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SELECTED ISSUES OF MODELING THE ACCUMULATOR INJECTION SYSTEMS IN NAVAL COMBUSTION ENGINES

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Abstract

In the paper has been made an attempt to replace the conventional system of fuel dose control and injection passing angle with electronic control system, which has been realized this manner, that selected hydraulic accumulator, injector C-R type and fuel dose controller has been attached the marine engine.

It has been assumed, that by controlling the current impulse of controller it is possible to model whichever size of fuel dose, injected to the combustion chamber and to control the injection lasting time.

Fuel which feeds the engine must flow through differently formed/shaped channels of particular component elements of fuel system. In relation to this, there has been made an attempt of presenting the method of modelling fuel flow through the chosen sections of fuel system with different shapes of cross-section.

Keywords: *damping by the parallel surfaces, flow through the apertures, energetic loss*

1. Introduction

In the construction of model describing the course of complex phenomena occurring in the process of fuel injection in bunker systems it is essential to consider many specific problems connected with the flow of this liquid through channels with complex shapes.

In this paper there has been made an attempt of describing the fuel flow through the apertures, pipes with section different from the circular, also the manner of calculating the hydraulic resistance of two flat surfaces has been presented.

The significant problem at estimating the operation of hydraulic appliances, which are parts of gear feeding the vessel engines by fuel under high pressure, is the proper estimation of tightness of moving connections of parts cooperating with each other.

In the mobile joints between two cooperative parts (e.g. between piston and cylinder in the piston pump, between rotor and housing in the gear pump, between needle and atomizer cylinder) the narrow apertures always arise what is conditioned by defined constructional backlash. As far as the basic criterion for flows in the pipes is obtaining the lowest resistances, the aim is to obtain the highest resistance in the flows through apertures that guarantees possibly lowest flow delivery.

2. Flows through the apertures between needle and atomizer cylinder

Considering the very low size of one of cross dimensions of aperture, flow through the aperture is usually a laminar motion [1,2,3]. It was experimentally affirmed that Reynolds' critical ordinal for apertures (related to the minimal size of aperture), when the transition from laminar flow to turbulent flow occurs, ranges

$$Re = \frac{v_{\acute{s}r} \cdot s}{\nu} \approx 600 \div 1000,$$

where:
$v_{\acute{s}r}$ – mean velocity in the aperture,
s – minimal size of aperture,
ν – kinematic viscosity coefficient.

2.1. Frictional flow in the flat aperture at zero pressure gradient

Let us take under consideration the flow in aperture between two parallel flat plates with infinite span and one of them moves to other that is immobile at constant velocity $\mathbf{v_0}$ on the thin layer of liquid with thickness **s** (Fig.2.1.).

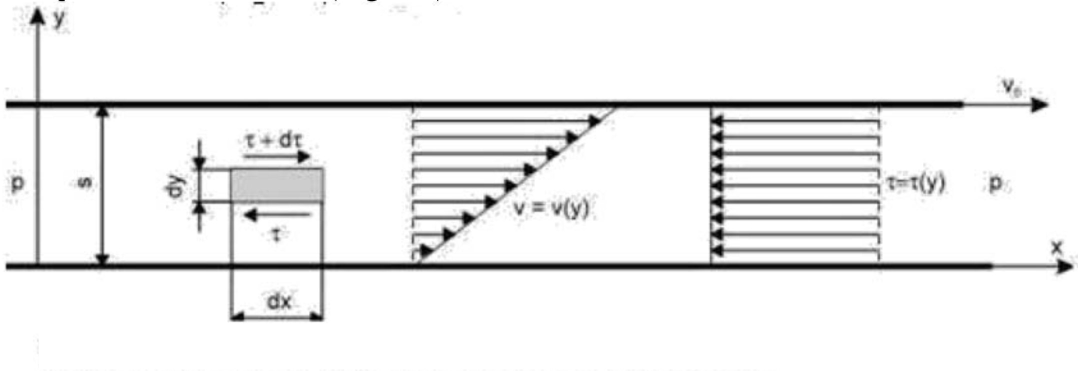

Fig.1. Flow in the flat aperture caused by the motion of upperplate

Pressure in the inlet section of such formed aperture is the same as in the outlet section (*dp/dx = 0*). Because of adhesion forces, the elements of liquid directly adjoining to plates remain in the release condition toward them. Whereas the indirect thin layers of liquid will translocate to each other, velocity gradient will be different from zero ($dv/dy \neq 0$) in normal direction to the pan motion. Assumption of infinite span of plates allows considering the flow in aperture as a flat flow. On the assumption that temperature in the layer of liquid is constant also its viscosity has not changed.

The infinitesimal element of prism form with section $dx{\cdot}dy$ and unit spam $dz = 1$ is taken under consideration in this layer of liquid. From the condition of force equilibrium on the direction of axis *x* we obtain:

$$p \cdot dy \cdot 1 - p \cdot dy \cdot 1 + (\tau + d\tau) \cdot dx \cdot 1 - \tau \cdot dx \cdot 1 = 0,$$

and further $d\tau = 0 \rightarrow \tau = C$ where *C* means constant.
Distribution of shear stress in considered flow in the aperture is homogenous.

Introducing Newton's formula of the dependence of shear stress on velocity gradient in normal direction we obtain

$$\tau = \mu \cdot \frac{dv}{dy} = C, \qquad \text{and after merger} \qquad v = \frac{C}{\mu} \cdot y = C_1,$$

Constant *C* i C_1 are defined on the basis of marginal conditions. For the approved system connected with the immobile plate is

$$for \quad y = 0 \rightarrow v = 0 \qquad y = s \rightarrow v = v_0 \quad so \quad C_1 = 0 \quad and \quad C = \mu \frac{v_0}{s}$$

Substituting defined values to (2.1) we obtain the relation on velocity distribution

$v = v_0 \cdot \dfrac{y}{s}$, so the velocity distribution in the aperture is linear distribution. Flow delivery through the section F=s·b is equal to $Q = \int_0^s v \cdot b \cdot dy = \dfrac{v_0}{2} \cdot s \cdot b$

It is obvious that mean flow velocity in the aperture $v_{śr} = \dfrac{v_0}{2}$

Previously introduced relations allow calculating the speed of shaft rotating at constant angular velocity around the axle which overlaps with the pan axle at presence of very small aperture in comparison to diameter of (Fig. 2.2) *s/D<<1*,where: *D* – diameter of shaft*; s* – size of radial aperture.

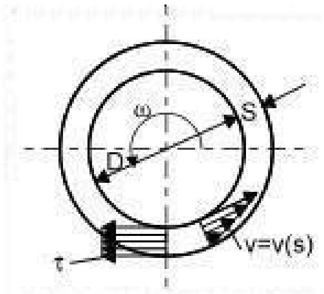

Fig. 2.2. Laminar flow in the ring aperture caused by the speed of shaft

At low relative aperture, the curvature of layer of liquid could be omitted and treated as flow in the flat aperture with linear distribution of velocity and homogenous distribution of shear stress. Defining the shaft length as *l*, moment of friction would be

$$M = \tau \cdot \pi \cdot D \cdot l \cdot \frac{D}{2} = \mu \cdot \frac{v_0}{s} \cdot \pi \cdot l \cdot \frac{D^2}{2},$$

where: $v_0 = \omega \cdot D/2$ – speed of shaft, so:

$$M = \mu \cdot \omega \cdot \pi \cdot l \cdot \frac{D^3}{4},$$

It should be emphasized that the laminar flow in aperture between the shaft and co-axial pan occurs for the Reynolds' critical ordinals $Re \leq 30 \cdot \sqrt{\dfrac{D}{s}}$.

2.2. Flow in the flat aperture under the influence of pressure differences

The second flat flow in the aperture, which has broad practical application, is the flow between parallel immobile plates under the influence of pressure difference. Such flows occur in the flat and cylindrical distributor, in the piston pump and hydraulic engines in piston–cylindrical opening and in the valves and other hydraulic sets. Considering very small nominal dimensions of apertures, flow velocities in the apertures have not reached such high values, to cause the turbulent flow in them. In principle, they will be the laminar flows.

The laminar stationary liquid flow in the flat aperture between two parallel plates is considered, omitting the phenomena, which occur on the edges of aperture and thermal processes.

Receiving reference standard (axis *x* agrees with the flow direction and places/situates in the longitudinal centre of aperture, axis *y* in perpendicular direction to the walls of aperture) was isolated the elementary prism with $2y \times l \times 1$ dimensions (Fig. 2.3) [1,2].

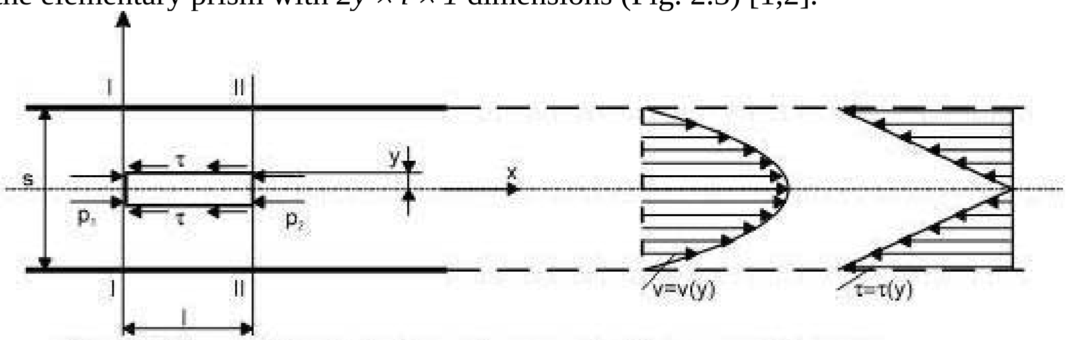

Fig. 2.3. Scheme of flow in the flat aperture caused by the pressure difference

From the condition of force equilibrium influencing on this elementary prism we obtain

$$p_1 \cdot 2 \cdot y \cdot 1 - p_2 \cdot 2 \cdot y \cdot 1 - 2 \cdot \tau \cdot l \cdot 1 = 0$$

where:

p_1 and p_2 – pressure in sections I – I and II – II,

τ – shear stress on the lower and upper wall (considering flat flow the stress on the side walls gives resultant equal to zero).

$$\tau = -\mu \cdot \frac{dv}{dy} \rightarrow \frac{dv}{dy} < 0,$$

After substitution and reduction we obtain

$$\left(p_1 - p_2\right) \cdot y = -\mu \cdot \frac{dv}{dy} \cdot l,$$

defining $$\frac{p_1 - p_2}{l} = \frac{\Delta p}{l}, \qquad dv = -\frac{\Delta p}{2 \cdot \mu \cdot l} \cdot y \cdot dy,$$

Integral of this equation $$v = -\frac{\Delta p}{2 \cdot \mu \cdot l} \cdot y^2 + C,$$

From the marginal conditions $$y = \pm \frac{s}{2}, \qquad v = 0,$$

was determined constant *C* of above equation

$$C = \frac{\Delta p}{2 \cdot \mu \cdot l} \cdot \frac{s^2}{4},$$

The formula for velocity distribution in the aperture will take the final form

$$v = \frac{\Delta p}{2 \cdot \mu \cdot l} \cdot \left[\frac{s^2}{4} - y^2\right],$$

Velocity distribution in the aperture is parabolic.

Velocity reaches the maximal value in the longitudinal centre of aperture

$$v_{max} = \frac{\Delta p}{8 \cdot \mu \cdot l} \cdot s^2,$$

Mean velocity of the aperture

$$v_{\acute{u}r} = \frac{2}{3} \cdot v_{max} = \frac{\Delta p}{12 \cdot \mu \cdot l} \cdot s^2,$$

Flow delivery through the aperture with unit breadth

$$Q = \frac{\Delta p}{12 \cdot \mu \cdot l} \cdot s^3,$$

and for the aperture with breadth, b>>s

$$Q = \frac{\Delta p}{12 \cdot \mu \cdot l} \cdot b \cdot s^3,$$

Pressure decease at velocity $v_{\acute{s}r}$

$$\Delta p = \frac{12 \cdot \mu \cdot l \cdot Q}{b \cdot s^3} = \frac{12 \cdot \nu \cdot l \cdot \gamma \cdot v_{\acute{u}r}}{g \cdot s^2},$$

Flow loss through the flat aperture amounts to

$$h_l = \frac{12 \cdot \nu \cdot l \cdot v_{\acute{u}r}}{g \cdot s^2},$$

or in analogous form to the formula for flow loss in the pipelines

$$h_l = \lambda \cdot \frac{l}{D_h} \cdot \frac{v_{\acute{u}r}^2}{2 \cdot g},$$

where:
$D_h = 2 \cdot s$ – hydraulic diameter,
$\lambda = 96/Re$ – flow loss coefficient for the aperture,
$Re = (v_{śr} \cdot D_h)/\nu$ – Reynolds' critical ordinal.

Power lost at the flow through aperture is equal to work which is necessary to translocate the liquid through aperture at pressure difference equal to pressure loss

$$N_p = Q \cdot \Delta p = \frac{s^3}{12 \cdot \mu} \cdot \frac{(\Delta p)^2}{l} \cdot b,$$

Above considerations and introduced formulas are correct on the assumption of constant viscosity in aperture. In reality/as a matter of fact the viscosity of liquid depends on temperature and pressure, which change along the aperture. For practical equations with sufficient accuracy is accepting the mean arithmetical value of kinematic viscosity coefficient

$$\nu_{úr} = \frac{\nu_1 + \nu_2}{2},$$

where: ν_1 and ν_2 define values of kinematic viscosity coefficient which matches temperature in the inlet/outlet of aperture.

The most common flow in the hydraulic appliances is the flow occurring between two parallel walls, which is caused by the pressure difference at simultaneous parallel translocation of walls.

The examples could be: the flow between rotating toothed–wheel rim and gear casing and the flow between cylinder and atomizer needle at their mutual relative motion in the engine atomizers.

If it is considered the laminar flow between two flat parallel walls and one of them moves with constant velocity v_0 at pressure gradient different from zero $(dp/dx)=(\Delta p/l)\neq 0$, it could be regarded as total of two flows: fractious flow with linear velocity distribution and flow caused by the pressure difference with parabolic velocity distribution.

When wall moves according to the direction of flow presented in Fig 2.4., velocity distribution of flow in the aperture to the co-ordinate system in the longitudinal centre plate is analytically defined by the formula:

$$v = v(y) = v_0 \cdot \left[\frac{y}{s} + \frac{1}{2}\right] + \frac{\Delta p \cdot s^2}{8 \cdot \mu \cdot l} \cdot \left[1 - \frac{4 \cdot y^2}{s^2}\right],$$

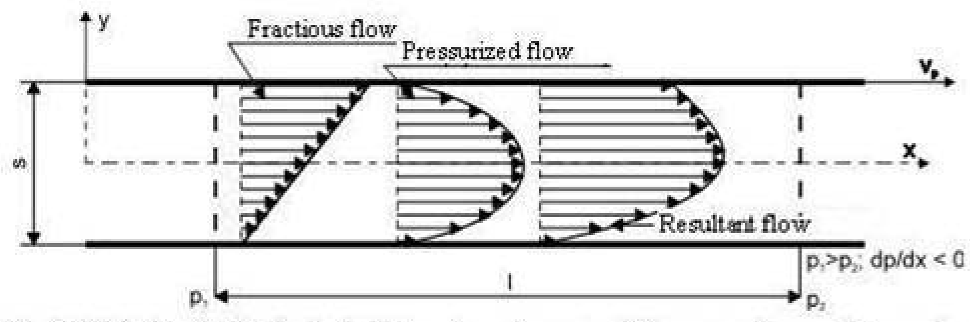

Fig. 2.4. Velocity distribution in the flat aperture at pressure difference and upper plate moving according to the direction of pressurized flow

If the wall moves in the opposite direction to the flow caused by the differential pressure, velocity in the cross-section of aperture will be

$$v = v(y) = -v_0 \cdot \left[\frac{y}{s} + \frac{1}{2}\right] + \frac{\Delta p \cdot s^2}{8 \cdot \mu \cdot l} \cdot \left[1 - \frac{4 \cdot y^2}{s^2}\right],$$

Velocity distribution in this case presents Fig. 2.5.

Knowing the velocity distribution, the value of mean velocity and the flow delivery is easy to define

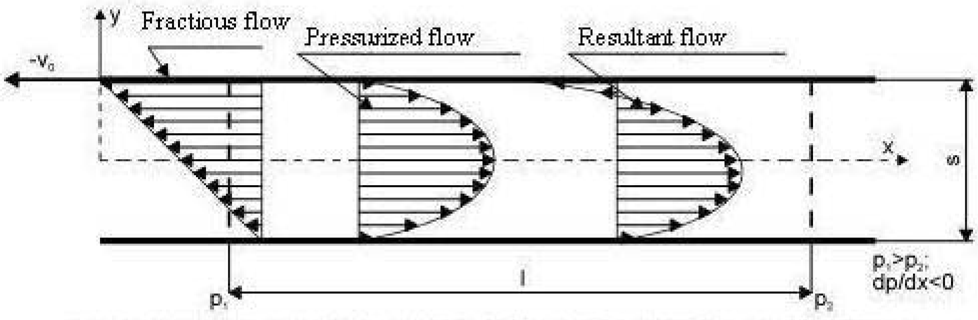

Fig.2,5. Velocity distributionin the total pressurized flow in the aperture at motion of upper plate opposite to the direction of pressurized flow

$$v_{\acute{u}r} = \frac{\Delta p \cdot s^2}{12 \cdot \mu \cdot l} \pm \frac{v_0}{2},$$

sign „ + ” – when plate moves in direction of lower pressure,
sign „ − ” – when the direction of plate is opposite.

The relation defines flow delivery through the aperture with span b

$$Q = \left[\frac{\Delta p \cdot s^3}{12 \cdot \mu \cdot l} \pm \frac{v_0}{2} \cdot s \right] \cdot b,$$

(sign „ ± ” like at defining mean velocity).

2.3. Hydraulic loss in the pipes with section different from circular. Hydraulic radius

The loss in the rectiaxial pipes with circular cross has been previously considered [1,2,3].

There should be put a question to what degree the presented considerations and formulas could be correct in solving the practical tasks in case of pipes with section different from the circular and pipes with circular section.

At calculating the hydraulic loss, the significant source of energy loss is shear stress appeared on the pipe walls, which was caused by liquid viscosity. Loss caused by the liquid friction will be the higher, the higher is circumference of washed by the liquid in relation to pipe section. At the same delivery and the same section, the flow resistance in the pipe with rectangular section will be higher than flow resistance and it in turn will be higher than the resistance in pipes with circular section. It is because circle has the lowest relation of circumference to section among all plane figures. Pipes with circular sections are the most beneficial taking under friction loss. So it is not difficult to draw to conclusion that measure, which is characteristic of values of loss occurring in pipes with different sections, is the relation of liquid section F_s to wetted circumference l_{zw}, i.e. to the circumference where liquid meets with walls.

This relation is called the hydraulic diameter $r_h = \frac{F_s}{l_{zw}}$

Fig. 2.6 presents examples of hydraulic diameters for cases, which are most common in practice.

For the flat apertures, where the width of aperture *a*, is significantly higher in comparison to high *s*, the values of hydraulic radius $r_h = \frac{s}{2}$.

Although there can not be the dynamic similarity of flows in the pipes with circular and non-circular-section, still on the basis of experiments it could be assumed that the character of variation of coefficient λ in pipes with different sections will be analogous at the same Reynolds' critical ordinals.

$$r_h = \frac{\frac{1}{4}\cdot\pi\cdot d^2}{\pi\cdot d} = \frac{d}{4} \qquad r_h = \frac{F_s}{l_{zw}} \qquad r_h = \frac{a^2}{4\cdot a} = \frac{a}{4} \qquad r_h = \frac{a\cdot b}{2\cdot(a+b)},$$

Especially at lack of experimental data for not very accurate calculations of flow loss in the pipes with sections different from the circular could be used formulas given for circular flow without fear of making glaring mistakes.

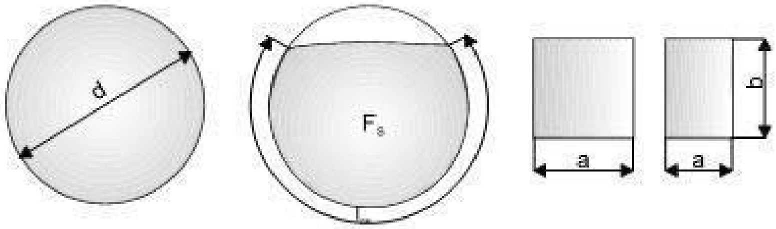

Fig. 2.6. The examples of hydraulic radiuses for the most common pipe sections

Because hydraulic diameter for the circle-section $r_h = d/4$, formula on calculating the hydraulic loss for pipes with any section will has following form $h_l = \lambda\cdot\frac{l}{4\cdot r_h}\cdot\frac{v_{śr}^2}{2\cdot g}$

where loss coefficient λ is defined for the particular zones on the basis of formulas for sections with circular-section, where Reynolds' critical ordinal is following

$$Re = \frac{4\cdot r_h\cdot v_{śr}}{\nu},$$

2.4. Hydraulic resistance of two parallel round surfaces immerge in liquid and approaching

In the present point the motion of two parallel plates immersed in liquid and approaching to each other will be considered, what concerns the sections of motion of atomizer needle in extreme positions [4].

Two parallel flat round plates with diameter *R* are located one above other in low distance from each other; space between them is filled with liquid. By pushing out the liquid the plates approach to each other at constant velocity u. Task lies in calculating the resistance, which plates meet. [4].

To calculate above task the installed set of parallel round plates was oriented to cylindrical coordinates with the beginning of set in the middle of bottom plate (which is assumed as immobile). The motion of liquid is axially symmetrical and because the liquid layer is very thin, in fact it is the motion along the radius ($v_z \ll v_r$), where $\frac{\partial v_r}{\partial r} \ll \frac{\partial v_r}{\partial z}$.

So the motion equation has form:

$$\eta\cdot\frac{\partial^2 v_r}{\partial z^2} = \frac{\partial p}{\partial r}, \quad \frac{\partial p}{\partial z} = 0 \quad \text{(a)} \qquad \frac{1}{r}\cdot\frac{\partial(r\cdot v_r)}{\partial r} + \frac{\partial v_z}{\partial z} = 0 \quad \text{(b)}$$

with following marginal conditions:

$$v_r = v_z = 0, \text{ when } z = 0; \qquad v_r = 0, \quad v_z = -u, \text{ when } z = h; \qquad p = p_0, \text{ when } r = R$$

where η – dynamic viscosity, *h* is the distance between plates and pressure p_0 is the exterior pressure. From the equations (a) we obtain:

$$v_r = \frac{1}{2 \cdot \eta} \cdot \frac{dp}{dr} \cdot z \cdot (z - h),$$

Merging equation (b) after dz we obtain $u = \frac{1}{r} \cdot \frac{d}{dr} \cdot \int_0^h r \cdot v_r \cdot dz = -\frac{h^3}{12 \cdot \eta \cdot r} \cdot \frac{d}{dr} \cdot \left(r \cdot \frac{dp}{dr} \right),$

from which $p = p_0 + \frac{3 \cdot \eta \cdot u}{h^3} \cdot \left(R^2 - r^2 \right),$

Total resistance force influencing on the plate is equal to $F = \frac{3 \cdot \pi \cdot \eta \cdot u \cdot R^4}{2 \cdot h^3}.$

4. Conclusions

Simulation calculations for the chosen sections of feeding system has been presented on the basis of theoretical and empiric relations without verification and tests on the real feeding installation in fuel system C – R. type.

As it results from the initial observations of measure results on the stand, the above-proposed relations are the output basis for detailed analysis of phenomena, which occur in the selected sections of fuel gear.

To verify the approved model, the experimental tests should be conducted on the test stand, which is replica of feeding system1SB and the obtained result should be used to adjust the approved calculation model.

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