1984

# Analysis of Dynamic Gas Forces on Reciprocating Compressor Valves

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ANALYSIS OF DYNAMIC GAS FORCES ON RECIPROCATING COMPRESSOR VALVES

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ABSTRACT

This paper describes a method by which the Kerr [1] mathematical model of a reciprocating compressor having rigid valve plates may be adapted to investigate the dynamic response of flexible valves of certain geometries. The method is restricted to flexible valves which can be modelled by spring-mass systems. Also, the section of the valve plate which is directly over the valve port must remain substantially parallel with the valve seat during motion since the effect of plate tilt on the gas force is unknown at present. The curve of gas force versus time for the moving valve plate is derived by the Kerr model from measured values of gas force applied to model valves, in a specially designed test rig which produces curves of gas force versus displacement for a range of pressure differences. A Fourier analysis is carried out with a view to determining whether or not any valve natural frequency is likely to resonate with a harmonic component.

THE RIG FOR MEASURING GAS FORCES

The rig shown diagrammatically in Fig. 1 is designed to measure gas forces exerted normal to the bottom surface of the valve plate. The gas force data fed into the computerised compressor model is derived from the gas forces measured in this rig. The rig is capable of measuring the variation of the gas force with time as the valve plate is rapidly pulled from its seat to release the pressurised gas in the plenum chamber to atmosphere.

Clearly it is also capable of measuring the gas force caused by the steady flow of gas through the valve at fixed increments of plate displacement.

The differences between the gas forces resulting from transient motion of the valve plate and those which result from the steady flow of gas through the valve at a series of fixed increments of valve lift are discussed elswhere [2] and the influence of these differenes on certain aspects of compressor behaviour have also been investigated [3].

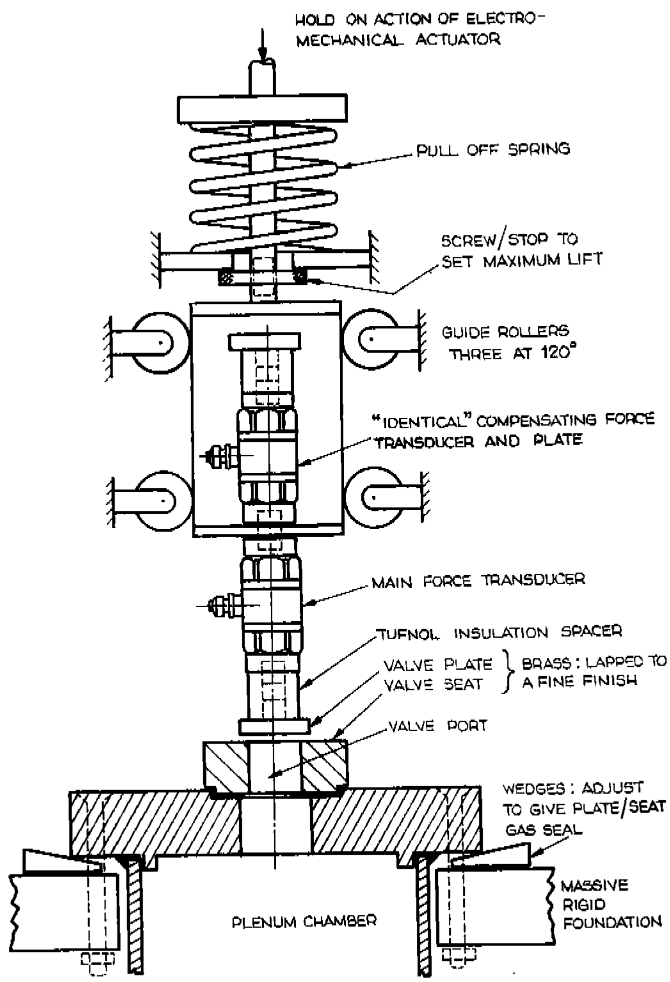


FIG.1. DIAGRAM OF DYNAMIC GAS FORCE MEASURING RIG.

The mechanical principles and electrical detail of the gas force rig have been described elsewhere [4] and are not repeated here.

With the valve closed a chosen plenum pressure above atmosphere is set and the valve is then rapidly pulled from its seat. During the valve motion the gas force and plate displacement signals are captured on a two-channel digital storage oscilloscope. The data is transferred to a microcomptuer and re-plotted in the form of gas force versus displacement for the series of excess plenum pressures used. Fig. 2 shows the result.

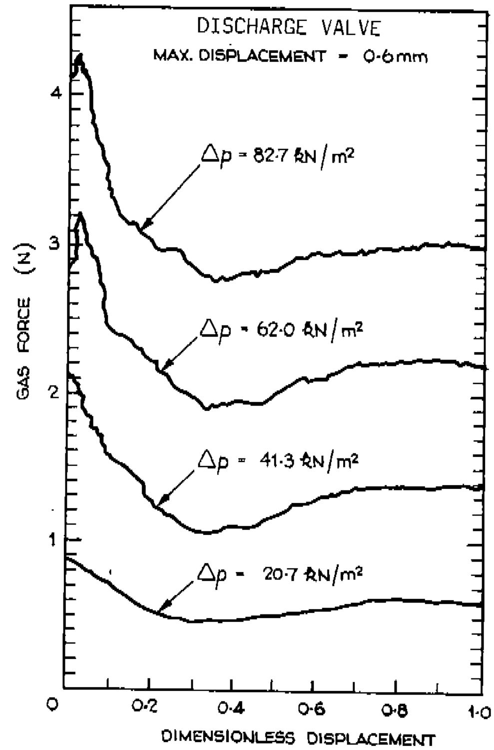


FIG.2. GAS FORCE v DISPLACEMENT.

It should be made clear that, although the authors contend that the gas force data derived from transient pull-off tests is closer to reality for the opening phase than data derived from force measurements made during steady flow through the valve at a succession of fixed increments of valve lift, they are aware that the true behaviour of the gas force acting on a valve in a compressor while running, may not be described exactly by the curves of Fig. 2. Differences could be caused by the fact that as the valves in a real compressor open, the pressure difference across the valve is changing: in the test rig the pressure difference across the valve is kept constant during valve motion. Furthermore, for the valve closing phase, the steady flow data measured at fixed increments of valve lift may be more appropriate than the transient data, given the fact that during the valve closing phase in an operating compressor, the flow through the valve is fully developed and akin to the steady flow conditions in the test rig. For simplicity, a single set of (transient) gas force data has been used in the work reported here for both the opening and closing phases of the valve, since the main purpose of this work is to demonstrate the analytical technique.

## THE USE OF THE GAS FORCE DATA

The mathematical model described in Ref. 1 was used to investigate the behaviour of the gas forces applied to the valves of a reciprocating test compressor under construction in the University of Strathclyde. Model valves having the same geometry as the suction and discharge valves were manufactured and inserted in turn into the gas force rig and curves like those shown in Fig. 2 were determined for each. (The curves shown are for the discharge valve).

The relationship between gas force, displacement and pressure difference for the opening and closing phases of the valve motion in the test compressor are assumed to be described by the gas force data given in these curves. The data was inserted into the mathematical model in the form of a matrix. Intermediate values of gas force were determined by interpolation and values beyond the measured range by extrapolation.

Fig. 3 shows the predicted variation with time of the gas force acting on both the suction and discharge valves during the opening phases of the valves i.e. during the time taken for each valve to travel from the fully closed position (sealed against the seat) to the fully open position (i.e. stationary against the stop). During this time interval each valve is a mass/elastic system driven by the force versus time history shown in Fig. 3.

## MODELLING THE VALVE OPENING PHASE

The opening phase ends with the valve abruptly ceasing to be a mass/elastic system (i.e. it becomes infinitely stiff against the stop) and the gas force becomes constant at the value appropriate to the maximum lift. The initiation of the opening phase is the reverse of this process in that the valve plate, infinitely rigid when in contact with the seat, becomes part of a mass/elastic system at the instant at which it separates from the seat. Modelling the initiation of the valve opening phase presents difficulties, because in a real reciprocating compressor oil contamination of the working fluid always exists to some degree and this gives rise to oil stiction effects on the valve, which make each valve opening a unique event [5]. While no two valve openings will be exactly the same, many will be very alike for a constant degree of oil contamination so that "typical" valve behavior seems a reasonable thing to propose and study.

The unmodified oil-free model [1] assumes that the gas instantaneously and uniformly occupies the entire gasket area at full pressure at the instant at which the plate separates from the seat. (The "gasket area" is the area of overlap contact of the valve plate on the seat). The model determines the instant of separation as being the instant (crank angle) at which the force tending to open the valve (pressure difference x port area) equals the force opposing the opening of the valve (the spring pre-load). In a real valve an oil film (probably incomplete) would also be present. The rupture and blowing away of the oil film would require a finite time so that the gas would in effect advance across the gasket area during this time and apply force to the plate in a (steep) ramp-like fashion and not as a step. The programme has been modified to achieve this and the initial value on the "ramp" at time zero in the opening process, was chosen to be equal to the final gas force value at the instant at which the plate hits the stop.

At first sight this assumption may seem unwarranted, but when one examines all other possible courses of action, one finds that they amount to very much the same thing. An attempt to illustrate this is shown in Fig. 4(a). The controlling factor is the degree of incompleteness of the oil film, i.e. the fraction of the gasket area which is "dry" and in contact with the high pressure side, for this determines the magnitude of the step increase in force which follows the instant at which the gas force becomes equal to the valve spring pre-load. In the unmodified model, all of the gasket area is occupied by gas at full pressure instantaneously, whereas in the modified model, only part of the gasket area is occupied instantaneously. The time rate of change of cylinder pressure and therefore the rate of change of gas force applied to the valve plate is determined by compressor speed so that the gas force versus time graphs for three different oil films should look like those shown in Fig. 4(a) over the first few microseconds of the opening process for the same crankshaft speed. A is the most complete oil film and C the least complete. A and B are virtually identical. The only difference is that A results in a very slightly shorter period for the quasi-transient than B. C requires a step from the zero datum before the ramp can begin. Curves A, B and C will give rise to different frequency spectra, but the differences should be small and confined to the high frequency components, so the easiest model to implement was chosen, that is B.

The gas force applied to the valve is in the form of a transient pulse. It is true that this pulse is applied once per revolution of the crankshaft, but between pulses the valve motion is brought to a halt by coming into rigid contact with the stop or the seat. Therefore, the valve, during its life as a mass/elastic element, has no "memory" of previous mass/elastic "lives", so a particular mass/elastic "life" has to be treated as a one-off transient as far as dynamic response is concerned. Of course, the model, fed with unique deterministic data, predicts every "life" as being the same.

For the purpose of carrying out a Fourier analysis of a transient, the periodic function shown in Fig. 4(b) has been used to simulate a quasi-transient. Clearly, the approximation improves as the space to mark ratio increases, but the cost of computing time is a limitation. Fortunately, the solution converges rapidly. As can be seen in Fig. 5, space to mark ratios of one and three give almost identical frequency spectra. The amplitude of the components for a space to mark ratio of three is half that for a ratio of one due to the arithmetic of the Fourier transform requiring a division by twice the number of points.

## APPLICATION OF FREQUENCY ANALYSIS

The two main operating variables of interest to the designers and users of reciprocating compressors are pressure ratio (governing delivery pressure) and speed (governing throughput).

Fig. 6 shows how the harmonic components of gas force applied to the valves change when the compressor speed is changed. Fig. 7 shows the effect of changing the pressure ratio. The space to mark ratio used to simulate a transient pulse is three in both cases. Worthy of comment perhaps is the fact that halving the speed from 1000 rpm to 500 rpm (Fig. 6) greatly increases the amplitudes of the low frequency components, especially in the case of the discharge valve. This is supported by the fact that the displacement versus time behaviour of the valves treated as rigid plates controlled by a linear spring is very violent, rebounding being predicted by the model. The spring-mass natural frequencies are 42 Hz for the suction valve and 171 Hz for the discharge valve.

Fig. 7 indicates that increasing the pressure ratio has two effects:

1. a general increase occurs in the amplitudes of harmonic components,

2. in the range 10 to 20 kHz the frequencies at which two groups of significant harmonic components occur are shifted upwards.

## REFERENCES


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[2] Fleming, J.S. Gas Force Effects on Compressor Valves in the Early Stages of Valve Opening, PhD Thesis, University of Strathclyde, Scotland, 1983.

[3] Shu, P.C., Fleming, J.S. & Brown, J. Analysis of Dynamic Gas Forces on Reciprocating Compressor Valves, XVIth International Congress of Refrigeration, Paris, 1983.

[4] Fleming, J.S., Brown, J. & Davidson, R. Gas Forces on Disc Valves, Compressor Technology Conference, Purdue, Lafayette, Indiana, 1980.

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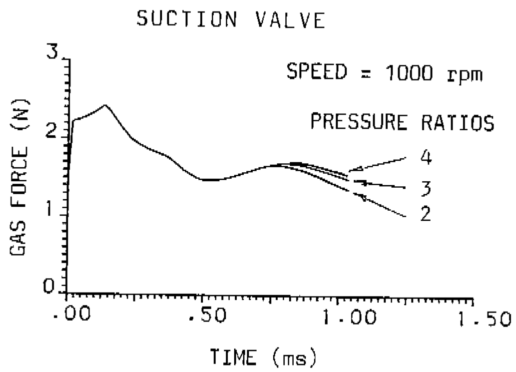


FIG. 3(a) GAS FORCE V TIME

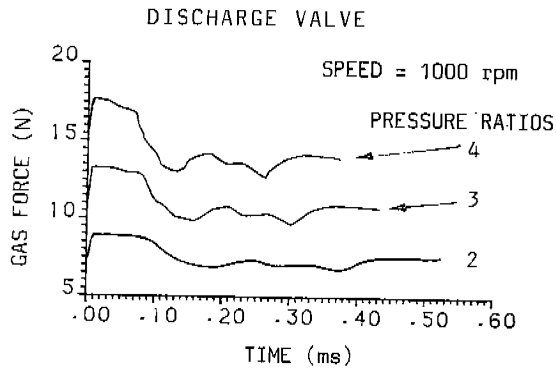


FIG. 3(b) GAS FORCE V TIME

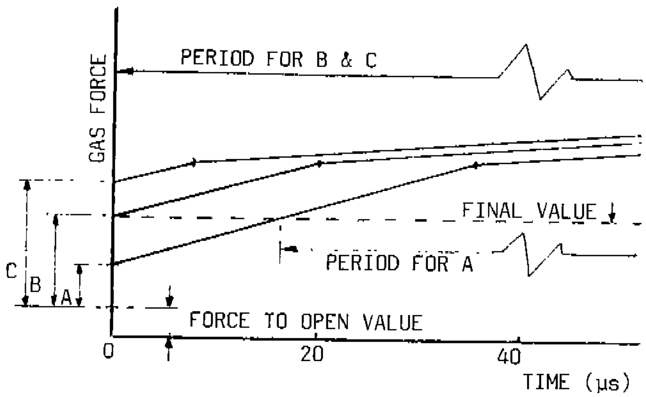


FIG. 4(a) MODELLING THE INITIATION OF VALVE OPENING

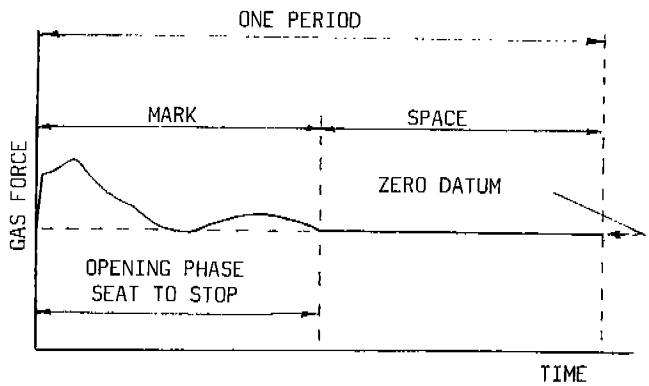


FIG. 4(b) SIMULATION OF TRANSIENT GAS FORCE ACTION
(SPACE/MARK RATIOS = 0, 1, 3)

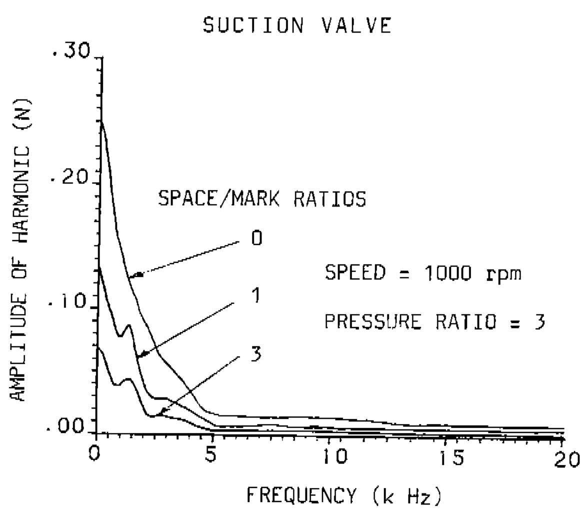


FIG. 5(a) GAS FORCE FREQUENCY SPECTRA
EFFECT OF SPACE/MARK RATIO

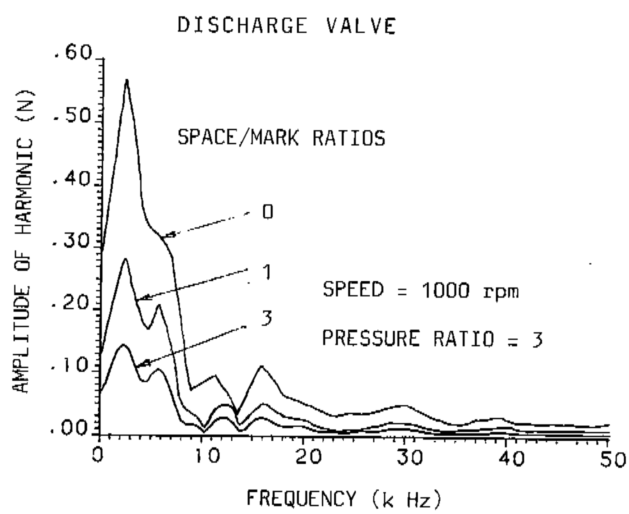


FIG. 5(b) GAS FORCE FREQUENCY SPECTRA
EFFECT OF SPACE/MARK RATIO

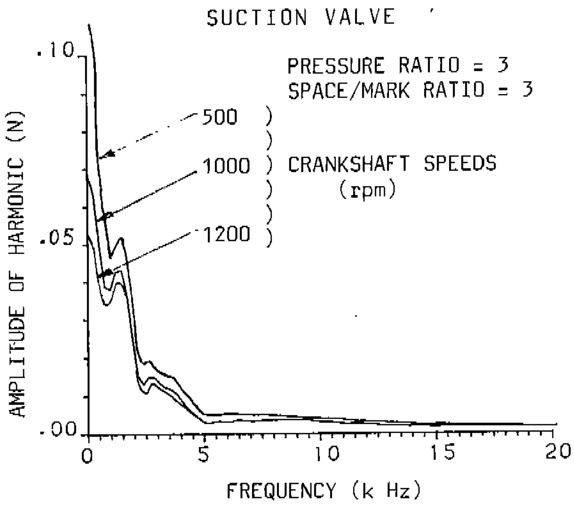


FIG. 6(a) GAS FORCE FREQUENCY SPECTRA
EFFECT OF CRANKSHAFT SPEED

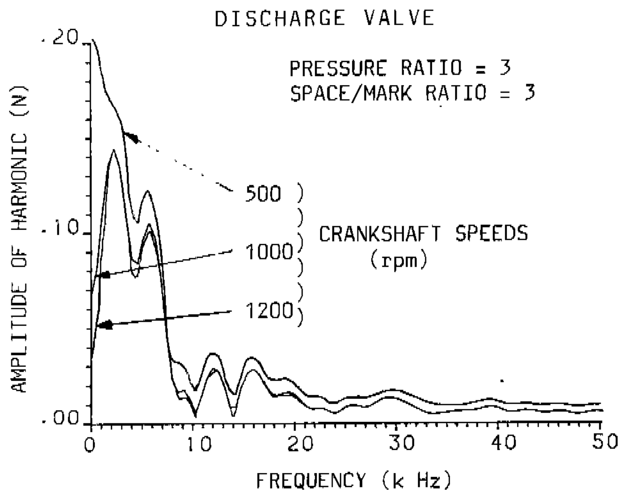


FIG. 6(b) GAS FORCE FREQUENCY SPECTRA
EFFECT OF CRANKSHAFT SPEED

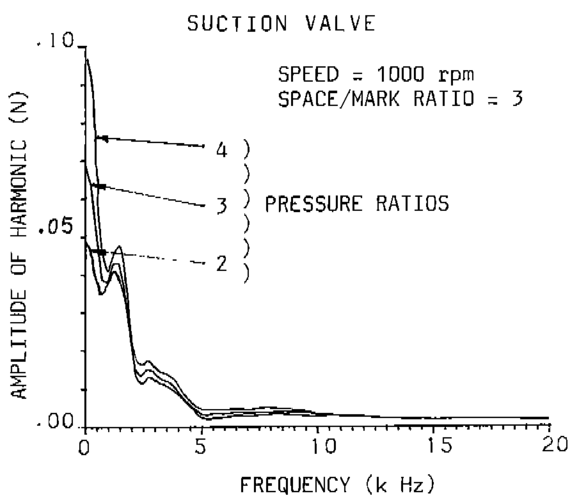


FIG. 7(a) GAS FORCE FREQUENCY SPECTRA
EFFECT OF PRESSURE RATIO

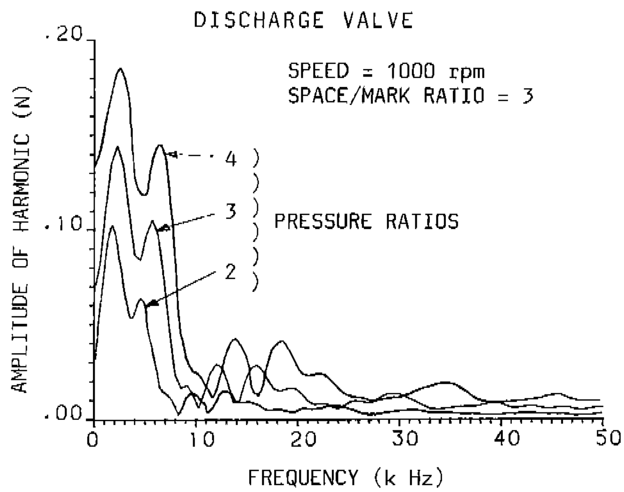


FIG. 7(b) GAS FORCE FREQUENCY SPECTRA
EFFECT OF PRESSURE RATIO