On Finite Element Modeling Of Valve Dynamics: Impacts, Oil Stiction, Gas Flow, ...

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ABSTRACT

The overall performance (efficiency, noise, gas pulsation, …) of piston compressors is strongly influenced by valve dynamics. In general, the dynamic behavior of the valve is modeled using finite element methods. While the free vibration of the valve can be accurately investigated using this technique, the simulation of the overall dynamics of the valve has to include a number of non-linear phenomena which influence the behavior. The most important are: the impacts, the oil stiction, the gas flow through the ports, the gas pulsation. An accurate numerical model of the dynamic behavior of valves has to account for all these phenomena. This increases the accuracy of input data, but increases the computation time to such an extent that it limits the use of such simulation software in the valve design process.

This paper deals with the development of a numerical model of valve behavior aimed for use in the design office. The valve dynamics and the thermodynamics of gas under compression are modeled simultaneously. The model functions in the time domain. The non-linear behavior of the valve due to the impacts, the oil stiction and the gas flow are modeled as external forces. These are in general functions of valve displacement and velocity. The valve body is modeled using beam finite elements. A restrained number of modes are used to compute the valve response.

An existing valve is modeled using the developed technique. The results obtained are compared with experimental results on a simple valve test rig in air. The measurements are carried out using a LASER vibrometer. The measured results are used to improve the accuracy of the model by modifying the input impact parameters. The number of modes needed for an accurate computation is discussed at the end of the paper.

INTRODUCTION

The valves are critical components of compressors and their dynamic characteristics have a big influence on the performance. This paper presents a model to represent the behavior of automatic reed valves. The simulation takes into account the effects of contact and the differences in pressure which are at the origin of the movement. A thermodynamic model of the compressor is therefore required. Other factors influence the dynamics such as the impacts of the valve on the seat or retainer. The effects of the oil film also play a major role during valve opening. These effects are strongly coupled and accurate modeling of all these aspects is essential.

NUMERICAL MODEL

The problem may be reduced to a series of differential equations describing the physical phenomena. In order to reduce the dimensions of the problem the valve is simplified and represented by a 2D FE beam model. The resolution of these equations is carried out by with direct step-by-step integration in the time domain using a 4th order Runge-Kutta routine. Either the inlet or the outlet valve can be represented by the diagram presented in figure 1.

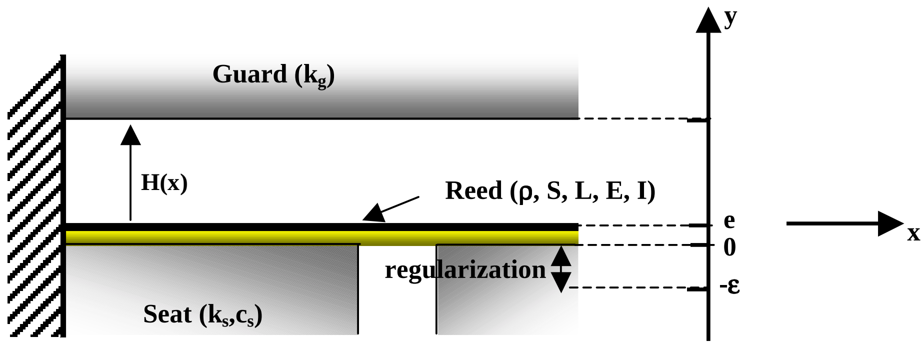

Figure 1 : Representation of the model

Structural Model

The valve structure is modeled using a plate composed of variable inertia beam elements, each element having two degrees of liberty for the vertical displacement and the rotation. The system can be approximately described as a fixed-free beam with a length L a cross section S density ρ, Young's modulus E and quadratic moment I when it is not in contact with the seat.

The matrix equation of the motion of the valve can be expressed in terms of the various forces acting on the system as follows:

$$M\,\ddot{Y} + C\,\dot{Y} + K\,Y = F(t, Y, \dot{Y})$$

where M, K, C are respectively the mass, stiffness and damping matrices of the structure. Y is the vector of the displacements of each degree of liberty and F the vector of generalized forces.

The forces F acting on the valve can be separated into several types. There are two principal categories which are:

- The pressure forces acting either side of the valve via the ports.
- The forces which can be classified as non-linear including:
  - The contact forces between the valve and retainer,
  - The contact forces between the valve and seat,
  - The stiction forces due to the oil film between the seat and the valve.

Thermodynamics

The piston generates the pressure pulsations due to its movement and these variations are in turn controlled by the movement of the valves which are directly linked to their static and dynamic characteristics. It is essential to have an accurate estimation of the pressure in the compression chamber. This is in turn is affected by the piston movement. To simulate the pressure variations a cubic law was employed. This equation to calculate the pressure p is typically in the form of a Van Der Waals type:

$$p = \frac{RT}{V-b} - \frac{a}{V^2}$$

where V is the volume of the compression chamber taking into account the reciprocating movement of the piston. R is the gas constant and T is the temperature of the chamber. The parameter is an approximate evaluation of the force of attraction between the molecules and the parameter b is related to the size of the molecules involved. This formulation allows the co-existence of several different phases. [1]

The chosen formulation the R.K.S law is the most popular cubic law form and introduces a form of temperature dependence in the attraction parameter, a.

The thermodynamic model of the compression chamber is therefore constructed using the R.K.S. law and the geometrical variation in volume of this chamber.
The compression is assumed to be isentropic. To integrate the calculation of pressure and temperature in the model the entropy is expressed as a function of the state variables which are the temperature and the molar volume.
The temperature of a defined state is calculated with the aid of the difference of entropy. The value of the pressure is given from the R.K.S. law.

The variations de pressure and temperature in the chamber of compression produce exchanges of matter with the adjacent chambers while the valve is open. These fluctuations of quantity of matter in the chambers are taken into account with the aid of a differential equation to achieve a molar balance. The quantity of matter flowing through the ports is determined with the aid of the following empirical law:

$$\Delta p = \frac{1}{2}\xi\ \rho u^2$$

where Δp is the pressure difference between each side of the port, ρ is the gas density u is the velocity of the gas at the entry to the port and ξ is an empirical coefficient defined with the aid of reference [3]

Impact

There is a test to determine if there is contact at each step of the calculation at the nodes of the structural model. The presence of an impact is simulated using a non-linear resulting force. [2] The valve seat is characterized by its geometrical profile with a function H (x). The seat has an intrinsic stiffness and damping noted k_S and c_S respectively. In the model a regularization layer ε is introduced. This numerical artifice permits the damping and the stiffness to evolve in a continuous manner between y = H(x) et y = H(x)-ε. The law of regularization chosen was of cubic form and is noted f_{reg}. (cf figure 2)

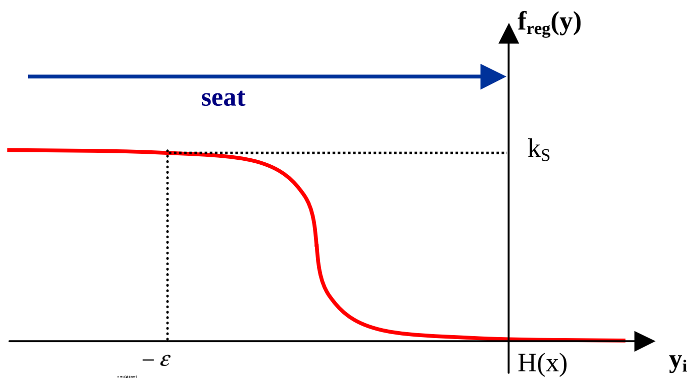

Figure 2: Representation of the regularization function

The introduction of two diagonal matrices K_S et C_S is necessary. They represent the values of damping and contact stiffness respectively at each node. Not all the nodes are involved simultaneously by the phenomenon of contact. In addition only the degrees of liberty that are in movement are affected by this force. All the values on the diagonals associated with the degrees of liberty in rotation or those in free flight are unaffected and set to zero.

If contact exists at a node the contact stiffness is expressed in the following form:

$$K_{Si} = k_S\, f(y_i)$$

The formulation for the damping is of exactly the same form.
This results in the following expression for the contact force:

$$F_S = -K_S Y - C_S \dot{Y}$$

The contacts on the retainer are modeled in a similar way.

Oil stiction

The oil is an important factor in determining the lifetime of the valve. Its characteristics the quantity and the distribution in the system are extremely important parameters. The oil is however only taken into account in the form of the fine film present between the seat and the valve. The heterogeneous nature of the milieu that its presence produces in the compressor is neglected at this stage. The model applied here is based on plate like valves and the oil is presumed to be incompressible. [4]

To describe the force which is produced by the oil film between the seat and the valve a simple law of flow through a capillary is used. Poiseuille's law permits the volume flow Q to be expressed as a function of several parameters to relate it to the true system:

$$Q = \frac{2\pi R}{12\eta} y^3 \frac{dp}{ds}$$

where R is the port radius, y is the valve displacement, η is the dynamic viscosity of the oil (depend on temperature), p is the pressure difference between the two faces of the valve and s is the curve abscissa in the direction of the flow.

The approximation of a linear pressure gradient is proposed at the interior of the film. This hypothesis permits the variation of the thickness, e, of the oil film to be described in terms of the variation of the volume of oil at the moment when the valve loses contact with the seat. The resulting expression includes the effects due to the geometry and to the debit.

$$\frac{de}{dt} y = -e\frac{dy}{dt} - \frac{y^3}{12\eta}\Delta p\, \frac{h^3}{e}$$

All these formulations are then coupled together and resolved simultaneously.

RESULTS AND DISCUSSION

Simulation Results

The computation permits to analyse the dynamics valves behavior of reed-valve system during the compression cycle. In order to illustrate how this tool can be used, in figure 3(a) are represented the compression chamber and the two reed valves. In the same figure, the pressure variation is presented as a function of the time noted (b), and the Pressure-Volume diagram for the cycle of the compressor is presented in curve (c).
The compressor rotates at a frequency of 50 Hz or 3000 rpm and the piston in the diagram is represented by a simple rectangle.

The suction or inlet valve is situated to the left side of the figure and the discharge or outlet valve is on the right. A cursor on the pressure/ time curve helps the user to see at which moment of the compression cycle the valves open or close. Fine analysis of p-V diagram can be carried out using an integrated marker.

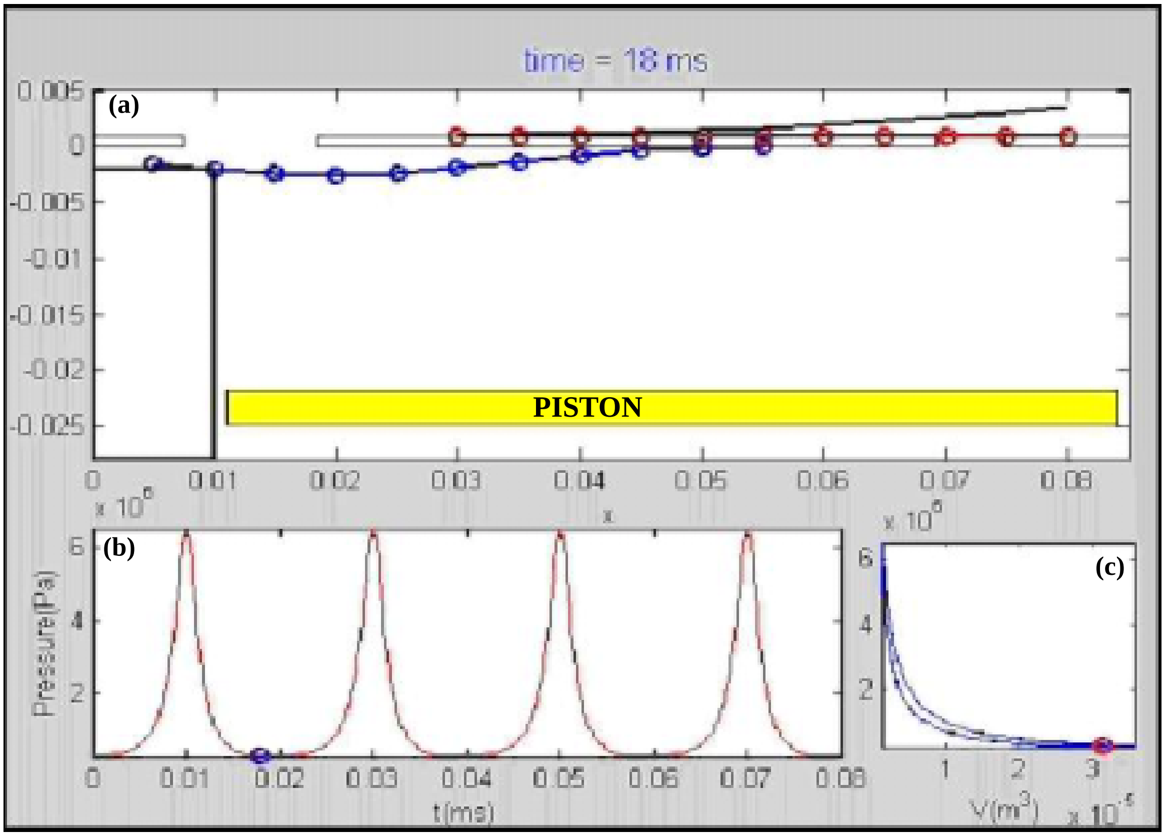

Figure 3: Representation of compression cycle at one given time.

The contact model used in the simulation tool gives an accurate estimate of overshoot and opening of the suction valve.

Experimental results

Tests have been made in order to validate the model in an initial phase. In the next phase, experimental test results will be used to tune the involved dynamic parameters. The experimental set-up consists of suction chamber, cylinder chamber, valve's seat, retainer and suction valve reed excited by compressed air at a controlled pressure. The contactless measurements are carried out using a vibrometer laser.

The valve reed is fed with air at constant pressure to obtain a specific opening position and the fluctuations caused by the turbulence induces vibrations at the natural frequencies of the system. The vibrometer signals enable the spectra of vibration velocity to be established around any steady state condition. In the figure below, the resulting spectra of the reed valve is shown over the frequency range 0-2500 Hz. This particular valve is 4.3 cm long and 1.8 cm wide at the clamped end with a thickness of 0.38 mm.

Measurements points are distributed at different points on the valve surface. Figure 4 shows the location of 2 of 18 points on the reed.

The first point is on the main axis of symmetry. The second at the same distance from the clamped end, but on one edge to detect torsional vibration components.

The vibration response measured at the point 1 can be use to investigate 2 first eigen frequencies. Those correspond to flexural modes at 186 Hz and 990 Hz, whereas the computation gives the first value at 170 Hz and the second at 1054 Hz. The simulation readings confirm the accuracy of the simulation technique described.

The vibration response measured at point 2 give a third eigen frequency. This supplementary frequency correspond to the first torsional mode at 1181 Hz.

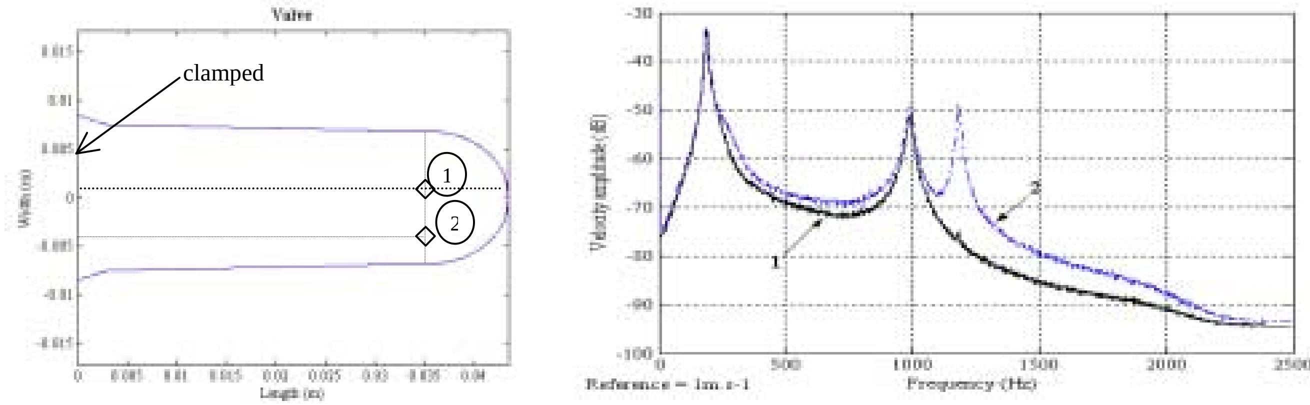

Figure 4: Diagram of the valve and results of vibration spectra at 2 points.

CONCLUSIONS

The model described in the paper takes into account many phenomena which according the literature influence the valve dynamics. The model consists of:

- A *finite element method* for the description of the valve structure,
- A thermodynamic simulation of the compression chamber with a *cubic law*,
- A contact model with a *penalty method*,
- An oil stiction model with a *Poiseuille's law*.

In the future work, new physical phenomena such as backflow around the ports are to be computed with the use of a fluid finite volume method and included in the model. Acoustics of the suction and discharge chamber are also to be modelised.

If all these phenomena are taken into account the computation time became a very important issue. The computation can be made for example on a modal basis when the reed of the valve is in free flight and is not submit to any nonlinear phenomena.

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