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An Approach to Computer Modelling of Reciprocating Compressors

S. Lawson

R. J. L. McLaren

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S. Lawson
Senior Engineer

R.J.L. McLaren
Engineering Director

Prestcold Limited, Reading, U.K.

ABSTRACT

Compressor performance is dependent on pressure drops, clearance volumes and valve timing. Historically, volumetric efficiency was worked out using simple theoretical equations modified by empirical factors and power by calculating valve pressure drops based on the ratio of valve port to cylinder area, and taking into consideration piston velocities. Early computerised performance techniques retained some or all of these methods. Subsequent computer models took into consideration the dynamics of the compressor valve systems and pressure drops across the valve by consideration of simple orifice calculations modified by coefficients of discharge and drag. The coefficients used in these models were empirically derived and often modified to match the prediction of the model with results from compressors.

This paper describes a model which uses text-book coefficients which do not require modifications once the geometry of the compressor and valve system have been established. The computer model developed allows detailed calculation of pressure drops through the various flow elements described by the position of the suction and discharge valves and piston, and from this allows the forces acting on the valves to be derived. The dynamics of the reed are modelled by techniques which allow the reeds to be described as beams, plates or cantilevers, whose motion can be described as a point mass, the dynamics of which are described by consideration of the configuration and stiffness of the valve being modelled. This technique allows for the full dynamic motion of various valve forms to be simulated using assumptions similar to those described in Reference (1). The interaction between valve systems and stationary stops is taken into consideration by the model, thus allowing a better definition of valve dynamics than would otherwise be the case.

INTRODUCTION

This paper describes a computer model which was developed to predict the performance of reciprocating compressors. It was intended that the model should be as simple as possible, while being valid for any valve configuration which may be used. The model was also to operate without the need for flow factors.

Some of the criteria used within the model are as follows:-

a) The full dynamics of the valves are considered.

b) For pressure drop purposes the valves are considered as a series of restrictions and the gas forces calculated based on instantaneous pressures.

c) No heat transfer within the cylinder is considered, but, in the case of suction cooled compressors, the effect of heat transfer from the motor is computed.

d) While the valves are closed the process is Isentropic.

e) The gas, when flowing through valves, is considered incompressible.

f) Pressure drops through all significant restrictions are considered.

g) The gas is defined by index of compression, specific heat (Cp) and gas constant R.

THE MODEL(s)

Fig. FC1 shows the outline of the compressor model. In order to correctly predict compressor performance the computer must first be provided with the physical characteristics of the compressor and the operating parameters, Fig.1. Inputs such as the valve reed mass and stiffness are obtained from a separate model.

INPUTS

Operating speed.
Motor efficiency.
Number of cylinders.
Number of cylinders per bank.
Cylinder bore.
Stroke.
Conrod length.
Clearance at T.D.C. (mm)
Clearance volume.
Head inlet area.
Head outlet area.

Suction Valve:-
Number of assemblies.
Effective mass in 2 modes.
Stiffness in 2 modes.
Flow path around ports.
Port minimum section.
Port plan area.
Area under reed around port.
Reed tip lift.
Reed lift at port centre when tip hits.
Stiction factor.

Discharge Valve:-
Number of assemblies.
Effective mass (or mass as func. lift).
Stiffness (or stiff. as func. lift).
Retainer effective mass.
Retainer stiffness.
Flow path around ports.
Port minimum section.
Port plan area.
Area under reed around port.
Retainer lift at port centre.
Stiction factor.

Conditions:-
Refrigerant.
Suction pressure.
Discharge pressure.
Suction temperature.

Fig.1

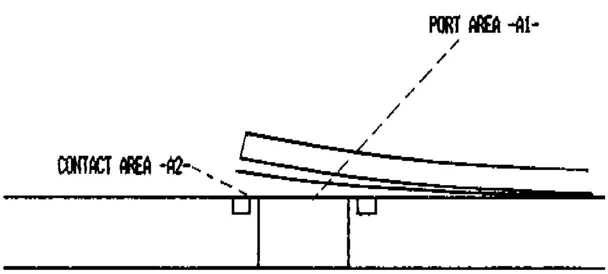


Fig 2 DISCHARGE CONTACT AREA

Of the inputs in Fig.1 the only area where the data is not self evident is the stiction factors. For the purpose of this model the stiction factor is defined as the ratio of cylinder pressure to head pressure when the valve starts to open. This ratio is best measured on test, but this is obviously difficult at the design stage. The factor may be calculated by considering the area of the reed exposed to the port pressure and the contact area. For the discharge valve, Fig.2:-

$$SF=(A1+A2)/A1 \qquad (1)$$

The value of SF obtained using equation (1) will tend to be too large, but for most of the compressor's operating range this will not significantly effect flow or power prediction. The use of over correcting stiction factors will, however, over predict the initial valve velocities, impact and bending stresses, causing the designer to build in safety factors. Experience of testing and modelling will allow modification of SF depending on valve thickness, operating speed, valve geometry and surface finish; reference (2) may be of interest.

THE VALVE MODEL

The main compressor model considers the valves to act as point masses and therefore the value of this mass must be established. The mass of a flexing reed is calculated on the basis that the momentum of the point mass is equal to the sum of momentums for elements along the length of the reed.

From the profile and thickness of the valve reed the valve model calculates the deflected form of the valve reed for the required bending modes. The valve stiffness is defined as the force per unit deflection of the reed at the port centre and is obtained directly from the deflected form. The effective mass of the valve reed for each bending mode is calculated:-

$$Mp= \Sigma_0^{\ell}(Mi \times Yi) /Yp \qquad (2)$$

this assumes that in any single bending mode the relative velocities of elements is proportional to their relative deflections.

If Mp is inserted into:-

$$Fn=1/(2 \times pi) \times SQR(Sp/Mp) \qquad (3)$$

then the natural frequency is given for the specified bending mode.

Where:-

Mi is the mass of a reed element.
Yi is the deflection of a reed element.
Yp is the deflection of the reed at the valve port centre.
Mp is the effective reed mass.
Sp is the reed stiffness.
Fn is the natural frequency.

Having computed the deflected form, the valve model is able to define the valve stresses as a function of reed lift at port centre.

MAIN COMPRESSOR MODEL

SUCTION REED DYNAMICS

When a rigid ring valve is used on a compressor the calculation of valve stiffness and mass is simple, the mass is the actual mass of the ring plus 1/3 of the spring masses, stiffness is the stiffness of the springs. When the plate hits the stop it is reasonable to assume that the valve ring either stops dead or rebounds.

If a cantilever valve reed with tip stop is used, Fig.3, the motion becomes more complex. The reed moves away from the valve plate until it reaches the tip stop, at this instant some energy is transferred to the stop thus reducing the reed velocity, the reed also goes into a second bending mode causing a change in the effective mass, resulting in a further change in velocity. The change in velocity is accompanied by a change in reed stiffness.

The model considers the moving valve member as acting as a point mass, of stated stiffness, concentrated at the port centre. If a stop is reached the values of mass and stiffness are updated and the velocity recalculated; Fig.FC2.

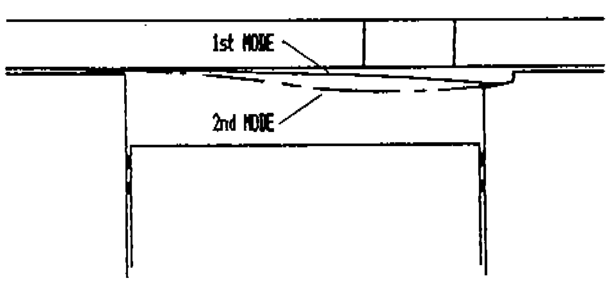


Fig 3 CANTILEVER SUCTION REED

Throughout the model every effort has been made to use only simple formulae and therefore the change of velocity of a point mass as the reed hits the stop poses problems.

The first observation is that if the effective mass of the reed changes when the reed hits the stop and there is no energy transfer, then in order to conserve momentum:-

$$M1 \times U1 = M2 \times U2 \qquad (4)$$

but we know that if the mass of the reed were concentrated above a stop of infinite mass then:-

$$U2 = 0 \qquad (5)$$

On the other hand, if the mass of the reed were concentrated at the reed root, then there would be no energy transfer, hence (4).

If a reed had a mass concentrated at the reed tip, then when the tip hits the stop:-

$$Yp/Yt = 1 \qquad (6)$$

If the mass were concentrated at the root, then when the tip hits the stop:-

$$Yp/Yt = 0 \qquad (7)$$

The best simple model which was considered for the change in velocity as the reed hits the stop was:-

$$U2 = U1 \times (M1/M2) \times (1 - Yp/Yt) \qquad (8)$$

Where:-

- M1 is the reed mass in first mode.
- M2 is the reed mass in second mode.
- U1 is the reed velocity prior to impact.
- U2 is the reed velocity after impact.
- Yt is the reed tip deflection.
- Yp is the reed deflection at port centre on impact.

By using the ratio of deflections, rather than the distance of the centre of mass from the tip, the model can cope with both flexing valve reeds and rigid plates.

When the valve member leaves the tip stop it is assumed that it does not stick to the stop and that there is no energy transfer between the stop and valve member.

While remaining either on or off the stop, Newton's law of motion is applied.

DISCHARGE REED DYNAMICS

The discharge dynamics are treated in one of two ways, Fig.FC3.

a) the valve reed wraps onto its stop.
b) there is an impact between the reed and its stop.

In case a) the stiffness of the valve reed increases as the reed wraps onto the stop and therefore the model must take this into account when calculating the restoring force. If the reed wraps onto the stop the model considers the stop stiffness to be infinite and that there is no energy exchange between the reed and stop.

In case b) the reed is considered to move freely until it impacts on the stop, when conservation of momentum is applied:-

$$M3 \times U3 = U4 \times (M3 + M4) \qquad (9)$$

Where:-

- M3 is the reed mass.
- M4 is the retainer (stop) mass.
- U3 is the velocity prior to impact.
- U4 is the velocity after impact.

While the valve reed is in contact with the stop the acceleration of the reed and stop combination and the acceleration of the reed alone are computed. When the acceleration towards the valve plate of the reed alone is greater than that of the reed and stop combination, the reed is considered to break free with no velocity change. Once the reed has become detached from the retainer, the retainer is considered to be at rest in its equilibrium position.

GAS FORCES

The criteria which has been used to model the valve dynamics can only be valid if the gas forces on the reeds are correctly predicted.

From the outset it was intended that any pressure drop coefficients used would not be varied depending on the configuration of the particular machine being evaluated. In order to retain a fixed set of coefficients, the valve systems are divided into several sections, Fig.4. Passage from one section to another represents a change of velocity and hence a pressure drop which, using incompressible flow equations, is given by:-

$$PD = k \times \rho \times (U5-U6)^2 \qquad (10)$$

In order to reduce the number of calculations within the model, the pressure drop is redefined as follows:-

$$PD = \left| k \times \rho \times (U5^2-U6^2) \right| \qquad (11)$$

where:

k is a constant dependant on the type of change (expansion/contraction/change of direction).

U5 is the gas velocity in section 5
U6 is the gas velocity in section 6
PD is pressure drop.

The redefined equation is considered suitable as the value of PD obtained will tend towards the value obtained using (10) if U6 is small when compared with U5 and the pressure drop will be insignificant as U5 approaches U6.

As the gas velocity is inversely proportional to the flow area, the pressure drop is more conveniently expressed:-

$$PD \propto \left| k \times (1/A5^2 - 1/A6^2) \right| \qquad (12)$$

By considering the relative positions of the piston, valve plate and valve reed, the correct values of k can be selected.

Referring again to Fig.4, this depicts a discharge valve where the port diameter is small when compared with the cylinder area. As the gas passes through the valve it undergoes several changes in the velocity, for each of these changes the standard text book values of k are used.

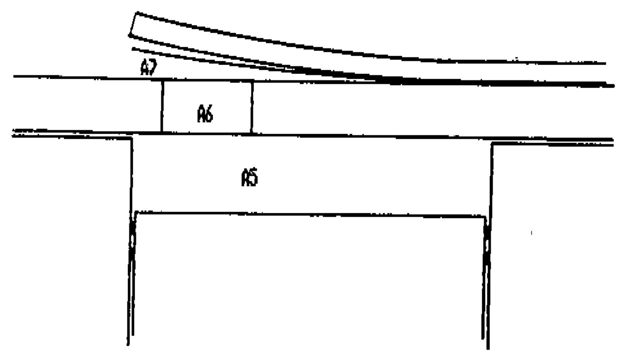


Fig 4 DISCHARGE FLOW AREAS

The gas first enters area A5 and this is considered to be a sudden contraction thus:-

$$PD1 \propto 0.5 \times (1/A5^2) \qquad (13)$$

There is then a change of direction as the gas enters A6:-

$$PD2 \propto 0.5 \times (1/A5^2) \qquad (14)$$

As the gas enters A6 there is either a contraction or an expansion. If we consider the piston to be near T.D.C. there is an expansion into A6:-

$$PD3 \propto 1.0 \times (1/A5^2 - 1/A6^2) \qquad (15)$$

The gas then leaves A6 and there is a change of direction, thus:-

$$PD4 \propto 0.5 \times (1/A6^2) \qquad (16)$$

The gas next expands or contracts into A7. If the valve reed is not fully open then this is probably a contraction:-

$$PD5 \propto 0.5 \times (1/A7^2 - 1/A6^2) \qquad (17)$$

The gas then expands from A7:-

$$PD6 \propto 1.0 \times (1/A7^2) \qquad (18)$$

From these pressure drop elements the overall pressure drop can be calculated, as can the pressure drop across and therefore the force on, the valve reed.

$$PD = (PD1 + \ldots + PD6) \qquad (19)$$

$$PDr = (PD4+PD5+PD6) \qquad (20)$$

where:

PD(1 to 6)	are the pressure drops in elements of the valve.
PD	is the total pressure drop.
PDr	is the pressure drop across the valve reed.
A(5 to 7)	are the flow areas of elements of the valve.

The values of k used would vary if the ports are chamfered and the port entry and exit profiles can easily be included in the programme inputs.

The values of k used in the model and the order in which events such as changes in direction and expansions occur are open to debate, but if gas forces are to be reasonably predicted, the valve must be divided into its elements.

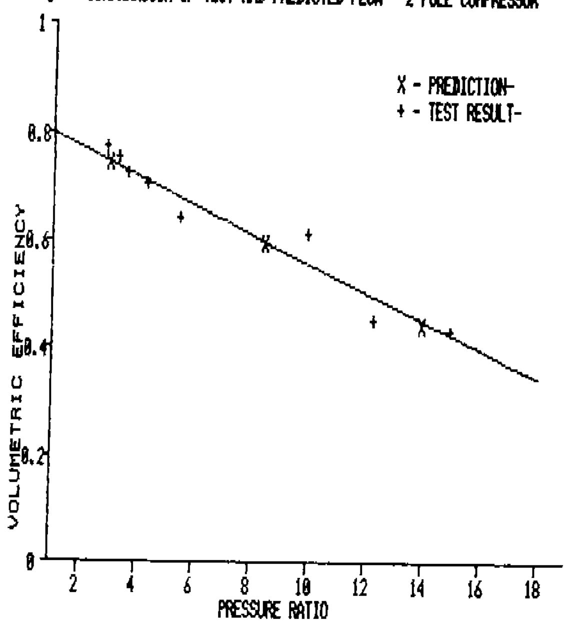
HOW ACCURATE? COMPARISON WITH TEST DATA.

To be of general use a model must suggest behaviour which is similar to the actual event and so the accuracy of the model is of paramount importance. The accuracy of the model should primarily be considered with regard to the gas flow, shaft power and valve stresses.

GAS FLOW AND POWER.

Initial comparisons were made using a family of six semihermetic compressors running on 50Hz and 60Hz electrical supplies. These compressors were each tested with two refrigerants at several conditions. As shown in Fig.5, the predictions of flow were consistantly within 5% of the test value. The power predictions had a similar accuracy for much of the operating range, but for pressure ratios of less than 2:1 could be up to 10% high.

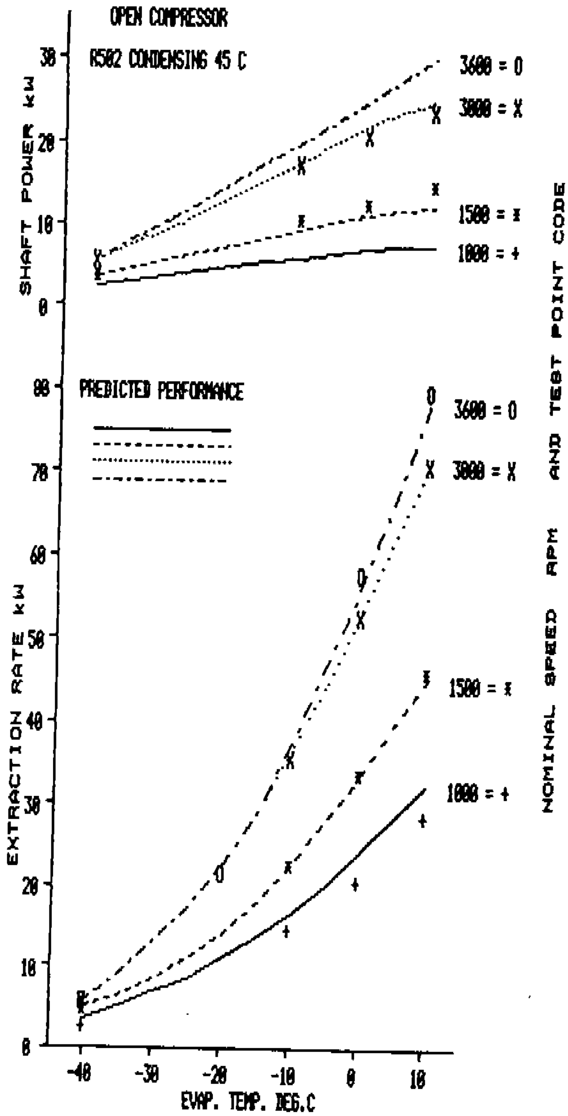
Fig 5 COMPARISON OF TEST AND PREDICTED FLOW - 2 POLE COMPRESSOR



To further evaluate the model, tests were carried out on an open compressor operating at speeds of between 1000rpm and 3600rpm. The results of these tests are shown in Fig.6 and again show good agreement.

Tests on a compressor from another family of machines initially suggested a flaw in the model. When test results and predictions were compared for high pressure ratios a 40% error was noted in the gas flow figures. Being unable to detect any unexpected errors within the model, the compressor was checked and the discharge reed retainer found to deviate from specification, Fig.7. When the valve reed stiffness and mass used by the model were redefined to suit the equipment on test, the revised predictions closely resembled the test results.

Fig 6 COMPARISON OF TEST AND PREDICTED PERFORMANCE



VALVE STRESSES

By using the valve displacement figures obtained from the model, the valve stresses can easily be calculated by reference to the separate valve model. When the predicted values are compared against stress values obtained by the use of strain gauges on the valve reed a typical correlation would be as shown in Fig.8.

For a typical cantilever suction reed with tip stop, the predicted stress levels are generally within 15% of test values. Altering the stiction factor, within reasonable limits, will change the stress predictions by about 10%.

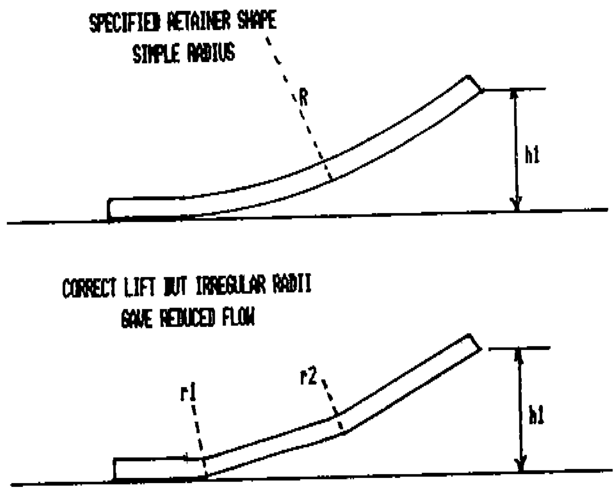


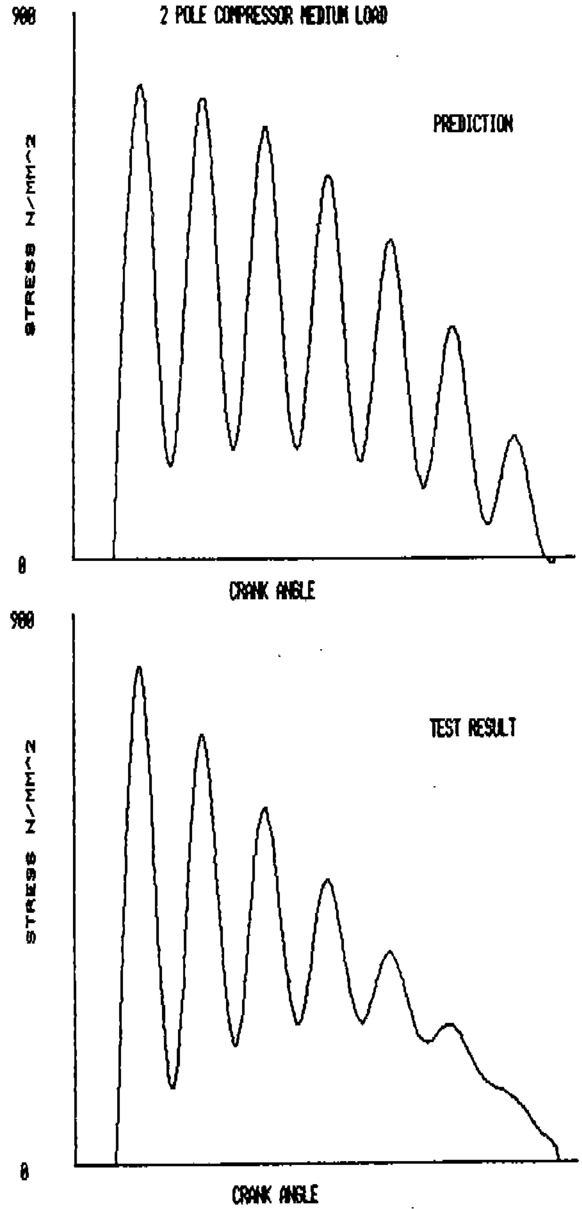
Fig 7 COMPARISON OF RETAINER SHAPES

IN CONCLUSION

The compressor and valve models which have been described ignore some of the events which undoubtedly occur within a reciprocating compressor. However, the comparisons which have been made with test data suggest that events such as the transfer of heat within the cylinder, although important in gaining a full understanding of the process, are not essential for predicting the overall performance. On the other hand, breaking the valve down into its various elements appears to be essential if the use of variable flow factors is to be avoided.

While evaluating the model it became apparent that for reciprocating refrigeration compressors the single most critical factor in obtaining high flow rates is the discharge valve dynamics. A model such as the one described allows the optimisation of any existing valve form, or, where required, the confident design of new valve forms.

Fig 8 COMPARISON OF TEST AND PREDICTED SUCTION REED STRESS



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Purdue Compressor Technology Conference 1982.

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XVIth International Congress of Refrigeration September 1983.

Fig FC1 - COMPRESSOR MODEL

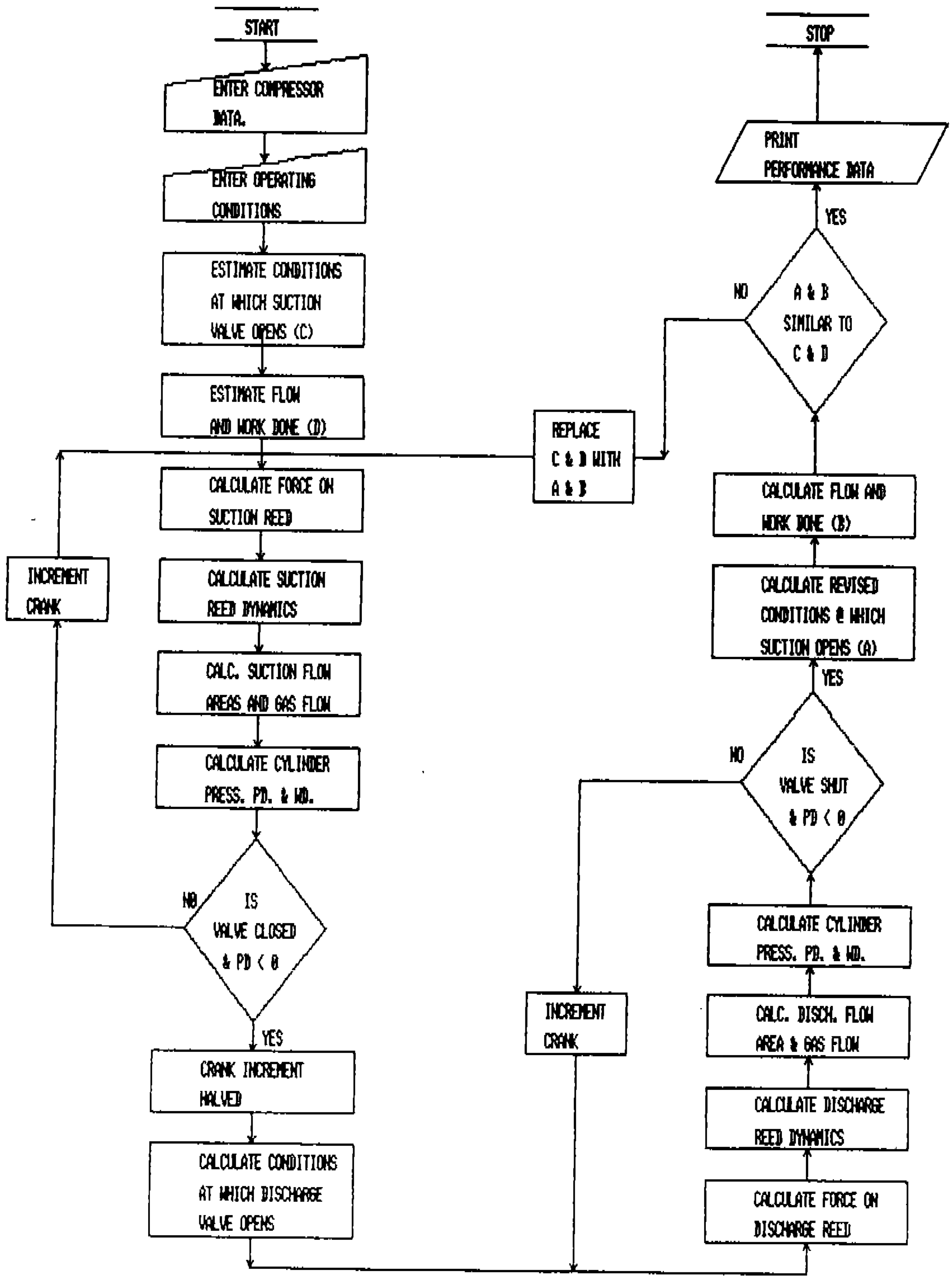

Fig FC2 - SUCTION VALVE DYNAMICS

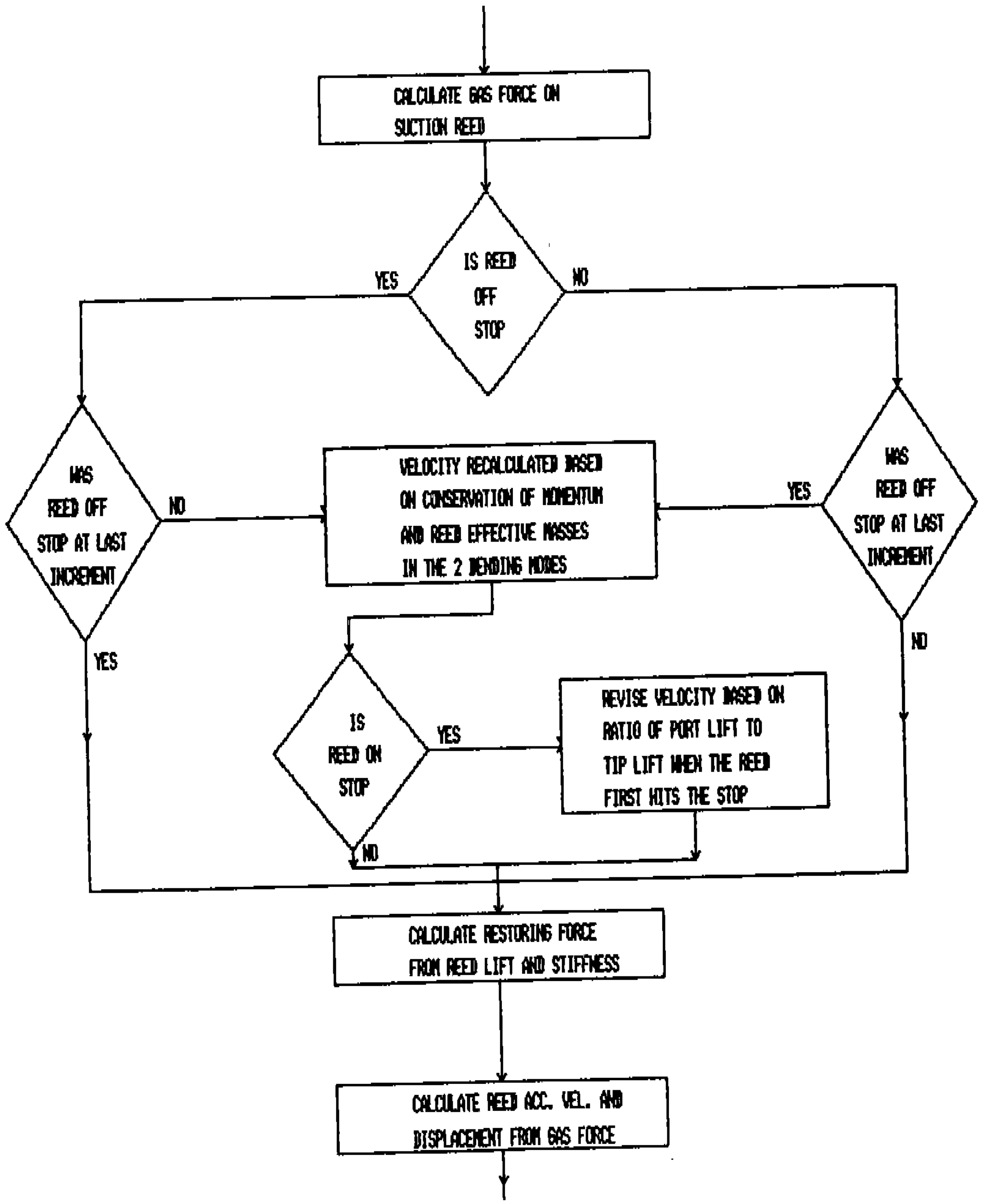

Fig FC3 - DISCHARGE VALVE DYNAMICS