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XI

INTERNATIONAL SCIENTIFIC AND PRACTICAL CONFERENCE "INTEGRATION OF SCIENCE AS A MECHANISM OF EFFECTIVE DEVELOPMENT”

Helsinki, Finland

November 28 - December 01, 2023

ISBN 979-8-89238-623-4

DOI 10.46299/ISG.2023.2.11

INTEGRATION OF SCIENCE AS A MECHANISM OF EFFECTIVE DEVELOPMENT

Proceedings of the XI International Scientific and Practical Conference

Helsinki, Finland
November 28 - December 01, 2023

UDC 01.1

The 11th International scientific and practical conference “Integration of science as a mechanism of effective development” (November 28 - December 01, 2023) Helsinki, Finland. International Science Group. 2023. 475 p.

ISBN – 979-8-89238-623-4
DOI – 10.46299/ISG.2023.2.11

MATHEMATICAL MODELING OF REED VALVE OPERATION IN ENGINES WITH PERIODIC WORKFLOW

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The intake reed valve is a common design element of various engines and units (Fig. 1). This type of the check valve is used mainly in 2-stroke gasoline engines with crank-chamber purging [1], where this type of valve is quite well developed both experimentally [2] and theoretically [3]. In addition, reed valves are widely used in air compressors [4], and a significant number of works have also been devoted to their research [5]. In addition, reed valves are used in elements of hydraulic systems [6] and also in pulse jet engines [7].

Interest in modeling a reed valve operation is associated with three reasons. On the one hand, it is necessary to correctly select the dimensions and parameters of the valve when designing an object (compressor or engine), and then the valve operation can be studied separately from the object. On the other hand, the valve model must be integrated into a complete mathematical model of an object of varying complexity, since the reed valve is an integral element of the intake system. Then the operation of the valve is considered as part of the operation of the entire object.

Finally, it should be noted that the reed valve operates under conditions of significant dynamic loads. As a result, fatigue failure to the valve is possible (Fig. 1a, b, c), and the most serious damage occurs with additional temperature exposure (Fig. 1d). In this case, the task of modeling the valve operation can be carried out in order to determine or clarify the cause of destruction.

When creating the valve mathematical models, several fundamentally different approaches are used. Thus, in some works [8] the problem of static loading of a valve petal from a constant pressure drop is considered. In this case, the valve petal deflection is assumed to be quasi-static (quasi-stationary) and is determined only by the pressure drop. However, the vast majority of studies address dynamic problems of valve motion.

Based on this, the purpose of the study is to evaluate the effectiveness of using simple models of a reed valve installed in the intake channel adjacent to the control volume (working cylinder or combustion chamber), as a component of 0-dimensional closed thermodynamic models of the engine working cycle or as an independent model when investigating the failure causes.

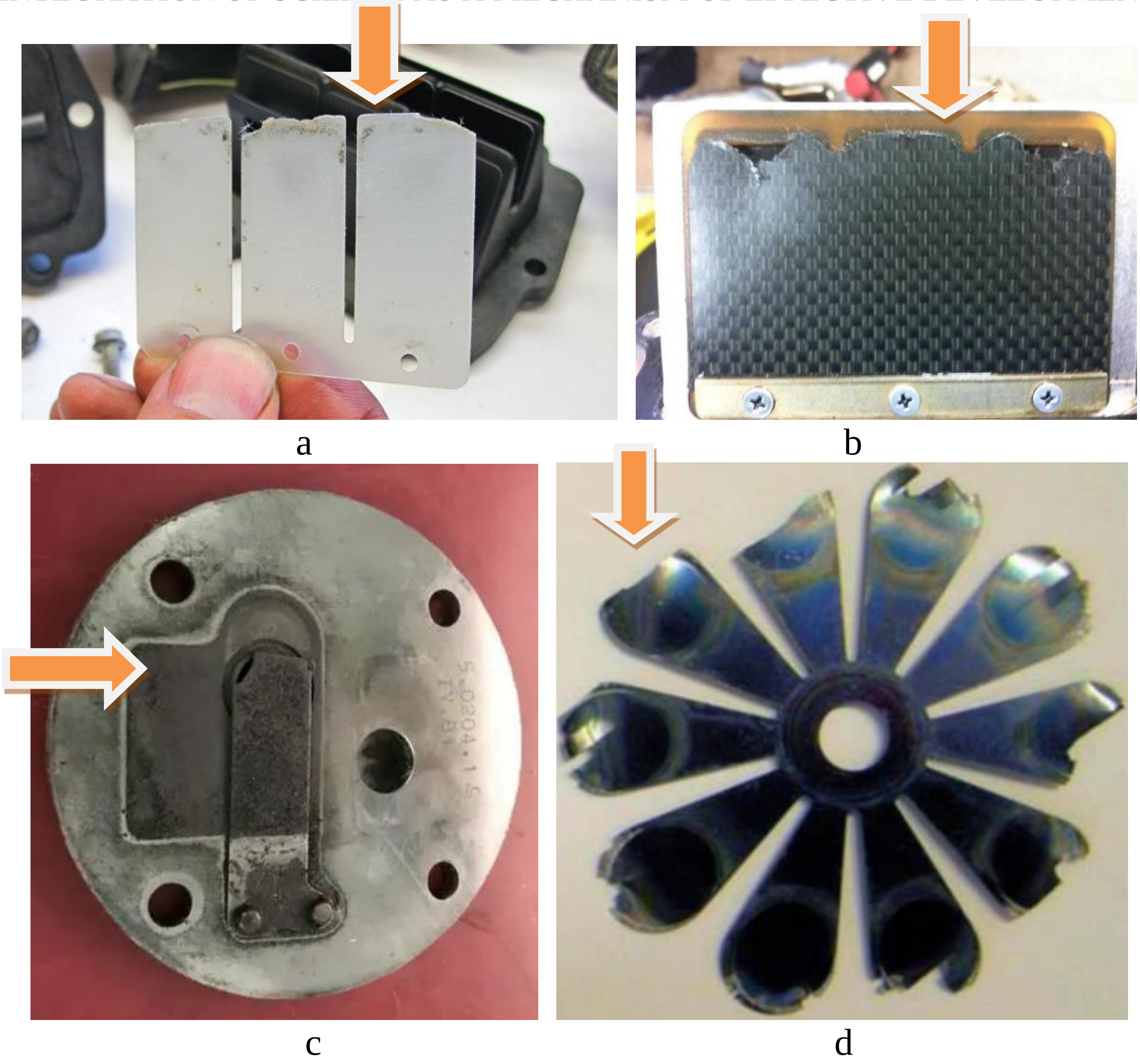

Figure 1. Reed valves and their damage in operation: cold fatigue cracking of metal (a) and plastic (b) valve petals in two-stroke gasoline engines and compressors (c), damage as a result of complex thermo-mechanical effects in small-sized pulse jet engines (d)

To achieve the goal above, it seems necessary to do the following tasks:

- consider the engine operating cycle model and identify the influence of the intake valve on the operating process,
- determine air flow rate as a parameter by which valve models can be compared,
- develop the valve models; perform mathematical modeling of the valve petal motion,
- based on the data obtained, conduct a comparative analysis of the accuracy and reliability of the models in relation to the type of engine under consideration, as well as evaluate the limits of their applicability.

In the general case, an engine with periodic workflow represents a certain control volume (cylinder, combustion chamber), to which the intake and exhaust systems with the corresponding channels (pipes) are connected [9]. In this case, the inlet pipe ends with a reed valve, which allows air into the volume in the forward direction (with a positive pressure drop between the environment and the volume), but closes with a negative one.

Air flow rate is a parameter that integrally characterizes the process of air flow through the reed valve under consideration [10]. If we do not take into account the dynamic pressure at the inlet (in some types of engines with periodic workflow it is not present, i.e. equal to ambient pressure), then for the dimensionless instantaneous air flow rate we can obtain the formula:

$$\frac{dm_e}{dt} = \frac{d\bar{m}_e/d\bar{t}}{\bar{\rho}_0\bar{a}_0F_a} = \frac{1}{\sqrt{1+\xi_\Sigma}}\Phi p^{\frac{1}{k_o}}\sqrt{\frac{2}{\gamma_0-1}\left(1-p^{\frac{\gamma_0}{\gamma_0-1}}\right)}. \quad (1)$$

where $\Phi = F_e/F_a = (D_e/D_a)^2$ is dimensionless parameter; D_e, D_a are diameters of the input channel and resonance pipe of an engine with periodic workflow; ξ_Σ is coefficient of hydraulic resistance [11] of the pipe and valve (valve system); γ_0 = 1,4 is heat capacity ratio; $p = \bar{p}/\bar{p}_0$ is dimensionless pressure in the control volume; $\bar{a}_0$ is air speed of sound; $\bar{p}_0, \bar{\rho}_0$ are ambient pressure and density; $t = \bar{t}\bar{a}_0/L$ is the dimensionless process time, L is the characteristic size of the engine (the overbar indicates dimensional values).

For a quasi-stationary setting, the valve petal lift is completely determined by only two factors: the distributed force (pressure drop Δp) and the valve petal stiffness. If the valve petal is a beam rigidly fixed on one side, which is acted upon by a distributed force $\bar{q}$ (Fig. 2a), then according to [11], the dependence of the lifting of the valve petal end on this force looks like this:

$$y = \frac{\bar{y}}{l} = 1{,}5\,\frac{\Delta \mathrm{p}}{\mathrm{S}}\,, \quad (2)$$

where l is the length of the valve petal; $\Delta \mathrm{p} = \Delta\bar{\mathrm{p}}/\bar{p}_0$ is the dimensionless pressure drop across the valve, and the dimensionless parameter S actually determines the rigidity of the valve petal:

$$S = \frac{\bar{E}}{\bar{p}_0}\left(\frac{\delta}{l}\right)^3. \quad (3)$$

where $\bar{E}$ is the elastic modulus of the material; δ is the valve petal thickness.

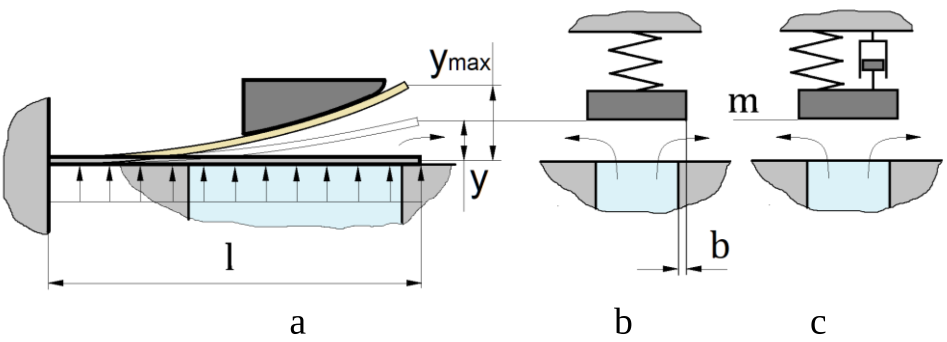

Figure 2. View of reed valve in the form of a beam with a fixed seal (a), in the form of a spring-loaded mass (b) with damping (c).

In general, the valve petal is affected by the forces of gravity, pressure and elasticity. Excluding the force of gravity, which is negligible compared to others, this allowed us to write the equation for valve reed motion in the form:

$$m\frac{d^2\bar{y}}{d\bar{t}^2} = -K\bar{y} + \Delta\bar{p}A\,, \tag{4}$$

where K is the valve stiffness; A is valve petal area (to the 1st approximation, it can be equal to the area of the hole opened by the petal).

To solve equation (4), it is necessary to reduce its order. This was done by transforming it into a system of equations for the 1st order derivative of velocity (acceleration) and the 1st order derivative of valve lift (velocity):

$$\begin{cases} \theta\dfrac{du}{dt} = -\dfrac{2}{3}Sy + \Delta p \\ \lambda\dfrac{dy}{dt} = u \end{cases}, \tag{5}$$

where $\theta = \gamma_0\rho_M\lambda\left(\frac{\delta}{l}\right)$ is dynamic coefficient; $\rho_M = \bar{\rho}_M/\bar{\rho}_0$ is dimensionless density of the valve blade material; $\lambda = l/L$ is relative length of the valve petal.

System (5) was solved numerically with initial and boundary conditions. Initial conditions were: y = 0, u = 0 at t = 0. Boundary conditions determine the rebound of the valve petal from the valve block during landing and from the stop plate at maximum lifting: if $y < 0$, then $y = -\beta y$, $u = -\beta u$, and if $y > y_{max}$, then $y = 2y_{max} - \beta y$, $u = -\beta u$ is accepted.

The coefficient of recovery (COR) lies in the range of β = 0 – 1, which corresponds to the range from completely inelastic to completely elastic impact. The calculations assumed an analogy of the interaction of a steel ball with a steel plate, in which the value of the coefficient β is 0.4 [12].

In the calculations, other quantities included in the equations were taken equal δ = 0.00025 m, l = 0.025 m, L = 0.80 m, y_{max} = 0.2, D_e= 0.035 m, D_a = 0.04 m, which approximately corresponds to some engines of the type under consideration [13].

The pressure drop across the valve was modeled by specifying pulsations (forced oscillations) of air pressure in the control volume according to a sinusoidal law. The oscillation frequency varied from $f = Sh$ = 0.1-0.7. The amplitude of oscillations was assumed to be constant and equal to $\Delta p = 0.3$. The air flow rate was calculated based on the condition that 10 reed valves are installed in the valve block according to Fig.1d.

The results of calculating the valve petal lift using the quasi-stationary model are presented in Fig. 3a. For comparison, Fig. 3b shows the results of calculations of the valve petal lift using a dynamic model at the same frequency of forced oscillations.

As follows from the obtained curves, in dynamics, the valve petal has a rebound from the stop plate and from the valve block, which causes the air backflow after impact on the valve block.

When modeling at high frequencies, a dynamic lag was revealed both when opening and closing the valve petal with increasing process frequency (Fig. 4). There is also a significant increase in the amplitude of petal speed fluctuations and especially the speed at the moment the valve petal impacts the valve block (Fig. 4a), which can affect reliability and durability. With increasing frequency of forced oscillations, the difference between the models grows rapidly (Fig. 4b). In addition, with increasing frequency, the phase lag of the petal lift and air flow curves increases.

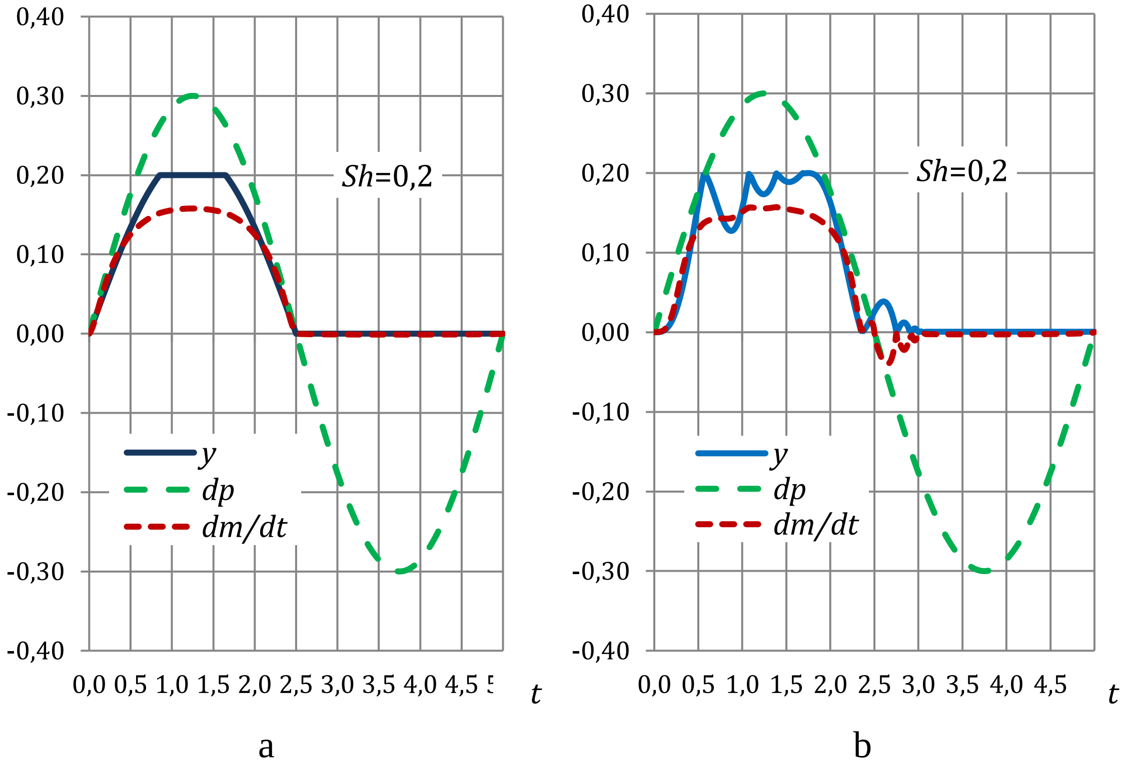

Figure 3. Comparison of the results of modeling a reed valve: a – quasi-stationary model; b – dynamic model

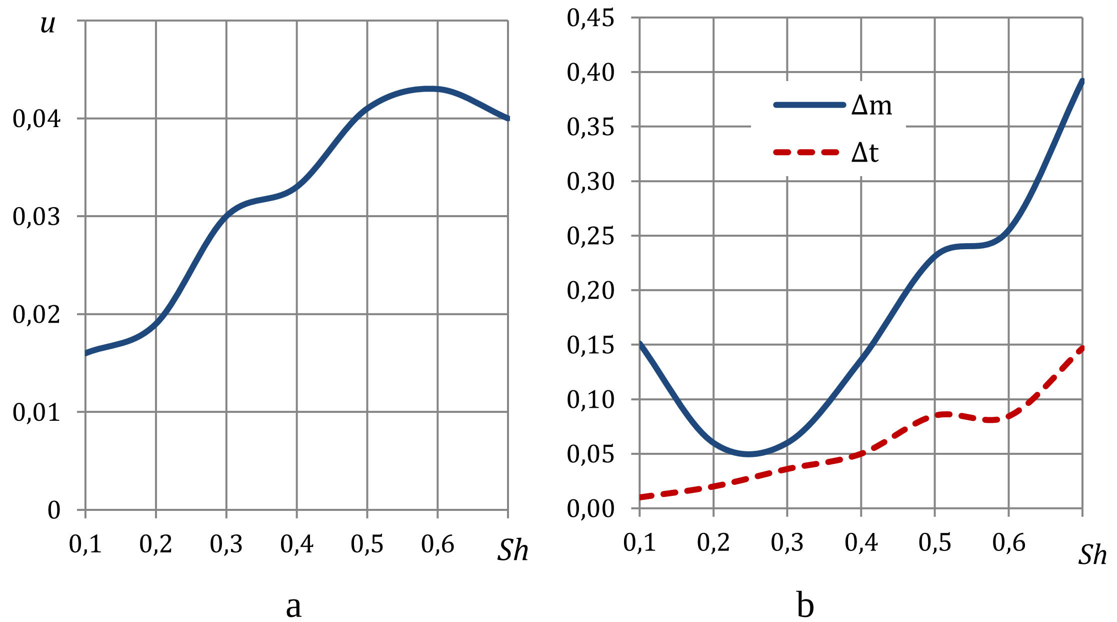

Figure 4. The resulting curves of parameters depending on the frequency of the process (Strouhal number): a – the speed of the valve petal at the moment of impact on the valve block; b – error of the quasi-stationary model in terms of air flow rate and oscillation phase lag

The found nature of the change in parameters leads to the fact that at frequencies with *Sh* below 0.1-0.15 the difference in air flow rate between models reaches 15%. Additionally, at *Sh* above 0.4-0.5 it exceeds 25% or more. In practice, this means that the quasi-stationary model is not applicable in the indicated ranges - it does not take into account dynamic phenomena and simply gives an excessive error, overestimating the air flow rate. In this regard, it is important to determine the realistic frequency range corresponding to real engines.

As an example, we consider one of the known types of engines with a periodic workflow – a valved pulse jet engine. As follows from Table 1, the Strouhal number for this type of engine is quite tightly grouped around the value of 0.30. It is near this value that the difference in air flow rate for the reed valve models under consideration is minimal (according to Fig. 4b, in the range of the Strouhal number Sh = 0.2-0.3 the difference is only 5-6%).

For 2-stroke gasoline internal combustion engines with crank-chamber purge [1, 3], there is a certain analogy to the working process of the reed valve with pulse jet engines, since the frequency values given in Table 1 approximately correspond to rotation speed in maximal modes. Then, with similar valve petal sizes, the operating range of rotation speeds will be located on the left side of the diagram (Fig. 4b), where the error of the quasi-stationary model is minimal or small.

It follows that the quasi-stationary reed valve model for these engines is quite applicable and can be used, at least at the preliminary stage of creating and debugging a general 0-dimensional thermodynamic model of the engine operating cycle.

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