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ANALYSIS OF THE FLOW FORCE IN THE FUEL COMPONENTS SUPPLY VALVES OF THE AIRCRAFT ENGINES

Asgat G. Gimadiev, Dmitry M. Stadnik and Dmitry S. Bratchinin
Samara State Aerospace University, Samara, Russia

ABSTRACT

With the use of the numerical modeling the process of opening of the main fuel valve of the aircraft propoulsion system is investigated. The impact of the flow force and throttling channels on the transient characteristics of the valve have been taken into account. There has been stated the possibiity of use of the developed valve design for calculation of the dynamic characteristics of the aicraft engines (ACE).

Keywords: ACE, main fuel valve, flow force, calculation (analysis), characteristic curves, static and dynamic characteristics.

1. INTRODUCTION

The fuel component supply valves are mounted in the ACE to ensure the pump filling before starting and component supply at the start directly. One of the most important issues is the timing of the actuation thereof. This is especially relevant if we take into consideration different pressures (flows) of the components and hydraulic characteristics of the fuel supply pipelines. In this regard the detailed analysis of the valve dynamic properties and assessment of the sensitivity thereof to the deviation of some or other parameters is required. As of today there are methods of calculation of the parameters of such units and analysis of their static and dynamic characteristics [1] which are considered to be approximate because of the inaccurate allowance for the hydrodynamic forces appearing in the shut-off-and-regulating elements. In the paper provided by the authors the specified gap is filled by the example of the analysis of the main fuel valve with the use of the software suites MATLAB/Simulink and ANSYS.

2. MATERIALS AND METHODS

The scheme of the valve under consideration is provided in the Figure-1, where 1 is the body; 2 - feedback line; 3 - mechanical stop; 4 - “cap”; 5, 6 - seal; 7 - nozzle; 8, 10 - spring;; 9 - dynamic seal;. By derivation of the equation of motion of the valve moving part the following allowances have been accepted: the bulk flexibility of the valve body is not taken into account because of its smallness; the hydraulic losses in the throttling elements are accounted according to the quasi-stationary model; there is no dry friction in the moving parts; the leak between the moving parts of the valve is negligible.

With account for the allowances accepted the derivation of the valve equations is presented on the basis of which its static and dynamic characteristics may be derived.

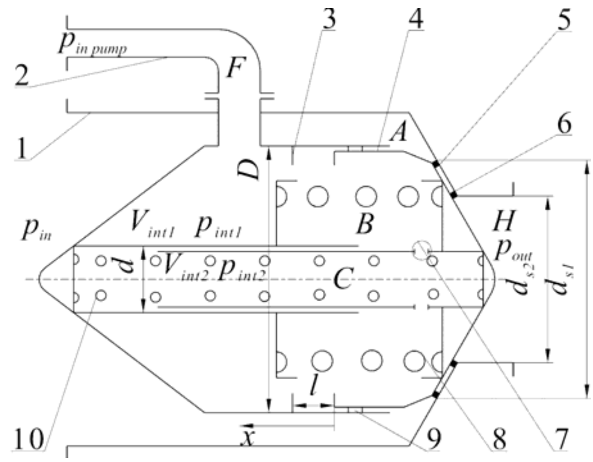

Figure-1. Analytical model of a differential valve.

The equation of motion of the moving valve part is of special interest in terms of analysis of the dynamic valve characteristics:

$$M \cdot \frac{d^2x}{dt^2} + \lambda_{fric} \cdot \frac{dx}{dt} + \gamma_{spr} \cdot x = N_{flow} - N_{spr} - \\ - F_{int\,1} \cdot p_{int\,1} - F_{int\,2} \cdot p_{int\,2}, \tag{1}$$

where M – the reduced mass of the moving part of the valve, kg; x - valve coordinate along its axis, m; t - time, s; $\lambda_{fric} = \mu \cdot l_{seal} \cdot b_{seal} / \delta_{seal}$ - viscous friction factor; μ - dynamic fluid viscosity, $Pa \cdot s$; $l_{seal} = \pi \cdot D$ - perimeter of the seal circle, m; D_{seal} - seal diameter, m; b_{seal} - seal width, m; δ_{seal} - gap between the seal and the body, m; γ_{spr} - total spring stiffness, N/m; N_{flow} - hydraulic force acting on the “cap” from the side of the valve flow channel, N; N_{spr} - total spring preload, H; $F_{int\,1} = \pi \cdot D^2 / 4 - \pi \cdot d^2 / 4$ - area of the cavity B, m^2; p_{int1} - pressure in the cavity B, Pa; $F_{int\,2} = \pi \cdot d^2 / 4$ - area of the cavity C, m^2; p_{int2} - pressure in the cavity C, Pa.

The hydraulic force arises as the result of action of the fluid flow passing through the valve on the valve "cap" and makes the sum of the hydrostatic and hydrodynamics forces:

$$N_{flow}=c_{hydrostat}\cdot p_{out}\cdot F_p+c_{hydrodyn}\cdot F_p\cdot \rho\cdot v^2/2 \tag{2}$$

where $c_{hydrostat}$ is the hydrostatic force coefficient; p_{out} - pressure at the valve output, Pa; $F_p=\pi\cdot D^2/4$ - area of the valve "cap", m^2; $c_{hydrodyn}$ - hydrodynamic force coefficient; ρ - fluid density, kg/m^3; v - average flow velocity at the "cap", m/s.

The coefficient $c_{hydrostat}$ accounts for the distribution of the static pressure over the "cap" surface. The coefficient $c_{hydrodyn}$ is similar to the coefficients derived in hydromechanics for determination of forces acting on a stream-lined body. The coefficients $c_{hydrostat}$ and $c_{hydrodyn}$ depend on the structure of the flow around the valve "cap".

The average rate of the fluid flow around the "cap" is calculated based on the flow of the fluid through the valve:

$$v=G/(\rho\cdot\mu\cdot F) \tag{3}$$

where G - mass rate of flow through the valve flow passage, kg/s; $\mu\cdot F$ - average effective area of the flow passage around the "cap", m^2.

The equations describing the fluid flow in the under-cap cavities by the "cap" motion:

$$F_d\cdot\rho\cdot\frac{dx}{dt}=\frac{V_{int\,2}}{c^2}\cdot\frac{dp_{int\,2}}{dt}+ +\mu_{tr}\cdot F_{tr}\sqrt{2\cdot\rho\cdot(p_{int\,1}-p_{int\,2})}, \tag{4}$$

$$F_D\cdot\rho\,\frac{dx}{dt}+\mu_{tr}\cdot F_{tr}\sqrt{2\cdot\rho\cdot(p_{int\,1}-p_{int\,2})}= =\frac{V_{int\,1}}{c^2}\cdot\frac{dp_{int\,1}}{dt}+G_{pipe}\;, \tag{5}$$

$$L_{pipe}\cdot\frac{dG_{pipe}}{dt}+R_{pipe}\cdot G_{pipe}=p_{int\,1}-p_{in\;pump} \tag{6}$$

where V_{int2} - capacity of the under-cap cavity C, m^3; c - velocity of sound in the fluid, m/s; $\mu_{tr}\cdot F_{tr}$ - total effective area of nozzles, m^2; V_{int1} - capacity of the under-cap cavity B, m^3; G_{pipe} - fluid flow through the pipeline, kg/s; $L_{pipe}=l_{pipe}/F_{pipe}$ - specific acoustic inductance of the pipeline, $1/м$; l_{pipe} - pipeline length, $м$; F_{pipe} - pipeline passage area, m^2; $R_{pipe}=128\cdot\nu\cdot l_{pipe}/\left(\pi\cdot d^4_{pipe}\right)$ - friction of piping, $1/(m\cdot s)$; ν - kinematic viscosity of fluid, m/s^2; d_{pipe} - pipeline passage diameter, m; $p_{in\;pump}$ – upstream pressure, Pa.

The fluid flow through the valve passage is determined by the dependence:

$$G=\mu_x\cdot F_x\sqrt{2\cdot\rho\cdot(p_{in}-p_{out})} \tag{7}$$

where $\mu_x\cdot F_x$ –is the effective valve area, $м^2$; p_{in} – pressure at the valve inlet, Pa.

Thus, the system of the differential and algebraic equations (1)… (7) is the mathematic model of the valve on the basis of which its dynamic characteristics may be calculated and the design parameters of the valve for meeting the requirements set may be selected.

3. RESULTS

In the mathematic model of the valve the main difficulty is the analytical description of the hydraulic force acting on the "cap". This is because, as was mentioned above, the hydraulic force depends on the structure of the flow around the "cap". At the same time the channel formed by the "cap" section and body in real-life structures may be of a rather complex shape which makes the task of calculating the relevant hydraulic force even more difficult. Let's consider the analysis of the hydraulic force and valve throttling characteristics with the use of the numerical modeling methods. For this purpose in the software suite NX the geometrical model of the valve is built and on the basis thereof the hydraulic domain of the valve passage is built on with account for the requirements to the analytical model.

In the software suite ANSYS the finite-element model has been developed and the hydrodynamic parameters have been analyzed. The example of the computed parameters is presented in the Figure. where 1 is the reverse-flow area.

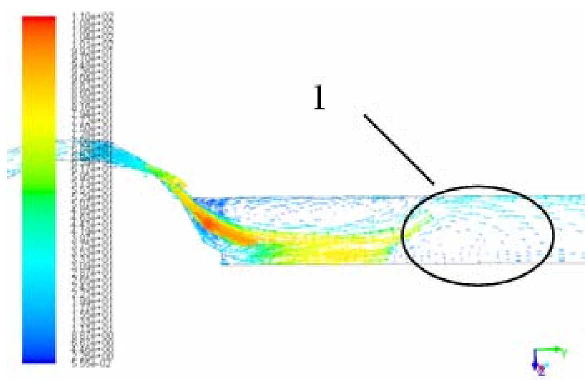

Figure-2. Computed velocity fields for the valve opened by 1/3.

On the basis of the computed parameters the throttling characteristics of the valve in the completely opened and in the two transient positions have been modeled. By the full-open valve the estimated and experimental characteristics match completely which gives evidence of the adequacy of the analytical model. The non-coincidence of the estimated and experimental characteristics in the valve open positions is explained by the appearance of secondary losses in the valve passage which has not been observed at the full-open valve.

Since the estimated throttling characteristics, as was mentioned above, are adequate to the real-world process the same conclusion may be drawn in respect of the other estimated parameters as well including those for the hydraulic force (Figure-3).

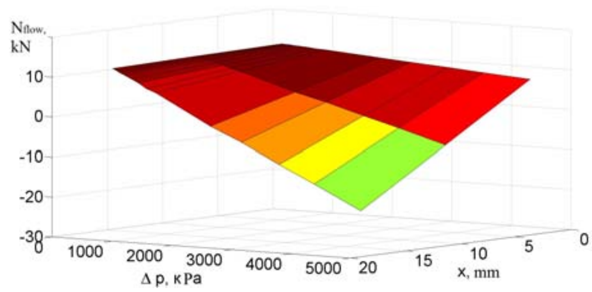

Figure-3. Dependence of the hydrodynamic force at the valve on the pressure ratio and “cap” stroke.

The given approximating dependence allows accounting for the complex nature of the fluid flow through the valve passage by analysis of its dynamic characteristics according to the above-mentioned equations (1)…(7) with the use of the numerical methods in the software suite MATLAB/SIMULINK.

The procedure for analysis of the valve parameters that has been developed by the authors is the basis for solution of the optimization tasks on the selection of the valve design parameters that will ensure the required static and dynamic characteristics.

The transient characteristic of the valve are calculated according to the control action for which the pressure difference at the valve inlet is taken. The analysis accounts for different nozzle diameters (Figure-1, Item 10).

In its initial position the valve is closed. By supply to the inlet of the pressure exceeding the opening pressure the valve is opened and the component is supplied to the consumer. The analysis was performed at the stepped increase in the pressure at the valve inlet from 0, 6 MPa to 22 MPa at the nozzle diameter 1, 2 and 6 mm.

By reduction of the nozzle diameter from 6 to 1 mm the valve opening time was increased from 5, 7 ms to 9, 3 ms. However, at the same time the peak of the mass flow rate at the outlet is shifted from 2, 2 ms to 2, 6 ms, the mass flow rate value is established after 4 ms. The transient pressure fall process shows an insignificant fibration. Thus, the change of the nozzle diameter allows performing the fine tuning of the valve opening time within a rather narrow time range. For a wider adjustment of the valve opening time the valve springs may be replaced, where it is not possible the throttle plate (orifice) may be mounted into the feedback line. By doing so the suggested mathematical model is slightly modified taking into account the presence of the plate in the pipeline.

5. CONCLUSIONS

The designed mathematical model of the valve may be used for analysis of the ACE dynamic characteristics as well as for forecasting of the change of its performance by changing of the valve parameters, for example, by solution of the optimization tasks or study of the impact of the parameter variety during the manufacturing process.

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