

The following method was used to determine, experimentally, the variation of pressure in the cylinder with respect to time. It was based once again on the fact that a disc on a knife-edged seat exhibits negligible time lag. The operation, in fact, was the method employed when using a Farnborough pressure indicator.

Because the disc moved when the pressure in the cylinder equalled the back pressure (both on suction and discharge), by measuring the time on the oscilloscope from a known datum for various back pressures, it was possible to plot the pressure variation in the cylinder vs time. The datum taken was the top dead centre marker which was also used for triggering the oscilloscope as explained earlier.

It/

It was found more convenient when measuring the time from the TDC, for both the suction and discharge modes, to display velocity of the disc instead of displacement. The reason being that it kept a constant straight line on the screen of the oscilloscope apart from the TDC marker and the time when the disc was moving from one seat to the other. Because the time for the disc to move was very short indeed, it meant that the acceleration and deceleration of the disc resulted in a sharp "blip" on the screen as shown in Fig. 3.7.3 for either suction or discharge. The distances of 'x' and 'y' varied depending on the back pressure applied to the disc.

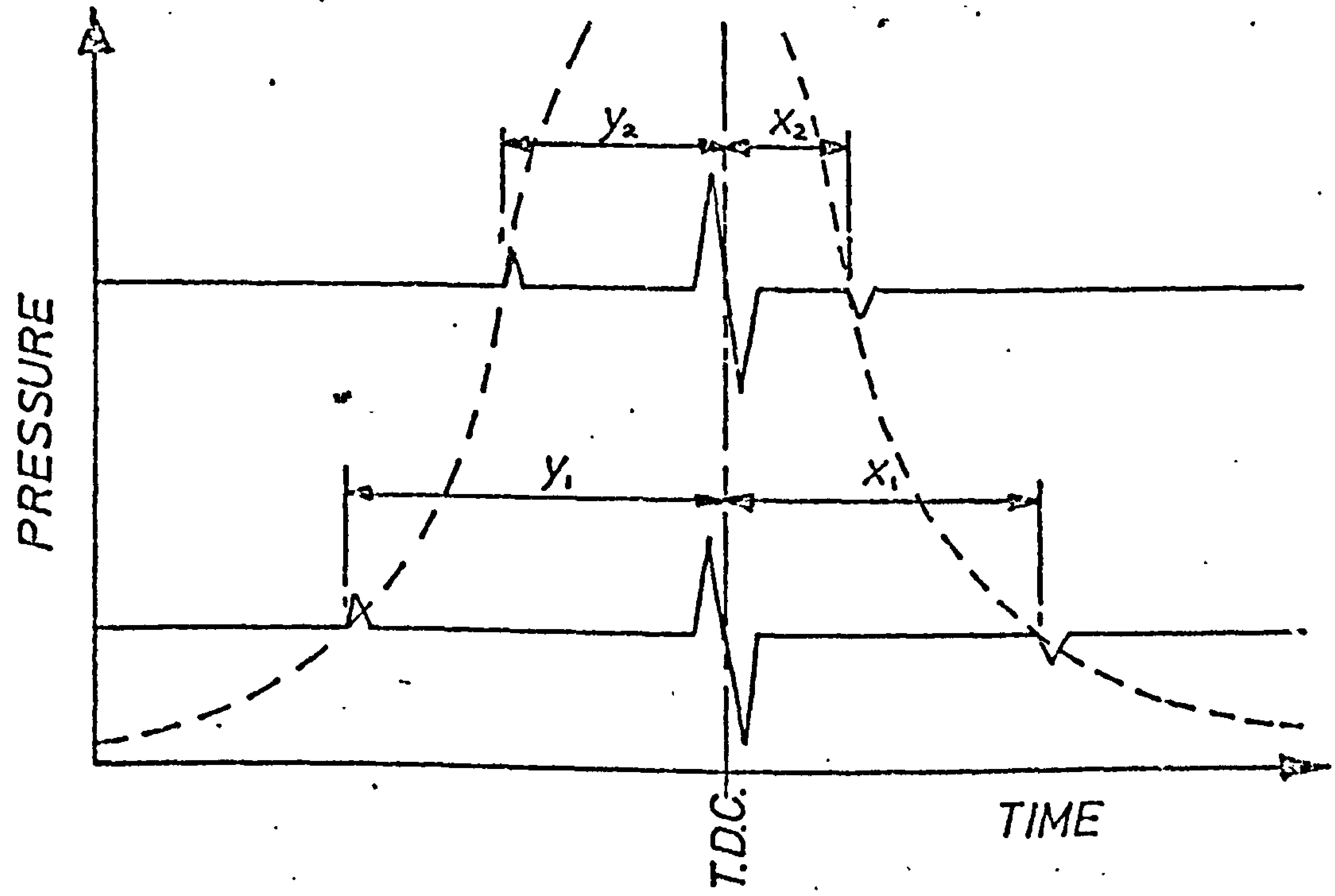


FIG. 3.7.3

A set of results for two particular speeds are shown in Tables (3.74) and (3.75) with accompanying graphs.

3.7.4.1 <u>Differentiation of Signal</u>

In order to display velocity instead of displacement on the screen, a capacitor was introduced between the F/M discriminator and/

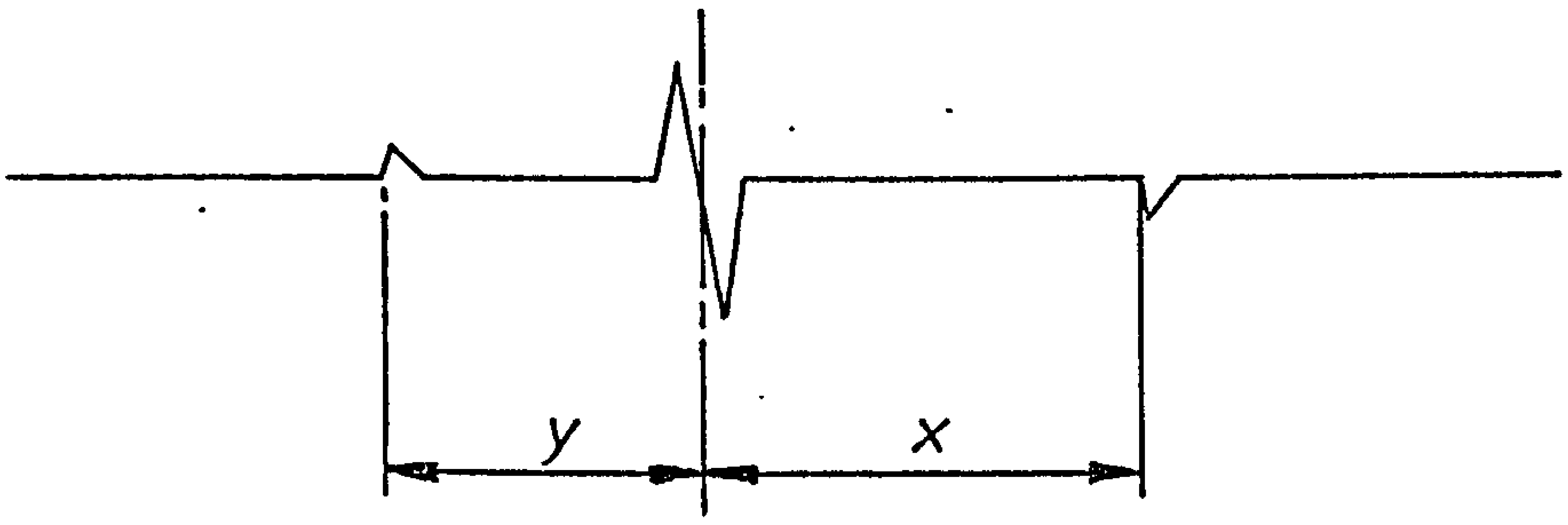
TABULATED RESULTS

To determine the pressure variation against time.

Compressor speed: 1140 RPM

Disc thickness: 0.25 mm

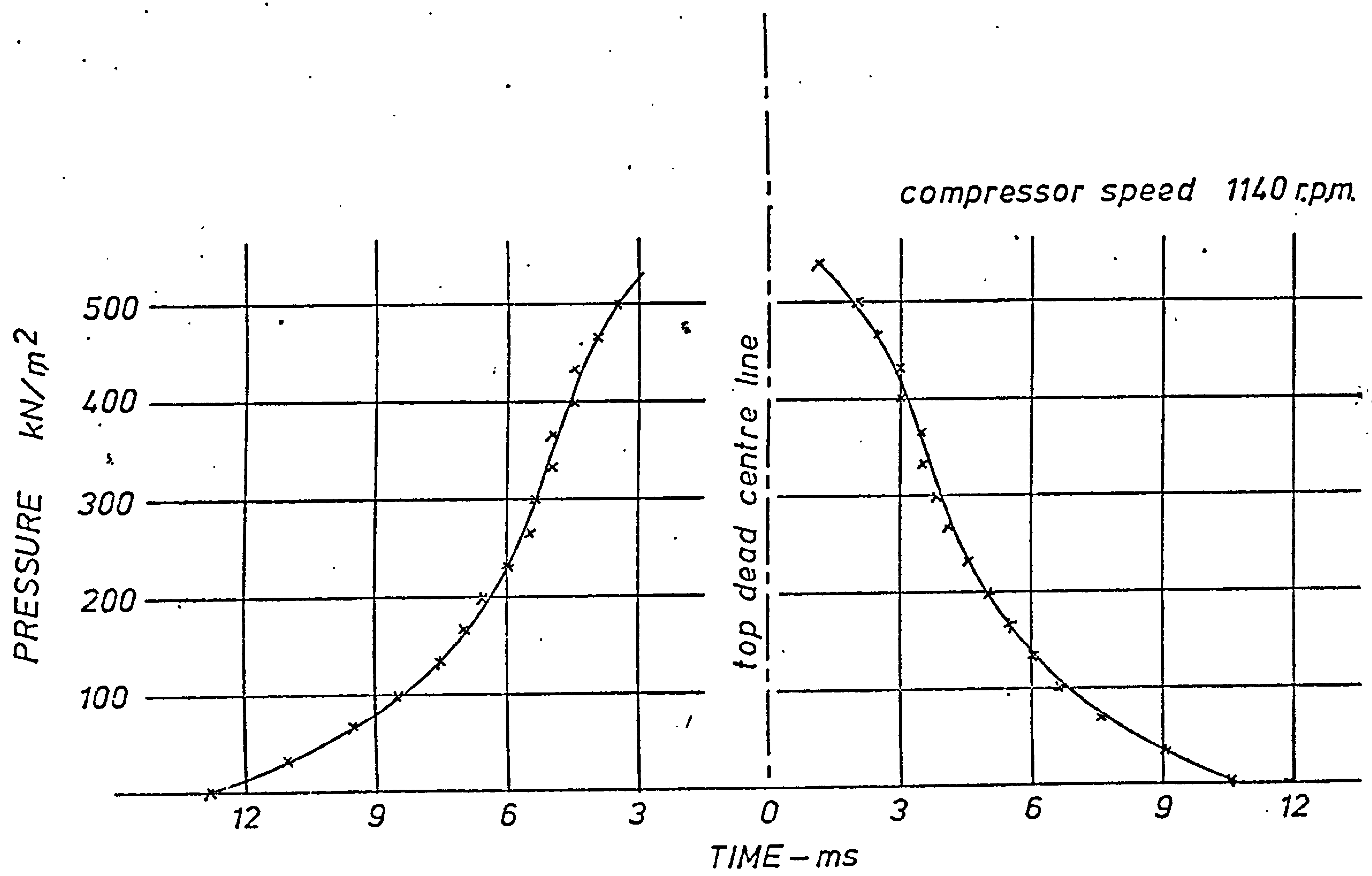
Valve seat: No. 4



Above is the "graph" on the oscilloscope.

BACK PRESSURE kN/m^2	'y' MILLISECONDS	'x' MILLISECONDS
0	12.5	10.5
35	11.0	9.0
70	9.5	7.5
105	8.5	6.5
140	7.5	6.0
175	7.0	5.5
210	6.5	5.0
245	6.0	4.5
280	5.5	4.0
315	5.5	4.0
350	5.0	3.5
385	5.0	3.5
420	4.5	3.0
455	4.5	3.0
490	4.0	2.5
525	3.5	2.0

TABLE 3.7.4



PRESSURE v CRANK ANGLE (TIME) TRACE-DETERMINED, EXPERIMENTALLY

FIG. 3.7.4

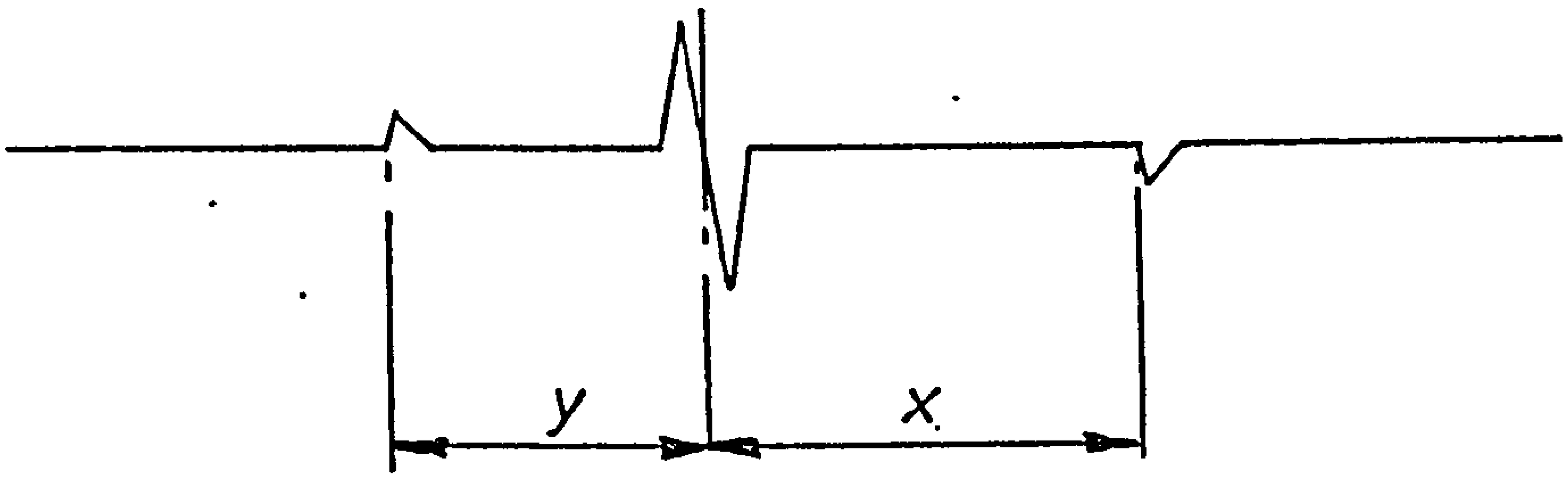
TABULATED RESULTS

To determine the pressure variation against time.

Compressor speed: 680 RPM

Disc thickness: 0.25 mm

Valve seat: No. 4



Above is the "graph" on the oscilloscope.

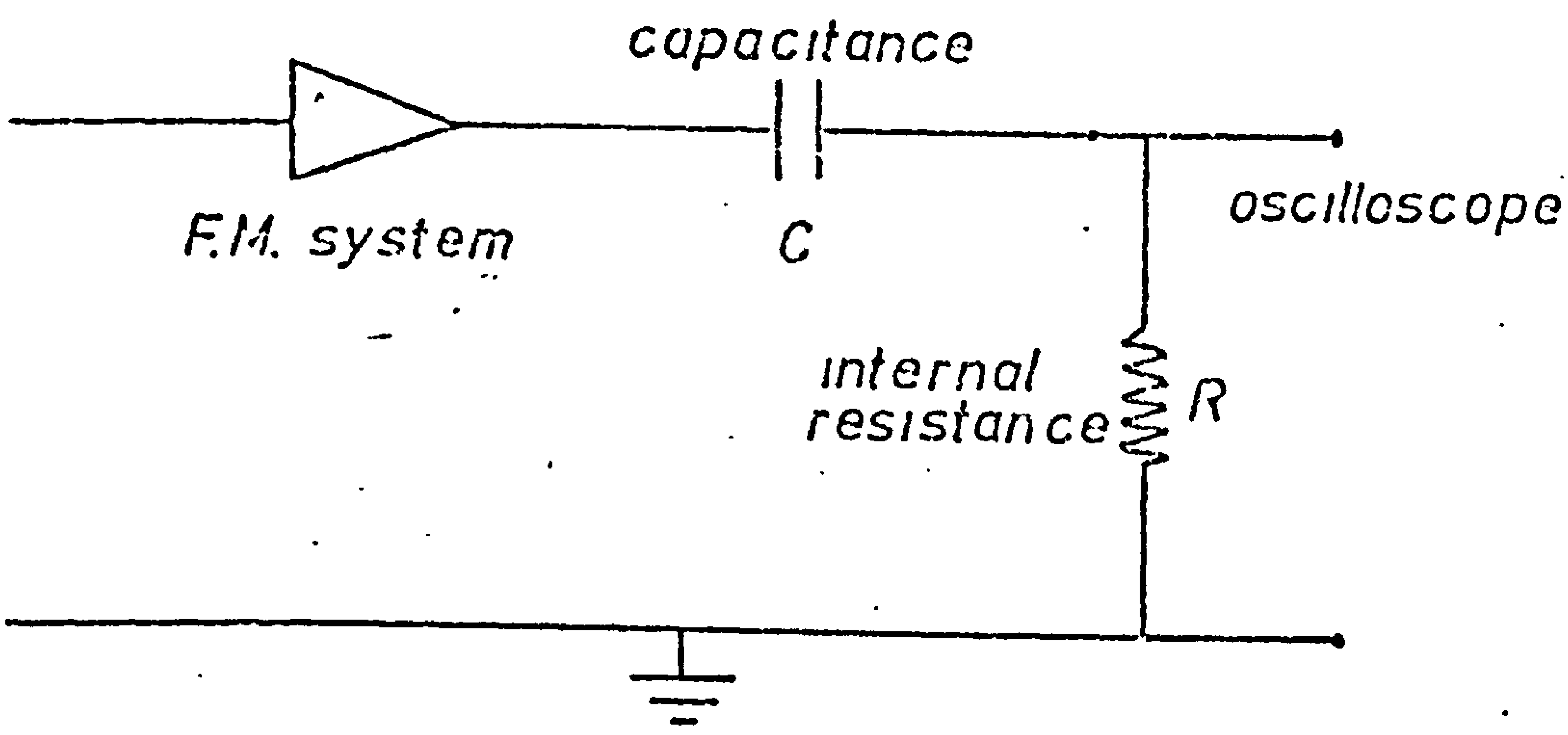
BACK PRESSURE kN/m^2	'y' MILLISECONDS	'x' MILLISECONDS
0	25.0	16.5
35	21.5	14.5
70	19.0	12.5
105	17.0	11.0
140	16.0	10.0
175	14.5	9.0
210	13.5	8.0
245	13.0	7.5
280	12.0	7.0
315	11.5	6.0
350	11.0	6.0
385	10.5	5.5
420	10.0	5.0
455	9.5	4.5
490	9.0	4.0
525	8.5	3.5

TABLE 3.7.5

PRESSURE v CRANK ANGLE (TIME) TRACE – DETERMINED EXPERIMENTALLY

FIG. 3.7.5

and the oscilloscope. This differentiated the displacement signal to show velocity. Fig. 3.7.6 shows that advantage was taken of the internal resistance of the oscilloscope to complete the differentiation network.



CIRCUIT FOR DIFFERENTIATION OF DISPLACEMENT SIGNAL

FIG. 3.7.6

3.8 Compressor Test Results

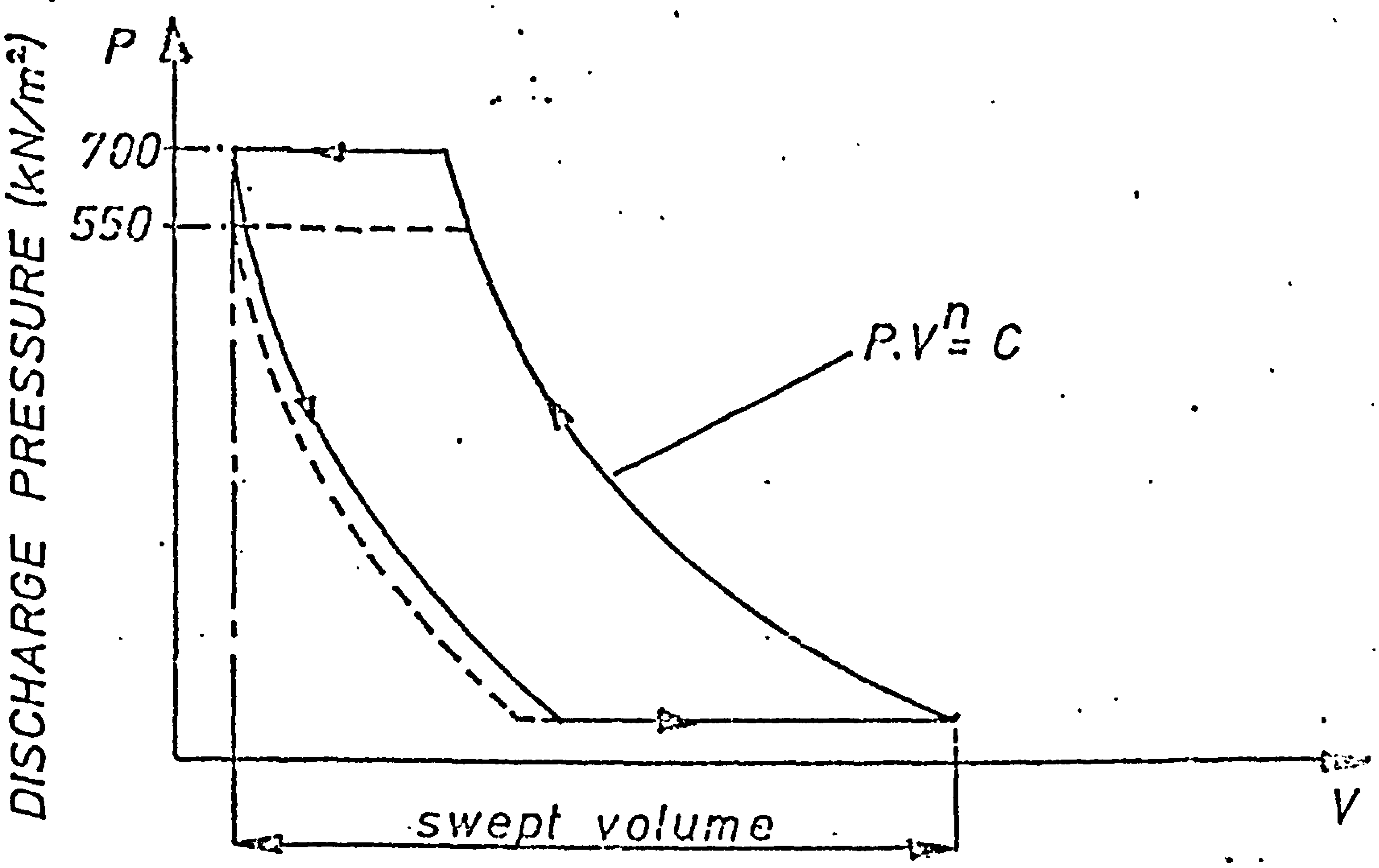
The results of tests carried out in the compressor apparatus are shown in Figs. 3.8.4 - 3.8.9. These results were all derived from a comprehensive series of tests carried out under various experimental conditions with different working fluids.

Fig. 3.8.3 shows a sample set of figures taken during one test run. It shows the experimental results as well as values derived from these. It can be seen that the discharge pressure was held constant for each test and also that the intermediate pressure remained constant. During the tests where air was the working fluid/

fluid, the storage cylinder was kept open to the atmosphere - hence the reason for the intermediate pressure being atmospheric. The oil temperature remained constant throughout the test.

Fig. 3.8.4 shows clearly how the time lag is affected by the rate of change of pressure in the cylinder. This rate of change of pressure is the average value during the opening sequence of the valve disc since it is found by dividing the pressure difference across the valve at opening by the time lag exhibited. As the rate of change of pressure increases, the time lag decreases. This increase of rate of change of pressure was effected by increasing the back pressure on top of the discs. This had the result of moving the balance pressure to a steeper part of the pressure v crank angle trace. No real difference was found when two different speeds were compared on a time lag v rate of change of pressure plot. Slight differences which were observed were attributed to the difference in viscosity caused by the different running speeds.

It can also be seen from Fig. 3.8.5 that no difference is observed when different discharge pressures are compared. This is possibly expected if Fig. 3.8.1 is considered.



IDEAL PRESSURE-VOLUME DIAGRAM

FIG. 3.8.1

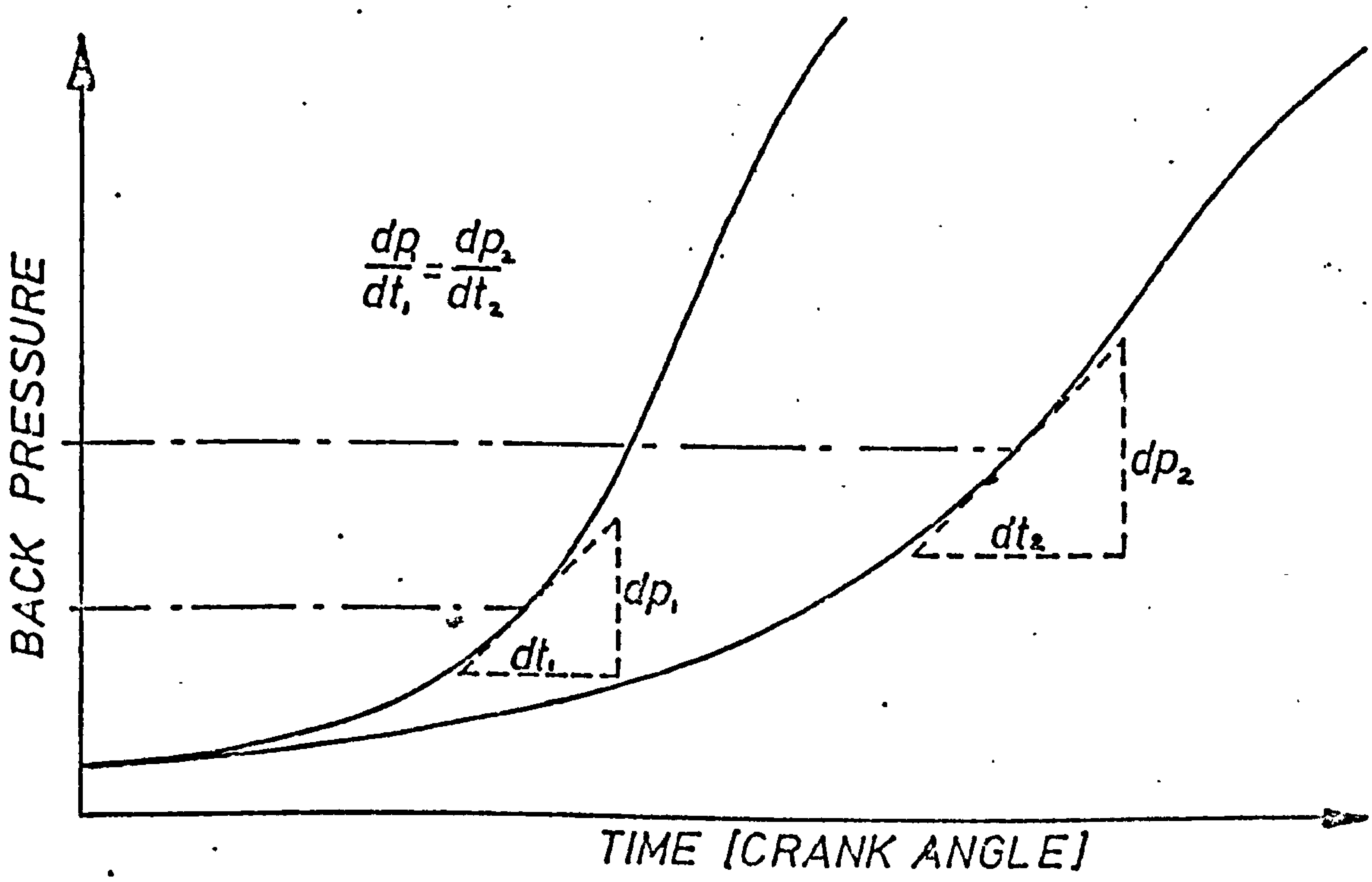
So long as the back pressure on top of the discs is less than 550 kN/m^2, it will make no difference to the time lag if the discharge valve of the compressor opens at 550 kN/m^2 or 700 kN/m^2.

In Fig. 3.8.6 comparison is made between the time lags for different compressor speeds. It can be seen that, within experimental error, there is virtually no difference in time lag when comparison is made with the rate of change of pressure in the cylinder.

As previously stated, the reason why different rates of change of pressure were achieved for one particular speed was because of the fact that the slope of the pressure v crank angle trace increased as the pressure in the cylinder increased. Therefore, by increasing/

increasing the back pressure on top of the discs, an increase in the rate of pressure change was accomplished.

However, for an increase in compressor speed, the back pressure does not need to be so great to achieve the same rate of pressure change. This can be shown diagrammatically in Fig. 3.8.2. Although the slopes are both the same, the pressures are not.



PRESSURE v CRANK ANGLE TRACES FOR TWO COMPRESSOR SPEEDS

FIG. 3.8.2

It can be inferred from Fig. 3.8.5 that the back pressure has little influence on the time lag. This means that the pressure increase with back pressure shown in Fig. 3.8.7 must be due to the increased rate of change of pressure associated with the increase of back pressure.

Fig. 3.8.8 shows the effect of the valve thickness on the time/

time lag exhibited. By decreasing the thickness, the time lag is also reduced. This difference is strikingly noticeable and was one of the reasons for introducing the flexiblity of the disc into the theoretical model. The theoretical model, in turn, ascribes this reduced time lag to the greater thickness of oil at the balance pressure for the thin disc and also to the earlier initiation of evolution.

Differences in time lag are also observed when two different working fluids are compared - air and Refrigerant 12 for the case of Fig. 3.8.9. Two possible reasons can be given to explain this. One is that the refrigerant, being more miscable with the oil than air, evolves vapour much more readily and so will have a larger evolution constant. The other reason is that the oil/refrigerant mixture has a much lower viscosity than oil by itself and so this will have an effect on the time lag. The theoretical model also corroborates this opinion.

TEST NO. 5A   DATE 24th APRIL, 1975   OIL USED CLAVUS 27   SEAT NO. 4

DISC THICKNESS 0.125 mm   SPEED 680 RPM   AMBIENT TEMP. 23°C

DISCHARGE PRESSURE	SUCTION PRESSURE	INTERMED PRESSURE	BACK PRESSURE	BACK PRESSURE		PRESSURE DIFFERENCE	NET TIME LAG	OIL TEMPERATURE	RATE OF PRESSURE INCREASE
				LEFT	RIGHT				
kN/m^2	kN/m^2	kN/m^2	kN/m^2	kN/m^2	kN/m^2	kN/m^2	MILLISEC	MILLIVOLT	$kN{*}10^3/m^2\,sec$
700	22.0	0	70	105	70	35	1.75	2.355	19.72
700	22.0	0	140	186	145	41	1.25	2.355	33.10
700	22.0	0	210	258	210	48	1.125	2.355	42.90
700	22.0	0	280	335	280	55	0.875	2.355	63.03
700	23.7	0	350	412	350	62	0.75	2.355	82.76
700	23.7	0	420	489	420	69	0.75	2.355	91.72

FIG. 3.8.3


COMPRESSOR SPEED 650 rev/min
WORKING FLUID - AIR
DISC THICKNESS - 0,2 mm
SEAT No. 4
TIME LAG (ms)
dP/dt (kN/m².s ×10³)
0
10
20
30
40
50
60
70
80
90
100
110
120
1
2
3
4
5
6

FIG. 3.8.4

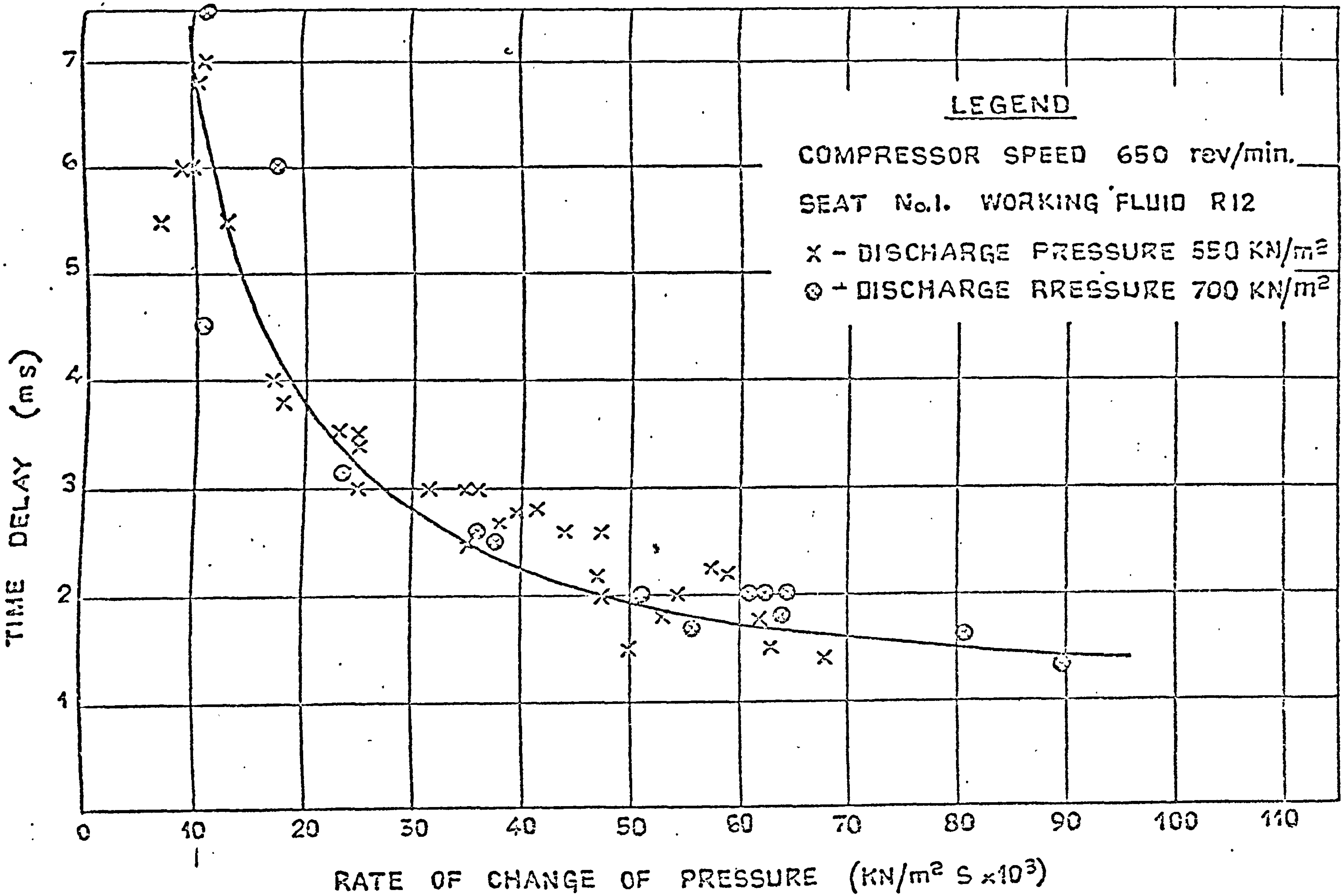
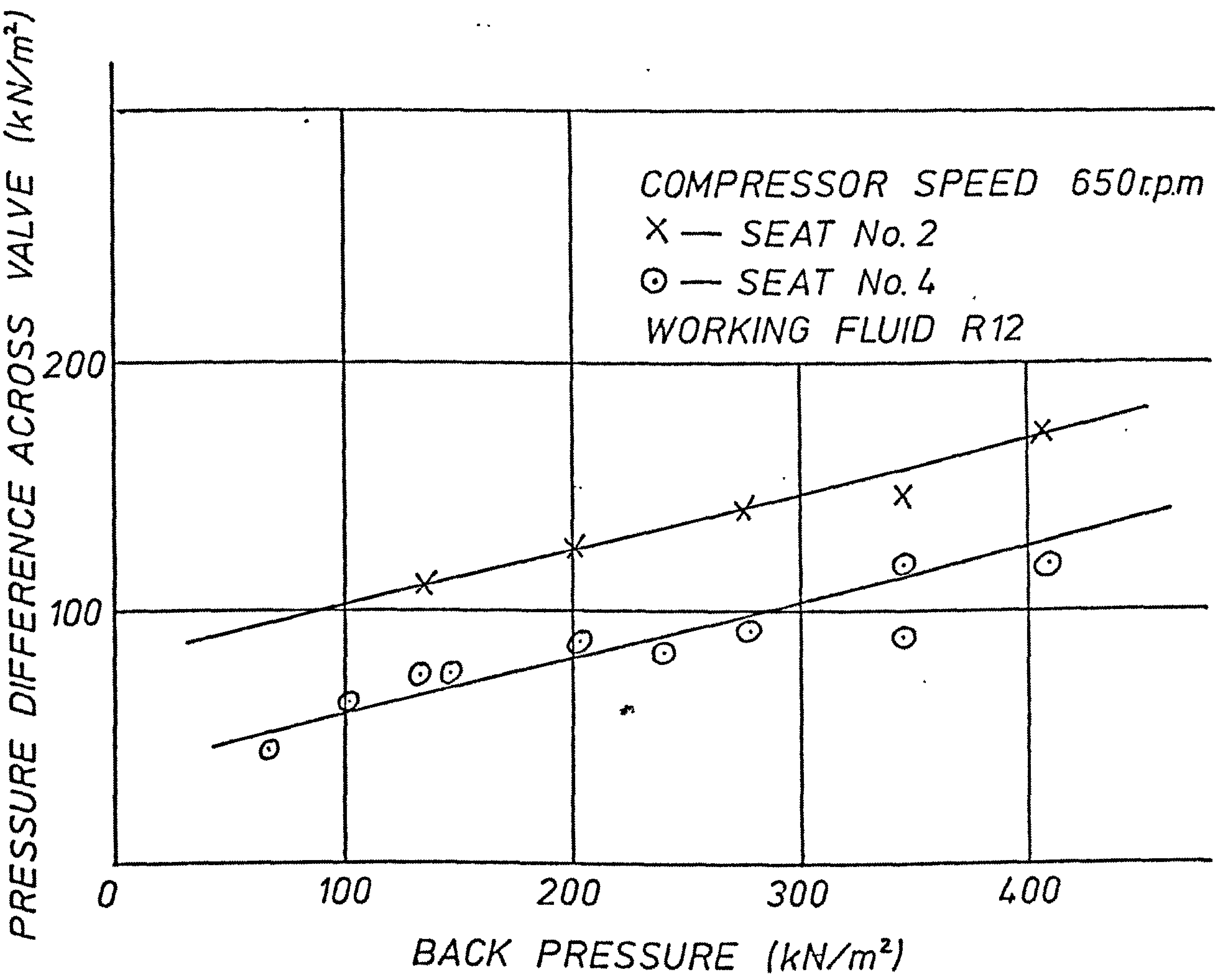

LEGEND
COMPRESSOR SPEED 650 rev/min.
SEAT No.1. WORKING FLUID R12
X - DISCHARGE PRESSURE 550 KN/m²
⊗ - DISCHARGE RRESSURE 700 KN/m²
TIME DELAY (ms)
RATE OF CHANGE OF PRESSURE (KN/m² S ×10³)
0
10
20
30
40
50
60
70
80
90
100
110
1
2
3
4
5
6
7

FIG. 3.8.5


COMPARISON OF TIME LAGS FOR DIFFERENT COMPRESSOR SPEEDS
× — COMPRESSOR SPEED 650 rev/min
⊙ — COMPRESSOR SPEED 1000 rev/min
⊡ — COMPRESSOR SPEED 1300 rev/min
No.4 SEAT 0,2 mm DISC THICKNESS
TIME LAG (ms)
$dP/dt \times 10^3$ (kN/m²s)
0
10
20
30
40
50
60
70
80
90
100
110
120
1
2
3
4
5
6

FIG. 3.8.6



PRESSURE DIFFERENCE INCREASING WITH BACK PRESSURE

FIG. 3.8.7

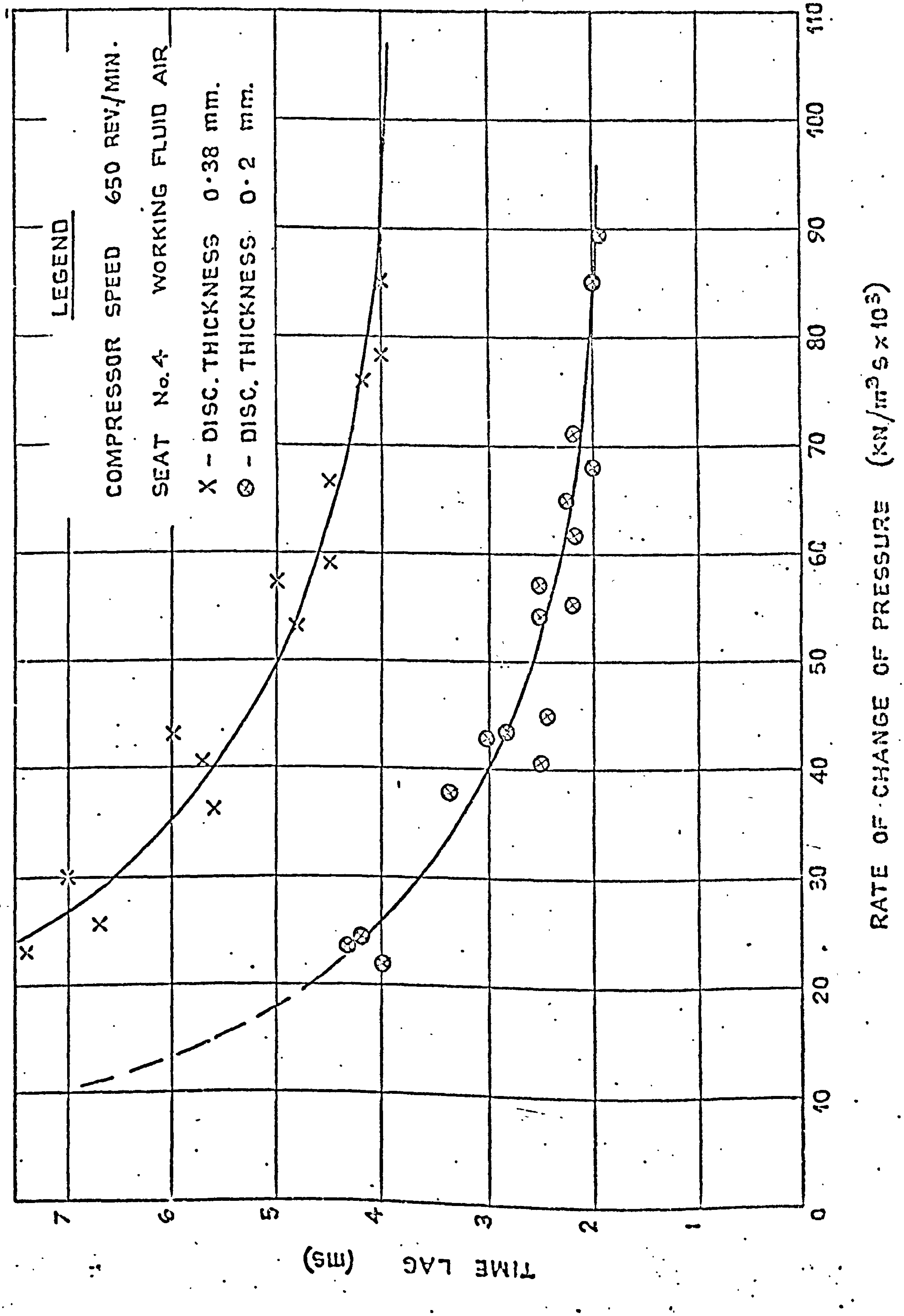
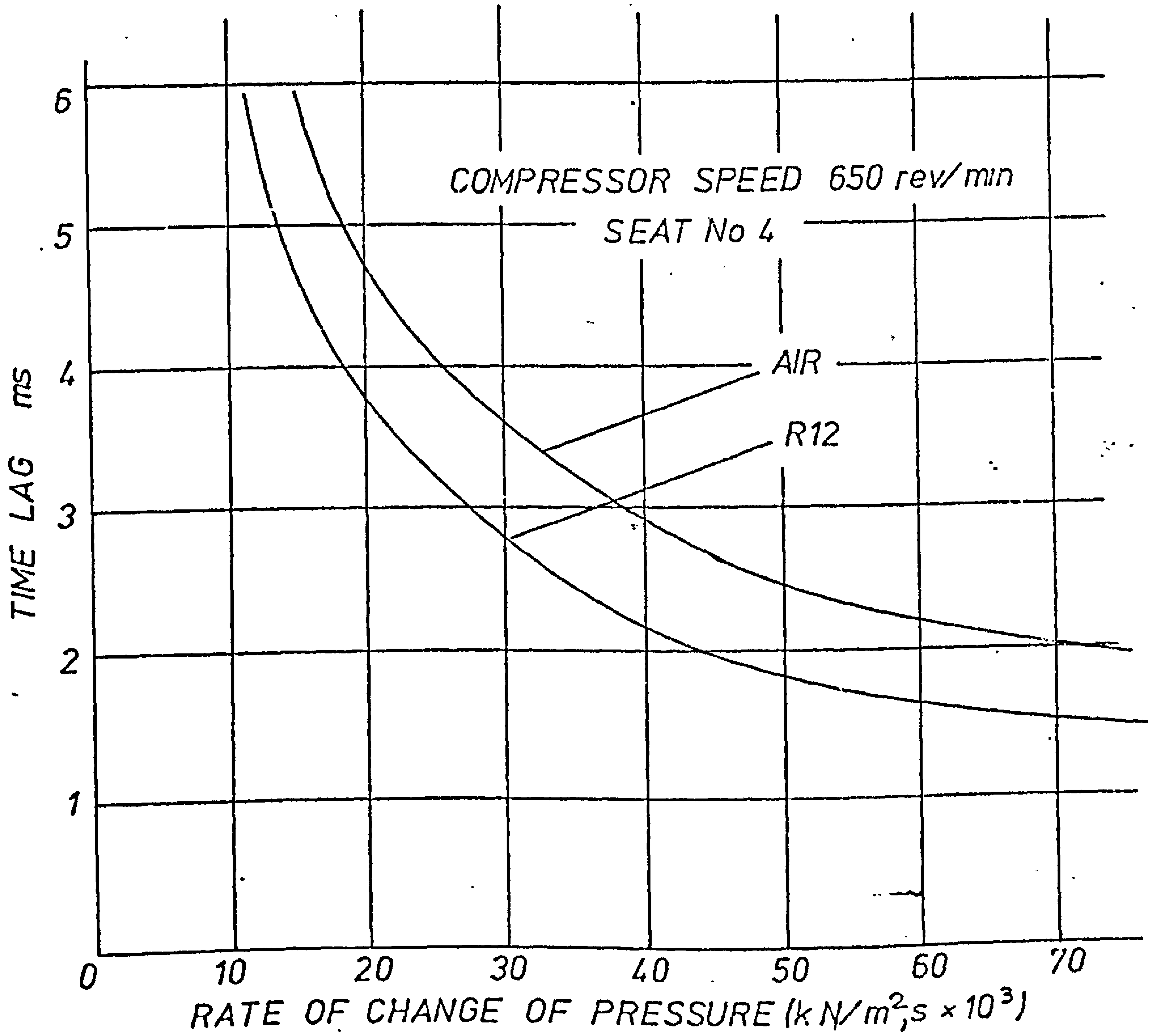

LEGEND
COMPRESSOR SPEED 650 REV./MIN.
SEAT No. 4 WORKING FLUID AIR
X - DISC. THICKNESS 0·38 mm.
⊗ - DISC. THICKNESS 0·2 mm.
TIME LAG (ms)
RATE OF CHANGE OF PRESSURE (kN/m³ s x 10³)
0
10
20
30
40
50
60
70
80
90
100
110
1
2
3
4
5
6
7

FIG. 3.8.8



EFFECT OF WORKING FLUID

FIG. 3.8.9

CHAPTER 4

DIRECT FORCE-PULL APPARATUS

4.1 Introduction

In parallel with the tests carried out on the compressor, an apparatus was designed and constructed which, it was hoped, would measure, in as direct a manner as possible, the stiction force on the valve. The compressor apparatus allowed the stiction force to be deduced from the pressure difference across the valve disc at opening. However, this force was really only an average value, so it was hoped that, by measuring the force directly, any variation during the opening sequence could be detected. The mode of operation of the apparatus also isolated the known parameters by eliminating the unknown effects related to the varying gas pressure in the compressor cylinder. In particular, these unknown effects were the influence of the pressure difference across the seat on the oil film and also the dependence on the pressure difference of the flexure of the valve disc. It was also possible to ensure, by inspection, that the oil film was complete before each operation.

It was during these tests that the variation in force with rate of valve displacement was observed. The possibility then arose of deducing empirically certain unknown coefficients in the theoretical model i.e. the time delay for the film to cavitate and the evolution rate of the gas in solution. The equipment was constructed to allow varying rates of pull-off force to be applied in order to observe any variation in the time for the film to collapse. The modification to the equipment also enabled the maximum applied force to be restricted. This will be shown to have allowed the determination of the pressure at which gas begins to be evolved.

The force pull apparatus gave the first real indication of the/

the effect of surface finish. If the seats and disc were not lapped, the peak force obtained was significantly reduced, as was the time for the film to rupture.

4.2 Description of Apparatus

A diagrammatic sketch is shown in Fig. 4.2.1 and a photograph of the complete set-up in Plate 3.

A valve seat, similar to the type used in the compressor apparatus, was mounted on a Kistler force transducer which in turn was rigidly fixed to a solid base.

A circular flat disc, which was meant to simulate a valve reed, was held on top of the seat and mechanically pulled from it by means of an electro-magnetic device.

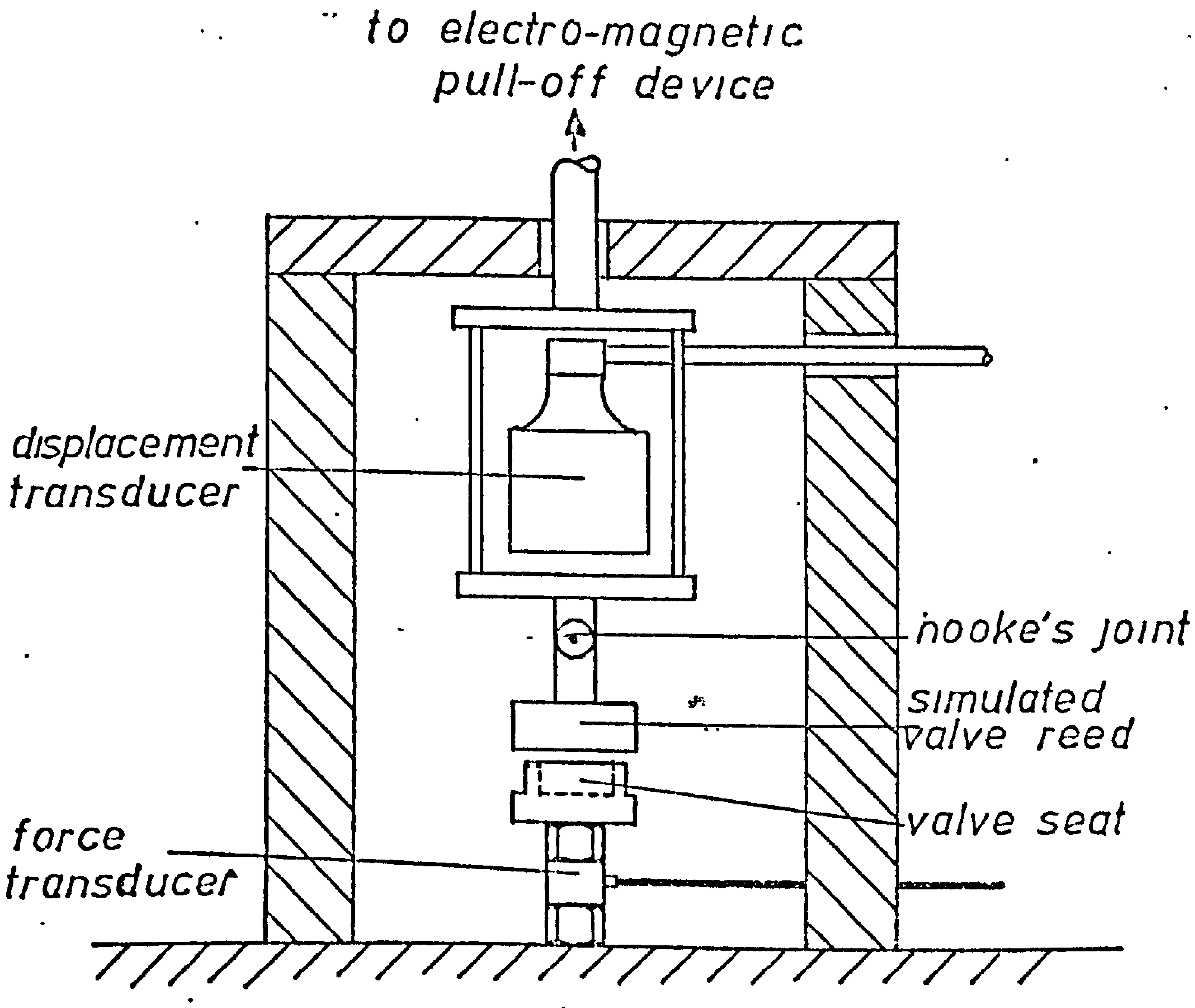
A Hooke's joint, placed between the pull-off device and the simulated valve disc, prevented any tilting of the disc when it was pulled off its seat and ensured that its motion was normal to the seat. If this did not happen a sliding action would occur, resulting in a reduction of the oil force holding the disc down.

The force transducer had been fixed to the base instead of to the moving disc. This was because the readings would be affected by the motion of the transducer which could act as an accelerometer.

In order to compare any variation of stiction force with valve displacement, a Wayne-Kerr displacement transducer was fitted to the apparatus.

As shown in the sketch, the displacement was measured from a flat surface immediately above the simulated valve.

A rigid, mild steel frame held the electro-magnetic device as/



DIRECT FORCE-PULL APPARATUS

FIG. 4.2.1

as well as the displacement transducer.

4.3 Electrical Circuit

A block diagram in Fig. 4.3.1 shows the method employed to apply power to the pull-off apparatus.

The electro-magnetic device referred to was, in fact, a Pye-Ling electro-magnetic vibrator. However, by applying a DC voltage to it, one direction of motion could be achieved.

The block diagram shows that two power supplies were applied to the vibrator. By this method, a suitable downward force could initially be applied to the valve and seat.

By closing the switch on the other power supply circuit, an increased voltage in the opposite direction enabled a sudden upward force to be applied, pulling the disc from its seat.

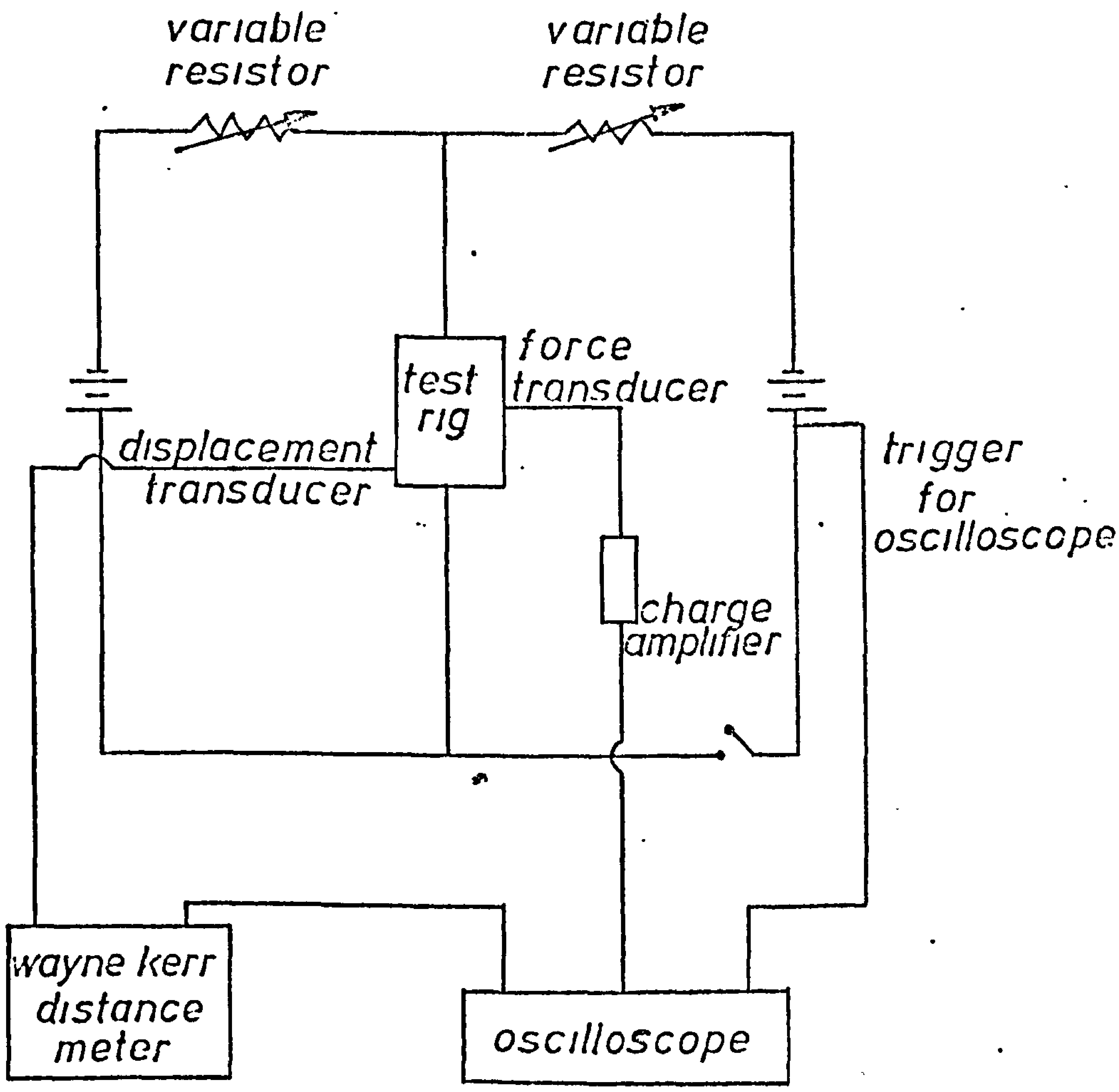
4.4 Instrumentation

4.4.1 Force Transducer

The oil stiction force holding the disc onto the seat was measured by a Kistler Type 9311 force transducer. This transducer was of the piezo-electric type in which the deformation of a quartz-crystal inside the transducer produces an electric charge. This charge was amplified using a charge amplifier which worked in conjunction with the transducer and produced a DC voltage which could be measured on an oscilloscope.

4.4.2 Displacement Probe

The displacement probe for detecting the motion of the valve disc was a Wayne-Kerr capacitative transducer. Instead of an/



BLOCK DIAGRAM OF FORCE PULL-OFF APPARATUS

FIG. 4.3.1

an output proportional to the inductance between the probe and the valve disc, as was the case of the displacement probes used in the compressor apparatus, the voltage developed here was proportional to the capacitance between the probe and the measuring surface.

The oscillator and amplifier used with this transducer were part of the same unit and the output of the 50 kc/s carrier frequency was amplitude-modulated by the valve disc displacement. A feedback circuit produced a demodulated signal proportional to the valve displacement.

4.4.3 Oscilloscope

Unlike the repetitive motion of the valve disc in the compressor equipment, the operation here was of a "one-shot" type. The signals from the force and displacement transducers had therefore to be saved in order that they could be analysed. Normally this would have involved triggering a camera to record the trace as it flashed across the screen of the oscilloscope. However, it was possible to store the oscillograms using the previously described storage oscilloscope. This had a special cathode-ray tube which could retain the trace for a long period, thus providing time to examine the traces before they were erased electronically to allow the next set of readings to be taken.

If desired, these traces could be photographed on 35 mm film for later analysis.

4.4.4 Triggering Operation

Since the valve motion lasted only a few milliseconds, the oscilloscope had to be triggered at the correct instant. A wire from the pull-off circuit of the equipment was connected to the triggering/

triggering channel of the oscilloscope. Thus, when the switch was closed, the current pulled the disc from its seat and simultaneously triggered the oscilloscope.

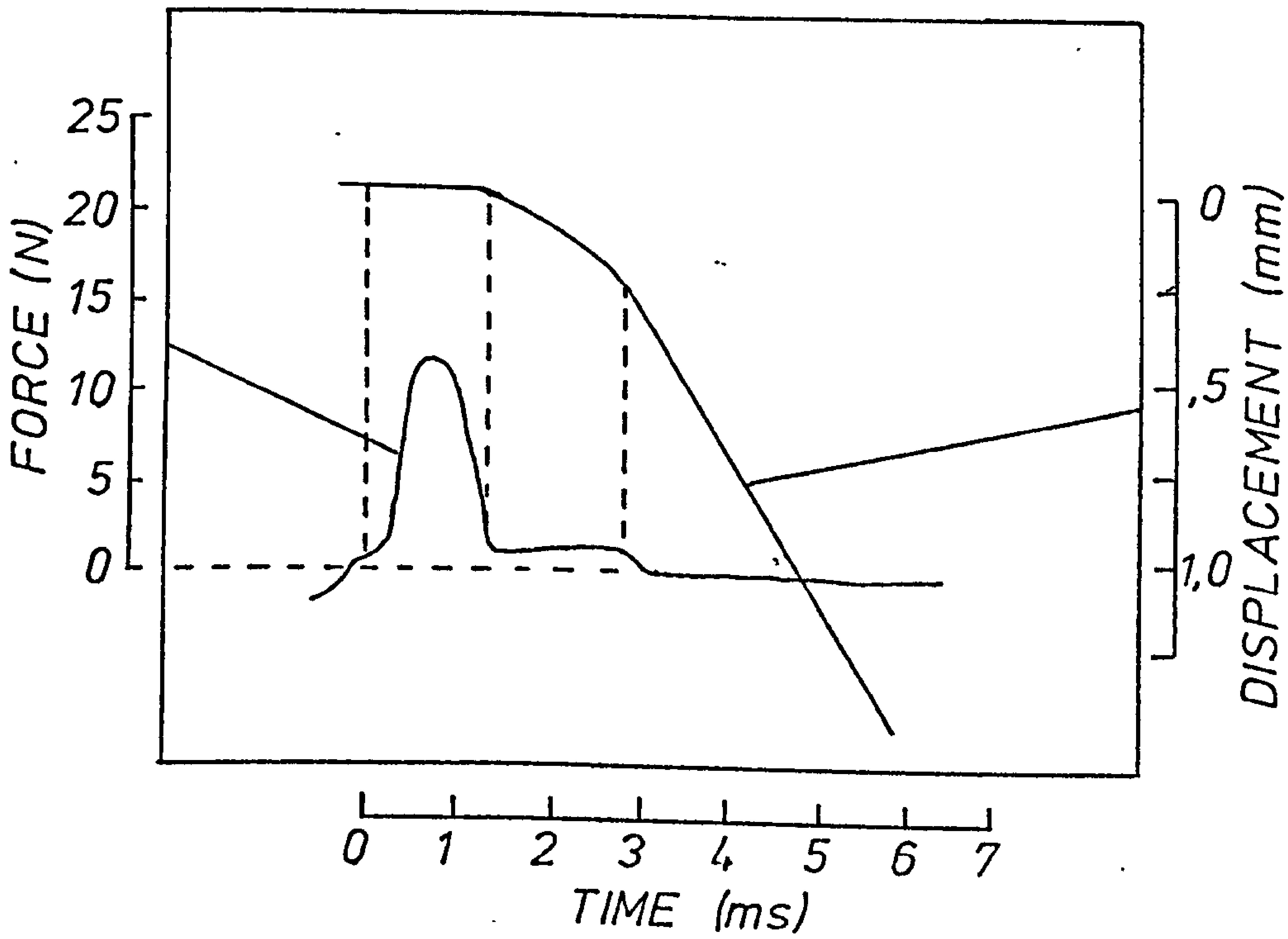
4.5 Force Pull-Off Results

4.5.1 General

A typical oscillogram from the pull-off apparatus is shown in Fig. 4.5.1. It shows a sharp rise in the oil tension when a force is applied to the valve disc, which then remains constant for a certain time. During this time, any motion of the valve disc is imperceptible. After this, the tensile force holding the disc falls to a much smaller value accompanied by a fairly rapid motion of the valve disc. The force then falls to zero indicating collapse of the film and the disc moves away at a much increased velocity.

It was found that the greater the tensile force in the oil caused by the increased pull-off rate, the shorter was the time the oil could sustain the peak force. This only applied when tests on similar seat profiles were compared. Different seats gave different times for a similar tensile force. However, if this time was plotted against minimum pressure generated in the film, there was found to be very little difference in the curves obtained from tests using different seats. The minimum pressure was found by plotting a curve of the oil pressure using a modified program and assuming a non-cavitating continuous film, as explained later in section 4.5.2. It was concluded therefore, that the time to film collapse was a function of the minimum pressure in the oil film.

The peak force was also observed to be dependent on the viscosity/



TYPICAL OSCILLOGRAM

FIG. 4.5.1

viscosity of the oil. The greater the viscosity, the greater was the peak tensile force.

Further analysis showed that the peak force had an effect on the rate at which the valve began to leave its seat prior to rupture. The greater the peak force, the greater was the velocity after the peak force collapsed.

These were the general observations made from a very large number of tests. Other points such as the dependence of a good surface finish on the time for the film to rupture corroborated very well with similar observations made on the compressor equipment.

It was decided to determine coefficients from these test results which could be incorporated into the theoretical model of a valve operating in a compressor.

4.5.2 Time Delay

By varying the pull-off rate, oscillograms could be produced which were of the form shown in Fig. 4.5.2. The effect of varying the rate merely varies the initial slope of the oscillogram.

Initially, the oil behaves in a purely viscous manner as if it were free from any dissolved gas (0 - D). At this time, small vapour bubbles in the oil rapidly expand in an attempt to regain an equilibrium condition and the force falls from the peak value of D to the lower value of E. It then holds this force steady for a finite time (E - B) and the force then collapses completely indicating that the film has ruptured (C).

The increase in oil film thickness, indicated by the displacement trace, is negligible for the time when the force is increasing. However, when the force falls to its lower value, the disc/

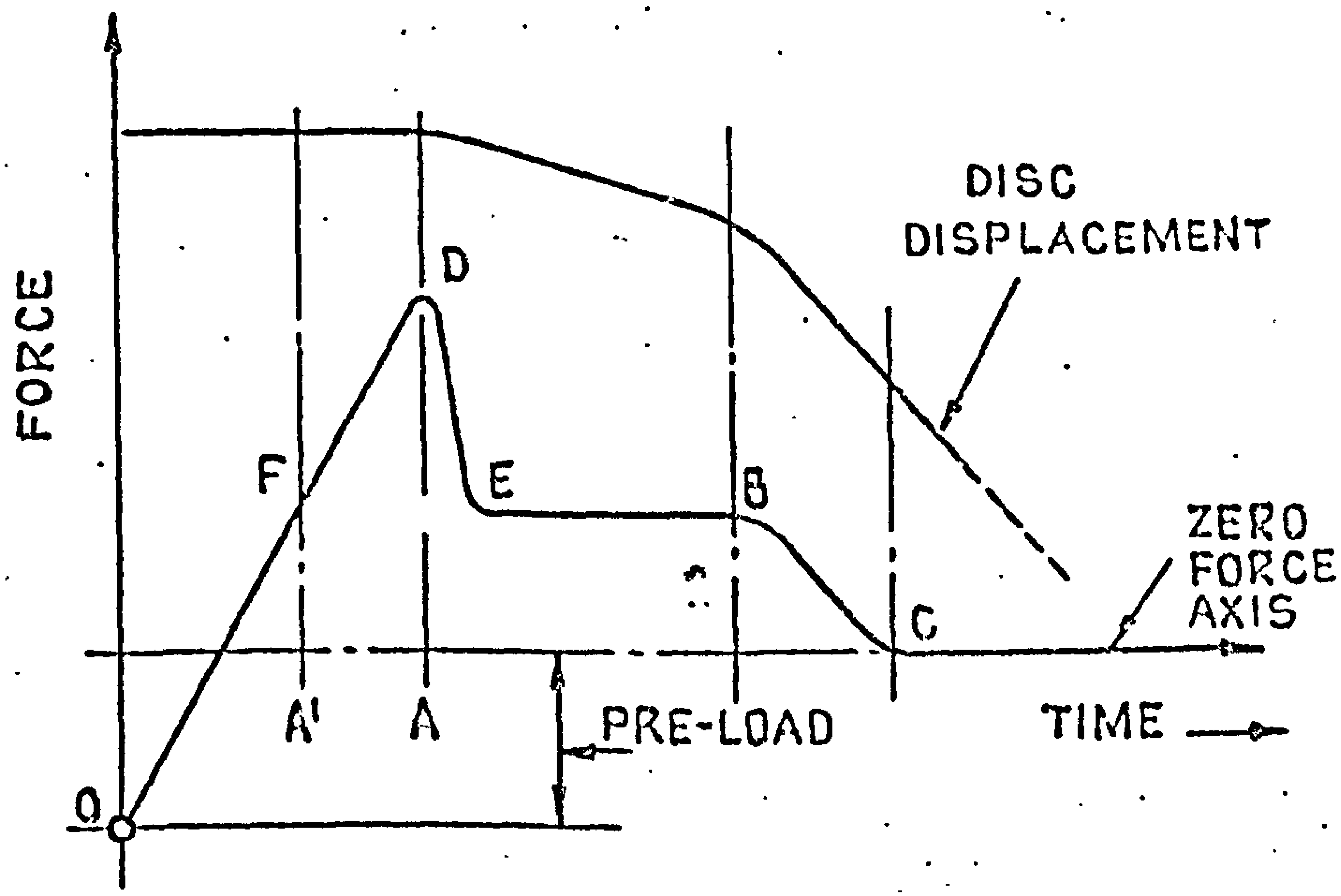

FORCE
DISC
DISPLACEMENT
D
F
E
B
C
ZERO
FORCE
AXIS
A'
A
PRE-LOAD
TIME
O

FIG. 4.5.2

disc moves rapidly from its seat until the film ruptures.

This would appear to agree with cavitation theory in which there is a time delay in the formation of bubbles during which time the oil can withstand large negative pressures. As gas is being evolved, the disc will tend to move away from its seat causing the oil film to "neck" resulting in a reduced force. When the oil film becomes too narrow to retain the disc, it will rupture.

4.5.3 Gas Evolution Pressure

When a tensile force is applied to the oil film between the valve disc and its seat, the pressure in the oil falls below the ambient pressure and gas is evolved. A small series of tests was carried out in which very low forces were applied to the oil film in order to determine if there was a minimum tension which could be applied without gas being evolved. The criterion used was that if the oil behaved in a viscous manner, i.e. took a time measured in seconds rather than milliseconds to rupture, then it would be assumed that no gas had been evolved.

During these tests, extremely small forces were applied - gradually being increased - during which time the oil film took a relatively long time to rupture. A force was reached where just a slight increase caused the film to rupture quickly. This applied force was still very small and, when the minimum pressure was calculated using the modified direct pull-off program, it was found to be of the order of 20 - 30 kN/m^2.

4.5.4 Determination of Time Delay

The general form of the pull-off force or negative oil film pressure v time graph is shown in Fig. 4.5.3.

The/

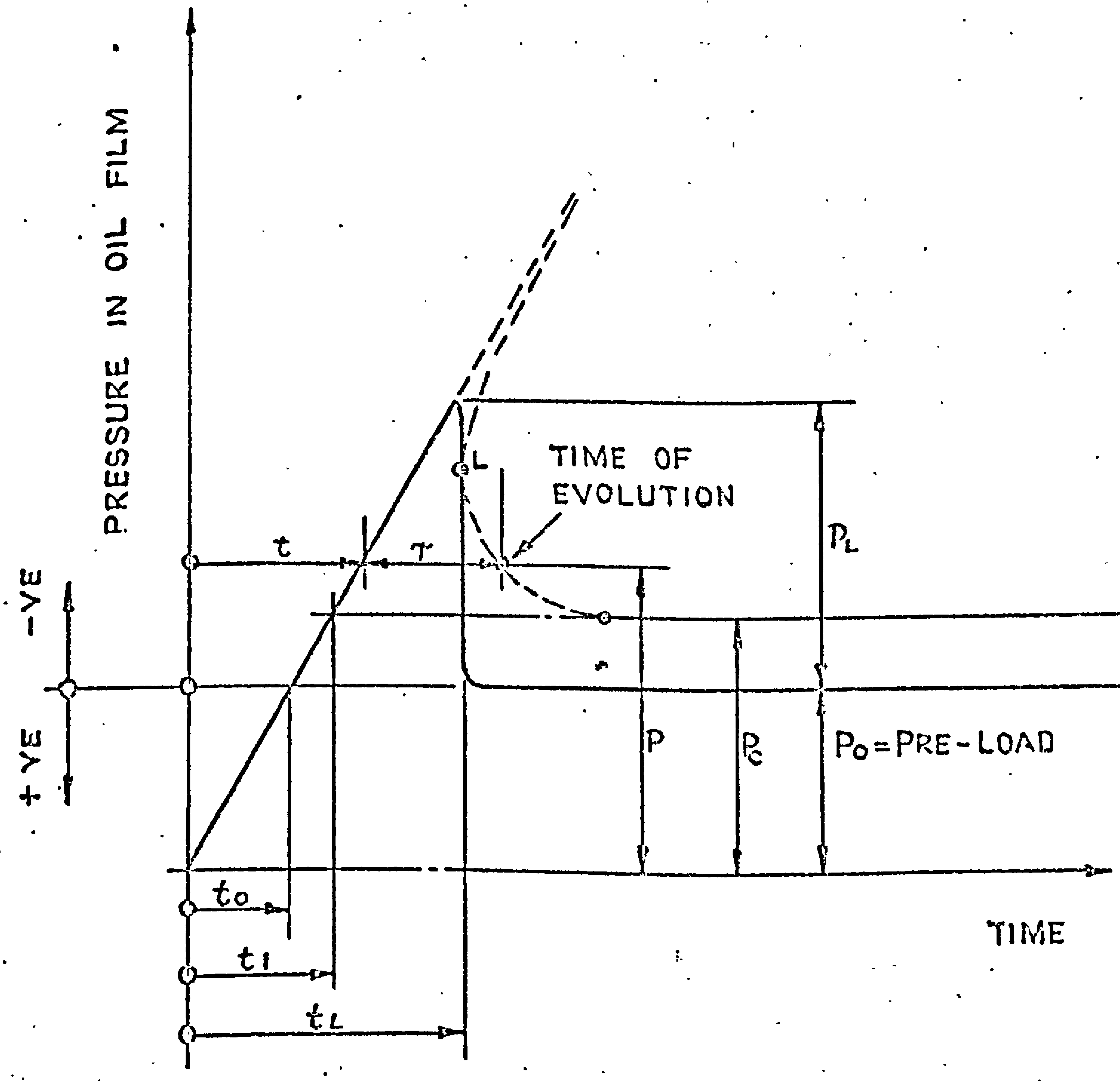

PRESSURE IN OIL FILM
-VE
+VE
t
τ
L
TIME OF
EVOLUTION
P_L
P
P_c
P_o=PRE-LOAD
t_o
t_1
t_L
TIME

FIG. 4.5.3

The pressure P_c is the minimum pressure determined experimentally at which a constant force could be applied for a lengthy time over which the film behaves viscously before rupture.

When the applied force produces a lower pressure than this, gas or vapour is evolved causing rapid collapse of the film. The growth of the bubble fromthe metastable state existing in the film proceeds rapidly after attainment of a critical dimension and could be looked upon as a simple "time delay" as shown diagrammatically in Fig. 4.5.4.

Referring to Fig. 4.5.3, an initial positive load or pressure is removed at a steady rate. At time t_o the load and hence pressure are zero. After time t_c bubbles grow until t_L when the film collapses.

If, after time t, the force is held constant, the time to collapse will obviously be greater, as the time delay was found to be inversely proportional to the applied force. The time to collapse will be time τ, shown by the dotted line.

At any time t,

$$\tau = \frac{K}{(P - P_c)} \quad \text{where K is a constant.}$$

$$P = \frac{dp}{dt} \times t \qquad \text{where } \frac{dp}{dt} = \text{rate of change of pressure}$$

$$\therefore \quad \tau = \frac{K}{\frac{dp}{dt} \times t - P_c}$$

Rapid evolution of gas occurs at time $t_e = (t + \tau)$.

As the pressure rises, this evolution occurs along the dotted curve. The limit is arrived at when the first rapidly-expanding bubbles disturb the metastable state.

If/

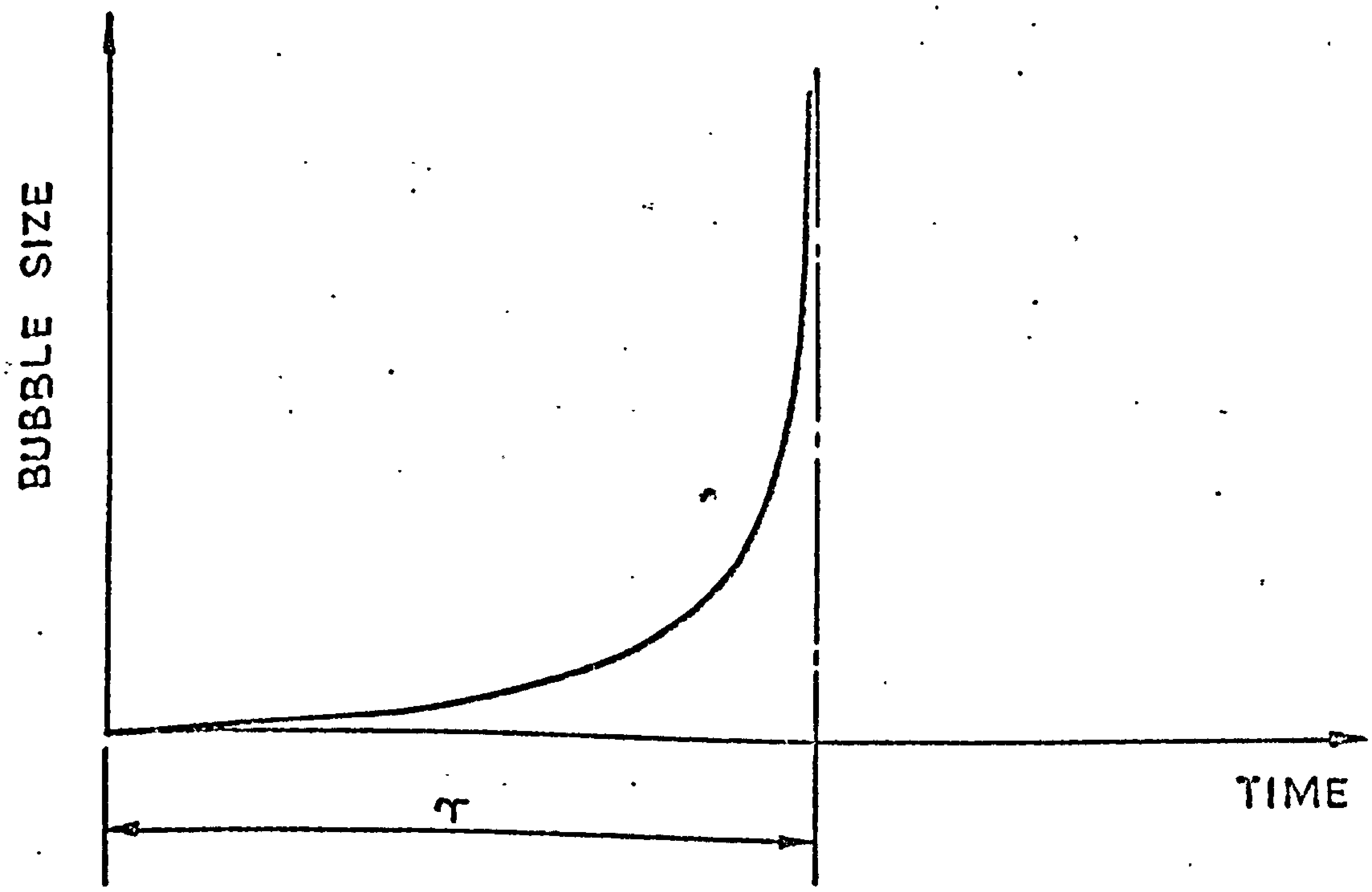


FIG. 4.5.4

If the limit occurs at t_L, t_L may be determined from

$$\frac{d}{dp}\left[t_e\right] = 0$$

i.e.
$$\frac{d}{dp}\left[\frac{P}{\frac{dp}{dt}} + \frac{K}{P - P_c}\right] = 0$$

Hence
$$\frac{dp}{dt} = \frac{\left[P - P_c\right]^2}{K}$$

and
$$t_L = \frac{P_c}{\frac{dp}{dt}} + \sqrt{\frac{K}{\frac{dp}{dt}}}$$

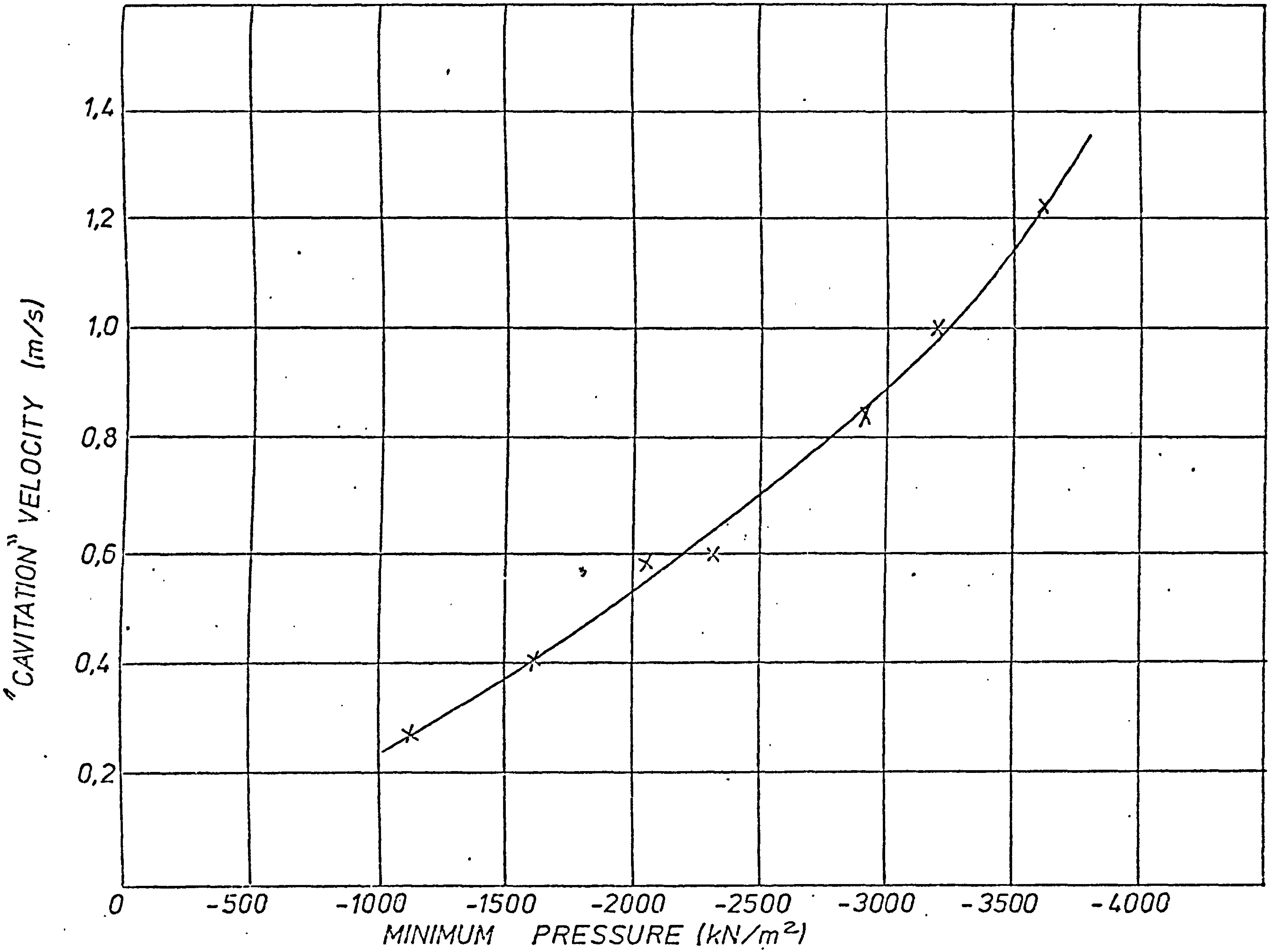
If the time is measured from P_c, a time for evolution is obtained which is inversely proportional to $\sqrt{\frac{dp}{dt}}$. Hence the peak pressure (as measured from P_c) is proportional to $\frac{dp}{dt}$.

4.5.5 Evolution Constant

As stated in Chapter 2, some method had to be devised to incorporate in the theoretical model the effects of gas evolution.

From the force pull-off tests, it was observed that, after the previously explained time delay, the simulated valve left its seat at a velocity which was somewhat less than the velocity after it had broken clear from the film. The velocity at which the valve left its seat after gas evolution is shown in Fig. 4.5.5 plotted against minimum pressure in the oil film. It shows that the velocity after evolution is proportional to the negative pressure in the oil film. Various values for an evolution constant were incorporated in the force pull-off program until one was found which gave a good approximation to the rate at which the simulated reed was leaving its seat before rupture.

Unfortunately, it was not possible to incorporate this value directly/

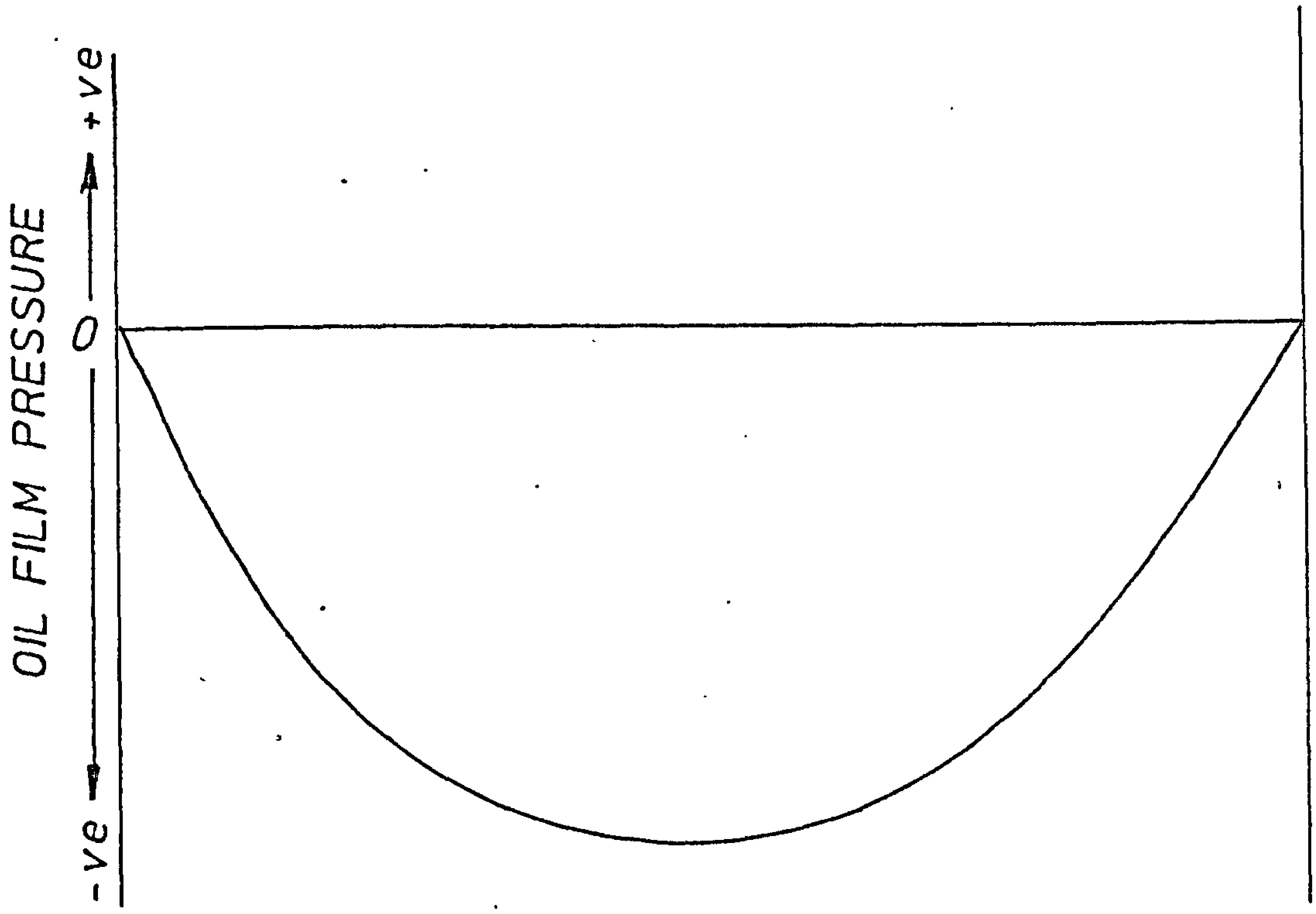


VARIATION OF "CAVITATION" VELOCITY WITH MINIMUM PRESSURE IN OIL FILM FIG. 4.5.5

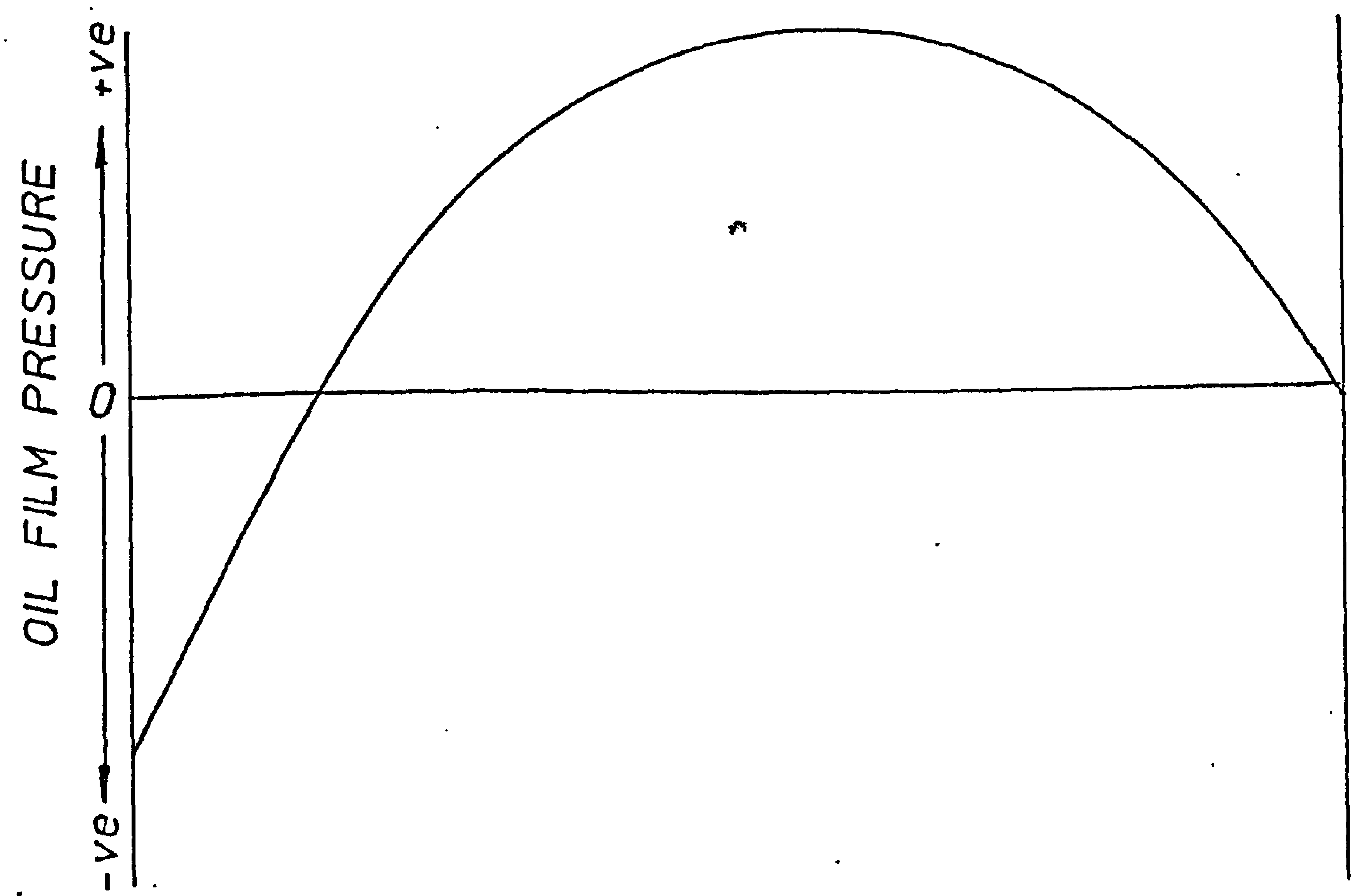
directly into the compressor program. The reason being that in the force pull-off equipment, the pressure profile over the valve seat is as shown in Fig. 4.5.6(a), whereas in the compressor, the pressure profile over the valve seat just before "cavitation" is as shown in Fig. 4.5.6(b). This, as explained previously, is due to the straightening out of the valve disc as the pressure difference decreases.

It can be seen in Fig. (b) that as the film at the inside "cavitates", a high pressure is maintained towards the outside edge of the seat. The parabolic shape of Fig. (a) will have a much greater area available for "cavitation" than for the case when the valve is flexing.

It may be inferred then that for the case of a flexible disc, evolution is confined in the early stages to a narrow strip near the inside edge of the seat. The evolution constant developed from the Direct Force-Pull experiments, must therefore be modified to allow for this reduced activity. To avoid unnecessary complication, a reduced evolution constant covering the full width of the seat was assumed. This reduced evolution constant was derived by proportion from the width of the seat initiating "cavitation". It was also made a function of the minimum pressure in the film as this was found to have an effect on the rate at which the disc left its seat.



(a) RIGID DISC



(b) FLEXIBLE DISC

COMPARISON OF PRESSURE PROFILES ACROSS OIL FILM IMMEDIATELY PRIOR TO "CAVITATION"

FIG. 4.5.6

CHAPTER 5

MATHEMATICAL MODEL

5.1 Introduction

As explained previously, a finite difference technique was used to develop a computer program to describe the processes involved in the initial stages of the valve leaving its seat. The parameters included in the theoretical model are:

a) The valve disc and seat dimensions.

b) The surface finishes of the valve disc and seat.

c) Oil viscosity.

d) Surface tension at the interface between oil and vapour or gas.

e) Rate of change of pressure across the valve.

f) Compressor suction and discharge pressures.

g) Gas or vapour evolution pressure.

h) Gas or vapour evolution delay time.

i) Rate of gas evolution.

As shown earlier in the thesis, many of these constants were determined experimentally. The need for determining them became apparent when discrepancies were observed between experimental results - from the compressor apparatus - and theoretical results - from the initial simulation model. In this way, the experimental programme influenced the development of the theoretical model, and the theoretical model, in turn, influenced the experimental programme - in the shape of the Force Pull-Off apparatus.

In order to simplify the working of the program, it is split into linked files. Flow charts of these files are shown in Figs. 5.1.1 - 3 and they broadly cover the following processes.

File 1: a) The input of the values of the above constants;

b) The determination of the point where the valve reed just begins to leave its seat (section 2.4).

File 2:/

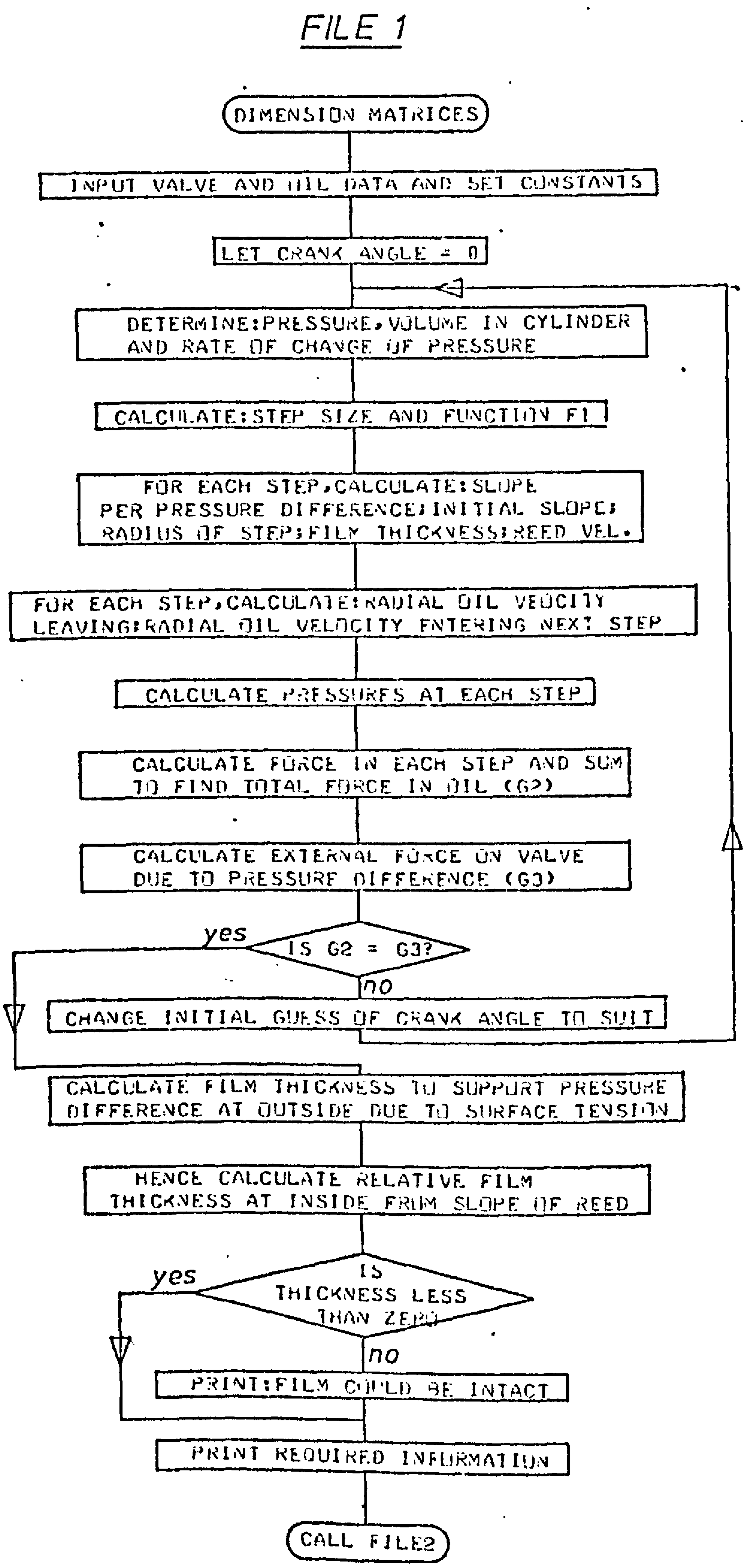

FILE 1
DIMENSION MATRICES
INPUT VALVE AND OIL DATA AND SET CONSTANTS
LET CRANK ANGLE = 0
DETERMINE:PRESSURE, VOLUME IN CYLINDER AND RATE OF CHANGE OF PRESSURE
CALCULATE:STEP SIZE AND FUNCTION F1
FOR EACH STEP, CALCULATE: SLOPE PER PRESSURE DIFFERENCE; INITIAL SLOPE; RADIUS OF STEP; FILM THICKNESS; REED VEL.
FOR EACH STEP, CALCULATE: RADIAL OIL VEOCITY LEAVING; RADIAL OIL VELOCITY ENTERING NEXT STEP
CALCULATE PRESSURES AT EACH STEP
CALCULATE FORCE IN EACH STEP AND SUM TO FIND TOTAL FORCE IN OIL (G2)
CALCULATE EXTERNAL FORCE ON VALVE DUE TO PRESSURE DIFFERENCE (G3)
yes
IS G2 = G3?
no
CHANGE INITIAL GUESS OF CRANK ANGLE TO SUIT
CALCULATE FILM THICKNESS TO SUPPORT PRESSURE DIFFERENCE AT OUTSIDE DUE TO SURFACE TENSION
HENCE CALCULATE RELATIVE FILM THICKNESS AT INSIDE FROM SLOPE OF REED
yes
IS THICKNESS LESS THAN ZERO
no
PRINT:FILM COULD BE INTACT
PRINT REQUIRED INFORMATION
CALL FILE2

FIG. 5.1.1

FILE 2(1)

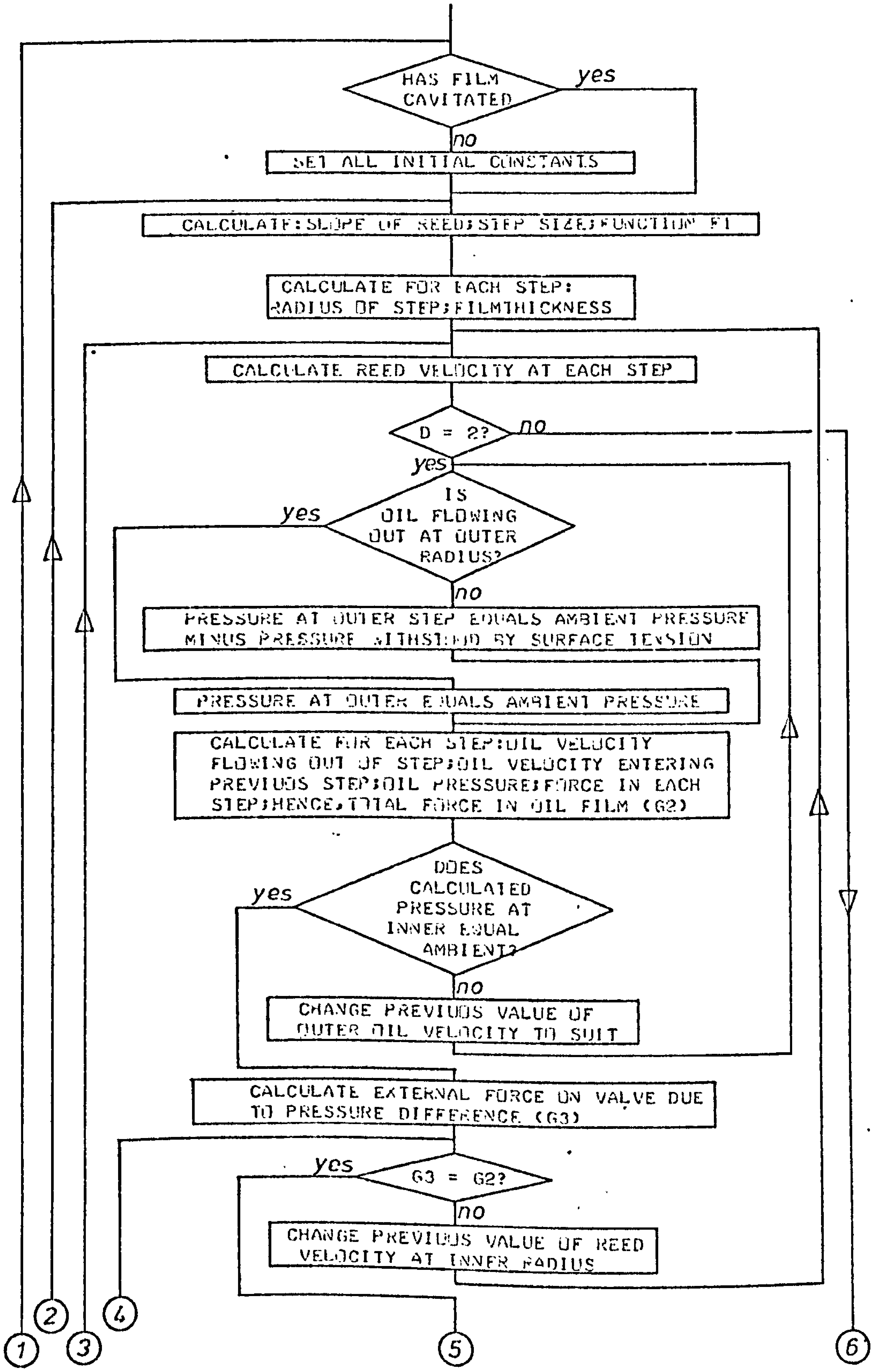


FIG. 5.1.2(1)

FILE 2(2)

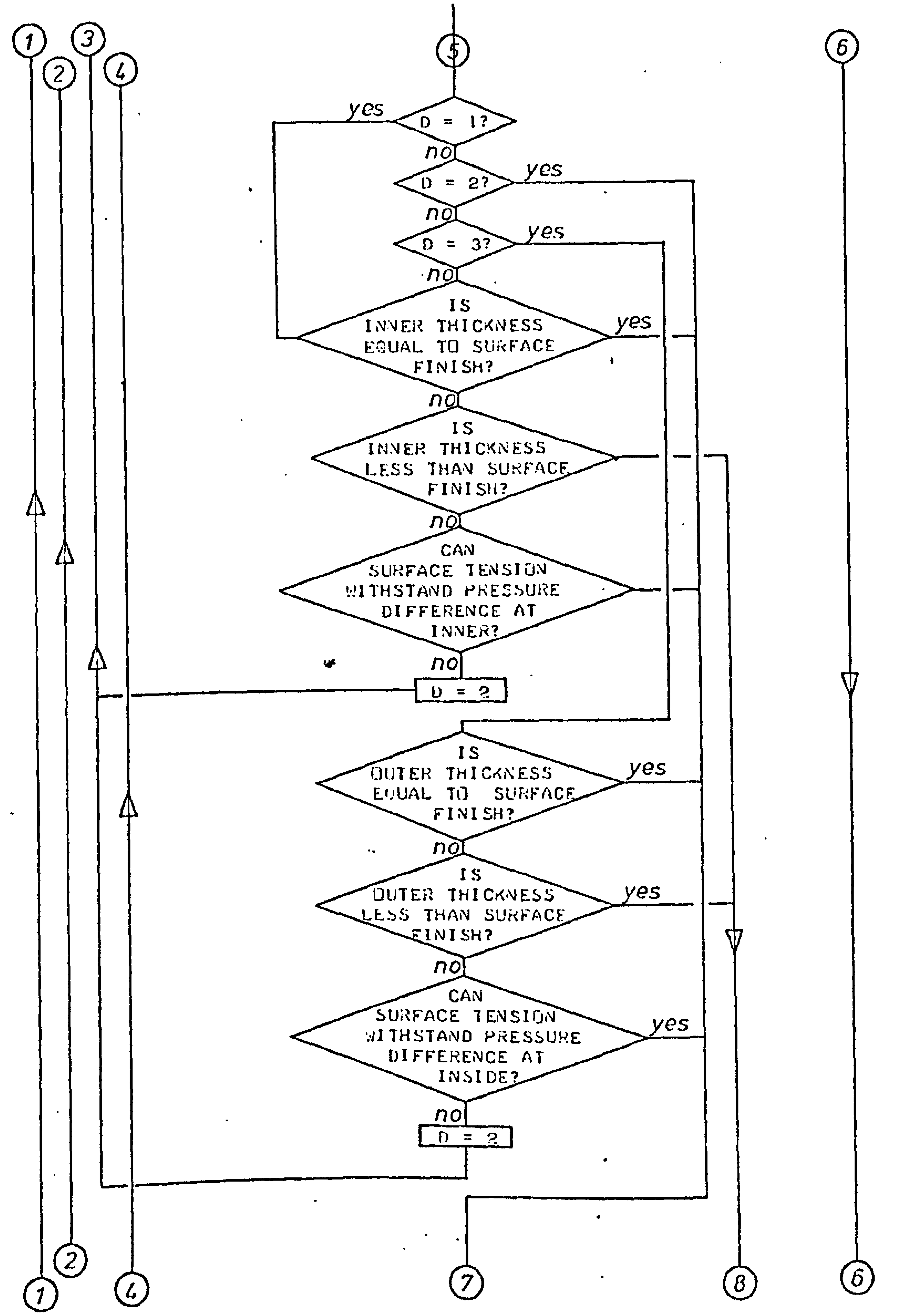


FIG. 5.1.2(2)

FILE2 (3)

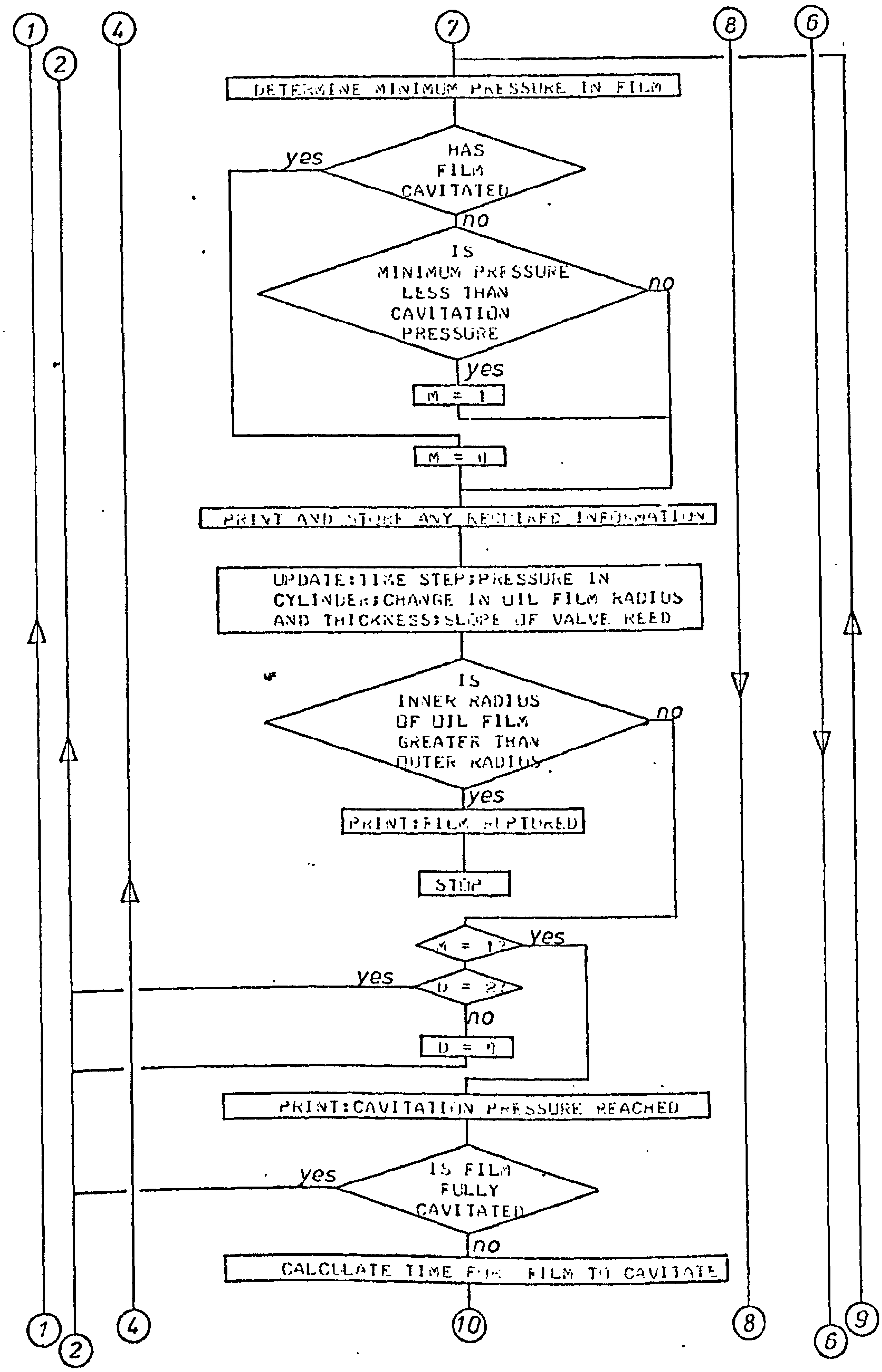


FIG. 5.1.2(3)

FILE2(4)

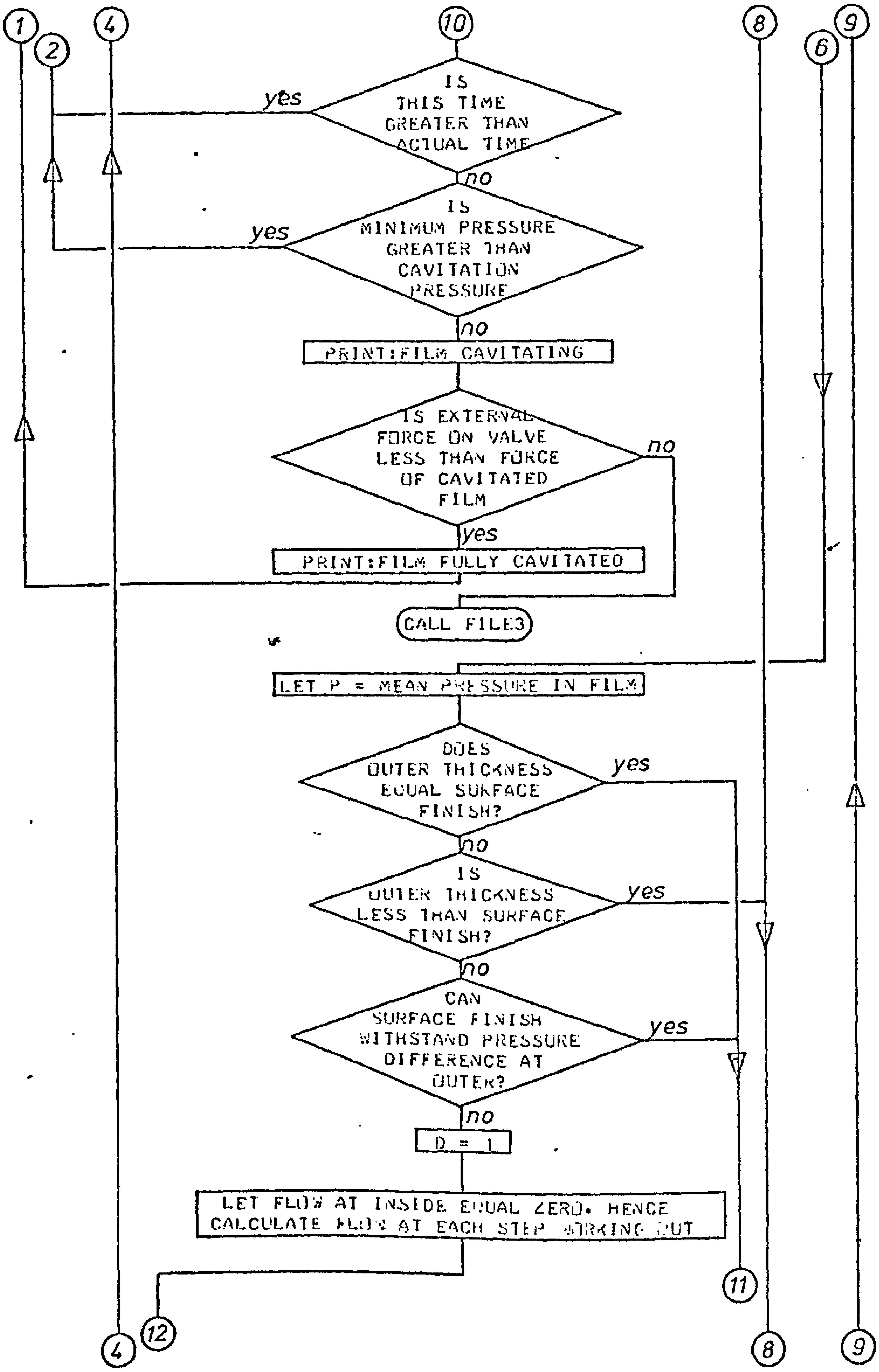


FIG. 5.1.2(4)

FILE 2(5)

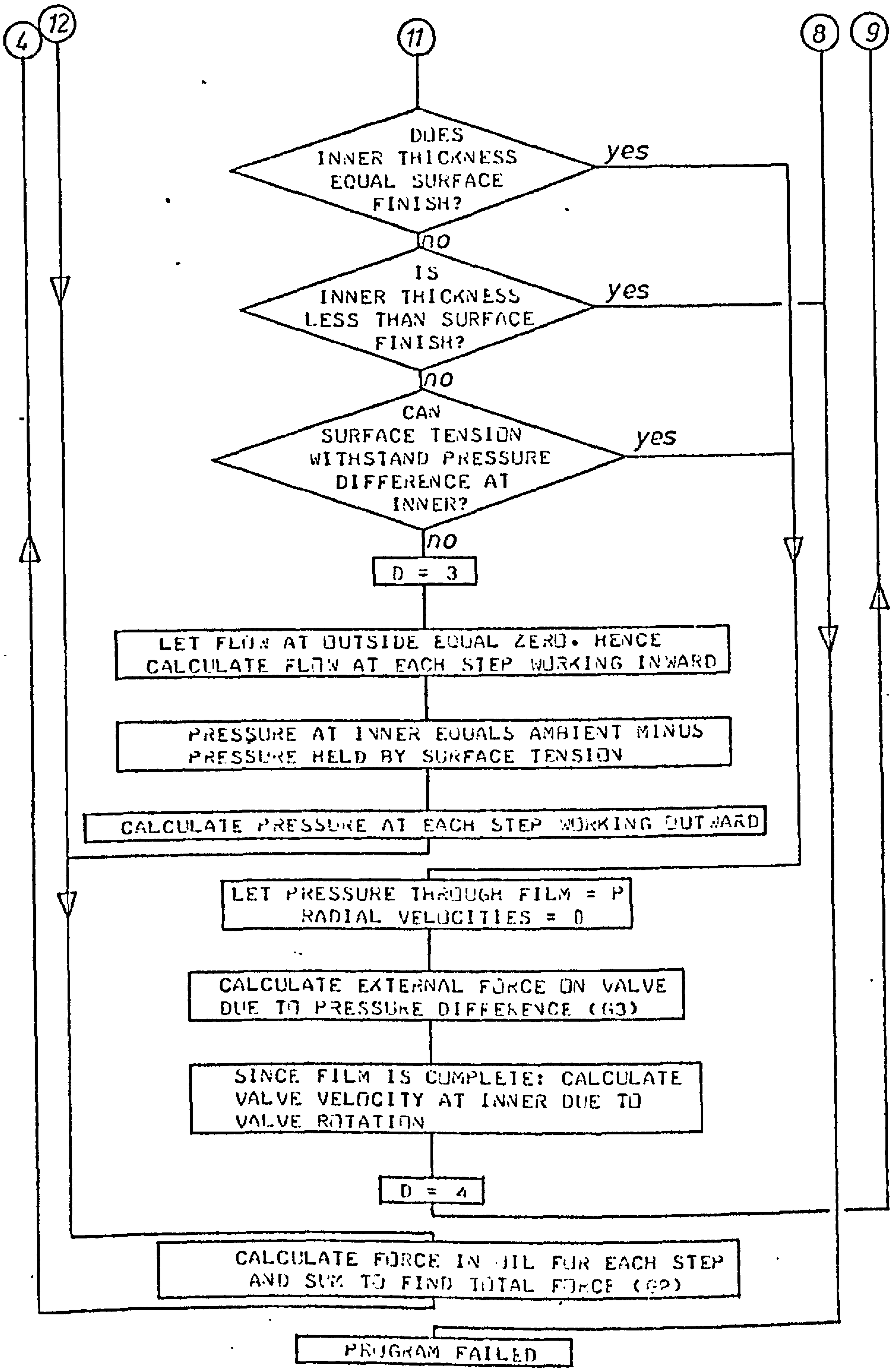


FIG. 5.1.2(5)

FILE 3

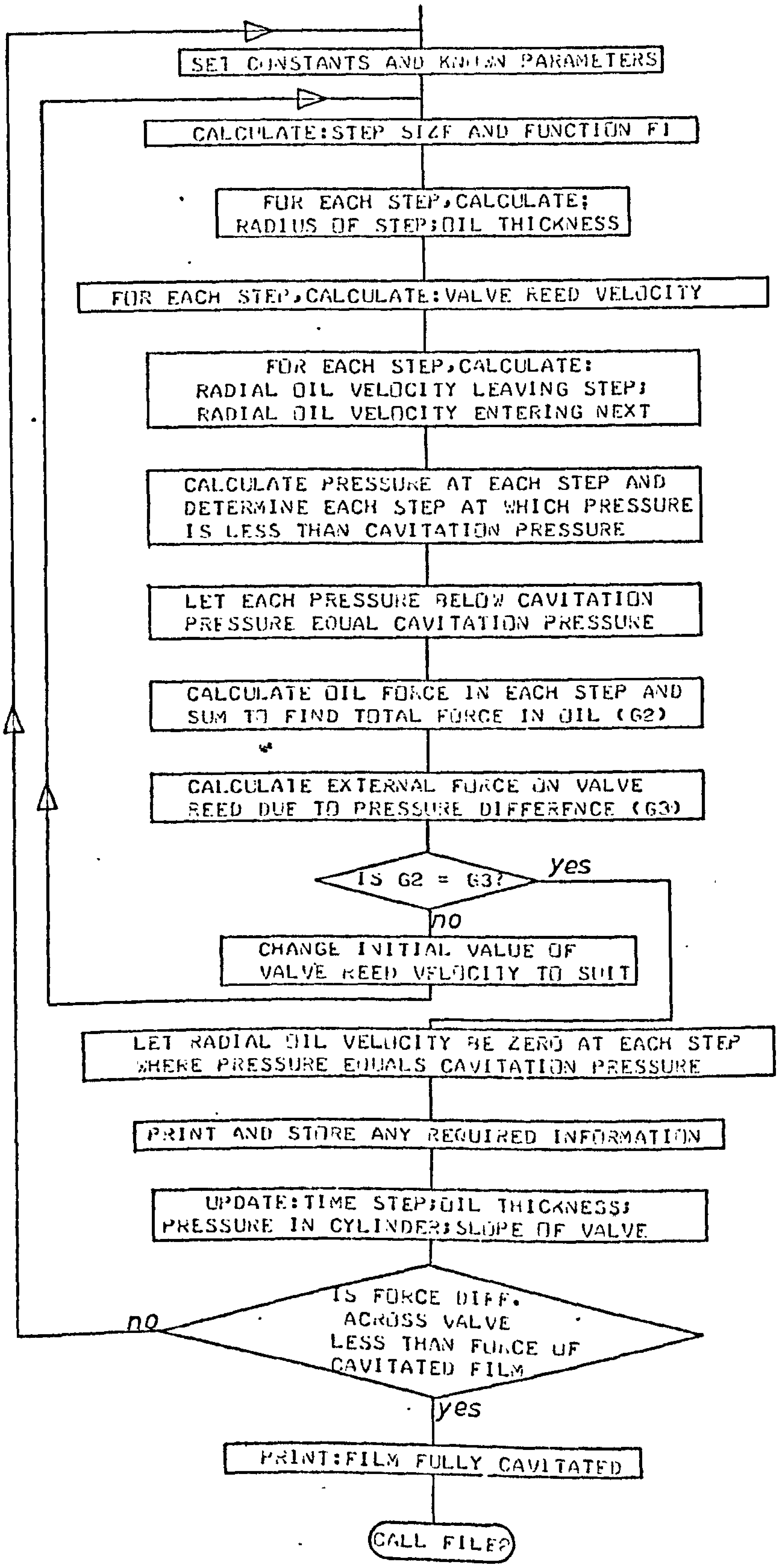


FIG. 5.1.3 .

File 2: a) The simulation of the valve operation while the reed moves with the oil acting in a viscous manner;

b) This file also simulates the valve operation while the reed moves with the oil acting in a viscous manner after the film has "cavitated" and gas or vapour is being evolved (section 2.8.1).

File 3: The simulation of the valve operation while the reed moves with gas or vapour being evolved, but before the evolution process has spread over the entire film. In the main, this process is confined to the period before pressure equilibrium occurs across the valve.

The computer model can handle either a suction valve or a discharge valve. However, again for clarity, only the part of the program concerning a discharge valve is shown.

5.2 The Valve Disc and Seat Dimensions

The processes described in the flow chart depend on the flexibility of the disc or reed. It is for this reason that the difference between a flexible and a rigid disc must be explained. Although the difference is purely comparative, a flexible disc must be very thin in comparison with its diameter whereas a rigid disc will be somewhat thicker. In practice, a reed or cantilever valve will almost certainly be flexible in order to allow it to function properly.

From the point of view of describing the processes involved, the criterion for a flexible disc is one in which the oil "cavitates" before pressure balance and hence incorporates File 3. Because a rigid disc does not initially deflect into the valve port and/

and therefore does not begin to evolve gas or vapour until after pressure balance it does not incorporate the process allowed for in File 3. The processes described by the theoretical model can be seen in Figs. 5.2.1 and 5.2.2. These diagrammatic interpretations will be referred to in the following descriptions.

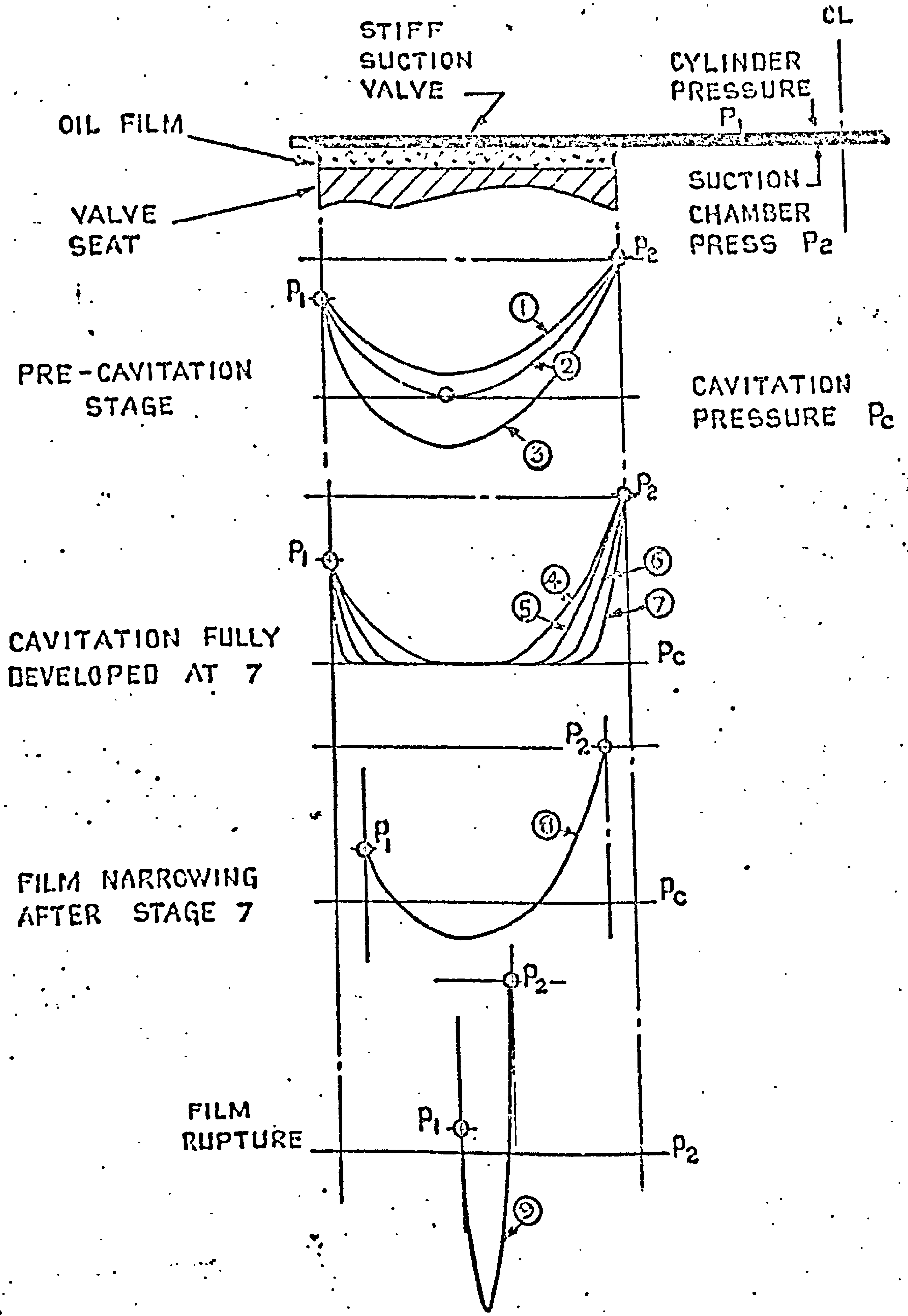
5.2.1 Rigid Disc

Fig. 5.2.1 shows, diagrammatically, the different stages when the valve is essentially rigid.

Initially, there is a positive pressure distribution across the seat - because of the greater pressure on top of the valve reed. This positive pressure distribution decreases as the pressure underneath the reed increases. After pressure equalisation, the pressure in the oil begins to fall to achieve balance. This part of the process is shown by stages 1 - 3, i.e. the "pre-cavitation stage". It should be stressed here once more, that the term "cavitation pressure" is used very loosely. What is really implied is the evolution pressure - that is, the pressure at which gas or vapour is evolved.

When the pressure falls below the cavitation pressure, P_c, gas is not evolved instantaneously. This is due to the time delay in bubble growth, previously described in section 2.8.2. After this time delay, the size of bubbles coming out of solution becomes significant. The pressure in the oil film therefore tends towards the evolution pressure. Because of the ever increasing gas pressure difference across the valve, the evolution pressure spreads quickly across the oil film in order to achieve a force balance. This is shown by stages 4 - 7.

Immediately/



DEVELOPMENT OF PRESSURE DISTRIBUTION IN OIL FILM FOR A RIGID DISC.

FIG. 5.2.1

Immediately after the film has cavitated, the computer "checks" to see if the film has completely cavitated. It does this by comparing the external force difference across the valve, due to the pressure imbalance, with the negative force of a completely cavitated oil film. If the external force is equal to or greater than the internal force, the oil film is considered to be completely cavitated. For a rigid disc, this is normally found to be the case and so stages 4 - 7 can be considered to occur almost instantaneously.

When the film is evolving gas or vapour, the valve accelerates off its seat with an accompanying reduction in the width of the oil film. The rate at which the valve leaves its seat, after complete cavitation of the oil film, is so great that the time step used in updating the varying parameters in the model is reduced by a factor of five. This narrowing of the width of the oil film, stage 8, results in the pressure in the oil film again falling in order to achieve force balance.

Rupture of the film occurs soon after this event. The film is considered to be ruptured when the inside radius of the oil film is greater than or equal to the outside radius.

5.2.2 Flexible Disc

When the valve reed is flexible, the processes just described are modified by the strain energy dissipated by the reed.

The reduction of the pressure difference across the valve in its closed position causes the reed to straighten out from its previously deflected shape. This has the effect of forcing oil from the outer edge of the valve. As previously mentioned in Chapter 2, the resulting pressure distribution in the oil film causes the/

the valve to lift from its seat before pressure equalisation. The lift of the valve is sufficient to generate a negative pressure near the inner edge of the seat as shown diagrammatically in Fig. 5.2.2. Stages 1 and 2 show how the pressure at the inner edge falls below the ambient but does not "cavitate". Once again this is due to the time delay in the evolution of gas. Because the oil film at the inside edge is very thin (the valve reed can in fact be touching the seat at the inner edge), surface tension effects are significant and the oil flow at the inside is zero. However, even if the surface tension is ignored, a negative pressure will still be produced near the inner edge as indicated by the dotted line of stage 1. A small oil flow will therefore also occur at the inner edge as well as the outer edge.

After the time delay in the evolution of gas, the negative pressure towards the inside edge will fall (algebraically) to the evolution pressure (stage 3). At this stage, however, pressure balance across the valve will still not be reached. It is at this point that the mathematical model incorporates File 3. File 3 calculates the rate at which the evolution pressure spreads across the width of the film (stages 3 - 5). The reduction of the positive pressure to the evolution pressure across the film is caused by the pressure below the valve increasing and causing it to lift and straighten out simultaneously. By the time pressure balance is reached, the evolution pressure will have spread over almost the complete width of the film. When this happens, the film is considered to be "fully cavitated" and the model then reverts to File 2 again and acts in a similar manner to a rigid disc. Like the rigid disc, once the film is fully cavitated, rapid valve movement/

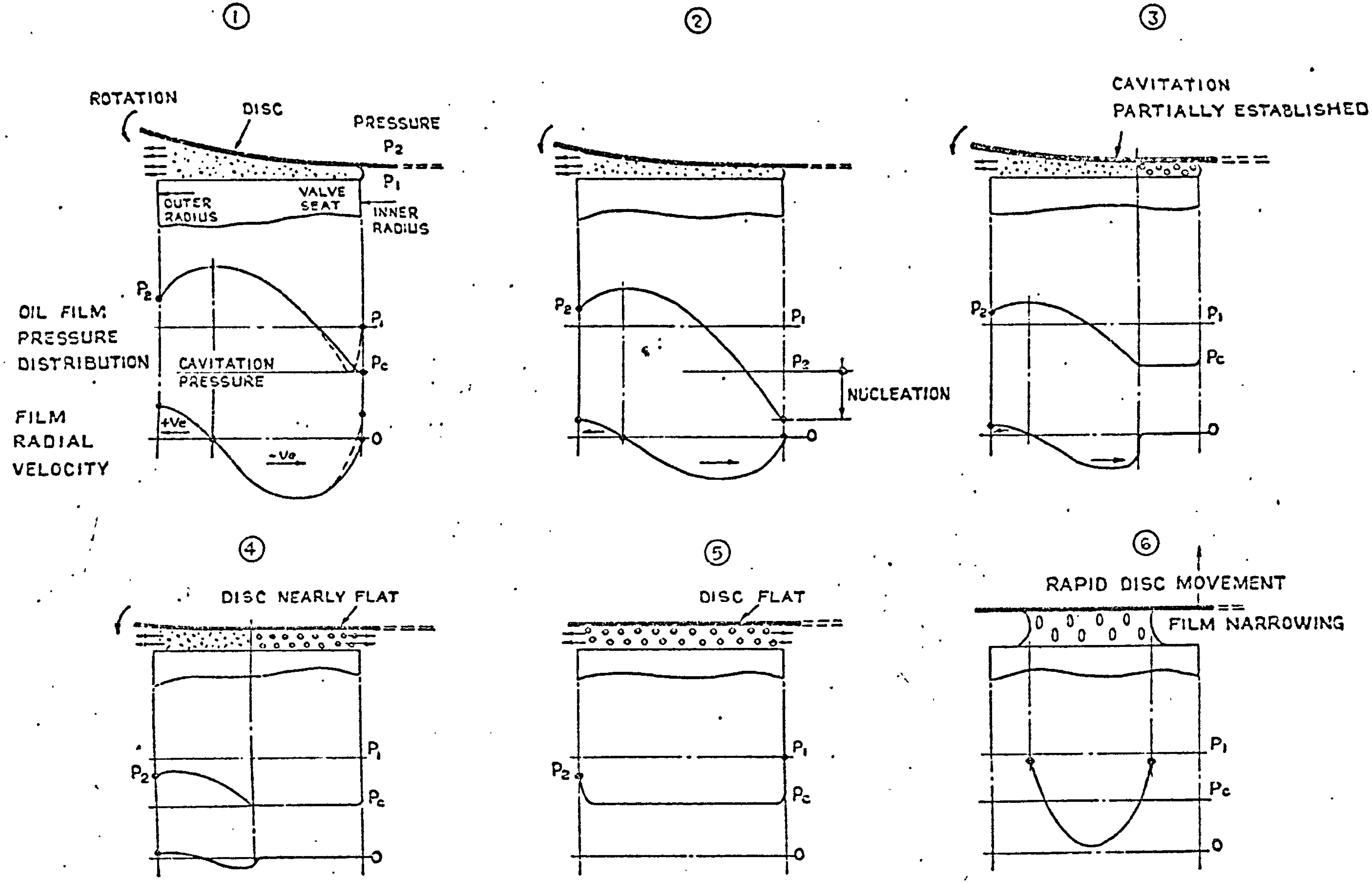


FIG. 5.2.2 CAVITATION PROCESS FOR A FLEXIBLE DISC.

movement takes place with subsequent early rupture of the film.

5.2.3 Comparatively Stiff Disc

If the disc is comparatively stiff, the processes involved have been found to be almost a mixture of the two previously described types.

For this type of disc, it is possible that the minimum pressure generated in the oil film as the disc straightens out may reach the evolution pressure. However, because of the time delay involved, evolution might not take place at this stage. There are two possible reasons for this: (1) Because the time delay is a function of pressure difference, the minimum pressure in the oil film may not be significantly enough below the evolution pressure to result in a short delay time; (2) The time taken for the minimum pressure in the oil film to fall to the evolution pressure may be so long that the pressure below the valve may almost have reached the balance point. In this case, once the time delay has been incorporated, pressure balance will have been accomplished and the computer will not call File 3.

In either of the two events above, evolution will be delayed until the pressure in the film again falls after pressure equalisation across the valve. Such valves therefore behave in a similar manner to valves fitted with rigid discs except that the comparatively stiff disc has a somewhat thicker oil film due to its movement in the early stages of the process.

Because of this thicker film, the time lag for the valve to open will be shorter than for the case of a rigid disc.

5.2.4 Seat Dimensions/

5.2.4 Seat Dimensions

If the width of a valve seat is small compared to its port diameter, the slope of the reed may be taken as linear over the seat width. However, inaccuracies will occur when calculating the film thickness if the seat width is large compared to its diameter. This effect and also the method of determining the film thickness will be explained more fully in Appendix B.6.

In simple terms, however, what happens is this. If the valve slope is considered to be linear for a wide seat, the calculated oil thickness is assumed to be much thicker towards the outside than it is in practice. This is shown in Fig. B.6.2. Again, because of the dependence of the time lag on the oil film thickness, the predicted time lag will be smaller than the practical case.

A wide seat will also exhibit a greater time lag than a narrow seat. This results from the extra pressure required below the valve to accomplish a force balance. However, the time lag is not increased pro rata with the ratios of the outside and inside seat areas. As has been shown previously, gas evolution and the minimum pressure developed in the film have a considerable effect .

5.3 Seat and Disc Surface Finishes

Because of the very thin nature of the oil film between the valve and seat, it is clear that the surface finish of both valve and seat is going to have a not inconsiderable effect on the oil thickness. The method described previously of taking the mean CLA value of the two surface finishes may not be an ideal method. However, in most practical cases in working compressors, both the surfaces/

surfaces of the valve reed and its seat have a very good finish, i.e. low CLA value, so any effect on the oil thickness is small.

The values which were used for the theoretical model were obtained experimentally. The CLA value for the disc was found to be 4 and for the seat, approximately 15. The seat values were found to vary a little for different seats, depending on how well they had been lapped. The discs, however, were punched-out pieces of shim steel and therefore did not require lapping as they already had a good surface finish.

5.4 Viscosity

By inserting purely arbitrary values of viscosity into the mathematical model, it became apparent that the viscosity of the oil would have a large influence on the time lag for the valve to open, and hence on the pressure difference across the valve at opening. It was found that any percentage increase in viscosity had almost the same increase in time lag. (This was when evolution of gas was ignored).

Therefore, in order that realistic comparisons between theoretical and experimental results could be made, the correct value of viscosity of the working fluid had to be inserted into the program. When air was the working fluid, figures for viscosity were obtained from Shell Oil Limited, since the oil used in the compressor, Clavus 27, was a standard type for which they could produce accurate values. Since the viscosity of oil is dependent on its temperature, the temperature of the oil at each particular running speed had also to be known. As explained in Chapter 3, this was obtained from a thermocouple situated in the compressor head/

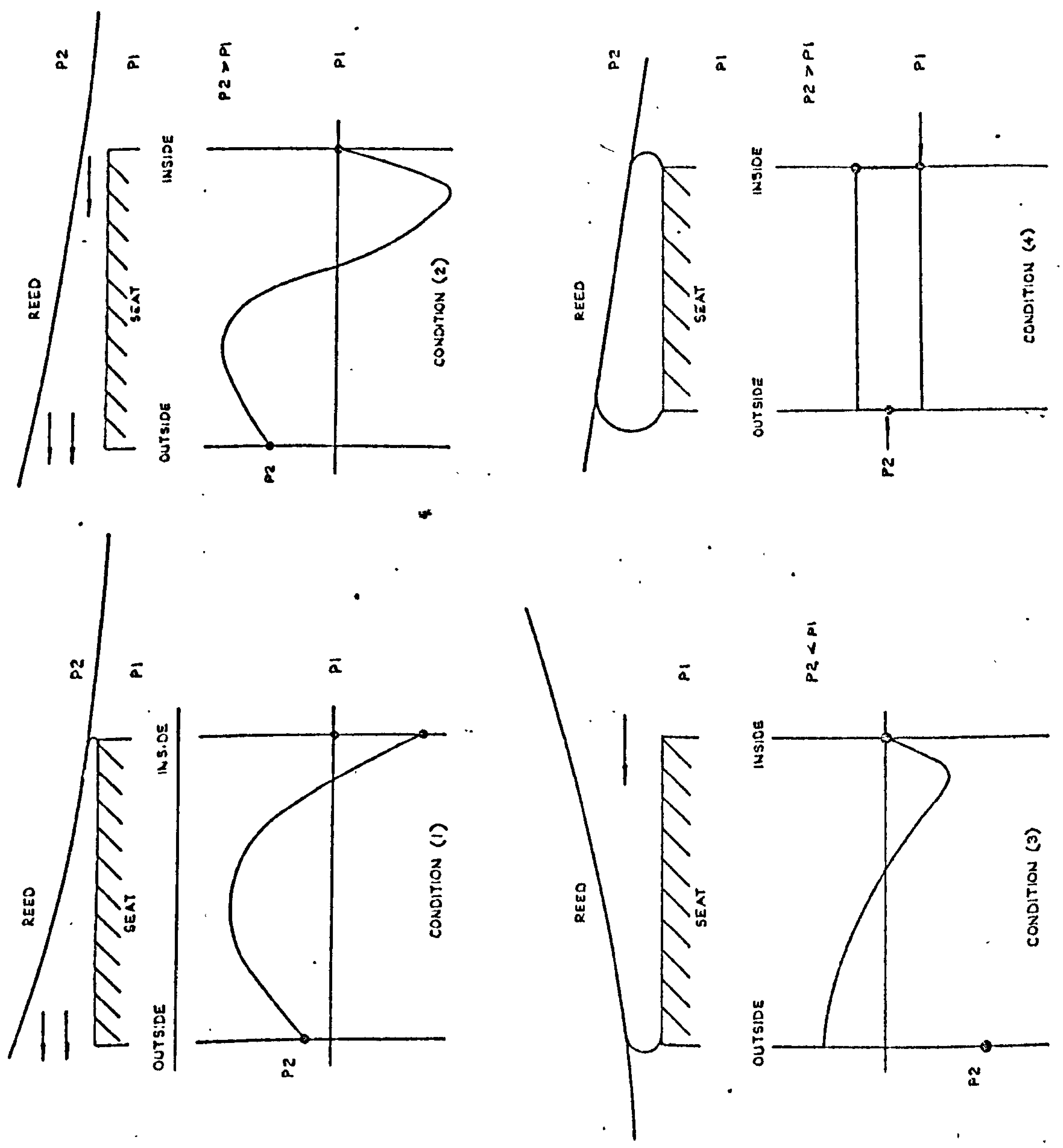
head. This temperature was found to vary very little for different running speeds in the range 500 - 1300 rev/min and was approximately 130°F. This temperature, in turn, corresponded to a viscosity of 16 centipoises.

This value had to be adjusted when comparison was made with results in which refrigerant 12 was the working fluid. Because R12 is miscible in oil and also because it has a lower viscosity, the figure used in the mathematical model, when a refrigerant/oil mixture was being used, had to be reduced.

One problem which arose was the fact that the amount of refrigerant in solution was not known. Although figures have been published which give the viscosity of refrigerant/oil mixtures at different temperatures, these figures still depend on the percentage mixture, by weight, of refrigerant being known. For the compressor apparatus used, determination of this ratio was extremely difficult - if not impossible - since any mixture removed from the sump of the compressor would have evolved refrigerant before any measurements could be made. What was done was to "estimate" the percentage mixture based on the experience of members of staff. The figure which was used was 15% which was equivalent to a value of 6 centipoises at the same running temperature. This value was obtained from the ASHRAE Guide and Data Book (3).

5.5 Surface Tension

Because of the possible effect of surface tension at the inner and outer edges of the valve seat, the four conditions which could arise had to be incorporated into the model. The four conditions are as shown in Fig. 5.5.1:



REED
SEAT
OUTSIDE
INSIDE
P2
P1
P2 > P1
P1
P2
CONDITION (2)
REED
SEAT
OUTSIDE
INSIDE
P2
P1
P2 > P1
P1
P2
CONDITION (4)
REED
SEAT
OUTSIDE
INSIDE
P2
P1
P1
P2
CONDITION (1)
REED
SEAT
OUTSIDE
INSIDE
P1
P2 < P1
P2
CONDITION (3)

FIG. 5.5.1

1) A closed film at the inner edge with oil flow at the outer edge.
2) Flow at both the inner and outer edges.
3) A closed film at the outer edge with oil flow at the inner edge.
4) A closed film at both inner and outer edges with subsequently no flow at all.

The surface film has been assumed to have zero angle of contact with the surface.

In practice, with a flexible reed, surface tension is only going to have an effect at the inner edge since the flexure of the valve reed will result in too great a gap at the outer edge. Although the direct force, due to surface tension, holding the valve on its seat is small, the surface tension initially restricts the flow of oil at the inner edge and results in an even greater negative pressure being produced at the inner edge. As the valve begins to lift, this surface film soon breaks and oil will begin to flow radially. This will cause condition 2 to result.

Because of the large gap at the outside edge with a flexible reed, it generally cannot sustain a surface film there until the pressure difference across the reed is quite small. By this time, however, the valve displacement is too great for this to happen. Condition 3 is therefore very unlikely to occur and is only added to the program for completeness.

A rigid reed can, however, have a surface film at both edges with practically uniform pressure within this film. This follows from the zero flow condition at the edges of the film.

5.6 Rate of Change of Pressure

The dependence of the time lag on the rate of change of pressure/

pressure is obvious, as a high rate of change of pressure will result in a reduced time lag. However, the rate of change of pressure is required not just at the balance point, but at all points on the compression or expansion curve. As explained previously in Chapter 2, although the rate of change of pressure can be considered linear over the opening sequence of the valve with no significant loss of accuracy in predicted time lag, the fact that it has an influence on the point at which the valve begins to lift, due to rotation, means that the mathematical expression for the true curve must be known. The derivation for $\frac{dp}{dt}$ which is used to update this parameter during the run of the program is shown in Appendix B.7.

In order to calculate $\frac{dp}{dt}$ at each time interval, the compressor characteristics of crank radius and connecting rod length, percentage clearance volume, gas constant and compressor speed must be inputed to the computer along with the valve and oil dimensions. The computer then begins at bottom dead centre (crank angle of zero radians) and calculates the rate of change of pressure at increasing crank angles until it determines the point at which the valve just begins to move. The crank angle is then advanced by fixed amounts after that depending on the time step chosen.

5.7 Suction and Discharge Pressures

These values could be obtained fairly simply since they were the suction plenum chamber pressure of the compressor and the back pressure respectively.

For the case when air was the working fluid, the suction port was open to the atmosphere and the suction pressure was therefore/

therefore just below atmospheric pressure. The difference in fact was too small to be detected on the pressure gauge used. A more accurate result could have been obtained using a manometer tube but this pressure was not really critical. The suction pressure gauge also measured the suction pressure when a closed circuit was used for R12. In this case, vacuums of a few inches of mercury could be detected.

The back pressure was also measured using a pressure gauge as explained previously in Chapter 3. This pressure had to be noted fairly accurately since this was the balance pressure for the disc on the knife edged seat and hence the pressure from which the time lag had to be measured. In a normal compressor, this pressure would be the discharge plenum chamber pressure.

5.8 Gas or Vapour Evolution Pressure

This pressure was determined from tests carried out on the Direct Force-Pull Apparatus described in Chapter 4. From these tests it was observed that the difference between the pressure in the oil film at which gas began to be evolved and the ambient pressure was very small. This pressure - referred to as the evolution pressure - is therefore closely associated with the ambient pressure.

If the ambient pressure is high, then the pressure of the gas in solution caused by this high pressure will be high also. Therefore, if the pressure in the oil film is dropped below the pressure of the surrounding gas, evolution will take place. When the valve is in its closed position, the ambient pressure will be the plenum chamber pressure - either suction or discharge depending/

depending on which valve is being considered.

If the discharge valve is once more considered, the computer assumes that the pressure in the oil film at which gas is evolved is the plenum chamber, or discharge, pressure less a fixed value of 30 kN/m^2. This figure of 30 kN/m^2 was the one determined experimentally from the force pull-off apparatus (see section 4.5.3).

This assumption of relating the evolution pressure to the discharge pressure is really the only possible one which can be made since a very low evolution pressure, for instance the cavitation pressure, would delay the valve opening very much more than is obtained in practice.

5.9 Gas or Vapour Evolution Delay Time

This component was also deduced experimentally from the Direct Force-Pull Tests. It was shown to be inversely proportional to the difference between the minimum pressure in the film and the evolution pressure. This relationship was approximately linear and was found to be insensitive to seat profile.

The method of determining the theoretical time delay has already been explained in section 4.5.4. The simplicity of this method meant that no differentiation was made between the case of a rigid disc, where the pressure profile is approximately parabolic, and the case of a flexible disc in which there is a sharp peak of negative pressure towards the inside edge caused by the valve reed straightening out. This meant that for a rigid disc, by the time the oil had "cavitated" at the point of minimum pressure, virtually all of the oil film across the seat width was below the evolution pressure, and so the "evolution constant" could be incorporated into the/

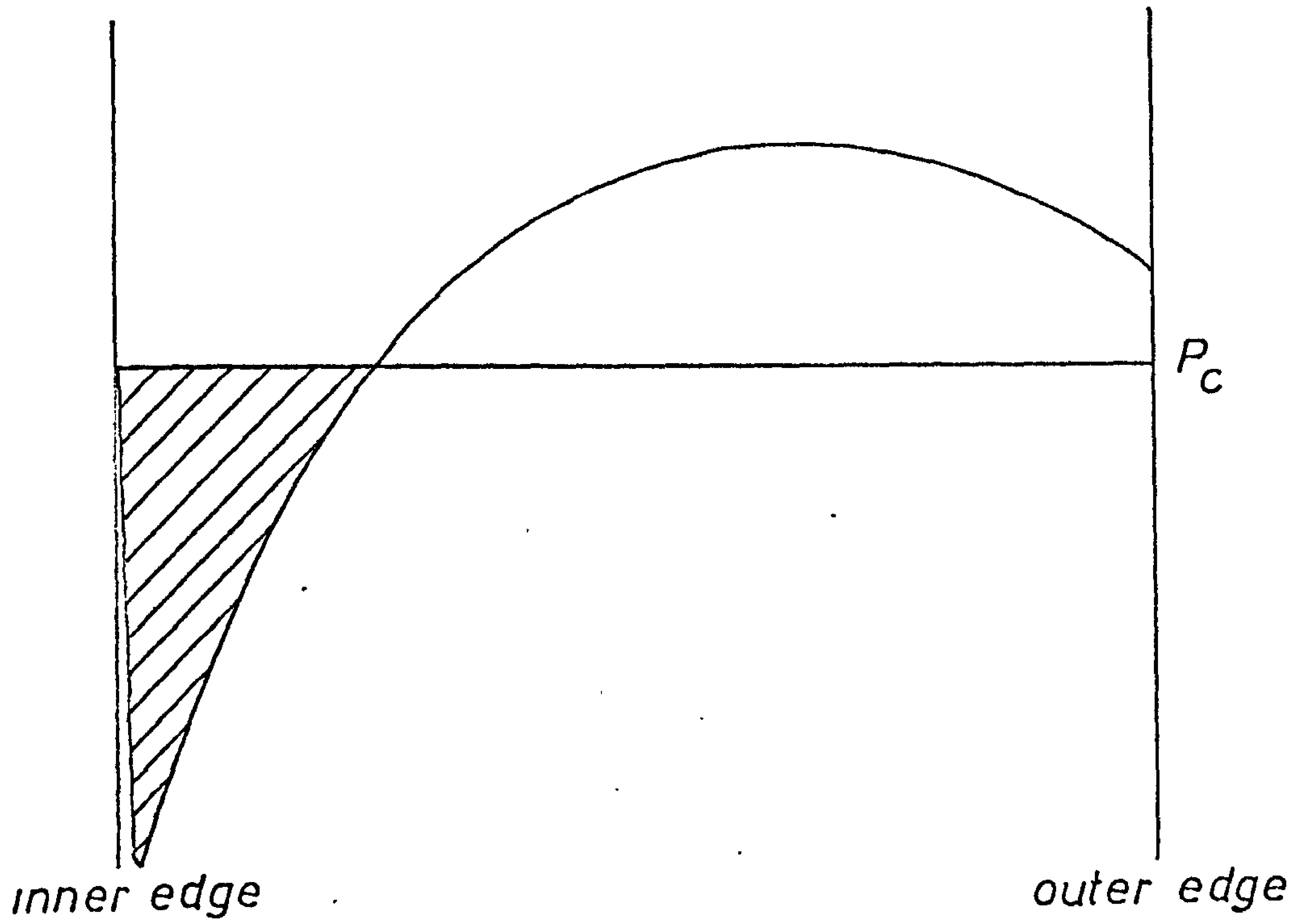
the mathematical model. However, for a flexible disc, whereas the oil at the inner edge will have "cavitated", the pressure towards the outer edge will still be positive. It would, therefore, be unreal to introduce the evolution constant at this point since the evolution constant assumes that gas is being evolved throughout the film. It is for this reason that File 3 is introduced. Once the model has deduced that the oil has cavitated at the inside edge but is still positive towards the outside edge, it accounts for the spreading of the gas evolution pressure across the seat width due to the reduction of the pressure difference across the valve.

5.10 Rate of Gas Evolution

The effect of increasing the reed thickness is to reduce the rate of rotation and slope of the valve with subsequent reduction in the thickness of the film when the cavitation pressure has spread over the entire area of the seat.

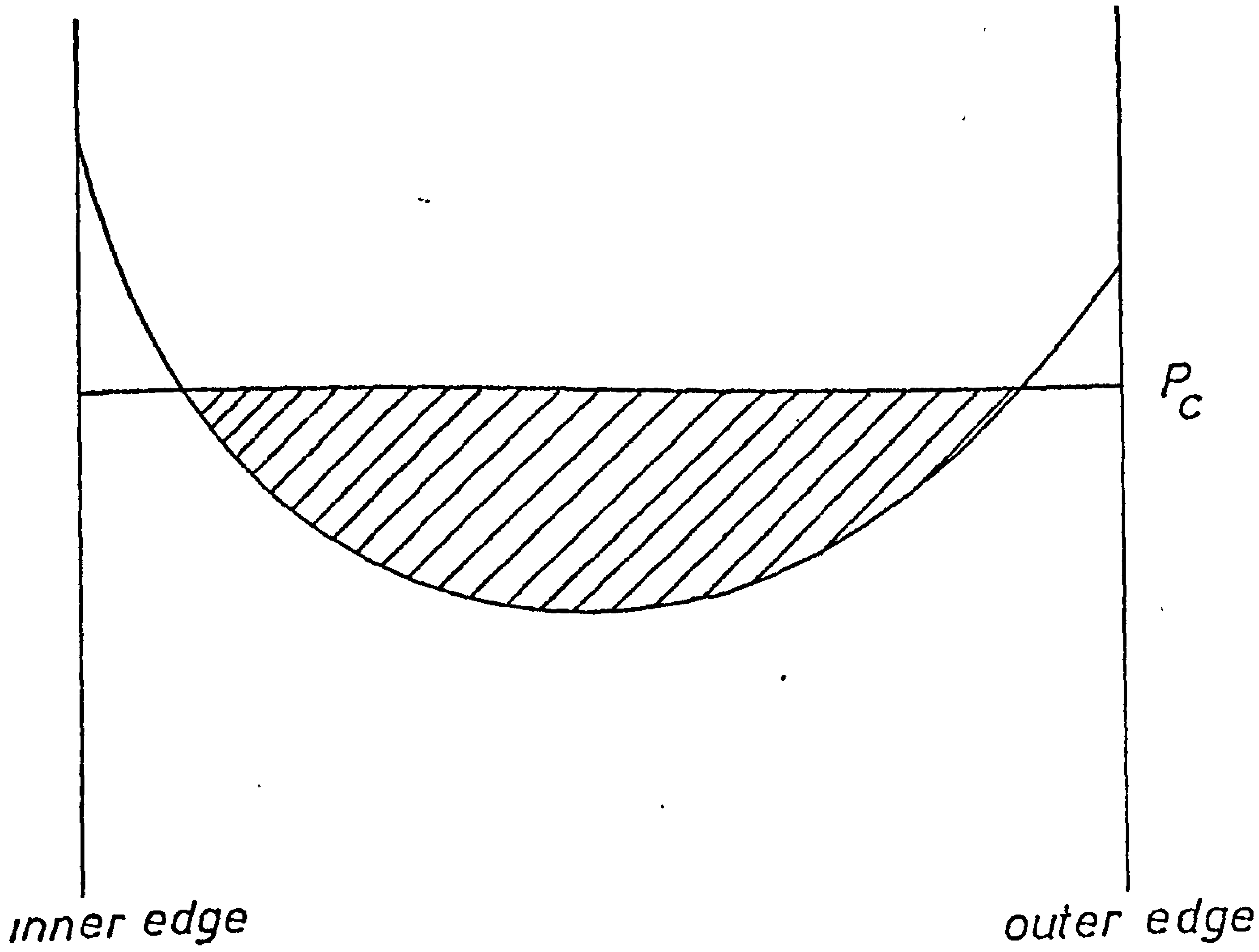
It is postulated that the rate at which gas evolves from the oil is related to the thickness of the film when it is fully cavitated. The simplest relationship is to assume this to be linear.

Fig. 5.10.1 shows a graph of the pressure in the oil film while the pressure difference across the valve is reducing. This corresponds to the diagram of Fig. 5.2.2 and clearly shows the spike of negative pressure at the inside of the seat. From Fig. 5.10.1 it will be seen that when the film cavitates, a high pressure is maintained towards the outside of the seat. From this it may be inferred that evolution is confined, in the early stages, to a fairly narrow strip near the inside edge of the seat. The evolution/



PRESSURE PROFILE ACROSS SEAT WIDTH FOR FLEXIBLE DISC

FIG. 5.10.1



PRESSURE PROFILE ACROSS SEAT WIDTH FOR RIGID DISC

FIG. 5.10.2

evolution constant developed from the Direct Force-Pull experiment must therefore be modified to allow for this reduced activity. The method of accomplishing this in the theoretical model is described here.

The pressure profiles of a rigid disc and a flexible disc are once more compared (Fig. 5.10.1 and Fig. 5.10.2).

After the delay time previously described, the rigid disc will have almost the complete width of its seat below the evolution pressure and therefore able to evolve gas. The flexible disc will have a much smaller area near the inner edge of the seat able to evolve gas. However, the negative pressure of the flexible disc will be much greater than that of the rigid disc. The rate of gas evolution which was calculated from the Force-Pull apparatus is therefore modified for a flexible disc in proportion to the area of the seat available to evolve gas. The relationship is taken as linear. This evolution rate is further altered by the value of the minimum pressures in the film at each step. In simple terms, the evolution rate is taken as a proportion of the shaded areas.

5.11 Mathematical Simulation of the Processes Involved

Computer plots of the processes described in sections 5.2.1 and 5.2.2 can be seen in Figs. 5.11.1 - 5.11.4. The reason for the curves not being smooth is because the valve seat is divided into only 12 sections. If more were chosen, the curves would obviously be closer to the correct profile. However, any increase in the number of steps would result in an increase in computer time. Twelve steps were found to be amply sufficient to give the correct blend between accuracy and computing time.

5.11.1 Rigid Disc/

5.11.1 Rigid Disc

The time steps involved in Fig. 5.11.1 are of 1 millisecond duration.

The initial high positive pressure (1) at the inside edge is due to the surface tension preventing the oil from flowing and therefore preventing it from falling to the gas pressure of the cylinder. It should be noted here that this disc is not completely rigid since there is oil flow at the outer edge. If it was completely rigid, surface tension effects would cause a high initial pressure at the outside edge also.

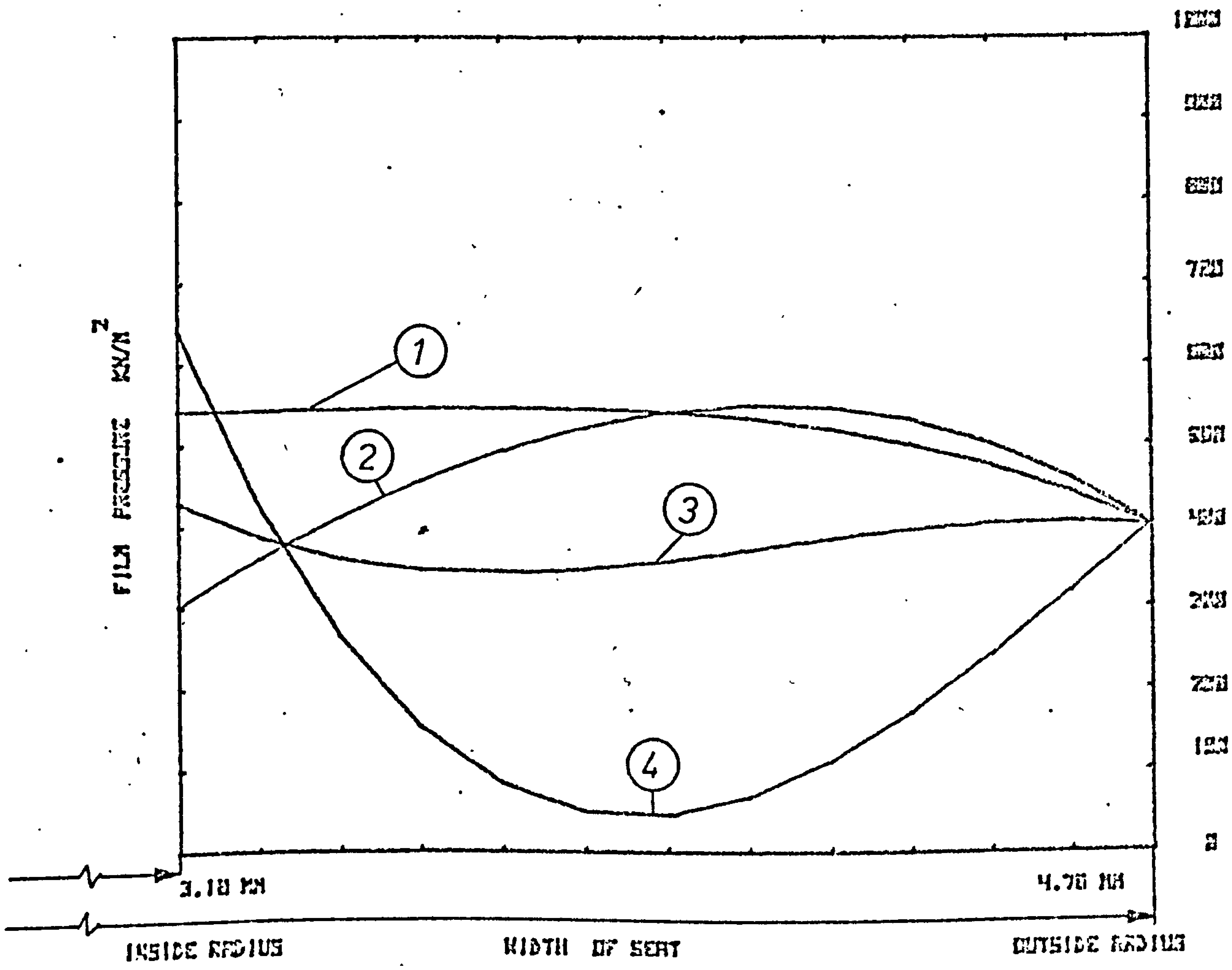
At stage 2, the gap between the disc and its seat has become too great for the surface tension to sustain the pressure difference between the gas pressure in the cylinder and the pressure of the oil film. The oil has therefore begun to flow and the oil film pressure has fallen to the cylinder pressure at the inside edge. Stages 3 and 4 show the gas pressure in the cylinder increasing and the negative pressure being developed in the oil film prior to gas or vapour evolution.

In Fig. 5.11.2, gas is being evolved and the disc is moving rapidly from its seat. (In this figure, the time step is only 0.2 milliseconds). The pressure in the oil film again falls with an accompanying reduction in the width of the oil film.

5.11.2 Flexible Disc

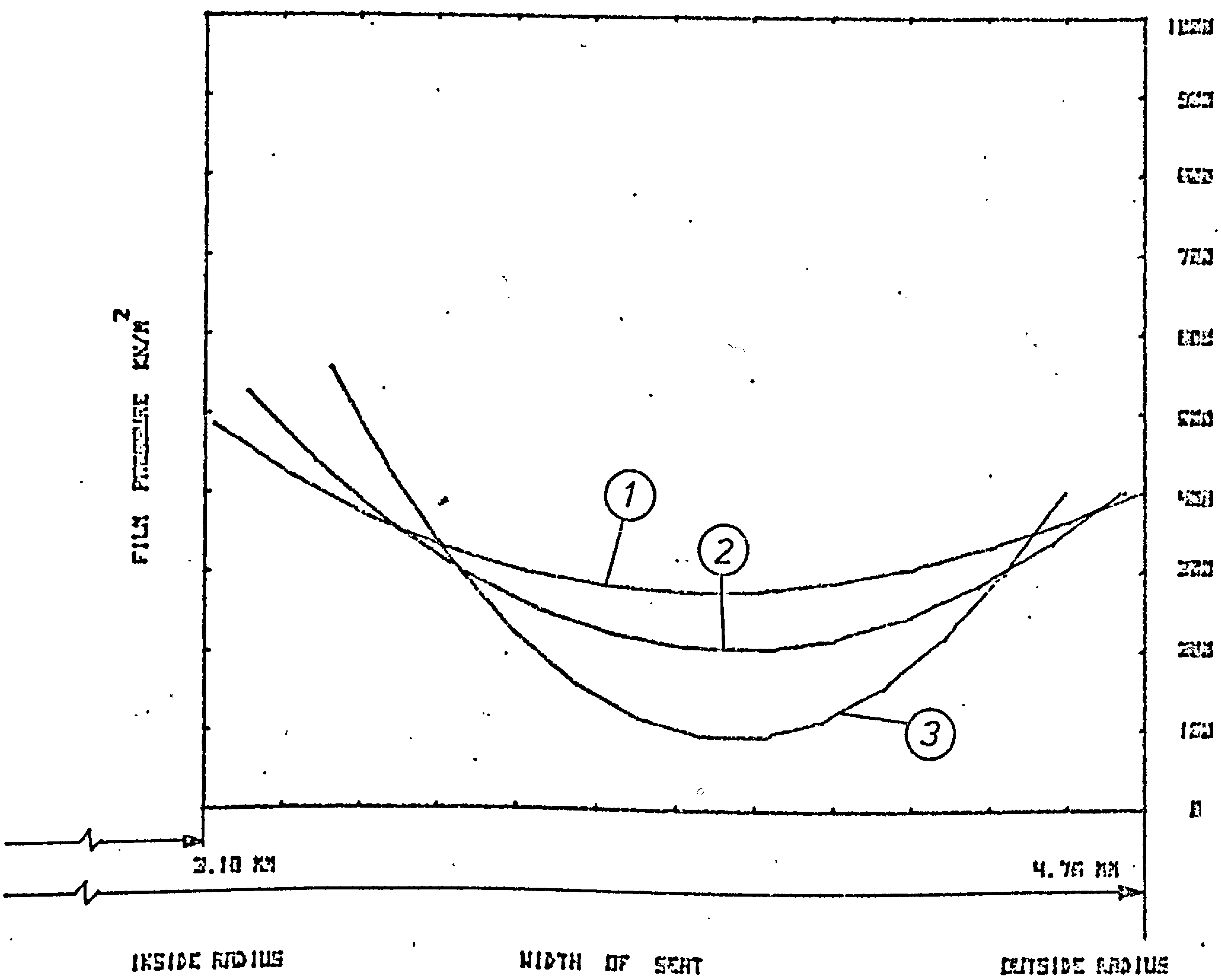
Computed pressure profiles of the processes involved for a flexible disc are shown in Figs. 5.11.3 and 5.11.4.

Like the rigid disc, there is a high positive pressure at the inside edge (1) and (2). The pressure difference between the oil/



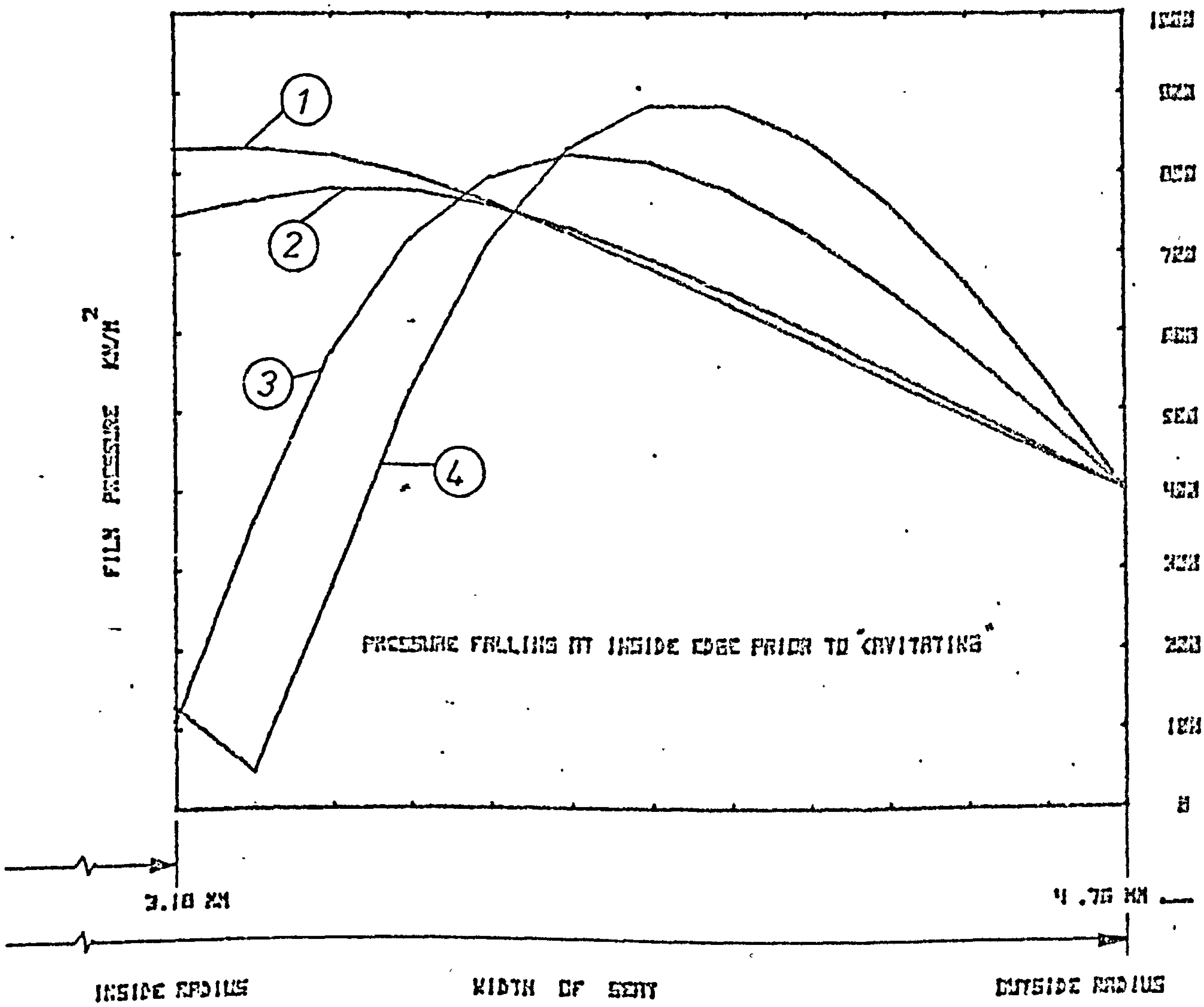
VARIATION OF PRESSURE DISTRIBUTION FOR RIGID DISC PRIOR TO "CAVITATION"

FIG. 5.11.1



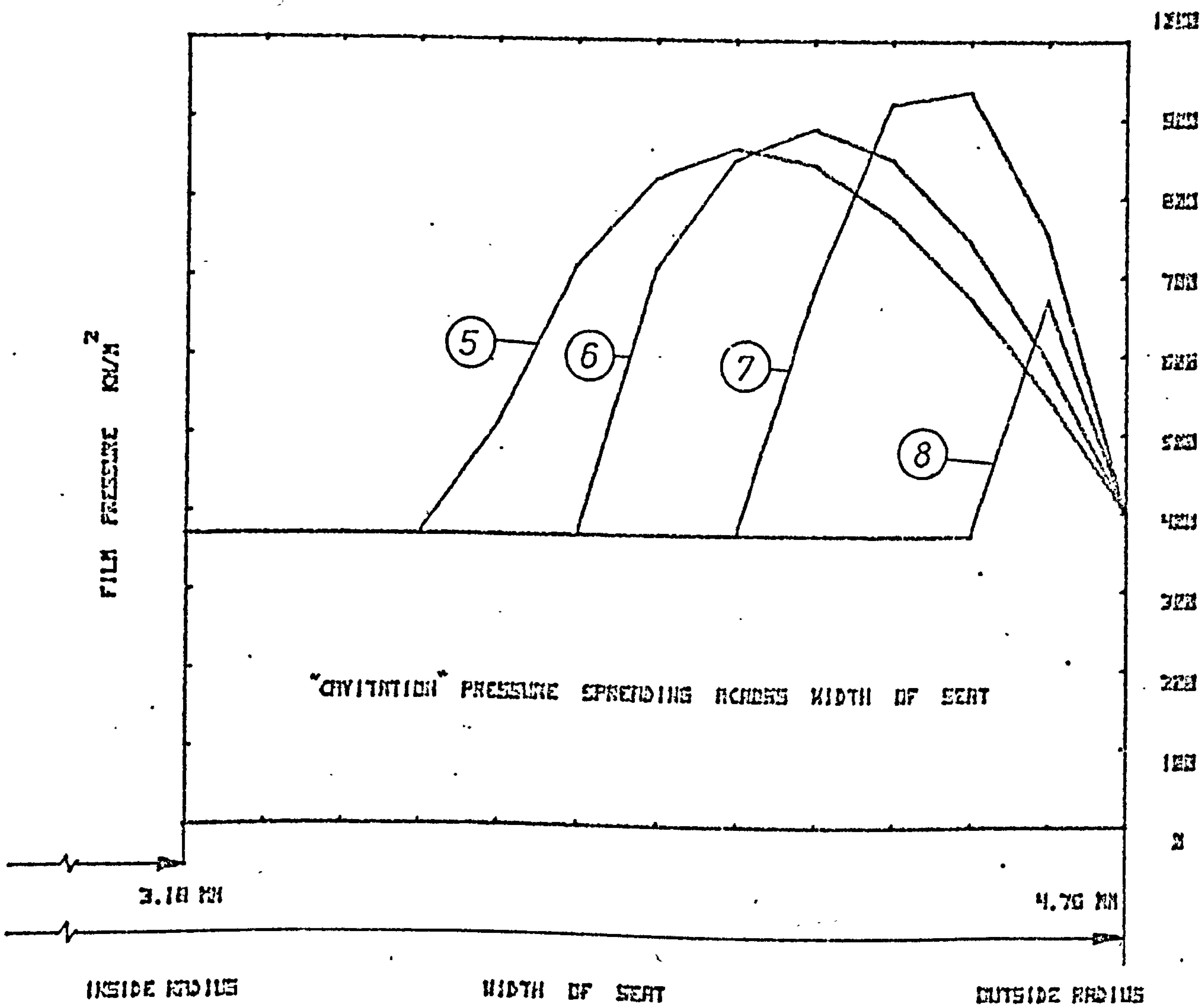
VARIATION OF PRESSURE DISTRIBUTION AFTER "CAVITATION" SHOWING THINNING OF OIL FILM ACROSS WIDTH OF SEAT

FIG. 5.11.2



VARIATION OF PRESSURE DISTRIBUTION FOR FLEXIBLE DISC
STAGE 1 (COMPUTED)

FIG. 5.11.3



VARIATION OF PRESSURE DISTRIBUTION FOR FLEXIBLE DISC
STAGE 2 (COMPUTED)

FIG. 5.11.4

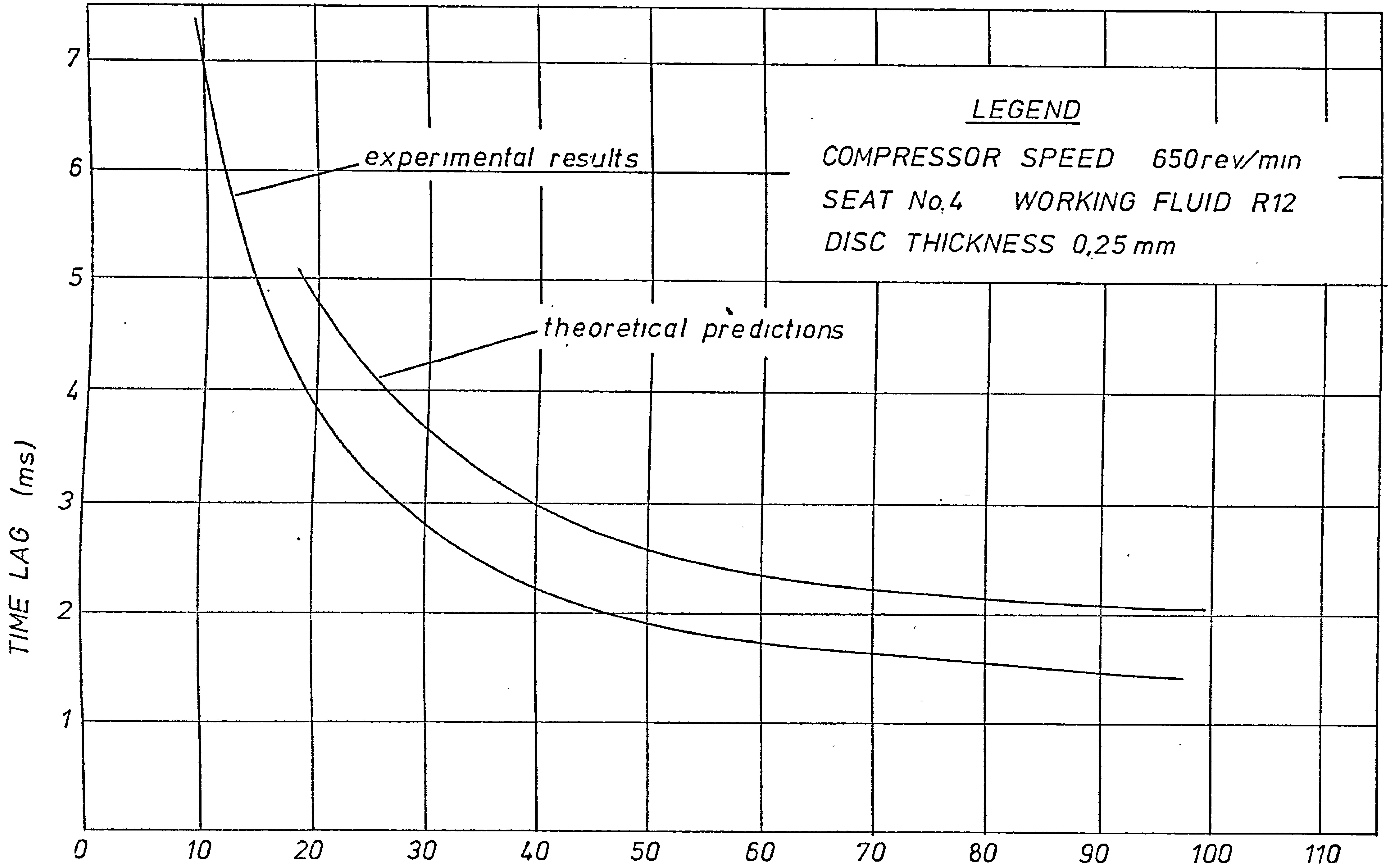
oil film pressure and the cylinder gas pressure is sustained by the surface tension. After the gap between the valve disc and its seat has become too large for the surface tension to withstand the pressure difference, the pressure in the oil film falls to the cylinder pressure with an accompanying oil flow (3). Because of the valve reed flexibility, however, a high positive pressure is still maintained towards the outside edge. Stage 4 shows that the pressure in the cylinder has risen slightly and that the oil pressure at the inside edge has risen accordingly. However, a negative pressure is still being generated near to the inside edge. It is at this point that gas or vapour begins to be evolved even although there is still a positive pressure towards the outer edge.

Fig. 5.11.4 shows the stages involved as each part of the oil film "cavitates" and the evolution pressure spreads over the width of the seat. When this is completed the oil film is considered to be "fully cavitated" and the "evolution constant" is introduced. The program then reverts back to File 2 and the stages shown in Fig. 5.11.2 then occur with the resulting reduction in the width of the oil film prior to film rupture.

5.12 Theoretical Predictions

In general, both the time lags and the pressure differences predicted by the mathematical model compare favourably with results obtained experimentally.

Fig. 5.12.1 shows comparison between these two sets of results for one particular valve seat. The curve for the experimental results is the dotted curve from Fig. 3.8.6. Although not exactly the same, the theoretical curve does exhibit similar trends/



RATE OF CHANGE OF PRESSURE ($kN/m^2.s$ $\times 10^3$)

FIG. 5.12.1

trends. The reason for choosing a theoretical compressor speed of 650 rev/min was to enable low rates of change of pressure to be obtained. Higher speeds would have meant that the early part of the curve would have been missing.

Fig. 5.12.2 shows how the theoretical model also predicts different time lags - and hence pressure differences - for different disc thicknesses, everything else being equal. The values of oil viscosity and compressor speed are therefore purely arbitrary. However, the viscosity was chosen as if air was the working fluid - with a resulting viscosity of 16 centipoises - and the compressor speed was made low in order to obtain low rates of change of pressure.

In Fig. 5.12.3, a theoretical comparison is made between two different working fluids similar to that of the experimental programme. The only way in which the two working fluids could be simulated was in the effect they had on the oil viscosity. As previously explained, the miscibility of refrigerant with oil resulted in a reduced viscosity. The comparison in Fig. 5.12.3 is therefore really only a comparison between oil with a viscosity of 16 centipoises (working fluid - air) and oil with a viscosity of 6 centipoises (working fluid - R12). No account is taken of the possibly larger evolution constant which would undoubtedly have an effect in further reducing the time lag.

It is hoped by these examples to show that the theoretical model does give results which agree in general with results obtained experimentally.

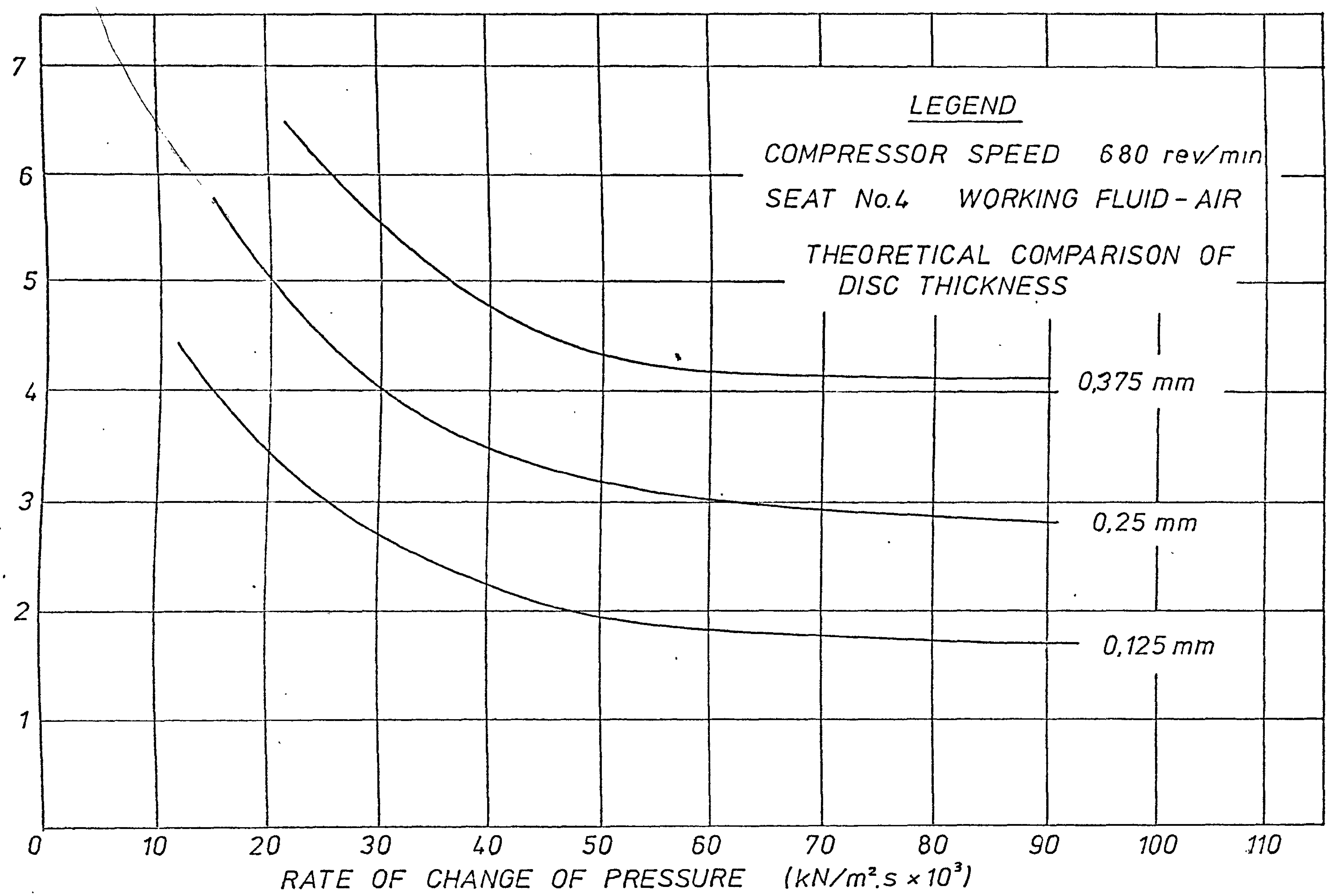

LEGEND
COMPRESSOR SPEED 680 rev/min
SEAT No.4 WORKING FLUID - AIR
THEORETICAL COMPARISON OF DISC THICKNESS
0,375 mm
0,25 mm
0,125 mm
7
6
5
4
3
2
1
0
10
20
30
40
50
60
70
80
90
100
110
RATE OF CHANGE OF PRESSURE (kN/m².s × 10³)

FIG. 5.12.2

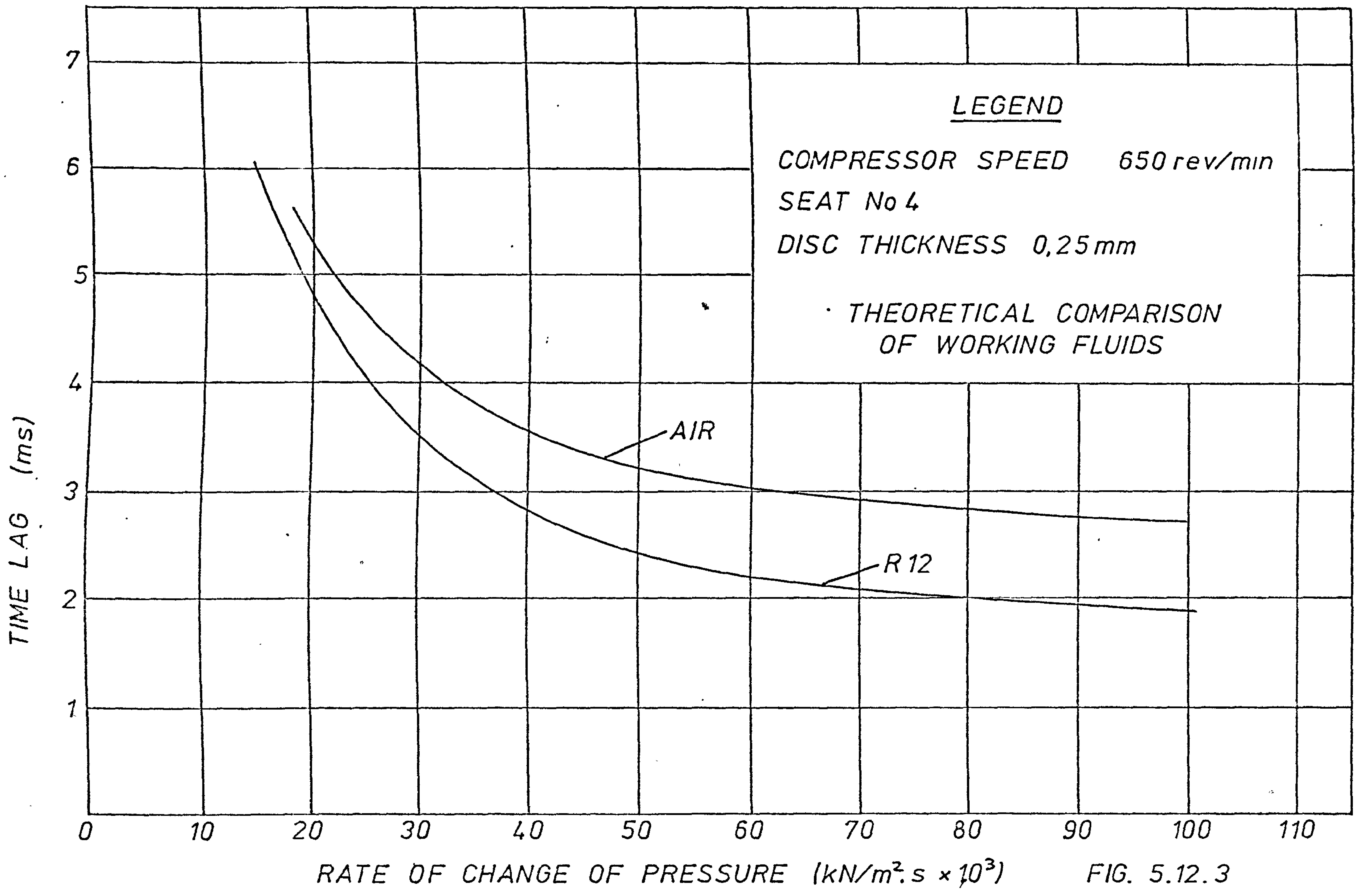

LEGEND
COMPRESSOR SPEED 650 rev/min
SEAT No 4
DISC THICKNESS 0,25 mm
THEORETICAL COMPARISON OF WORKING FLUIDS
AIR
R 12
TIME LAG (ms)
0
1
2
3
4
5
6
7
10
20
30
40
50
60
70
80
90
100
110
RATE OF CHANGE OF PRESSURE (kN/m².s × 10³)

FIG. 5.12.3

CHAPTER 6

DISCUSSION

6.1 Introduction

The purpose of this investigation was to determine experimentally the effect of oil stiction on the function of automatic compressor valves. Simultaneously, a theoretical model was developed to describe the stiction process.

At first, the investigation was more of an exploration into the topic rather than an attempt to validate a theory. This meant that both the experimental apparatus and, eventually, the theoretical model were expanded and refined as more knowledge of the subject was acquired. The theory developed was, therefore, partially influenced by the early experimental work. Similarly, the experimental programme was based on observations from theoretical predictions. In particular, the Force Pull-Off apparatus was initially designed to measure directly the oil stiction forces involved when a valve disc leaves its seat. This in turn gave a better appreciation of the cavitation processes involved and eventually the apparatus was used almost exclusively to determine the unknown coefficients required in the theoretical model.

As the work progressed, the number of significant parameters involved became so great that this process of parallel development was the only reasonable one. Hence, the mathematical model contains variables or constants developed experimentally. These were incorporated in the mathematical model and the predictions checked by results from the compressor tests.

6.2 Experimental Coefficients

One of the parameters investigated during this project was the rate at which gas or vapour was evolved from the oil film. This/

This evolution rate has been shown to have a large bearing on the time lag in the valve opening since it causes the rapid break-up of the oil film. Unfortunately, this was also one of the most difficult characteristics to measure because of the complexity of the event and the short time in which it occurs. To understand it fully, a major study would have to be made of the cavitation phenomenon which is outwith the scope of this present investigation. Simulation of the cavitation process was limited to what was required for an adequate model. The method described in Chapter 4, to obtain a value for this evolution rate, is therefore justified by the fact that the values deduced were in accordance with results obtained in the compressor apparatus.

This method was not suitable for determining an evolution constant when an oil/refrigerant mixture was used. The reason being that because R12 is gaseous at atmospheric pressure, it would come out of solution before any tensions could be applied to the oil film. However, because refrigerants are much less viscous than oils, any appreciable amount in solution causes a marked reduction in the viscosity of the oil. Figures published in the ASHRAE Guide and Data Book, show that only a 15%, by weight, refrigerant in oil mixture reduces the viscosity from 16 centipoises to 6 centipoises at 130^{o}F. It was found that by merely inserting this reduced value of viscosity in the mathematical model, theoretical predictions were very similar to those obtained in practice. The need to determine the evolution rate of vapour was therefore obviated.

Similarly, the value of the time delay in the evolution of vapour from the mixture could not be determined experimentally. However, in most practical cases where the valve reed is flexible, "cavitation"/

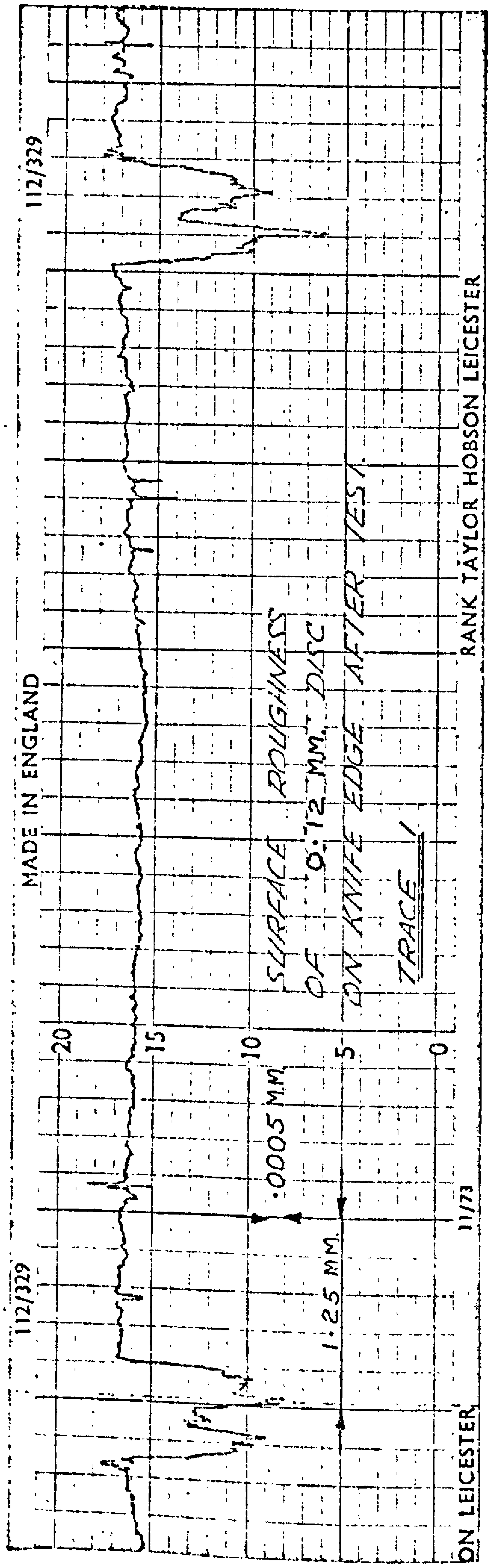
"cavitation" was initiated well before the gas belance pressure across the valve was reached.

6.3 Valve Flexibility

One of the most significant aspects of the experimental tests was the effect of the thickness of the valve disc. The time lag due to oil stiction could be reduced if the valve thickness was made less. Also, it might be noted that Nagaoka and Hotani (18) found that the volumetric efficiency of a compressor increased when thin valve reeds were employed. Their explanation of this increase in volumetric efficiency was not based on oil stiction effects but on the supposition that, when thin reeds were used, the valve lift would be greater and consequently the pressure drop through the valve would be greatly reduced. The improvement in performance must, however, be due in part to reduced oil stiction.

The thinner a valve disc or reed is, the greater will be the negative pressure "spike" produced towards the inside edge of the seat. This negative pressure "spike", as has been explained previously, occurs before the pressure balance condition across the valve and is caused by the valve reed straightening out as well as lifting. Also as previously mentioned, a "wire drawing" effect could consequently be produced at the inner edge of the seat because of the very small valve opening, in conjunction with the very large pressure difference. This wire drawing effect could conceivably cause damage to either the valve reed or the valve seat, especially if there were any scratches at the inner edge of the seat which could amplify the effect. This type of damage is not unknown to compressor manufacturers who have to peen the edge of the valve seat

to/



MADE IN ENGLAND
112/329
RANK TAYLOR HOBSON LEICESTER
SURFACE ROUGHNESS
OF 0.12 MM. DISC
ON KNIFE EDGE AFTER TEST
TRACE 1
20
15
10
5
0
·0005 MM
1.25 MM
11/73
112/329
ON LEICESTER

FIG. 6.3.1

to ensure that a good surface finish is obtained.

If a valve reed is made from very thin material, it is less likely to be able to withstand the continuous shock forces against the seat and stop. This can be seen in Fig. 6.3.1. It shows a measurement of the surface finish of a 0.12 mm test disc after only a few test runs. A very pronounced indentation has been made by the sharp edged seat which could lead to disc failure if the compressor was run continuously. This damage is not significant when a valve disc of, say, 0.35 mm is used.

A compromise must be made when choosing a valve reed thickness for a particular application between the reduction in stiction effect, when using a thin reed, and the limitations imposed by impact fatigue considerations.

6.4 Application of Mathematical Model

The experimental programme undertaken and the ensuing theoretical simulation have produced a model which is suitable for assisting in the design of automatic compressor valves. Although throughout this thesis reference has almost invariable been made to a discharge valve, the model can just as easily handle a suction valve or indeed the stiction effect of a valve disc or reed against its stop. With minor modifications it can also be applied to ring valves where the effect of oil stiction against the stop in particular can be quite pronounced.

The major difference between a suction valve and a discharge valve is in the rate of change of pressure across the valve at opening which has been shown to have a significant effect on the time lag. For a typical compressor running at 3000 rev/min, the/

the rate of change of pressure at opening of the suction valve is only 70 x 10^3 kN/m^2 s, whereas for a discharge valve it is 10^6 kN/m^2 s. The time lag in opening of the discharge valve will therefore be very much less than that of the suction valve. Although this small time lag will produce a pressure peak in the compressor cylinder, this effect will quickly disappear as soon as the valve opens. However, the significance of oil stiction is that the boundary conditions at the beginning of the valve opening sequence are considerably modified, and the output of a valve motion simulation program, in consequence, will be in error giving inaccurate predictions of such important parameters as impact velocities, valve closing times and valve bounce.

It is interesting to note here that a commercial compressor manufacturer has adopted this oil stiction program into their own overall compressor simulation program and satisfactory correlation with experimental results has been obtained.

6.5 Future Work

As mentioned in section 6.2, the Time Delay and Evolution Constants for refrigerant/oil mixtures have been assumed to be of the same order as the constants for oil alone. Although the theoretical predictions of the time lags in the motion of the compressor valve discs are fairly satisfactory, it is felt that research on this aspect of the stiction phenomenon should be continued.

One possible approach for determining these coefficients would be to situate the Force Pull-Off apparatus inside a pressurised chamber. The oil/refrigerant mixture, under pressure, would/

would then be fed to the valve seat, preventing any vapour evolution prior to the tensile force being applied to the oil film.

Stroboscopic holography, as used by Brown and Lough (4), could be developed to show the true deflected shape of a valve reed. Moreover, due to the sensitivity of such a method, the pressure distribution across the seat of the valve should be capable of derivation. From the pressure distribution, the "cavitation zones" could be mapped during the opening sequence.

APPENDIX A

SHOCK TUBE APPARATUS

A.1 Introduction

It was thought, initially, that a shock tube could be used to measure the response of a valve disc sitting on seats with various profiles. The response of the disc could then be determined when the seat was dry and when oil was present. It was hoped that these results could be related to the case of a compressor where the pressure difference across the valve was not a step function but an increasing rate of change of pressure.

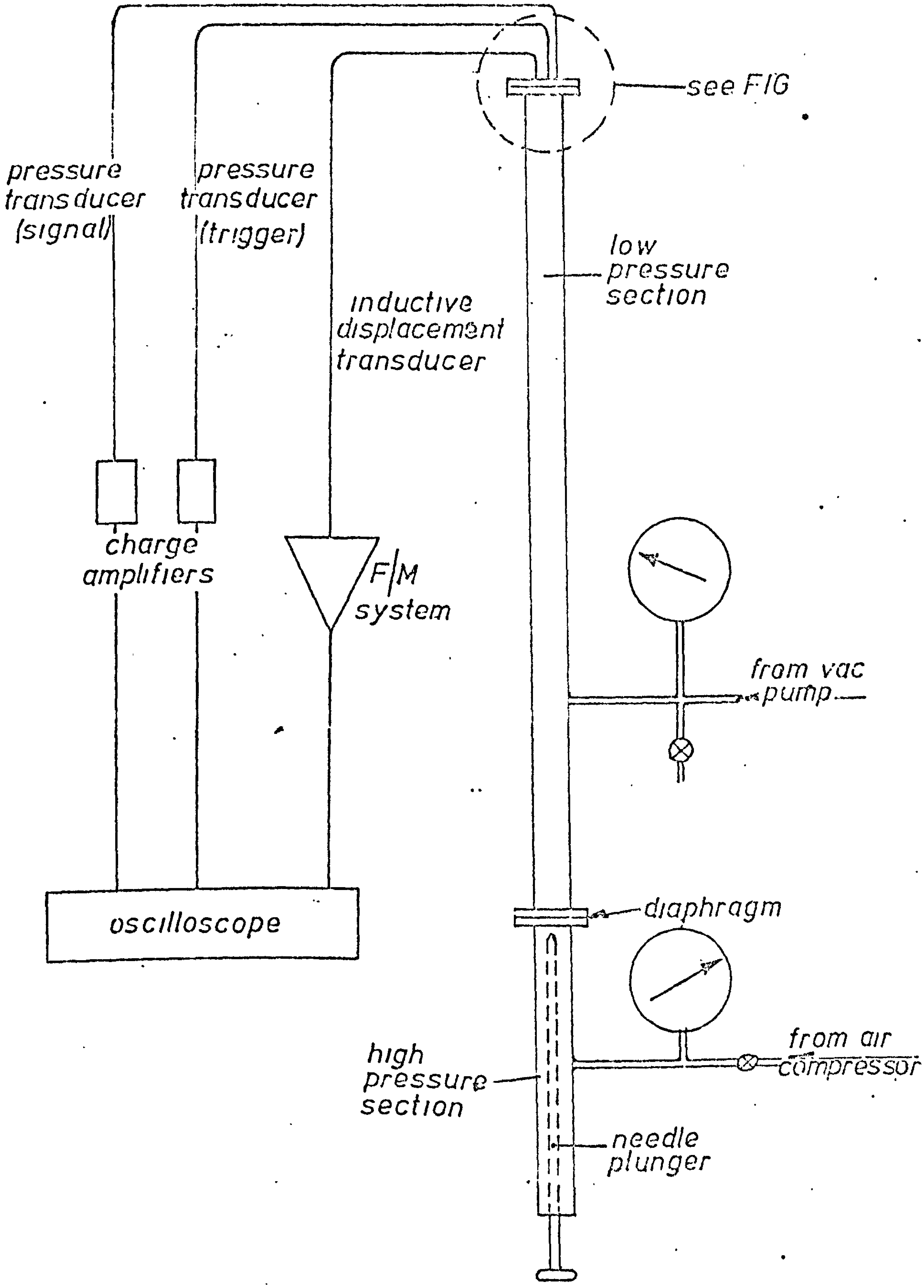
The fact that a shock wave produces virtually a step input meant that the force/time history was very similar to that in the pull-off apparatus when a high voltage was applied to the coil. The difference was that, in the shock tube, the valve disc deflected into the valve port under gas pressure, as in a compressor.

One of the main difficulties with the shock tube, which was not encountered with the other two pieces of apparatus, was in ensuring completion of the oil film between the disc and its seat prior to each operation.

A.2 Description of Shock Tube

The shock tube consisted basically of a 2" inside diameter steel tube mounted vertically. The reason for mounting it vertically rather than horizontally was to take account of any gravitational effects on the valve disc. It was supported by six tension springs to reduce any vibration effects.

The high pressure and low pressure sections of the tube were separated by a thin brass diaphragm. The thickness of the diaphragm could be varied between 0.025 - 0.1 mm depending on the pressure difference required. For very small pressure differences it/



SCHEMATIC DIAGRAM OF SHOCK TUBE AND ANCILLARY EQUIPMENT

FIG. A.2.1

it was found that a paper diaphragm was very successful.

A high degree of repeatability was also obtained for the pressure differences at which each thickness of brass shim, or paper, ruptured. However, if a particular pressure difference across the diaphragm was required, a sharp plunger could pierce it.

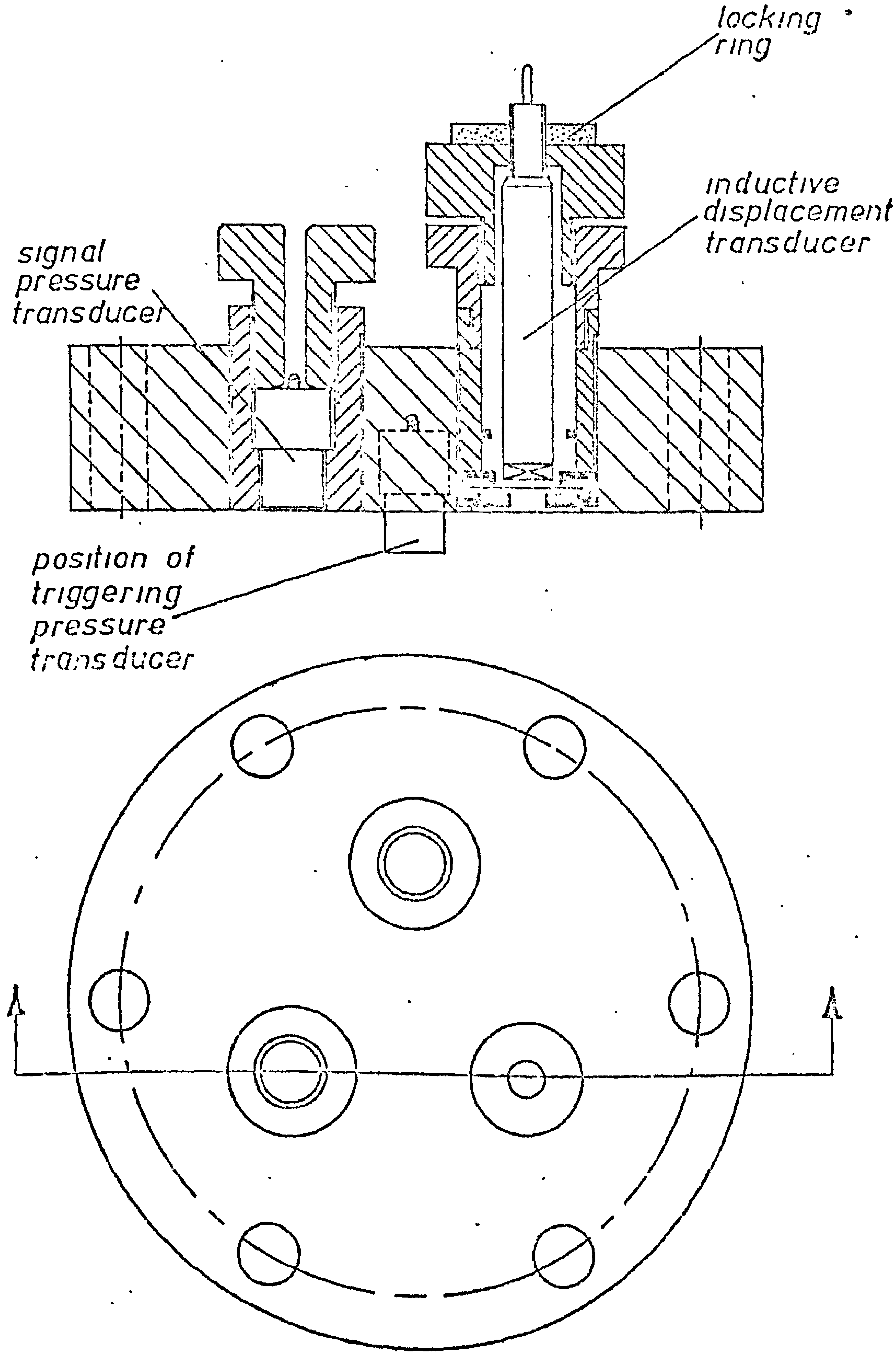
The top half of the shock tube was evacuated using a vacuum pump. A filter immediately after the outlet from the low pressure side of the shock tube prevented any small pieces of ruptured diaphragm finding their way to the vacuum pump.

A.2.1 Test Head of Shock Tube

Two Kistler Type 701A pressure transducers and the test disc on its seat were mounted at the top end of the shock tube as shown in Fig. A.2.2. One pressure transducer was used for measuring the shock wave. The other one was used for triggering purposes as will be explained in section A.3. The measuring pressure transducer and test seat were mounted flush with the end of the LP section of the tube in order to avoid connecting passageways which degrade the dynamic response.

The test seat was placed in the head, as shown, with the valve disc sitting on top of it. The limit on the motion of the disc was governed by a retaining seat and the complete unit was held in position by a locking device which was screwed down on top of it. This locking device also held the displacement transducer which measured the displacement of the disc. An explanation of the displacement transducer and its ancillary equipment has been given in section 3.6.

A.3 Triggering Operation/



POSITIONING OF TRANSDUCERS IN HEAD OF SHOCK TUBE

FIG. A.2.2

A.3 Triggering Operation

The signals from the pressure transducer and the displacement transducer were recorded on a storage oscilloscope (see section 4.4.3). Because of the very short time (about 100 μs) during which the operation occurred, some method had to be devised to trigger the sweep of the oscilloscope at the correct time.

At first, another pressure transducer was placed midway down the shock tube and the shock wave when passing the pressure transducer triggered the oscilloscope. However, the oscilloscope then had to be delayed a certain time after triggering to allow for the relatively long time (1 ms) that the shock wave took to travel up the rest of the tube. This was most unsatisfactory since any slight difference in the speed of the shock wave meant that the oscilloscope triggered either too early or too late.

If the measuring transducer was used to trigger the oscilloscope, it meant that the first part of the signal was missing, because of the small deflection needed in the oscilloscope input signal to cause triggering.

Another pressure transducer was therefore installed in the head of the shock tube but about 10 mm lower down. This meant that the oscilloscope was triggered as the pressure wave front reached this transducer and the time interval was found to be correct for the speed at which the shock wave was travelling.

A.4 Shock Tube Results

The mathematical model assumes that the strain energy in the reed, caused by its deflection into its port due to the pressure difference across the reed, is dissipated in displacing the oil between/

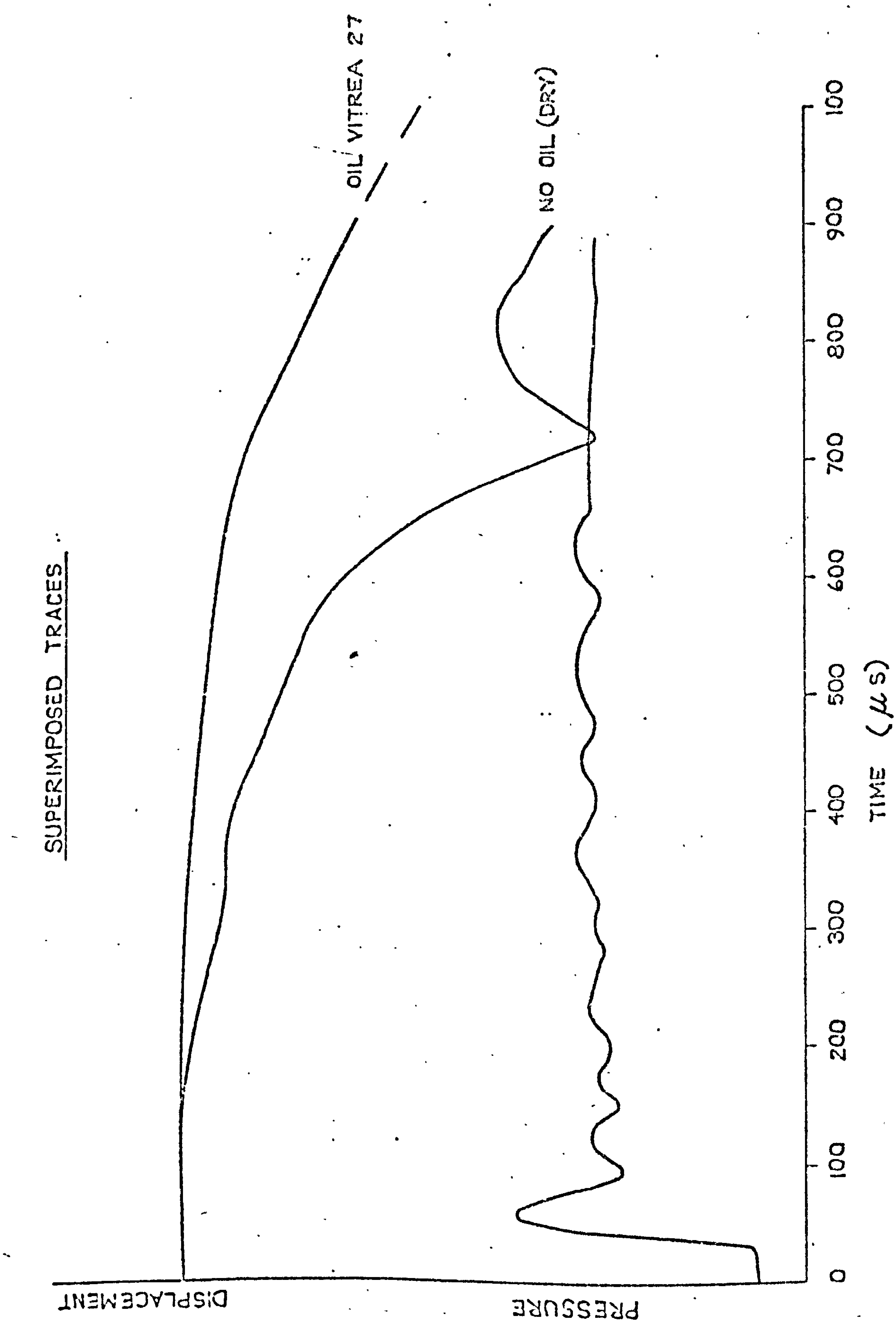

SUPERIMPOSED TRACES
OIL VITREA 27
NO OIL (DRY)
DISPLACEMENT
PRESSURE
0
100
200
300
400
500
600
700
800
900
100
TIME (μs)

FIG. A.4.1

between the reed and its seat.

This assumption is validated by results obtained in the shock tube equipment described.

Fig. A.4.1 shows superimposed oscillograms taken under identical pressure shock wave conditions for a valve operating (a) dry and (b) with oil present.

The lift/deflection trace for the valve in the dry condition shows the presence of an oscillatory component. This is absent when oil is present. The oscillatory component is due to free vibration of the reed and its absence in the other trace can only be accounted for by the absorption of the strain energy by the oil film.

The traces also show the increased time required for the reed to leave its seat when oil is present. This increased time is small, due to the very large step input across the valve when the pressure wave arrives.

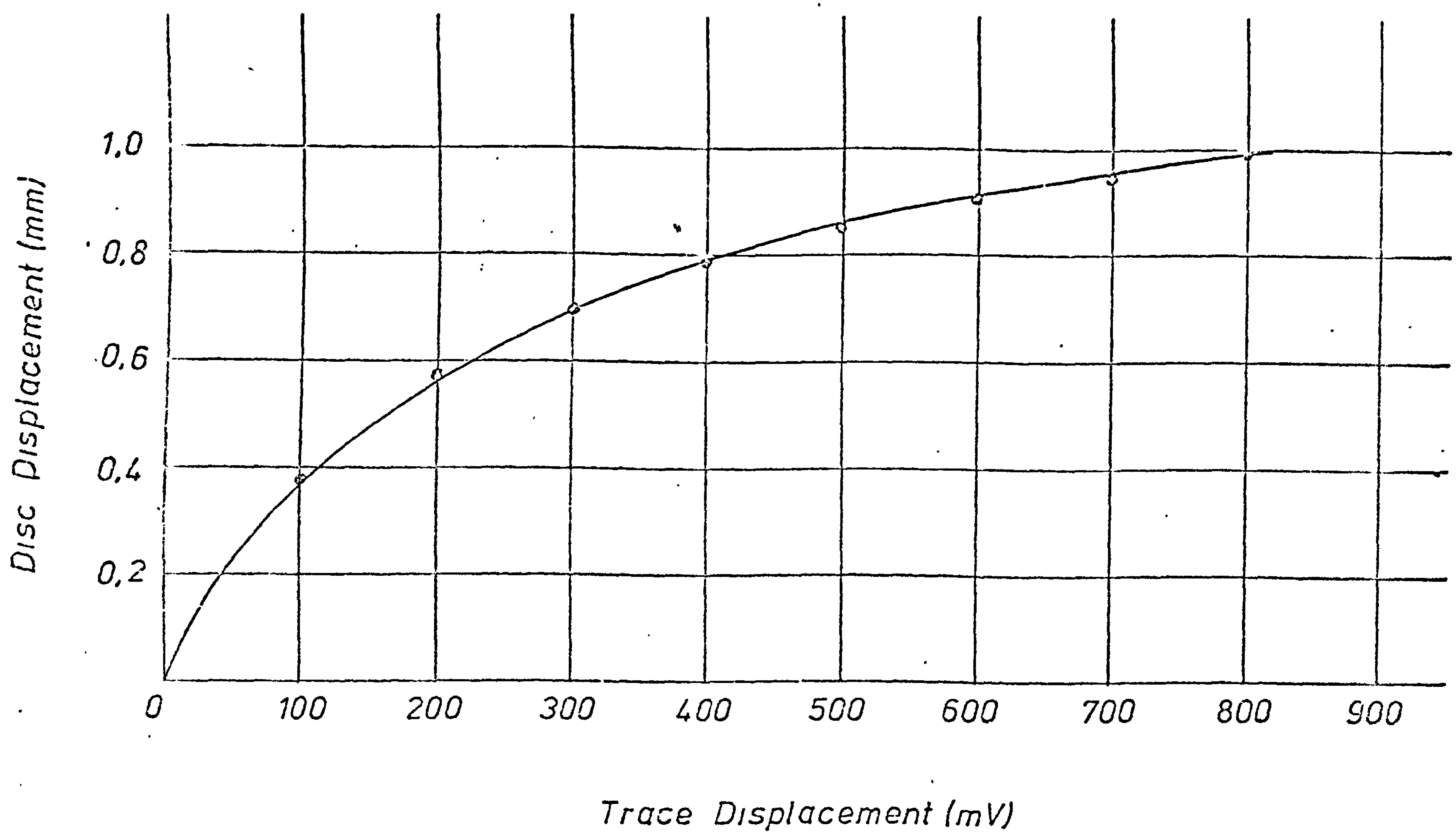
A.5 Calibration of Inductive Displacement Transducer

Purely for convenience, the inductive displacement transducer used in both the compressor and shock tube experiments was calibrated when it was situated in the shock tube head.

The method employed compared the deflection of a typical disc displayed on an oscilloscope screen with the known deflection using a depth micrometer.

When the disc was sitting on the bottom seat, it was raised vertically using the depth micrometer, the reading on the micrometer and the deflection on the oscilloscope being noted at regular intervals. Because the displacement transducer is influenced/

influenced by the inductance between itself and the disc, an insulating disc had to be fixed to the end of the micrometer in order that it did not have any effect on the reading. The calibration of the transducer is shown in Fig. A.5.1.



INDUCTIVE DISPLACEMENT TRANSDUCER CALIBRATION FIG. A.5.1

APPENDIX B

FINITE DIFFERENCE TECHNIQUE

B.1 Introduction

The finite difference technique used, and explained here, was an iterative method which was suitable for solution on a digital computer. The basis of the approach was to determine the values of velocity and pressure at various points in the oil film by initially guessing and then adjusting the unknown boundary conditions to suit the known ones.

Because the pressure did not vary circumferentially, all oil velocities were therefore radial. (This was one of the assumptions made in section 2.2 of the thin film approximations to the Navier-Stokes Equations).

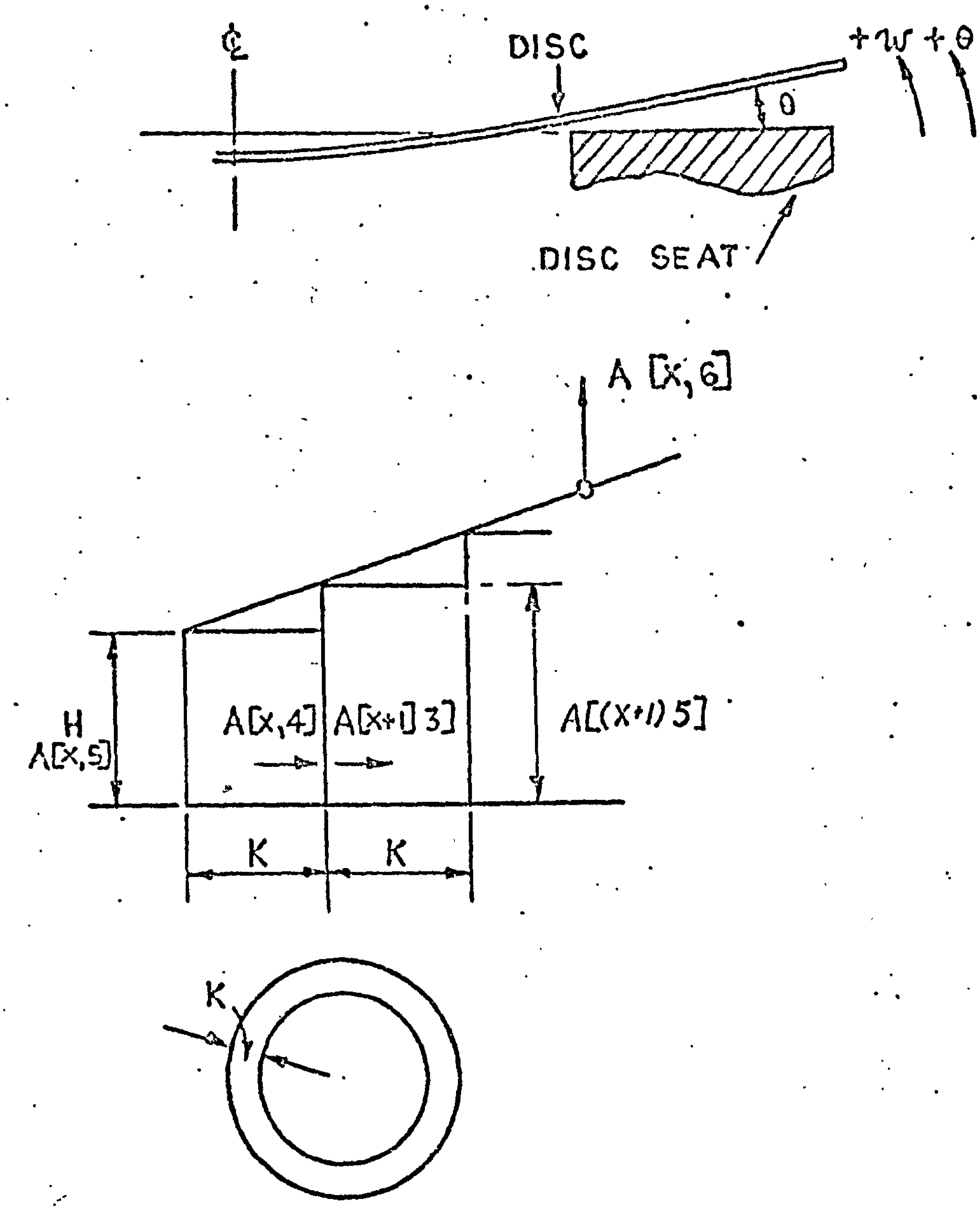
The oil film between the valve reed and its seat was first of all divided into a number of annular ring elements of equal width as shown in Fig. B.1.1. A section through one of these elements is shown in Fig. B.1.2. Because the width of each element was small, the oil thickness over it was considered uniform, immaterial of the seat surface profile or the valve reed slope.

The various values of oil pressure, velocity and thickness etc., between each element were stored in matrix form. An explanation is given here as to what the variables refer to.

The values of one particular variable e.g. oil pressure, were stored in one column of the matrix. The different values of each variable across the width of the seat were therefore stored in the different rows of the matrix. Because it was the value at the points between each element that was measured, there had to be one more row in the matrix than the number of elements, to account for the inside and outside edges of the element.

In Fig. B.1.2, if "X" refers to one particular point:

A/



FINITE ELEMENT SYMBOLS AND SIGN CONVENTIONS

FIG. B.1.1

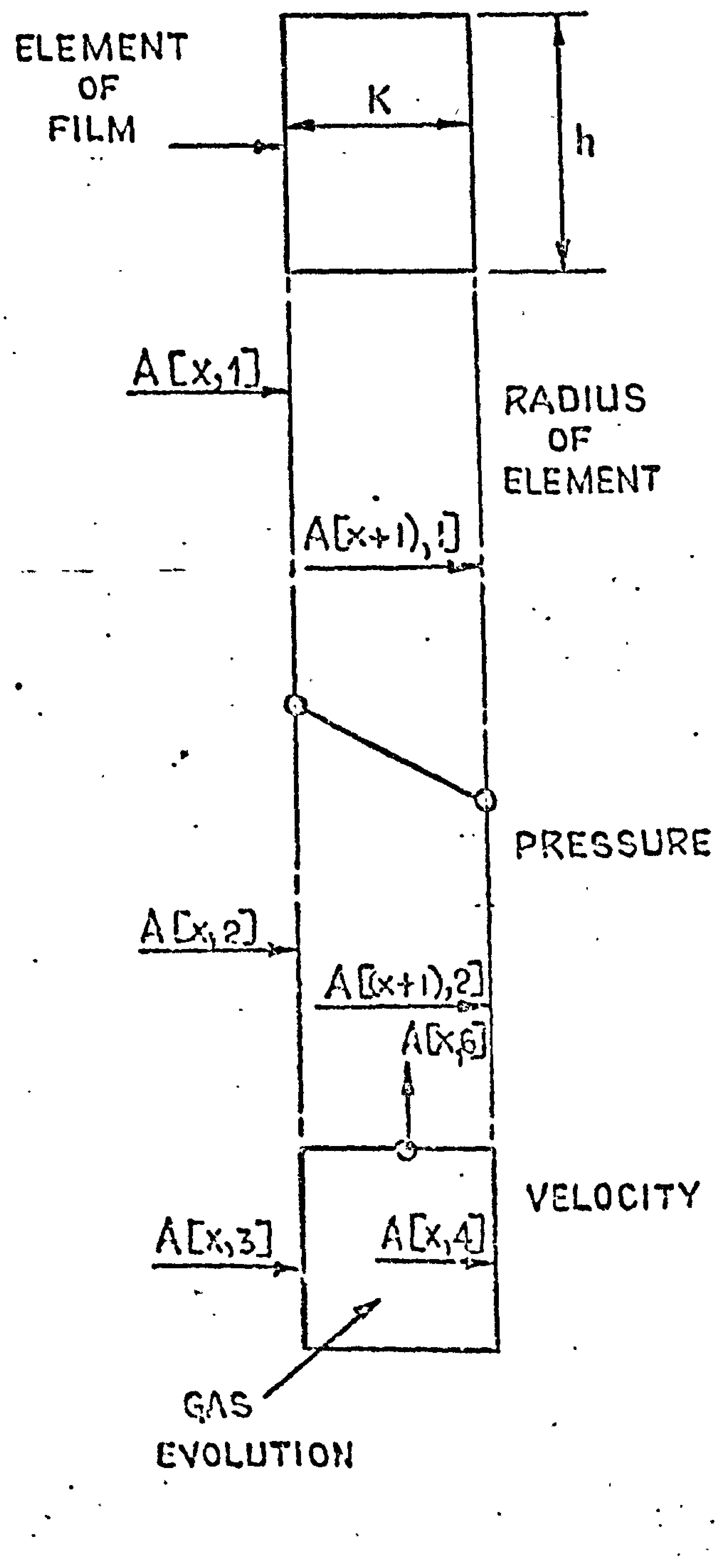

ELEMENT
OF
FILM
K
h
A[x,1]
RADIUS
OF
ELEMENT
A[(x+1),1]
PRESSURE
A[x,2]
A[(x+1),2]
A[x,6]
VELOCITY
A[x,3]
A[x,4]
GAS
EVOLUTION

FIG. B.1.2

A [X, 1] = inside radius element

A [(X + 1), 1] = inside radius of next element

A [X, 2] = pressure at inside radius of element

A [(X + 1), 2] = pressure at inside radius of next element

A [X, 3] = fluid velocity entering element

A [(X + 1), 3] = fluid velocity entering next element

A [X, 4] = fluid velocity leaving element

A [X, 5] = thickness of element

A [X, 6] = axial velocity of valve reed.

B.2 Calculation of Radial Oil Velocities

Fig. B.1.2 is again considered. If the velocity of oil flowing into the element was known, the velocity of oil flow out could be calculated by continuity of the mass flow volume.

The volume flowing into the element was equal to the circumferential area times the velocity. Using the appropriate symbols described previously, this was:

$$2.\pi.A [X, 1] \;.\; A [X, 5] \;.\; A [X, 3].$$

The volume flowing out of the element was therefore:

$$2.\pi.A [(X + 1), 1] \;.\; A [X, 5] \;.\; A [X, 4].$$

The outward flow of oil, however, had to be reduced because of the axial flow of oil due to the motion of the valve reed. This reduction was small because of the fact that axial velocities were shown to be approximately only one thousandth of the value of radial velocities.

The volume flowing axially was approximately equal to:

2/

$$2.\pi.A\,[X, 1] \;.\; K \;.\; A\,[X, 6]$$

By equating mass flow in and mass flow out, the value for oil velocity leaving the section was found to be:

$$A\,[X, 4] = \frac{(A\,[X, 1] \;.\; A\,[X, 3] - A\,[X, 6] \;.\; K \;.\; A\,[X, 1]\,/\,A\,[X, 5])}{(A\,[X, 1] + K)}$$

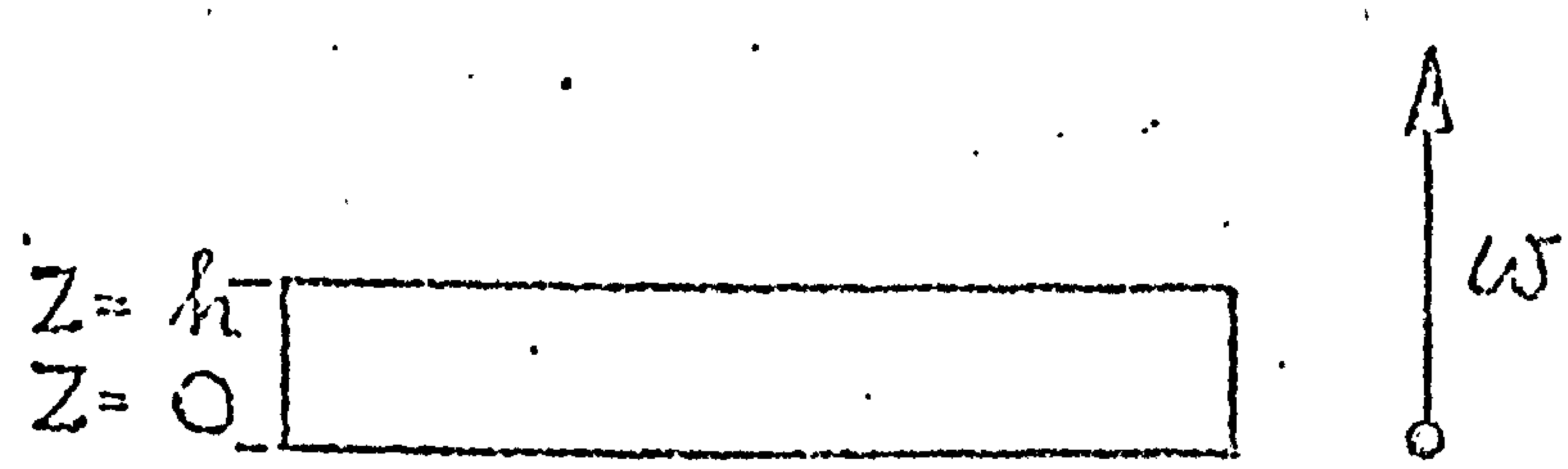
In order to determine the oil velocity entering the next element - $A\,[(X + 1), 3]$ - the value of the velocity of oil leaving the element was modified to account for the increase or decrease in film thickness due to the slope of the valve reed.

This modification was a simple linear relationship giving

$$A\,[(X + 1), 3] = \frac{A\,[X, 4] \;.\; A\,[X, 5]}{A\,[(X + 1), 5]}$$

B.3.1 Determination of Pressure Gradient

From the Navier-Stokes equations with thin film approximations as mentioned in section 2.2.

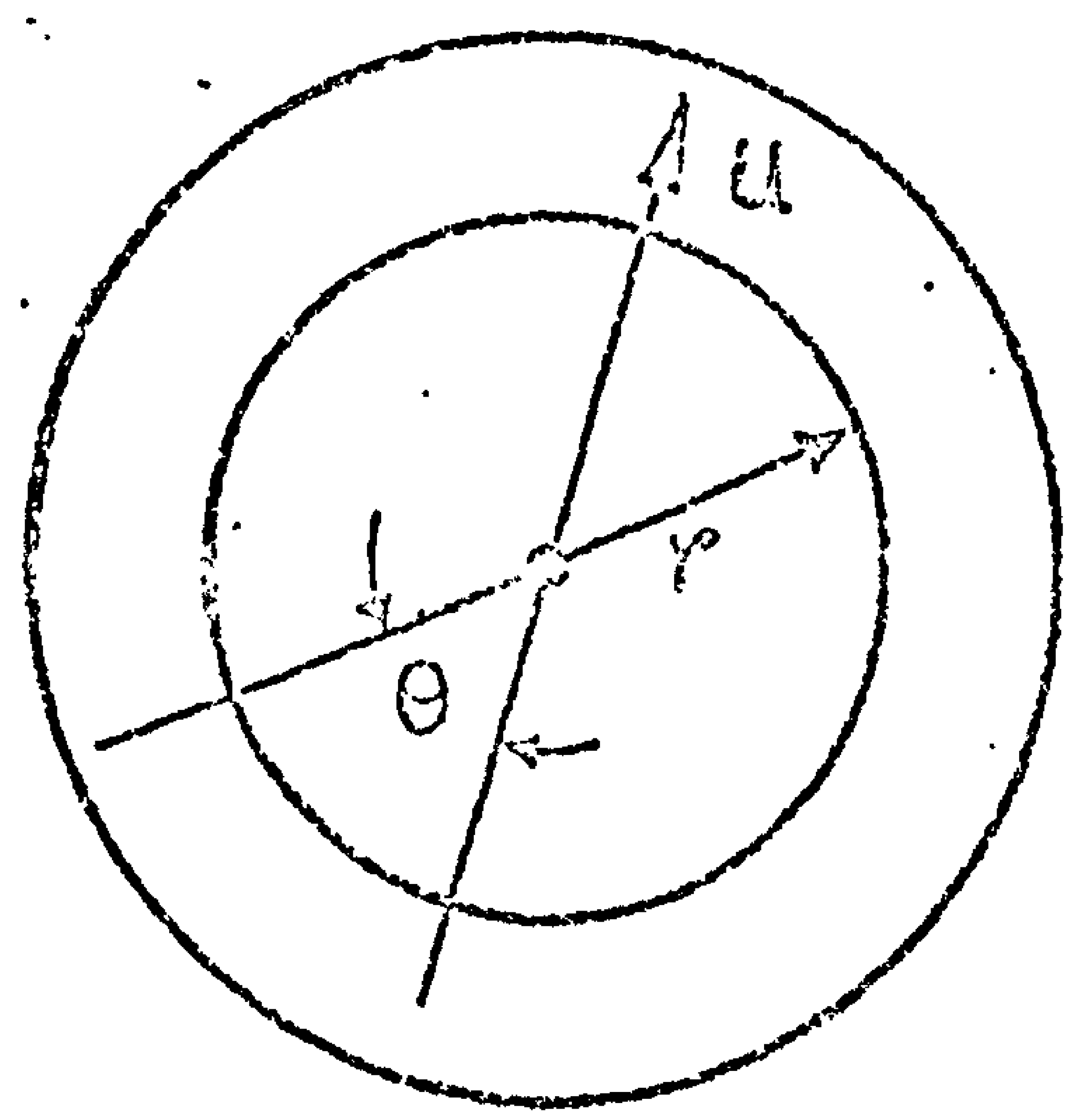


$$\frac{dp}{dr} = \mu.\frac{d^2u}{dz^2}$$

$$\frac{dp}{dz} = 0$$

$$\frac{dp}{d\theta} = 0$$

$$\frac{du}{dr} + \frac{u}{r} + \frac{dw}{dz} = 0$$



u = radial velocity
w = axial velocity

$$\frac{dp}{dr} = \mu \cdot \frac{d^2u}{dz^2}$$

$$\therefore \quad \frac{du}{dz} = z \cdot \frac{1}{\mu} \cdot \frac{dp}{dr} + C_1$$

$$\therefore \quad u = \frac{z^2}{2} \cdot \frac{1}{\mu} \cdot \frac{dp}{dr} + C_1 z + C_2$$

If the film thickness = h,

Since there was no slip at the solid surface,

When $z = 0$, $u = 0$ $\therefore$ $C_2 = 0$

When $z = h$, $u = 0$

$$\therefore \quad 0 = \frac{h^2}{2} \cdot \frac{1}{\mu} \cdot \frac{dp}{dr} + C_1 h$$

$$\therefore \quad C_1 = -\frac{1}{\mu} \cdot \frac{dp}{dr} \cdot \frac{h}{2}$$

The flow rate across any circumferential boundary can be expressed as $q = \bar{u}h$, where $\bar{u}$ was an average velocity. The flow rate was obtained by integrating the radial velocity, u, over the film thickness.

$$\therefore \quad \bar{u}h = \int_0^h u \cdot dz$$

$$= \left[\frac{z^3}{6} \cdot \frac{1}{\mu} \cdot \frac{dp}{dr} \cdot -\frac{1}{\mu} \cdot \frac{dp}{dr} \cdot \frac{h}{2} \cdot \frac{z^2}{2} \right]_0^h$$

$$\therefore \quad \bar{u}h = \left(\frac{h^3}{6} - \frac{h^3}{4}\right) \cdot \frac{1}{\mu} \cdot \frac{dp}{dr}$$

$$\therefore \quad \bar{u}h = -\frac{h^3}{12\mu} \cdot \frac{dp}{dr}$$

$$\therefore \quad \text{radial velocity} = -\frac{h^2}{12\mu} \cdot \frac{dp}{dr}$$

B.3.2 Calculation of Oil Pressures/

B.3.2 Calculation of Oil Pressures

Again referring to the Navier-Stokes equations, it was possible to evaluate the radial oil velocity as shown in the previous section, i.e. radial velocity $= - \frac{h^2}{12.\mu} \cdot (\frac{dp}{dr})$.

From Fig. B.1.2 it can be seen that for one particular element, $(\frac{dp}{dr})$ is equal to the pressure at one edge of the element minus the pressure at the other side of the element divided by the step width, i.e. $\frac{dp}{dr} = \frac{A[X, 2] - A[(X + 1), 2]}{K}$

The radial oil velocity in each element was taken to be the mean value of the flow velocity entering and the flow velocity leaving.

Again referring to Fig. B.1.2,

$$\text{radial velocity} = \frac{A[X, 3] + A[X, 4]}{2}$$

If "h", the height or thickness of the element is taken as $A[X, 5]$ then by substitution,

$$A[(X + 1), 2] = A[X, 2] - \frac{(A[X, 3] + A[X, 4])}{2 * A[X, 5] * A[X, 5] * F1}$$

where F1 is a function equal to $\frac{1}{12 * \mu * K}$

B.4 Iteration Technique

The calculation of the radial oil velocities and pressures at each point as explained in sections 3.2 and 3.3 depended on the initial inside boundary conditions of pressure, oil velocity and axial reed velocity being known. In fact, the only conditions known were the inside and outside pressures which, ignoring surface tension effects at present, were the cylinder and plenum chamber pressures for any point in the cycle.

Initial/

Initial small values were chosen for the inside oil velocity and axial reed velocity. Using these values, it was then possible to calculate the other fluid velocities and hence the fluid pressures at each point. The last value of oil pressure calculated should then equal the outside known value of pressure. If this was not the case, which it was virtually certain not to be, the initial guess of radial fluid velocity was altered and the process re-iterated until the two pressures agreed.

Once this was accomplished, a check had to be made to ensure that the forces on the valve due to the pressure difference across it were equal to the forces in the oil film due to the calculated pressures. (This is explained in section B.5.)

If not, the initial guess of the reed's axial velocity was changed and the whole process begun again. By changing the value of the reed's velocity, the effect was to alter the pressure profile in the oil film until its total area i.e. total force, balanced the external forces on the reed.

A flow diagram of the double iteration method can be seen as part of Fig. 5.2.

B.4.1 Instability During Iteration

It was found during the early stages of the development of the program that under certain conditions the process became unstable. That is, if the last value of pressure calculated was not equal to the known pressure, the correction made to the initial value of radial fluid velocity was too great. The computer, in attempting to correct this error, overcorrected in the opposite direction and so on until an infinitesmal value was produced in the computation. This problem was eventually solved in the following manner/

manner.

The inside radial velocity was given a very small value -0.01 mm/s. Because this would almost certainly be too small, it had to be increased for the next iteration. The increase was also made very small - usually 0.001. Because this next velocity would still be too small, it would have to be increased again. Obviously, if the increase was always going to be 0.001, the time to reach the true value would be very long. The first value was, therefore, remembered in the computer. If the next value was still too small, the increase was doubled. This doubling of the step increase carried on until the calculated value of pressure was greater than the known pressure. The step size was therefore halved and subtracted from the previous value.

Two possibilities could arise. The value could still be too large, i.e. on the same side of the datum, or too small, i.e. crossed the datum (the datum being the known value). If the computated pressure was still too large, the step size was doubled again and subtracted from the previous value. If the computated pressure was this time too small, the step size was halved again - since obviously, the initial radial velocity must have been close to the correct velocity - and added to the previous value. This process carried on until the correct figure was reached. The correct figure was considered as within 1% of the true value.

This method of quickly finding the radial oil velocities was also used when determining the axial reed velocities.

B.5 Check of Force Balance

The external force on the valve was due to the pressure difference/

difference across it.

For a discharge valve, the force above the valve was equal to the plenum chamber pressure times the area over the outside radius of the valve seat. (The pressure over the rest of the valve disc was balanced by an equal pressure under it). The force under the valve was equal to the cylinder pressure times the area of the valve port.

The external force on a discharge valve was therefore:

$$\pi \, . \, A[(s+1), \overset{1?}{2}]^2 \, . \, P2 - \pi \, . \, A[1, 1]^2 \, . \, P1 \qquad (B.5.1)$$

where P1 = cylinder pressure

P2 = plenum chamber pressure

s = number of finite elements.

The internal force in the valve was equal to the sum of the forces in each element.

For each element,

internal force = Mean Pressure x Area

Now, $$\text{Mean Pressure} = \frac{A[X, 2] + A[(X+1), 2]}{2}$$

$$\text{Area} \doteq 2 \, . \, \pi \times \text{Mean radius} \times \text{element width}$$

$$\doteq 2 . \pi . \frac{A[X, 1] + A[(X+1), 1]}{2} \, . \, K \, .$$

The total internal force of the oil in the valve was:

$$\sum_{X=1}^{s} \frac{\pi . K .}{2} \, (A[X,2] + A[(X+1), 2]) \, . \, (A[X, 1] + A[(X+1), 1]) \qquad (B.5.2)$$

For the check on the force balance, the total external force on the valve - equation B.5.1 - must equal the total internal force due to the oil pressure - equation B.5.2.

B.6.1 Initial Determination of Oil Film Thickness/

B.6.1 Initial Determination of Oil Film Thickness

Because the valve reed was initially considered to be rigid, an attempt was made to calculate the oil film thickness by determining the minimum thickness that the surface tension could withstand.

If Fig. B.6.1 is considered.

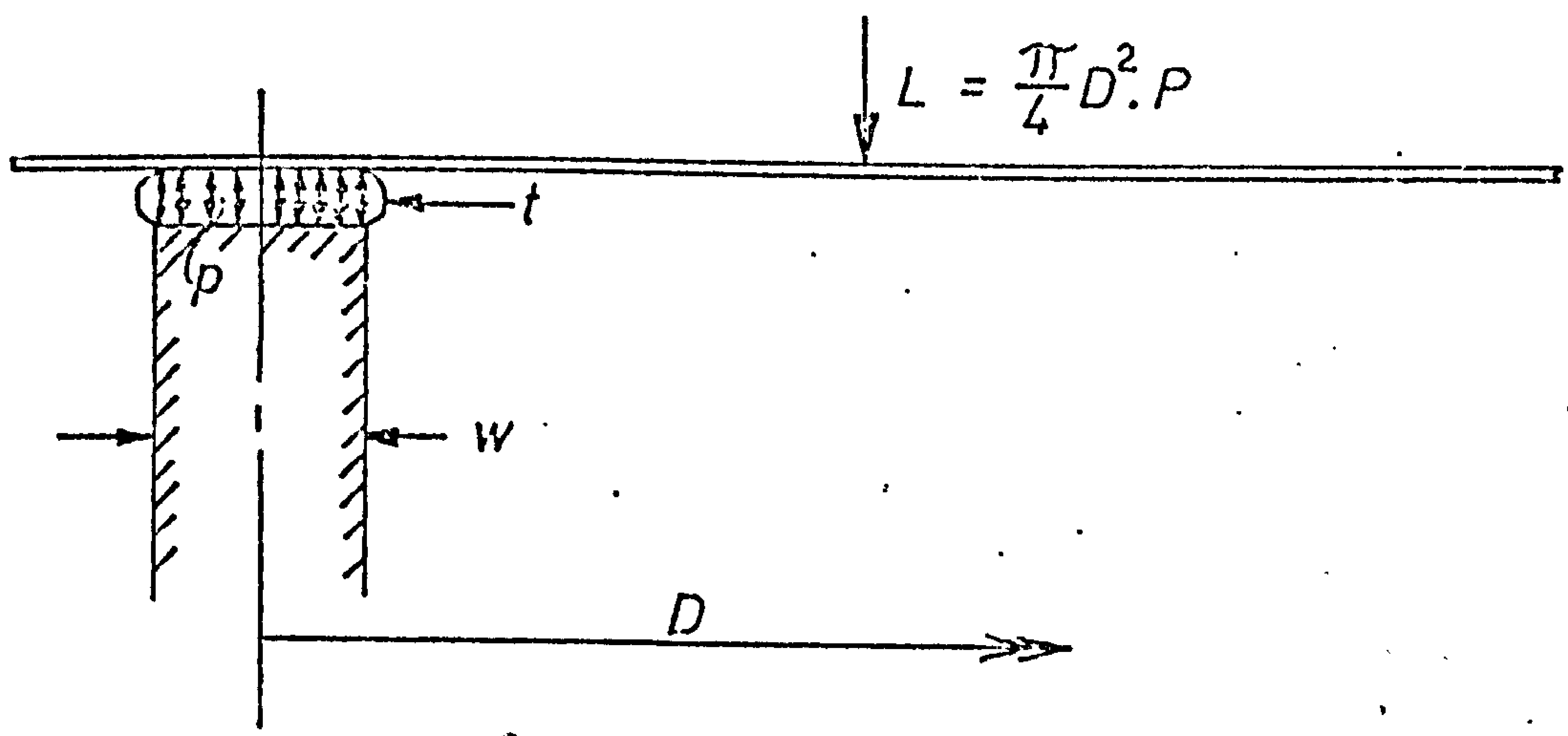


FIG. B.6.1

Internal pressure in oil film = $p = \frac{2 . \gamma}{t}$

where γ = surface tension.

Area of oil film $= \pi \times D \times W$

$\therefore$ Load supported by oil film $= \pi \times D \times W \times \frac{2 . \gamma}{t}$

Load applied to valve disc $= \frac{\pi}{4} \times D^2 \times p$

where p = pressure difference across valve.

$$\therefore \quad \frac{\pi}{4} \times D^2 \times p = \pi \times D \times W \times \frac{2 . \gamma}{t}$$

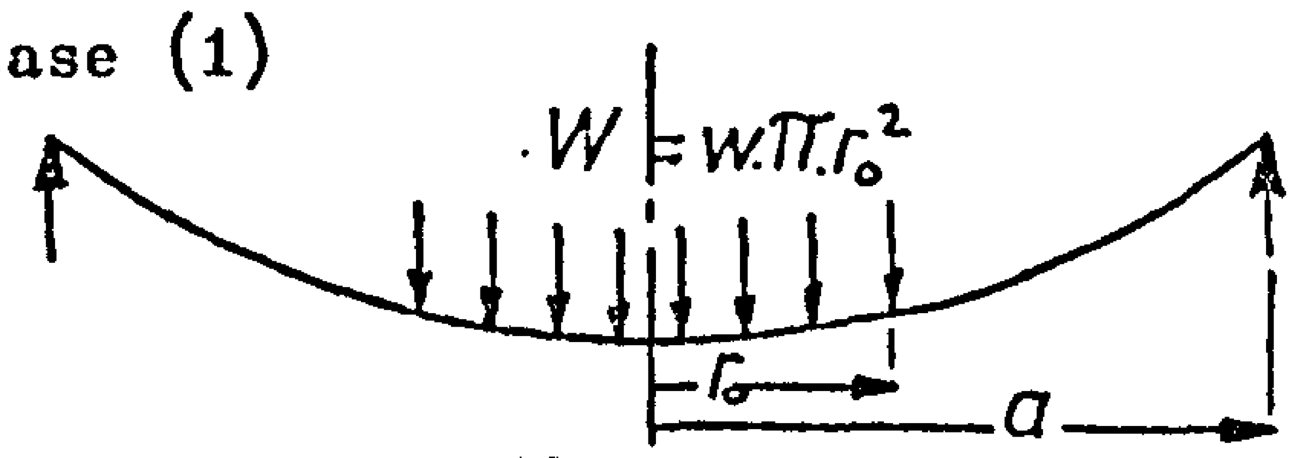
$$\therefore \quad \text{Thickness of film} = t = \frac{8 . W . \gamma}{p . D}$$

B.6.2 Calculation of Oil Film Thickness

This was accomplished by considering the oil to fill the gap produced by the deflected shape of the reed due to the pressure difference across it.

The deflected shape of the reed was derived by summing two of the standard solutions found in Roark (20).

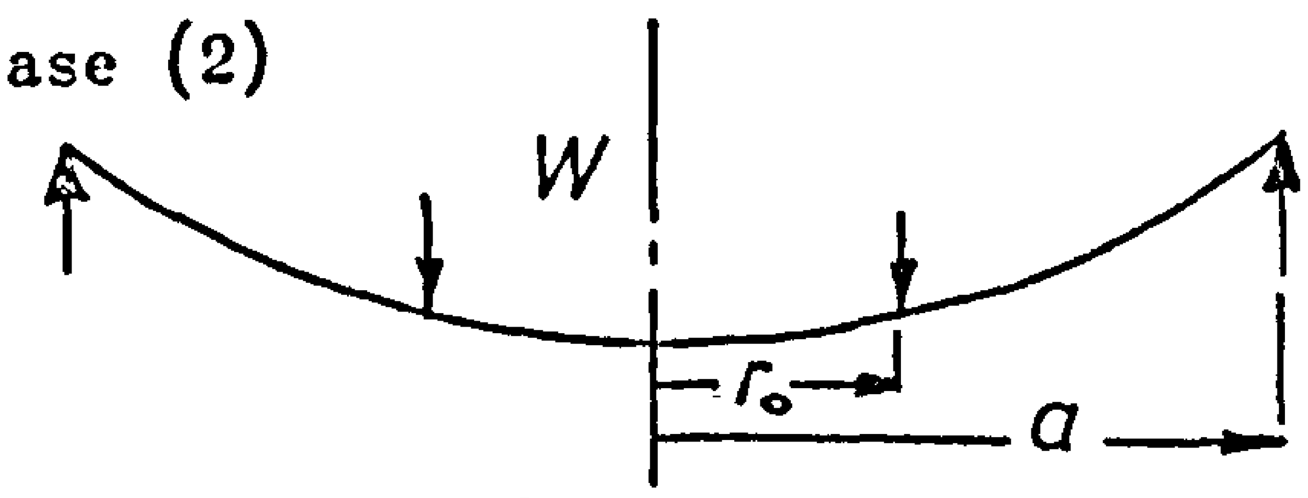
For case (1)



Edges supported. Uniform load over concentric circular area of radius r_o.

$$y = \frac{-3W.(m^2 - 1)}{16.\pi.Em^2.t^3}\left[\frac{(12\,m + 4)\,(a^2 - r^2)}{(m + 1)} - \frac{2\,(m - 1)\,r_o^2\,(a^2 - r^2)}{(m + 1)\,a^2} - (8r^2 + 4\,r_o^2)\log\frac{a}{r}\right]$$

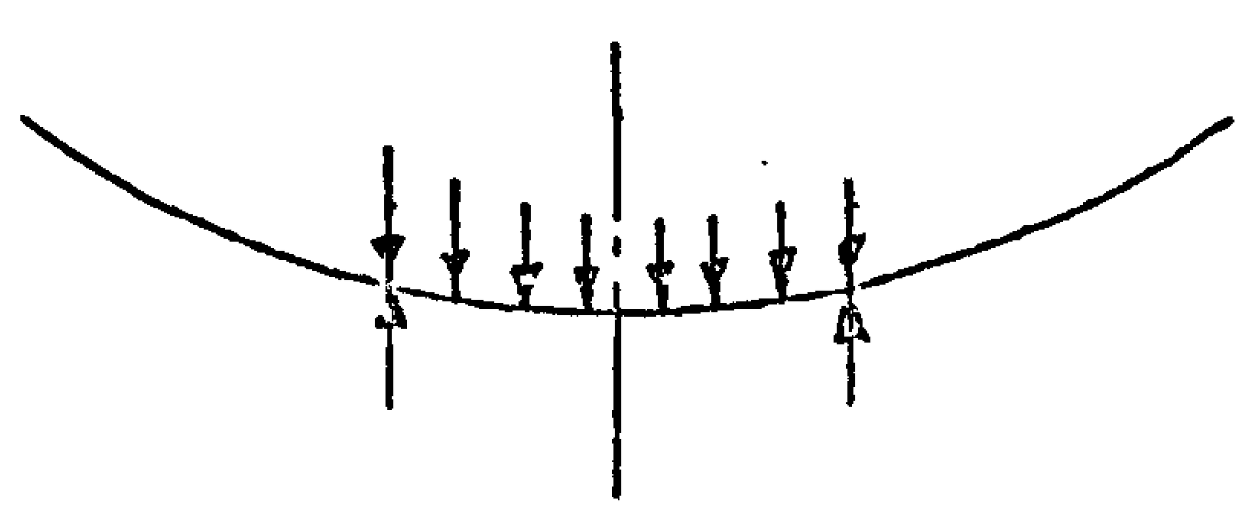
For case (2)



Edges supported. Uniform load on concentric circular ring of radius r_o.

$$y = \frac{-3W\,(m^2 - 1)}{2.\pi.Em^2.t^3}\left[\frac{(3\,m + 1)\,(a^2 - r^2)}{2\,(m + 1)} - (r^2 + r_o)\log\frac{a}{r} \quad \frac{-(m - 1)\,r_o^2\,(a^2 - r^2)}{2\,(m + 1)\,a^2}\right]$$

By substracting (2) from (1) we are left with the condition



Edges free. Uniform load over concentric circular area of radius r_o. Supported by uniform load on concentric circular ring of radius r_o.

Collecting/

Collecting terms gives:

$$y = \frac{-3W\ (m^2 - 1)}{2 \quad Em^2\ t^3} \quad \frac{(a^2 - r^2)}{(m + 1)} \quad \frac{(12\ m + 4)}{8} - \frac{(3\ m + 1)}{2}$$

$$+ \frac{(m - 1)\ r_o^2 \cdot (a^2 - r^2)}{4\ (m + 1)\ a^2}$$

$$- \left(r^2 + \frac{r_o^2}{2}\right) \log \frac{a}{r} + (r^2 + r_o^2) \log \frac{a}{r}$$

$$\therefore \quad y = \frac{-3W\ (m^2 - 1)}{8 \quad Em^2\ t^3} \quad \frac{(m - 1)\ r_o^2\ (a^2 - r^2)}{(m + 1)\ a^2} + 2\ r_o^2 \log \frac{a}{r}$$

y = Deflection measured from tip.

W = Pressure difference x r_o^2.

m = reciprocal of Poisson's ratio.

E = Young's modulus of elasticity.

t = thickness.

r_o = inside radius of port.

a = outside radius of disc.

r = any radius where $r_o < r < a$.

By knowing the maximum deflection, i.e. when $r = r_o$, it was possible to calculate the deflection up from the inside radius (and hence oil thickness) as: Deflection = y (max) - y (r).

Also, a constant must be added to account for the effect of the surface finish.

B.6.3 Slope of Reed

The equation for the valve reed deflection when subjected to a pressure differential was fairly complex as shown in section B.6.2.

In order to simplify the working of the program, the slope at/

at the inside edge could be calculated and this slope taken as linear over the width of the seat. For valves where the seat width is small, as is the normal case, then this approach gives a fairly good approximation as is shown in Fig. B.6.2(a).

If the valve seat is wide, however, this method is inaccurate because of the excess pressure over a larger area on top of the valve and because of the strain energy in the valve itself. Fig. B.6.2(b) shows the deflected form for a fairly wide valve seat.

The two seats compared are test seats No. 4 and 2. Although in most practical cases, this error is trivial, the fact that some experimental tests were carried out on valves with fairly wide seats meant that the real deflection form had to be included in the theoretical approach.

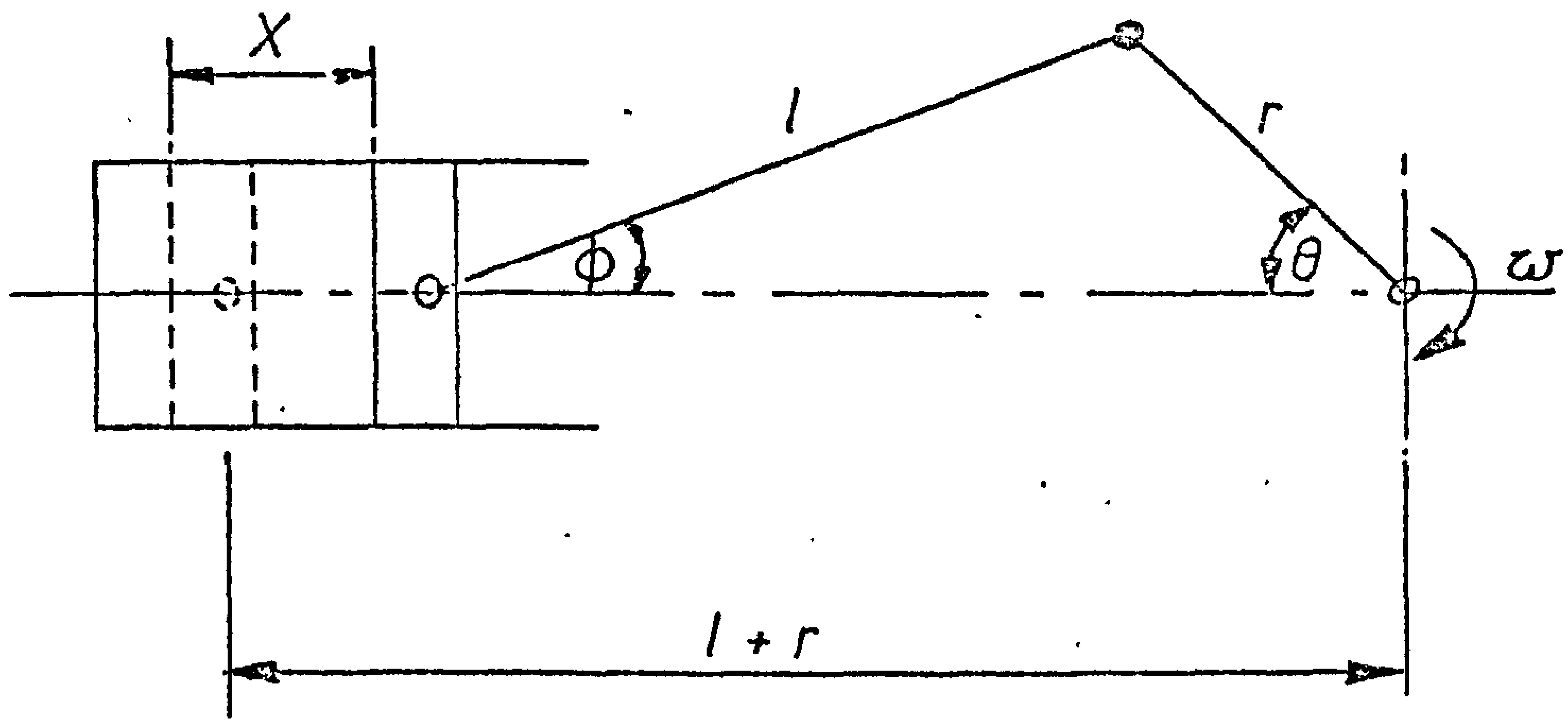
Because the errors in the deflected shape of a valve with narrow seats are negligible, the theoretical model can be applied not only to disc valves, but also to cantilever type valves. In cantilever type valves, the deflected shape is influenced by strain energy due to the length of the valve and is therefore not concentric. Over the small width of the seat, however, the difference is negligible and can be ignored.

B.7 Rate of Change of Pressure Across Valve/

COMPARISON BETWEEN TRUE DEFLECTION AND DEFLECTION WITH LINEAR DISPLACEMENT ASSUMED (TWO EXPERIMENTAL VALVE SEATS SHOWN)

FIG. B.6.2

B.7 Rate of Change of Pressure Across Valve



The movement from TDC (X) = $r + l - (r \cos\theta + l \cos\phi)$

$$\text{Now, } l \sin\phi = r \sin\theta$$

$$\therefore \quad \cos\phi = \sqrt{1 - \left(\frac{r}{l}\right)^2 \sin^2\theta}$$

$$\text{and } \frac{d \cos\phi}{d\theta} \doteqdot -\frac{1}{2}\left(\frac{r}{l}\right)^2 \sin 2\theta$$

$$\therefore \quad \frac{dx}{d\theta} = r.\left[\sin\theta + \frac{1}{2}\left(\frac{r}{l}\right) \sin 2\theta\right]$$

$$\text{and } \quad \frac{dx}{dt} \doteqdot \omega.r.\left[\sin\theta + \frac{1}{2}\left(\frac{r}{l}\right) \sin 2\theta\right]$$

Since this is approximate, we must modify the expression for piston movement by using,

$$x = \int \frac{dx}{d\theta} \cdot d\theta + \text{constant}$$

$$\therefore \quad x = -r.\left[\cos\theta + \frac{r}{4l} \cos 2\theta\right] + \text{constant}$$

when $x = 0$, $\cos\theta$ and $\cos 2\theta = 1$

$\therefore$ 0/

$\therefore \quad 0 = - r \quad 1 + \frac{r}{41} \quad + \text{constant}$

$\therefore \quad \text{constant} = r \quad 1 + \frac{r}{41}$

$\therefore \quad x = - r \quad \cos \quad + \frac{r}{41} \cos 2 \quad + \quad r \quad 1 + \frac{r}{41}.$

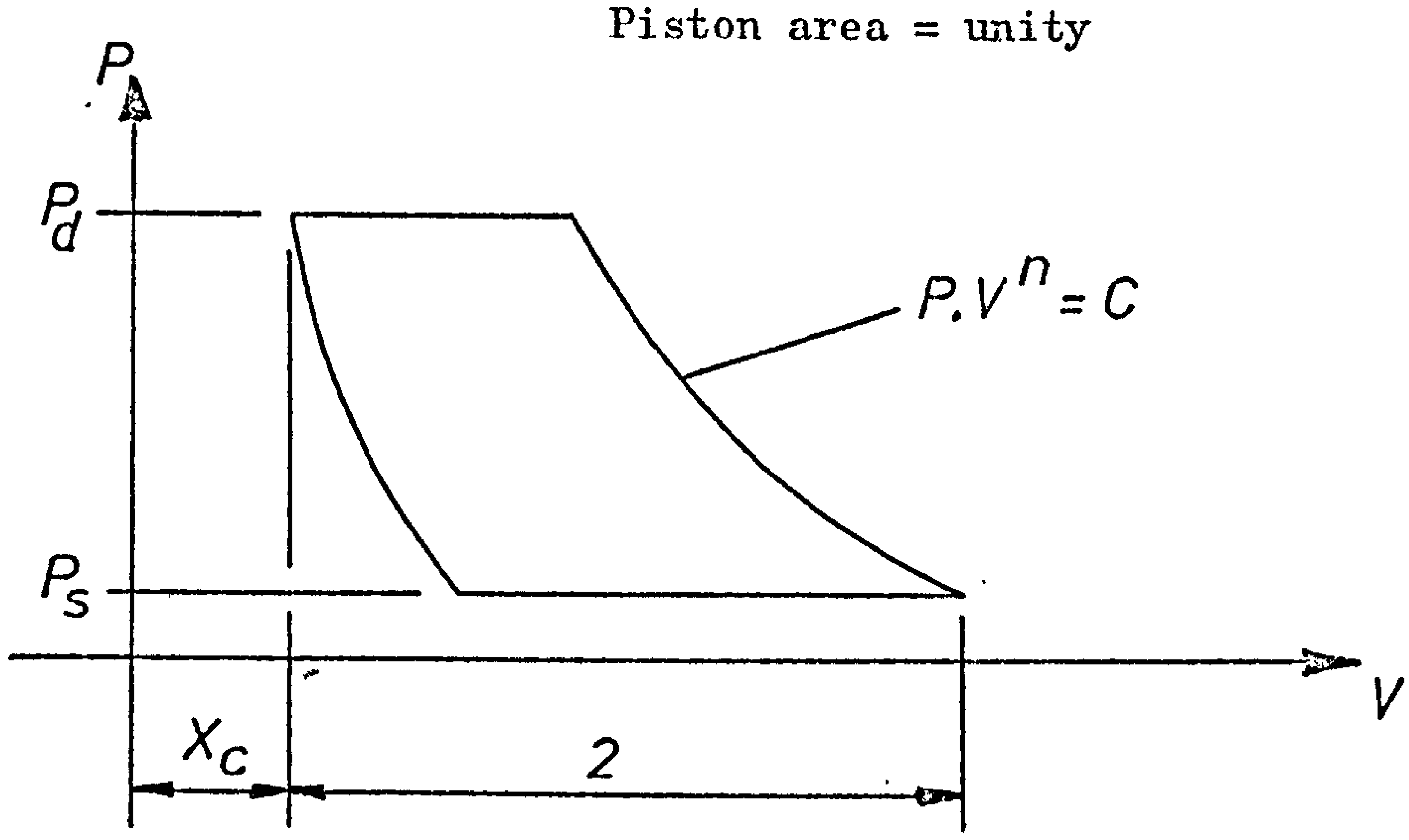
During compression and expansion,

$PV^n = C$ where C is fixed by initial conditions

Stroke = 2 units

x_c = 2 x clearance fraction

Piston area = unity



In the above diagram

$PV^n = P_s \times (x_c + 2)^n$ for discharge valve

$= P_d \times x_c^{\ n}$ for suction valve

and $V = [1 + \frac{r}{41}] - [\cos\theta \quad + \frac{r}{41} \cdot \cos 2\theta] + x_c$

since unity area and radius were assumed.

Knowing the initial conditions, P is easily computed from θ.

Also $\frac{dP}{dt} = \frac{dP}{dV} \times \frac{dV}{d\theta} \times \omega$

where $\frac{dV}{d\theta} = [\sin\theta \quad + \frac{r}{21} \sin 2\theta]$

$\frac{dP}{dV} = - n \; C \; V^{-(n + 1)}.$

B.8 Velocity of Reed (surface tension film complete)

When the film was complete at both inside and outside edges due to the effect of surface tension, the velocity of the reed at the inner edge was calculated as follows:

Fig. (a) represents the oil film trapped between the valve and its seat and Fig. (b) the case of a finite time, t, later. The assumptions made were that the valve reed slope was linear and the oil/gas boundary edges are straight.

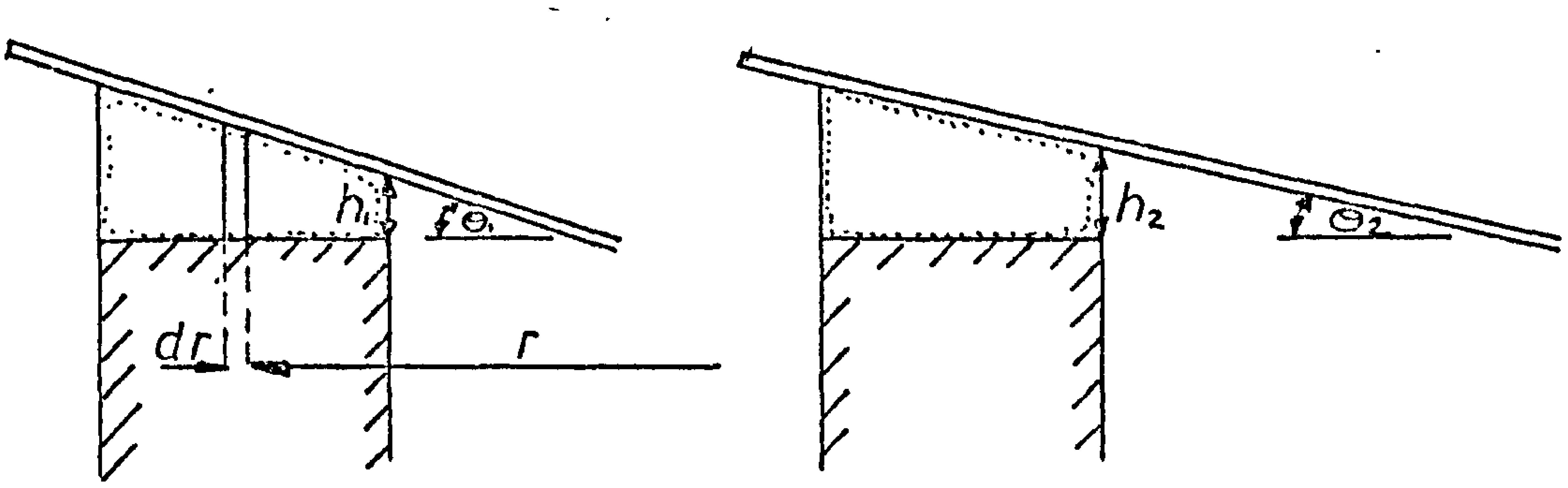


FIG. (a) FIG. (b)

For Fig. (a),

$$\text{Height of annular ring} = h_1 + (r - r_1).\theta$$

$$\therefore \text{ Volume of annular ring} = 2.\pi.r.\delta r.[\,h_1 + (r - r_1).\theta\,]$$

$$\therefore \text{ Volume of film} = \int_{r_1}^{r_2} 2.\pi.r.[\,h + (r - r_1).\theta\,]\ dr$$

$$= \int_{r_1}^{r_2} [\,2.\pi.r.h + 2.\pi.r^2\theta - 2\pi r.r_1.\theta\,]\ dr$$

=/

$$= \Big[\pi . r^2 . h + \tfrac{2}{3} . \pi r^3 . \theta - \pi . r^2 . r_1 \theta\Big]_{r_1}^{r_2}$$

$$= (\pi . r_2^2 . h + \tfrac{2}{3} . \pi . r_2^3 . \theta - \pi . r_2^2 . r_1 . \theta)$$

$$- (\pi . r_1^2 . h + \tfrac{2}{3} . \pi . r_1^3 . \theta - \pi . r_1^3 . \theta)$$

$$= \pi . h \quad (r_2^2 - r_1^2) + \pi . \theta \ (\tfrac{2}{3} r_2^3 + \tfrac{1}{3} r_1^3 - r_2 r_1)$$

Let $A = (r_2^2 - r_1^2)$ and $E = (\frac{2}{3} r_2^3 + \frac{1}{3} r_1^3 - r_2 r_1)$

$\therefore$ Volume of film $= \pi \ (h \times A + \theta \times E) = \text{constant}$

$\therefore \ A (h_2 - h_1) + E \ (\theta_2 - \theta_1) = 0$

$\therefore \ h_2 - h_1 = \frac{E}{A} (\theta_1 - \theta_2)$

For time t, rate of change of thickness i.e. inner velocity of reed, $= - \frac{E}{A} \times$ rate of change of slope.

$\therefore$ A (1, 6) $= - \frac{E}{A} \times$ rate of change of slope.

APPENDIX C

MISCELLANEOUS

C.1 Effect of Surface Tension

Tests on Clavus 27 oil, which was the oil used in the compressor apparatus, showed that the surface tension force at normal running temperatures was approximately 31×10^{-5} N/cm. The tests were carried out on a Cambridge-de Nouy Tensiometer. The apparatus is based on the instrument designed by Dr. P.L. Nouy for the measurement of the surface tension of liquids. It consists, in its simplest form, of an apparatus for pulling a 4 cm circumference platinum-iridium wire from a liquid and measuring the restraining force.

If a typical seat is, say 10 mm inside diameter, the surface tension force around the circumference holding a disc on the seat will be:- Surface Tension/cm x circumference (cms)

$$= 31 \times 10^{-5} \times \pi \times 1$$

$$= 10^{-4} \text{ N}$$

which is negligible when compared to a stiction force of the order of 10 N measured on the force pull-off apparatus.

C.2 Gravitational Effect of Disc

The test discs were all 16 mm diameter shim steel ranging in thickness from 0.13 - 0.38 mm. A typical disc of 0.3 mm thickness would have a volume of:

$$\frac{\pi}{4} \times 16^2 \times 0.3 = 60 \text{ mm}^3$$

The density of steel $= 7.9 \times 10^3$ kg/m^3

The disc would therefore have a weight of only 4.675×10^{-3} N. The pressure difference required to move this disc against gravity for a typical port diameter of 6 mm would be 0.16 kN/m^2 which is negligible/

negligible compared to a pressure equivalent to the stiction force of (typically) 50 kN/m^2.

C.3 Inertia Effect

The velocity of the valve disc when it is still in contact with the oil film has been observed in both the compressor apparatus and the force pull-off apparatus to be very small. However, considering the very worst condition, i.e. the acceleration from the bottom seat to the top seat when the disc has broken free from the oil film:

$$\text{Displacement of disc} = 0.25 \text{ mm}$$

$$\text{Time taken} = 0.5 \text{ ms}$$

$$\therefore \quad \text{Acceleration} = 2 \text{ m/s}$$

$$\text{Inertia Force} = \text{Mass x acceleration}$$

$$= \frac{4.675 \times 10^{-3}}{9.81} \times 2$$

$$= 9.53 \times 10^{-4} \text{ N}$$

This force is equivalent to a pressure difference across the valve of 0.03 kN/m^2 which is again negligible.

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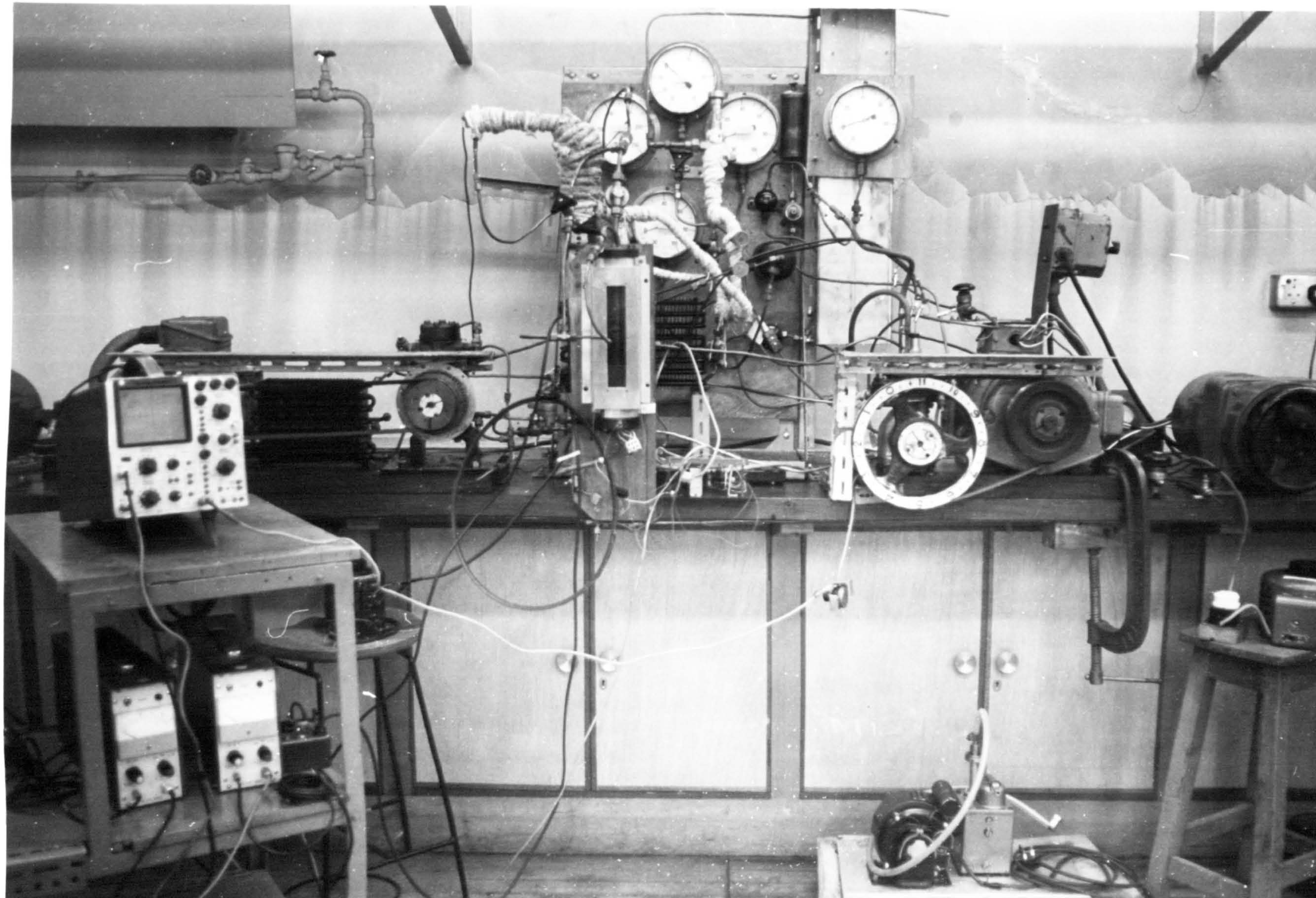
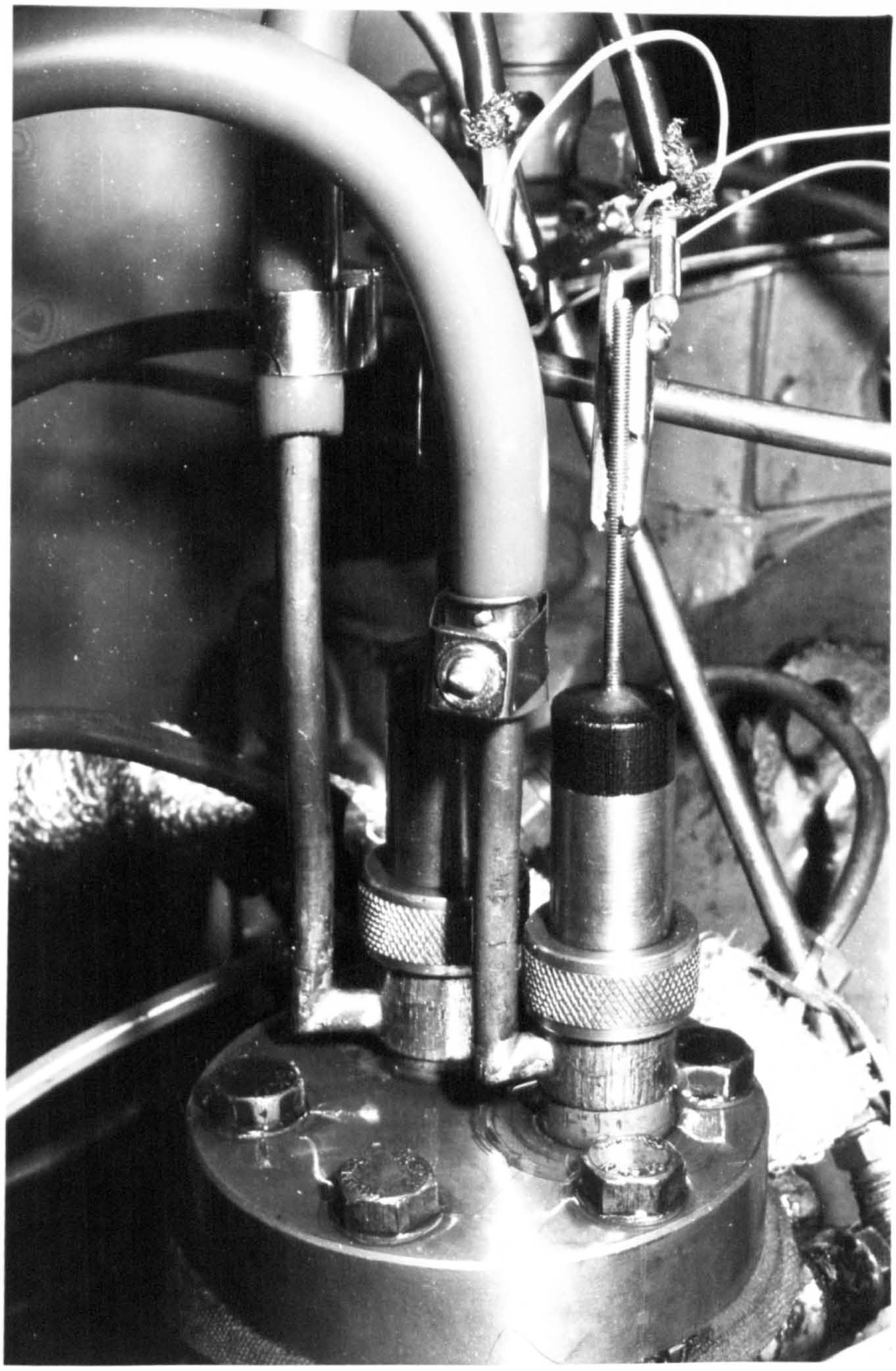
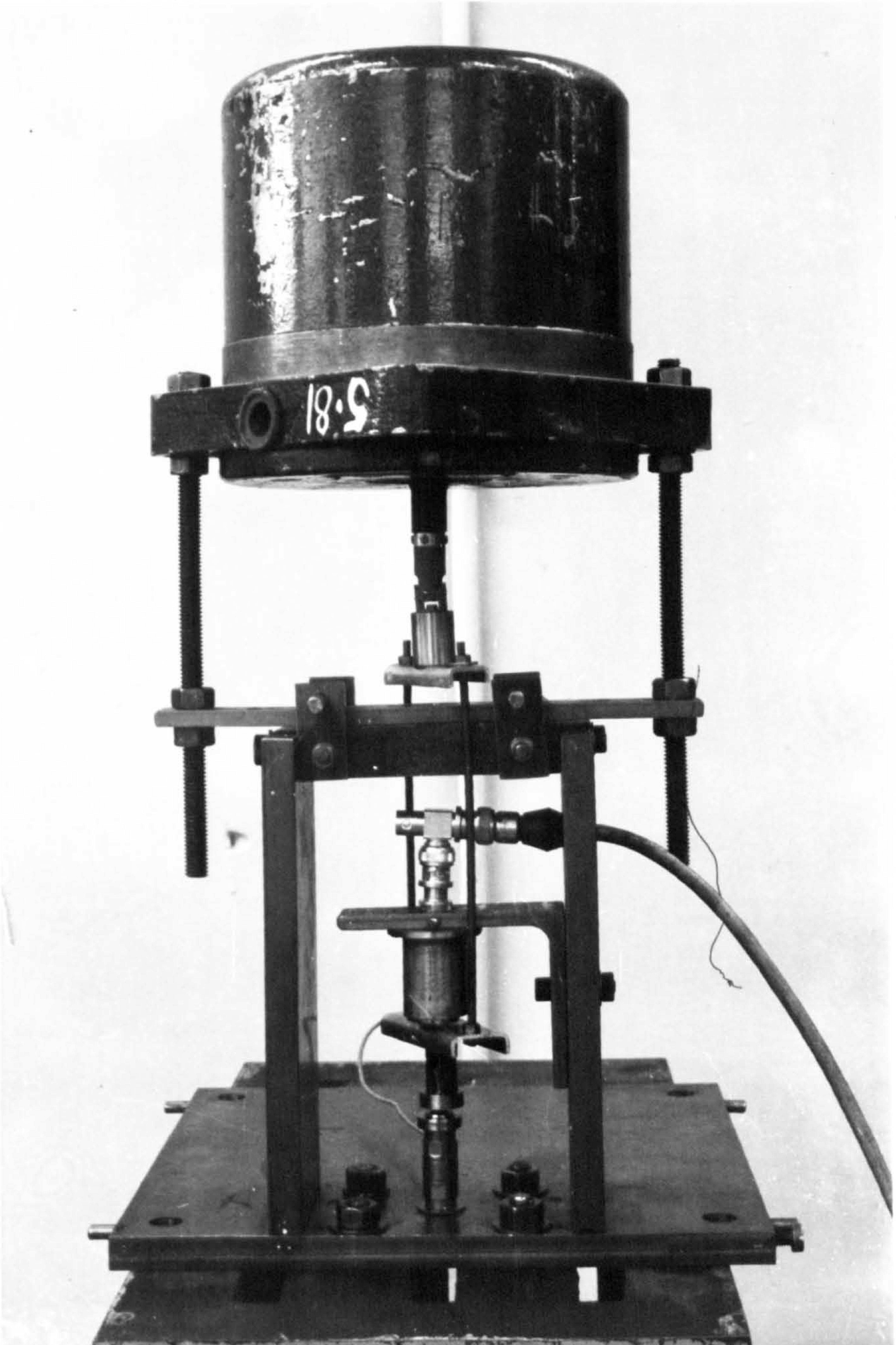


GENERAL VIEW OF COMPRESSOR TEST CIRCUIT PLATE 1



CLOSE-UP OF CYLINDER HEAD

PLATE 2



FORCE PULL-OFF APPARATUS

PLATE 3

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