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by

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ABSTRACT

This paper describes part of a programme designed to develop a CAD technique for the design of reciprocating compressors and their valves. Use has been made of the PHOENICS (1) computer program to solve for velocity flow field, streamlines and pressure distribution (and hence gas force applied to the valve plate) for the case of the flow of air through two simple valve types, namely a disc valve and an annular ring valve.

Even these simple shapes, possessing as they do sharp corners and abrupt changes in passageway orientation and flow cross-section, make the flow difficult to model. However, advances over earlier methods (2) have been achieved and the increasing availability of more economical computing power and software give cause for optimism about what is achievable in flow modelling.

Comparisons are made between computed and measured values of the force applied by the gas to the valve plate.

INTRODUCTION

The earliest mathematical models (3), (4), (5), (6), of reciprocating compressors were devised to predict the thermodynamic behaviour of the compressor, and this they did reasonably well despite the simplifying assumptions made concerning valve action and the assumed ideal nature of the working fluid. As computing power and modelling skills grew, mechanical complexities such as multi-cylinder arrangements and pipework systems of finite volume and complex characteristics were successfully modelled (7).

Valve life in service depends on the dynamic behaviour of the valve and on the properties of the materials from which the valve is made. Material properties are outside the scope of this paper. Modelling the dynamic behaviour of the movable element of a valve has proved to be a difficult exercise. Even for a valve with one degree of freedom an accurate version of the gas force versus valve lift curve is difficult to compute, so that the value of the impact velocity of the plate on the stop, which depends on this curve, often differs considerably from measured values.

The effect on the gas force of flow transients caused by the intermittent nature of the flow in reciprocating machines, has been shown to be worth considering only in small high speed compressors (8). In larger, slower machines an analysis based on steady state conditions for flow through a valve at a succession of fixed increments of valve lift is suitable.

The authors of this paper formed the opinion that the lack of agreement between measured and calculated values of gas force (2) was due to the mathematical model failing to take account of gas viscosity-related influences like wall friction and turbulence. It has been shown (9) that at very low values of valve lift where Reynolds numbers in the plate-seat gap are very low ($Re < 20$), failing to take account of wall friction gave computed gas force values which bore no sensible relationship with measured values at all, while introducing a reasonable mathematical model for wall friction gave rise to very good agreement.

At higher values of valve lift, where gas velocities are much higher, Reynolds' numbers are much higher and turbulent flow exists. Many difficulties are associated with modelling and measuring turbulent flow. Suffice to say that turbulence is still the subject of much active research.

In the work reported here a powerful computer code developed by Spalding et al (1) for solving heat transfer and fluid flow problems was used by the authors to investigate the influence of turbulence model on the agreement between computed and measured values of gas force. This program has effectively five different turbulence models from which to choose. It is hoped that the knowledge gained will assist with the development of simpler special-purpose programs which are economical to run.

The first objective is to achieve a realistic solution of the valve dynamics problem: the second is to incorporate this solution into a compressor model so that the same model can accurately relate compressor thermodynamic behaviour and valve dynamic stressing.

VALVE GAS FORCE MEASUREMENTS

Two simple valve types were chosen for the study. In both, the movable element (the "plate") was rigid and had only one degree of freedom. The first type consisted of a circular disc mounted concentrically over a round valve port, valves A, B and C (see figure 1).

The gas forces on the valve plate of this valve were measured in a specially constructed test rig which has been described elsewhere (10).

The second valve type is an annular ring valve, Valve D (see figure 2). This valve was fitted in an operating compressor as described at the IIR Congress in 1987 (11) and the gas force on the valve moving element was deduced from the measured values of acceleration and displacement as follows, friction and gravity forces being assumed to be negligible:

$$\text{gas force} = \text{inertia force} + \text{spring force} \qquad \ldots\ldots \quad [1]$$

Since the plenum and cylinder pressures were also measured the gas force coeffficient was calculated as follows:

$$\text{gas force coefficient} = \frac{\text{gas force}}{\text{plate area x pressure difference}}$$

In the case of the annular ring valve fitted to the operating compressor the pressure difference across the valve changes continuously with time and valve lift, but a particular valve lift is always uniquely associated with a particular pressure difference, because transient effects in a slow running machine like this one are negligible. The gas force computations using PHOENICS were carried out for a series of valve lifts, each associated with a pressure difference determined from the measured data (11), and compared with corresponding "measured" values of gas force as determined by equation [1] (see figure 6).

COMPUTED & MEASURED VALVES COMPARED

In figures 3, 4, 5 and 6, computed and measured values of the force exerted by the flowing gas on the valve plate are plotted against valve lift. The valve geometrics are shown in figures 1 & 2.

For valves A, B & C (figures 3, 4 & 5) the computations and measurements were made for the same combination of outlet pressure (atmospheric) and pressure difference (20.7 kNm^{-2}). In the case of valve D, each plotted value of gas force (figure 6) is associated with a different pressure difference due to the fact that these gas force values were derived from measurements made on a suction valve operating normally in a live compressor.

In all cases values of gas force were computed by PHOENICS by making use of the concept of "eddy kinematic viscosity" (E.K.V.), the simplest of the models of turbulence available. Values of E.K.V. between ten and ten thousand times greater than the laminar kinematic viscosity for the working fluid (air) were used as trial values and in each case the E.K.V. was assumed to be constant throughout the flow field. In reality, this cannot be true but some common trends in the relationships between the computed and measured values can be discerned from figures 3, 4, 5 and 6 which are of assistance in indicating the way in which more sophisticated turbulence models might lead to more reliable computed values of gas force.

The force applied by the gas to the valve plate is computed from the integral $\sum p\delta a$ where p is the pressure in a fluid volumetric element in contact with a plate surface element of area δa.

Quite clearly, changing the value of the E.K.V. changes the computed value of gas force. The best agreement between computed and measured values of gas force is obtained when the E.K.V. is in the region of 10^{-3} (figures 3, 4 & 5) and 10^{-2} (figure 6). The work presented here introduces a programme of work in which a range of geometrical parameters, Reynolds numbers and pressure drops is being investigated.

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Valve	A	B	C
R_1	3.17	3.17	6.35 mm
R_2	4.20	4.76	8.40 mm

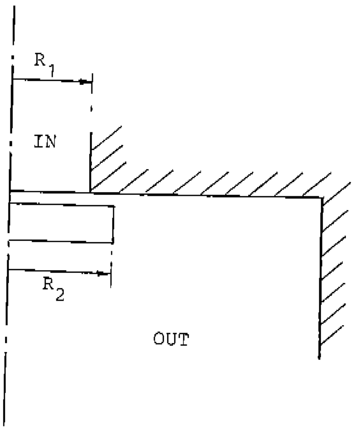

FIGURE 1 DISC VALVE
SYMMETRICAL HALF

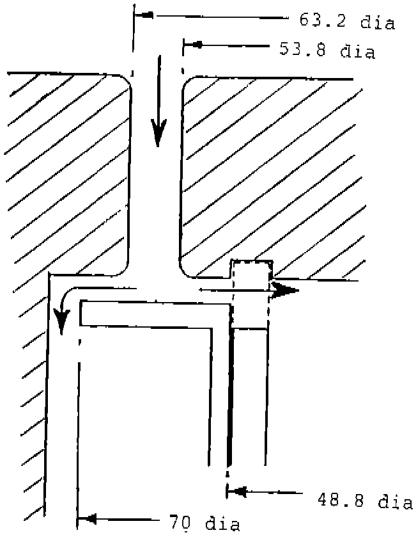

FIGURE 2 RING VALVE (D)
PORT-PLATE SECTION.
(all sizes mm)

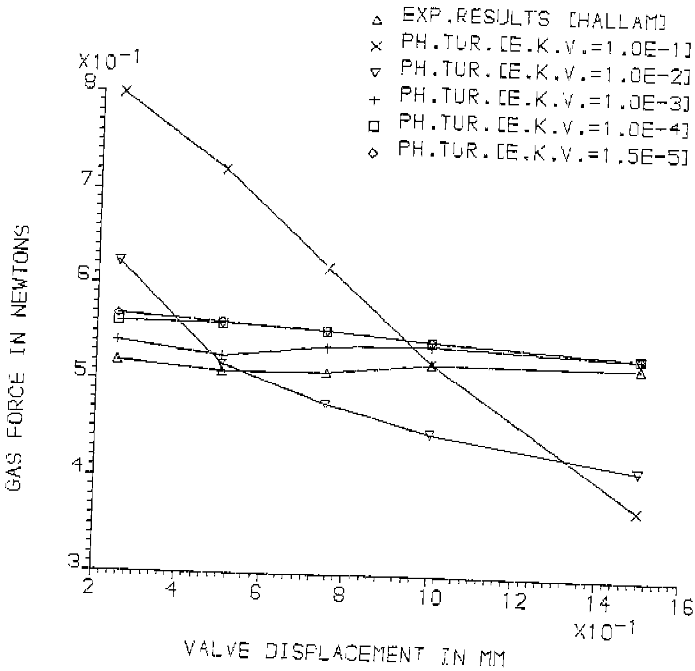

FIGURE 3 (VALVE A)

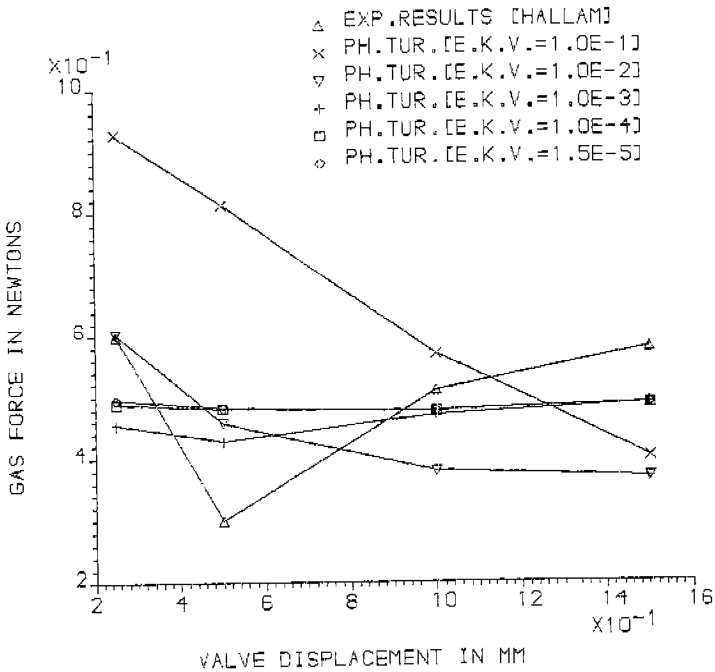

FIGURE 4 (VALVE B)

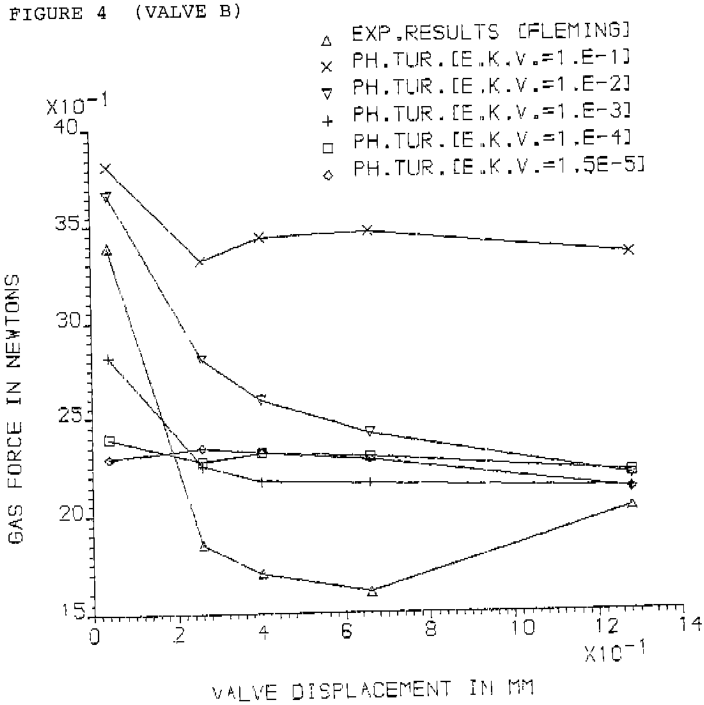

FIGURE 5 (VALVE C)

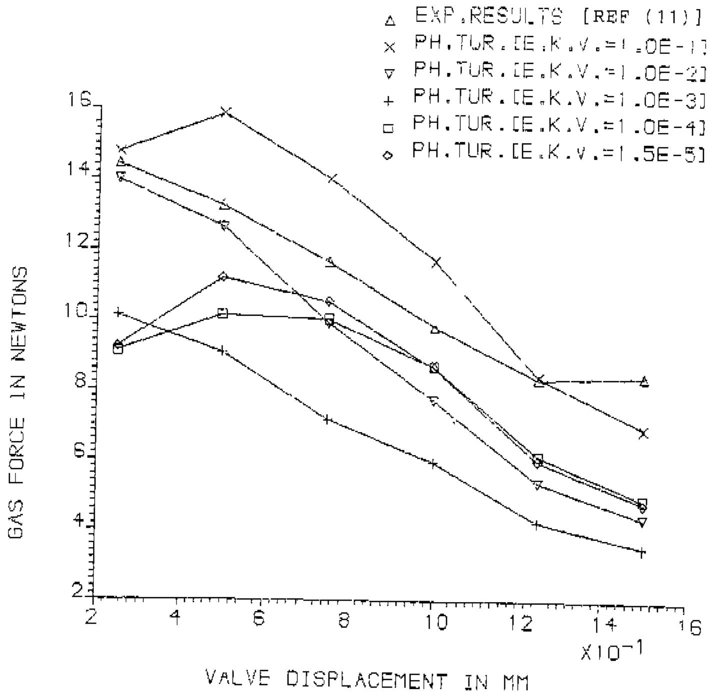

FIGURE 6 (VALVE D)