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A Compressor Valve Model for Use in Daily Design Work

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ABSTRACT

An outline of a procedure used to design cantilever valves is presented and discussed. The model differs from a simulation in that it simplifies many peripheral aspects and deals rigorously with only the valve and forces acting upon it. Inputs are limited to data describing initial conditions, pertinent geometry, operating conditions, and the structural characteristics of the valve and/or compressor. Output consists of graphs showing the deflection of the valve as a function of time and the mass flow history through the valve plate ports. Program execution typically requires less than an hour on a 32K minicomputer.

INTRODUCTION

Flapper valves are widely used in hermetic compressors, with valve size and shape governed by bore size and port distribution. The dynamic behavior of these components is determined by their mass and stiffness characteristics, the fluid forces acting on them and the physical constraints limiting travel. The adequacy of a given design is dependent on its ability to function quietly and efficiently for an extended period under a wide range of conditions.

The procedure outlined herein allows comparison of the performance characteristics of valve designs without the need to fabricate and test prototypes. It has the advantage of quickly providing a detailed microscopic prediction of time-dependent cyclic behavior. In addition, it allows the performance of new designs to be compared to existing valves. The time for a single evaluation thus becomes hours rather than months. The intent of this effort is to increase the success rate of designs placed on life test thereby reducing the time and cost required to qualify a new valve design.

MATHEMATICAL MODEL

The valve is analyzed as a variable width beam undergoing transverse vibration. The various mode shapes and natural frequencies associated with specific boundary conditions are determined via the procedure outlined in reference (1). The time-dependent behavior of the valve is described mathematically in references (2) (3) and (4) where the following form of the beam equation is used:

$$\frac{\partial^2}{\partial x^2}\left[EI(x)\,\frac{\partial^2 y(x,t)}{\partial x^2} \right] + m(x)\,\frac{\partial^2 y(x,t)}{\partial t^2} = F_j(t)\,\delta(x-x_j) \qquad 1$$

The solution of equation (1) requires a set of boundary conditions to describe the constraints on each end of the valve. Clamped-free boundary conditions apply for both discharge and suction reeds that are neither in contact with their seat nor stop (see Fig. 1).

Figure 1

Clamped-pinned boundary conditions describe a suction valve in contact with a stop at its tip (see Fig 2).

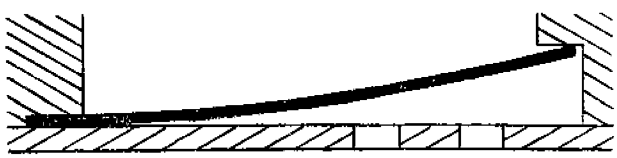

Figure 2

The following expressions describe these constraints:

Clamped-Free:

$$y(0,t) = 0 \qquad \frac{\partial y(0,t)}{\partial x} = 0$$

$$\frac{\partial^2 y(L,t)}{\partial x^2} = 0 \qquad \frac{\partial^3 y(L,t)}{\partial x^3} = 0$$

Clamped-Pinned:

$$y(0,t) = 0 \qquad \frac{\partial y(0,t)}{\partial x} = 0$$

$$\frac{\partial^2 y(L,t)}{\partial x^2} = 0 \qquad y(L,t) = 0$$

The procedure has the effect of lumping mass and stiffness into discrete elements distributed along the length of the valve, as shown in Figure 3.

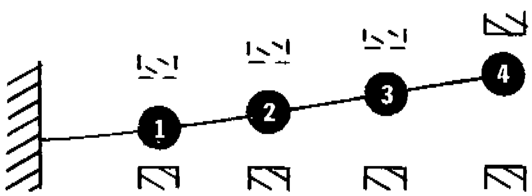

Figure 3

Each of the masses are constrained as follows:

$0 \leq$ displacement $\leq$ stop height (if stop exists)

The interaction of the valve with seat and stop is modeled by a "bounce"-type of analysis where the velocity after impact is related to the velocity before impact by the expression:

$$\dot{x}_{\substack{after\\impact}} = -CFR \cdot \dot{x}_{\substack{before\\impact}}$$

The transformation from clamped-free to clamped-pinned boundary conditions is dictated by the motion of the valve tip. A bounce against the stop in two successive time increments causes the program to switch from the first to the second analytical mode. The motion is modeled as clamped-pinned until the deflection of any part of the valve becomes less than its value at the last impact with the stop. When this occurs, the analysis reverts to the clamped-free mode and the tip either moves away from the stop or else impacts with it, thereby causing the model to continue in the clamped-pinned mode.

The thermo-fluids model developed in reference (2) is utilized here. The fluid network consists of three elements; the discharge cavity, cylinder volume, and suction plenum. The pressures in both volumes adjacent to the cylinder are held constant in order to simplify and shorten the analysis.

COMPRESSOR MODEL

The operation of the compressor is divided into 2° increments of crankshaft rotation and each cycle is treated as a series of 180 discrete time steps. In order to start the process, a set of initial conditions must be specified that "reasonably" reflect the state of the valve and the conditions within the cylinder at a given point in time. Discharge valve closing is an opportune starting point when a prediction of suction valve motion is sought and vice versa. The quantities to be supplied in this case and a reasonable first approximation for each are listed below:

	Suction	Discharge
Location of Valve Closing	30° ABC	10° ATC
Cylinder Pressure	Avg. Suction Plenum Pressure	Avg. Discharge Plenum Pressure
Cylinder Temperature	Avg. Suction Plenum Temperature	Avg. Discharge Plenum Temperature

Repeated alternate modeling of the suction and discharge valves will allow the designer to quickly determine the crankshaft location at which each valve closes.

Starting with this information, plus a geometric description of the compressor and the finitie-element mode shapes and natural frequencies describing the valve, the algorithm produces a step-by-step record of the following quantities.

1) Cylinder pressure
2) Valve displacement
3) Valve tip velocity
4) Pressure difference across the ports
5) Effective port area
6) Port velocity
7) Mass flow through the ports
8) Mass in the cylinder

In addition, the model outputs the following information once the valve has closed.

1) Average port velocity
2) Average pressure drop across the ports
3) Net mass transfer
4) Work expended during the time the valve is open

5) Duration of valve opening
6) A measure of valve efficiency

A typical set of output plots is shown in Figures 4-7.

CONCLUDING REMARKS

The algorithm described herein is a generalized technique that can be applied to a flapper valve of any desired shape. The model can be used to give an approximate indication of performance based on imprecise inputs or to study the effects of small tolerances applied to carefully determined data. Efforts to determine the accuracy of this approach by experimental means have shown measured and computed results to agree within 2 percent. Reference (5) discusses the measurements and techniques required to reach this conclusion and summarizes the work performed in this area.

LIST OF SYMBOLS

$\delta(x-x_j)$ - Kronecker delta equation; defined as 0 for $x \neq x_j$, equal to 1 for $x = x_j$

CFR - Coefficient of restitution

E - Modulus of elasticity

F_j - Amplitude of force at distance x_j from base of valve

I(x) - Area moment of inertia about the neutral axis

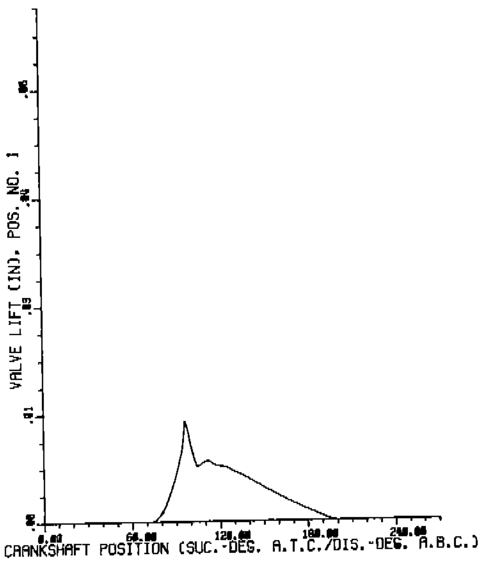

Figure 4

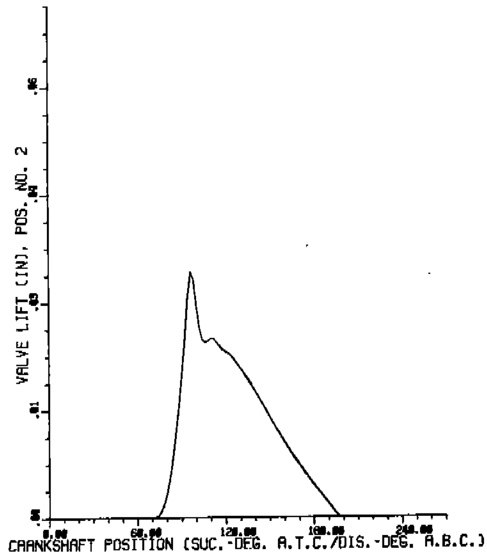

Figure 5

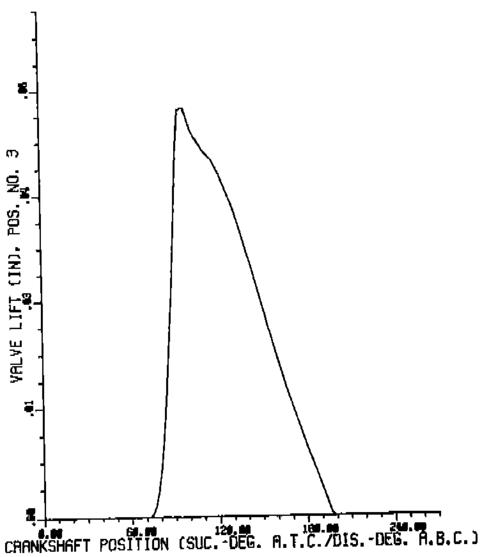

Figure 6

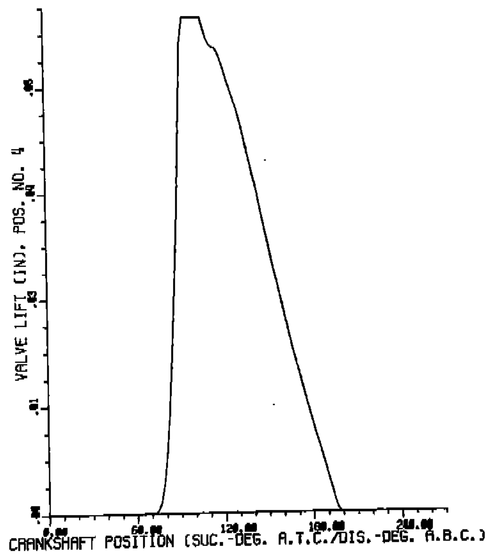

Figure 7

$m(x)$ - Mass per unit length

t - Time

x - Distance from base of valve

x_j - Distance from base of valve to force F_j

y - Valve deflection

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