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EVALUATION OF A LINEARLY DAMPED MODEL TO PREDICT THE CLOSING MOTION OF A SPRINGLESS FLAPPER VALVE

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ABSTRACT

A linearly damped model, based on the application of Newton's second law, was applied to predict the closing time for a springless flapper valve. Viscous damping forces were assumed to be a linear function of the flapper velocity. The model predictions, including a delay time to account for the pressure lag at the flapper valve, were in good agreement with experimental data for a pulse combustor flapper valve. Agreement was obtained by adjusting the dimensionless damping coefficient; however, data for two flappers, which differed only in mass, were found to require different values of the dimensionless damping coefficient. It was concluded that a universal dimensionless damping coefficient can not be determined; therefore, experimental investigations would be necessary to determine the coefficient for a given valve geometry. The potential danger of using a linearly damped model with an incorrect value of the damping coefficient is demonstrated by the range of closing times presented. The present study should be a useful contribution to future modeling of flapper valves and other springless valves.

NOMENCLATURE

A_{eff}	effective force area of flapper [m ²]	χ	dimensionless displacement (x/x_{max})
D	damping coefficient [kg/s]	$\dot{\chi}$	dimensionless velocity ($d\chi/d\tau$)
$f(\tau)$	function of dimensionless time, τ	$\ddot{\chi}$	dimensionless acceleration ($d^2\chi/d\tau^2$)
m	mass of flapper [kg]	Ψ	dimensionless pressure parameter
P	pressure [Pa]	ω	angular frequency [rad/s]
t	time [s]	Subscripts:	
x	displacement of flapper [m]	a	amplitude
\dot{x}	velocity of flapper (dx/dt) [m/s]	c	combustion chamber, closing
\ddot{x}	acceleration of flapper (d^2x/dt^2) [m ² /s ²]	eff	effective
β	dimensionless pressure ratio (P_m/P_a)	m	mean
δ	dimensionless damping coefficient ($D/m\omega$)	max	maximum
τ	dimensionless time ($t\omega$)	u	upstream (intake decoupler or atmospheric)
ϕ	phase angle [rad]		

INTRODUCTION

An analytical investigation of a linearly damped motion model was undertaken in order to determine the suitability of the model for predicting the closing times of springless flapper valves. The specific application investigated was the air flapper valve in a Helmholtz type pulse combustor. The pulse combustor operation is characterized by a periodic combustion process that drives and is in phase with a fluctuating pressure. The fluctuating pressure forces the combustion products out of the combustor and draws reactants into the combustor. The flapper valves limit back flow during the high pressure portion of the cycle and allow inflow of reactants (air and fuel) during the low pressure portion of the cycle. The primary characteristic of the valve is that it is pressure driven both during closing and opening. Previous research related to flapper valves has been limited. Numerous studies of compressor valves have been conducted; however, these studies have focused primarily on reed valves and spring loaded valves, both of which have restoring forces which result in different behavior. Several investigators have formulated models to predict the motion of flapper valves, but none of these efforts has included comparison with experimental measurements of the actual flapper motion. The present study provides a critical evaluation of a linearly damped model used to predict the closing motion of springless valves. The results of this study should be applicable to other pressure driven valves (i.e., springless ring type compressor valves).

BACKGROUND

The air flapper valve is an important component of a pulse combustor because it controls the flow of combustion air into the combustion chamber and restricts back flow. A typical configuration for an air flapper valve is presented in Fig. 1. The main component of the flapper valve is the flapper which moves freely between the front plate (valve seat) and back plate (valve stop). Two flappers were used in the experimental study (details to be reported elsewhere) and were modeled in the present work, a Teflon® coated fiberglass flapper (fiber) with a mass of 1.5 grams and a steel flapper (metal) with equivalent dimensions and a mass of 5.5 grams. All other dimensions of the valve construction were held constant.

The ideal relationship between the flapper motion and the combustion chamber pressure is depicted in Fig. 2. The combustion chamber pressure, P_c , upstream of the valve, exhibits approximately sinusoidal variation and has been represented as such. Downstream of the valve the intake decoupler pressure (low amplitude) or atmospheric pressure, P_a , acts. The behavior of the flapper during closing differs from the behavior during opening in that back flow exists until the valve has closed fully, whereas no flow occurs during opening until the flapper moves. In both cases the flapper motion is driven by the pressure forces; there are no restoring forces as in spring loaded and reed valves.

The basic experimentally determined behavior of the flapper during closing is presented in Figure 3. The flapper experiences a delay in closing of approximately 10% of a cycle from the rise in the combustion chamber pressure above zero gage pressure. The flapper then moves for approximately 10% of a cycle until it reaches the front plate (valve seat). The following motion is a combination of small scale bouncing and bending of the flapper as well as combined bending with the front plate. Closing times were consistently measured from the combustion chamber zero pressure crossing.

There have been few experimental studies of flapper valve behavior. Xu et al. (1990) conducted an acoustic investigation of a flapper valve assembly and determined that the flapper valve was acoustically equivalent to a hard termination. Keel and Shin (1991) conducted a crude measurement of the flapper displacement; however, their results did not indicate the actual time for the closing or opening processes. Other detailed experimental studies of flapper motion could not be located in the literature.

There have been a number of analytical investigations of flapper valve behavior. Two models have been used to predict the closing motion for the flapper: an undamped model and a linearly damped model. The undamped model is based on the application of Newton's second law to describe the motion of the flapper and neglects all forces that would damp the flapper motion. Griffiths and Weber (1969) used this approach. The second approach, a linearly damped model, includes a damping force which is defined as the product of the velocity and a damping coefficient. The linearly damped model approach, including a numerical solution, has been used by Dhar (1980), Lee (1983), Van Essen (1995), Morel (1991), and Zinn et al. (1989). The primary difficulty in applying this model is in establishing a value for the damping coefficient. None of the flapper motion modeling efforts located in the literature included validation through experimental measurement of the actual flapper motion.

There have been numerous studies of valve behavior in compressors; however, they have focused primarily on valves with restoring forces (reed valves and spring loaded valves). MacLaren (1972) presents a summary of models for compressor valve motion. The linearly damped approach has also been suggested by Soedel (1992) for modeling of compressor valves. These valves will have behavior which differs significantly from that of springless valves; therefore, the results of the studies can not be applied directly to the present work.

MODEL DEVELOPMENT

The linearly damped model has the same basic form as the work of Dhar (1980), Lee (1983), and Van Essen (1995); however, a new dimensionless form of the governing equation and an analytical solution for the flapper displacement will be presented. The model is based on the application of Newton's second law to describe the motion of the flapper, with a force term to account for the damping of the flapper motion due to viscous forces. These forces are assumed to vary linearly with the velocity of the flapper. Applying Newton's second law to the flapper yields

$$m\ddot{x} = P_{net}A_{eff} - D\dot{x} \quad (1)$$

where the coefficient D is a proportionality constant referred to as the damping coefficient. If the net pressure acting on the flapper, P_{net} , is considered to be zero at time t equal to zero, then the corresponding initial conditions for the flapper motion are

$$x(0) = 0 \quad \text{and} \quad \dot{x}(0) = 0 \quad @ \quad t = 0 \quad (2)$$

Assuming the pressure upstream of the valve, P_u , is approximately atmospheric pressure and the combustion chamber pressure has a sinusoidal variation, the net pressure acting on the flapper can be expressed as

$$P_{net} = P_c - P_u = P_m + P_a \sin(\omega t + \phi) \quad (3)$$

where P_m , P_a , and ω are parameters describing the operating conditions and are considered to be inputs (constant) when conducting an analysis. The phase angle ϕ is not an independent parameter and is determined such that the pressure given by Eq. (3) is zero at $t=0$ and the pressure gradient is positive for the valve closing process. Therefore, ϕ can be expressed as

$$\phi = \sin^{-1} \left[-\frac{P_m}{P_a} \right] \quad (4)$$

Inserting the expression for the net pressure into Eq. (1), the governing equation becomes

$$m\ddot{x} = [P_m + P_a \sin(\omega t + \phi)]A_{eff} - D\dot{x} \quad (5)$$

The assumptions for the model can be summarized as follows:

- (1) the flapper moves as a rigid body (single degree of freedom),
- (2) the damping of the flapper motion varies linearly with the flapper velocity,
- (3) the combustion chamber pressure (sinusoidal variation with time) acts downstream of the valve, and
- (4) the pressure upstream of the flapper valve is approximately atmospheric pressure.

Dimensionless variables are defined as follows:

$$\chi = \frac{x}{x_{max}} \quad (6)$$

$$\tau = t\omega \quad (7)$$

$$\psi = \frac{P_a A_{eff}}{m\omega^2 x_{max}} \quad (8)$$

$$\delta = D/m\omega \quad (9)$$

$$\beta = P_m/P_a \quad (10)$$

Applying the dimensionless variables, Eq. (5) becomes

$$\ddot{\chi} = \psi[\beta + \sin(\tau + \phi)] - \delta\dot{\chi} \quad (11)$$

with the initial conditions

$$\chi(0) = 0 \quad \text{and} \quad \dot{\chi}(0) = 0 \quad @ \quad \tau = 0 \quad (12)$$

Integrating Eq. (11) and applying the appropriate initial condition, an expression for the dimensionless velocity is obtained as

$$\dot{\chi} = \psi[\beta\tau - \cos(\tau + \phi) + \cos(\phi)] - \delta\chi \quad (13)$$

Eq. (13) is then rearranged to obtain the form

$$\dot{\chi} + \delta\chi = \psi[\beta\tau - \cos(\tau + \phi) + \cos(\phi)] = f(\tau) \quad (14)$$

This equation has a solution with the general form of

$$\chi = e^{-\delta\tau} \left[\int e^{\delta\tau} f(\tau) d\tau + \text{Constant} \right] \quad (15)$$

The terms in brackets are obtained, after applying the initial conditions and much algebra, as

$$\int e^{\delta\tau} f(\tau) d\tau = \left(\frac{\psi}{\delta} \right) e^{\delta\tau} \cos(\phi) + \left(\frac{\beta\psi}{\delta^2} \right) e^{\delta\tau} (\delta\tau - 1) - \psi e^{\delta\tau} \left(\frac{1}{\delta^2 + 1} \right) [\delta \cos(\tau + \phi) + \sin(\tau + \phi)] \quad (16)$$

and

$$\text{Constant} = \left(\frac{\beta\psi}{\delta^2} \right) + \left(\frac{\psi}{\delta} \right) \left(\frac{1}{\delta^2 + 1} \right) (\delta \sin(\phi) - \cos(\phi)) \quad (17)$$

The flapper motion is subject to limits imposed by the front and back plates. For the closing process, the dimensionless flapper displacement will vary from 0 to 1 where $\chi = 1$ corresponds to the closed position.

The primary limitation of this model is the need to establish a value for the damping coefficient. The damping coefficient is not easily related (analytically) to the physical mechanisms which cause the damping of the flapper motion; therefore, the value of the damping coefficient is estimated for most computations.

From Eqs. (15)-(17), and given that

$$\phi = \sin^{-1} \left[-\frac{P_m}{P_a} \right] = \sin^{-1}(-\beta) \quad (18)$$

the relevant dimensionless parameters which determine the flapper closing time based on the linearly damped model can be summarized as follows:

$$\tau_c = t_c \omega = \tau_c \left(\frac{P_a A_{\text{eff}}}{m \omega^2 x_{\text{max}}}, \frac{P_m}{P_a}, \frac{D}{m \omega} \right) = \tau_c(\psi, \beta, \delta) \quad (19)$$

RESULTS AND DISCUSSION

Model predictions of the flapper closing times are presented in Fig. 4, along with the experimental data. The model predictions presented are for a value of the dimensionless pressure ratio parameter of $\beta=0.15$; however, it can be shown that the results are essentially unchanged for $0 \leq \beta \leq 0.3$. The experimental data had a value of the pressure ratio in the range $0.1 \leq \beta \leq 0.2$. Model predictions are provided for several values of the dimensionless damping coefficient. The model underpredicts the closing times and does not account for the apparent delay in the flapper motion which is shown in Fig. 3. This behavior was found to be a result of the pressure at the flapper valve lagging the pressure at the combustion chamber at the time of the valve closing. A delay of approximately 11% of a cycle ($\tau=0.7$) was observed and apparently accounts for the delay in the flapper closing.

Because the combustion chamber pressure is normally available for analysis of the flapper valve behavior, an attempt was made to predict the flapper closing with the combustion chamber pressure and a delay time. A dimensionless delay time of 0.7 (11% of a cycle) was added to the model results to account for the pressure delay. The resulting flapper closing times are presented in Fig. 5. The model can be forced to produce the correct closing time by adjusting the dimensionless damping coefficient. For the metal flapper a value of approximately 10 is required for the dimensionless damping coefficient; while, a value of 25 is required for the fiber flapper. There apparently is not a universal value for this parameter, requiring that it be determined experimentally. There is also the need to establish a value for the delay time. A brief investigation of the dependence of the delay time on the air inlet pipe length indicated that it was not a strong function of this variable. It is believed to be determined by the boundary conditions imposed by the flapper assembly at the end of the air inlet pipe. A representative plot of the dimensionless displacement is presented in Fig. 6 for $\psi=180$ and $\beta=0.15$. The model results, which include a delay time, agree reasonably well with the experimental data.

CONCLUSIONS

The model predictions for the linearly damped model were compared to experimental data for the closing of a pulse combustor flapper valve. The model predictions were found to have poor agreement with the experimental data when the combustion chamber pressure was used as the driving force for the flapper motion. When the combustion chamber pressure was used as the driving force and a delay time was added to the results, the linearly damped model results were in good agreement with the experimental data when an appropriate value of the dimensionless damping coefficient was selected. This dimensionless coefficient was found to differ when the only physical parameter changed was the mass of the flapper. Therefore, it was concluded that a universal dimensionless damping coefficient can not be determined. This is the primary concern with using the linearly damped model; experimental measurements are necessary to determine the value of the damping coefficient. The results of this study emphasize the care that must be exercised when applying the linearly damped model to predict the closing time of springless valves.

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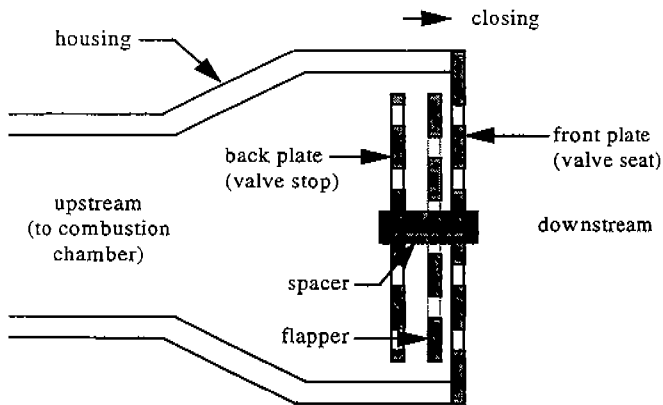


Figure 1: Air Flapper Valve Construction

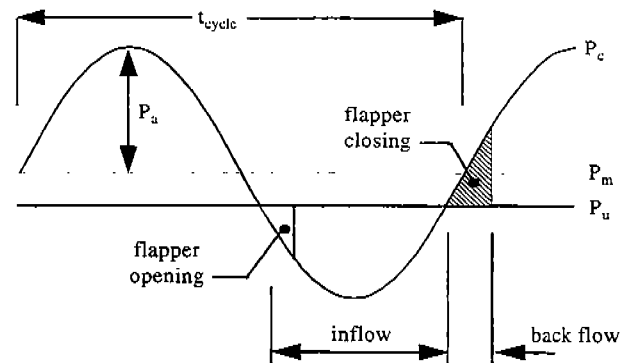


Figure 2: Idealized Pressure and Flapper Behavior

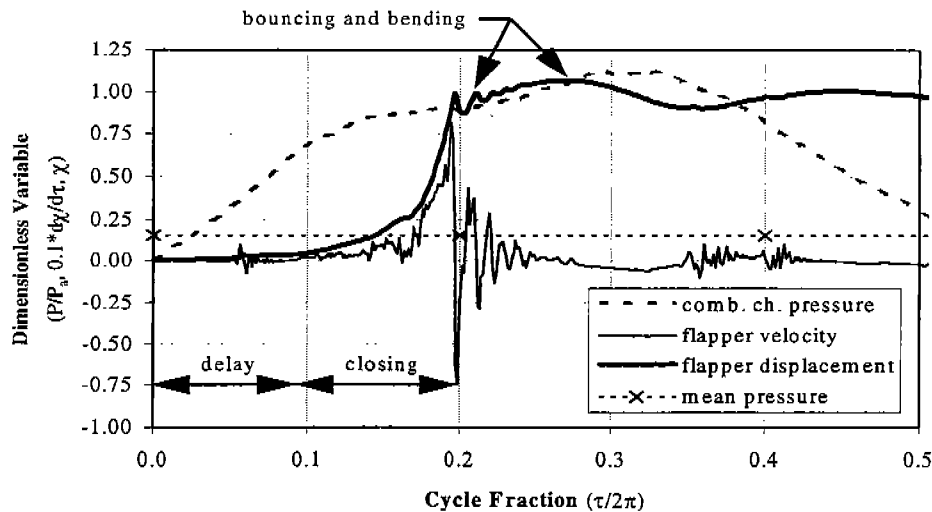


Figure 3: Typical Flapper Closing Behavior ($\psi = 180$, $\beta = 0.15$)

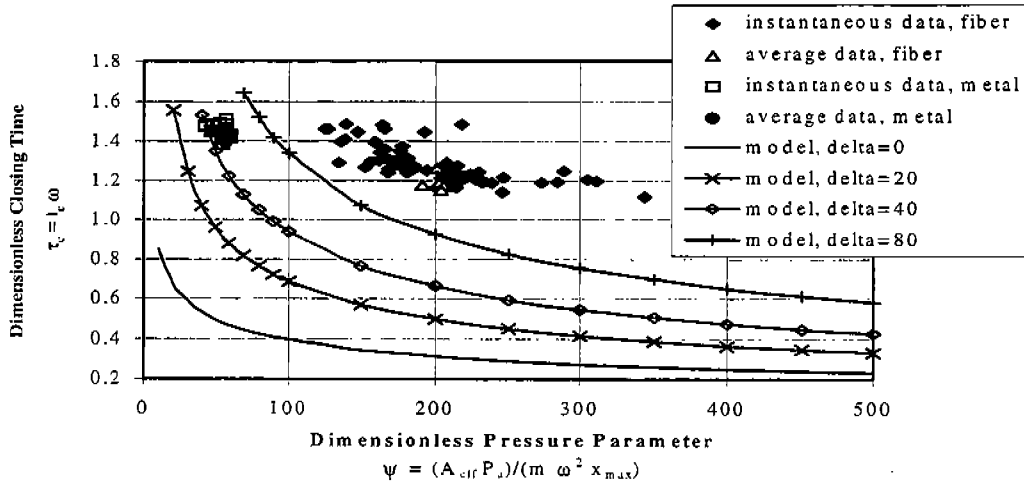


Figure 4: Flapper Closing Time, Experimental Data and Model Predictions

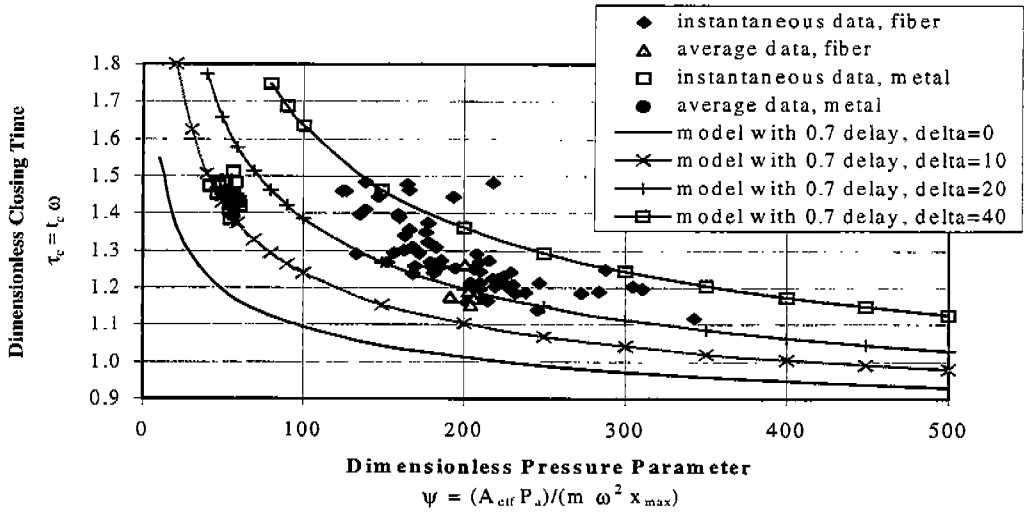


Figure 5: Flapper Closing Time, Experimental Data and Model Predictions with Delay

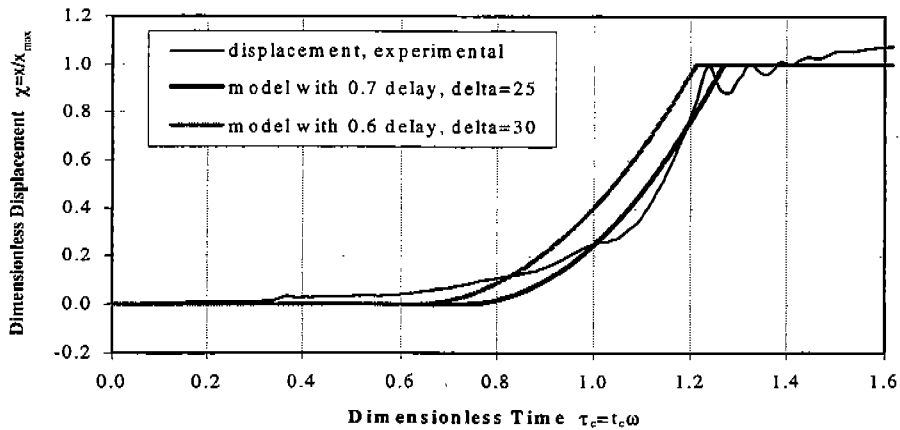


Figure 6: Flapper Displacement During Closing ($\psi = 180$, $\beta = 0.15$)