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PULSATORY PHENOMENON IN HYDROSTATIC DRIVING SYSTEMS

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ABSTRACT

In this paper were treated and briefly presented the particular aspects related to possible dynamic unstable behaviors of flow variation into the volumetric pumps. Dynamic characteristics for transitory state working regime were deduced and possible pressure resonances induced into the base system by the pump flow variation were defined

KEYWORDS: volumetric pumps, the pump flow, hydrostatic driving systems

1. INTRODUCTION

This study starts from the basic hypothesis that hydrostatic driving systems are very vulnerable ensembles capable to stability loss during its working time if external or internal disturbing actions are deliberately or accidentally produced. From the category of these deliberate factors the authors remind proper working and command transitory regimes, specific constructive aspects, etc, and from the accidental factors are mentioned pounce loads, involuntary commands, functional shocks, etc.

This paper defines a set of possible functional instabilities into the hydrostatic driving systems due to the structural aspects which have proper values for hydraulic units working state such as flows variation of the axial or radial pistons pumps.

2. FLOWS VARIATION

It is known from the scientific literature that volumetric hydraulic units with axial pistons (rotative pumps or motors), regardless ther structural configuration, have a characteristic of output flows variation. For a global estimation of the pumps flows variation the pulsation factor is used as a main parameter. This parameter is defined as follows

$$\delta = \frac{Q_{i\max} - Q_{i\min}}{Q_{med}} \cdot 100 \text{ ;[\%]} \qquad (1)$$

which means it is evaluated with respect to pump instantaneous flows parameters specific for each constructive configuration. The notations in Eq.(1) have the follving significances: Q_{imax} is the maximum instantaneous flows for one complete rotation of the pump rotor; Q_{imin} denotes the minimum instantaneous flows for one complete rotation of the pump rotor; Q_{med} is the average flows for one complete rotation of the pump rotor.

An intuitive image of the instantaneous flows variation for an axial pistons pump is depicted in Fig. 1.

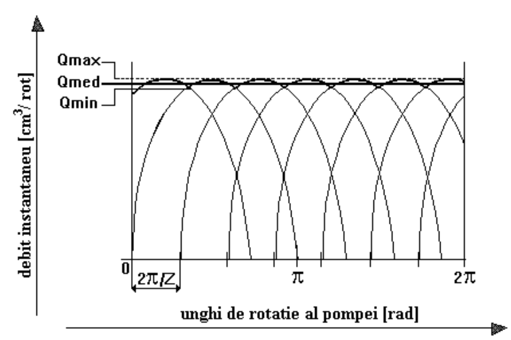

Fig. 1. The flows pulsation for a pump (vertical ax: instantaneous flows; orizontal ax: pump rotation angle)

3. THE PULSATORY CHARACTER OF THE PUMP FLOWS

If we consider the flows variation results, the pump flows Q_p have the folloving expression [1]

$$Q_P = \frac{q_o}{2\pi}\omega_I.\sin(\omega_I.t+\varphi_o); \quad (2)$$

where

$$q_o = 2.r.A_p.\frac{\sin(m\pi/7)}{\sin(\pi/7)}.\sin\alpha;$$

$$\varphi_o = (m-1)\pi/7; \quad (3)$$

for the axial pistons pumps with leaning block and 7 pistons, and ω_I is the angular speed at the pump shaft. The Eq.(3) denotes the mathematical model of the harmonic variation of pump flows. This phenomenon was integrated through the pump flows pulsation - δ. This parameter is a function of pistons amount (z = 3,5,7,9), number of pistons connected to the output (m = 1-2, 2-3, 3-4, 4-5), radius of pistons basic montage circle (r), the piston area (A_p), and the angle (α) of leaning block or disk.

4. DYNAMIC CHARACTERISTICS OF THE HYDROSTATIC SYSTEM

For the linear motor acting as working body with the reduced mass at the stem (M) and the resistant force (F_E), the motion equation is

$$M.\ddot{x}+\gamma_{ML}.\dot{x}+F_E = A.p; \quad (10)$$

For the rotative motor acting as working body with the reduced moment of inertia at the motor shaft (J) and the resistant moment (M_E), the motion equation is

$$J.\ddot{\varphi}+\gamma_{MR}.\dot{\varphi}+M_E = \frac{V_{oM}}{2\pi}.p \quad (11)$$

From Eqs.(8) and (9) applying the derivative operator, it results the following equations

- for the linear motor system

$$\ddot{x} = \frac{q_o\omega_I^2}{2\pi A}\cos(\omega_I t+\varphi_o) - \frac{\alpha_{ML}}{A}.\dot{p} - \frac{\beta_{ML}}{A}.\ddot{p}; \quad (12)$$

- for the rotative motor system

$$\ddot{\varphi} = \frac{q_o\omega_I^2}{V_{oM}}\cos(\omega_I t+\varphi_o) - \frac{2\pi\alpha_{MR}}{V_{oM}}.\dot{p} - \frac{2\pi\beta_{MR}}{V_{oM}}.\ddot{p}; \quad (13)$$

After replacing Eqs.(12) and (13) in Eqs.(10) and respectively (11), terms ordering and identification of the terms signification into the final equations results

$$\ddot{p}+2\xi\omega.\dot{p}+\omega^2.p = H+\Pi.\sin(\omega_I.t+\Phi); \quad (14)$$

The shape of Eq.(14) definus the oscillating character of the pressure variation into the hydrostatic driving systems, regardless the motor type of the system (linear or rotative). The flows pulsation of the pump acting on the entire system leads to a possible internal resonance phenomenon if the excitation pulsation ω_I becomes closer to the natural pulsation ω of the system.

In respect with the system type into Eq.(14) can be defined the essential parameters as follows

- for the systems with linear motor

$$\omega_{ML} = A\sqrt{\frac{2}{M.C_h}}; \quad (15)$$

$$\xi_{ML} = \frac{\omega_{ML}}{2.A^2}\left(\alpha_{SIS}M+\gamma_{ML}C_h\right); \quad (16)$$

$$H_{ML} = \frac{2.A.F_E}{(M+\gamma_{ML}).C_h}; \quad (17)$$

$$\Pi_{ML} = \frac{q_0}{\pi C_h}\sqrt{1+\left(\frac{\gamma_{ML}}{M.\omega_I}\right)^2}.\omega_I^2; \quad (18)$$

$$\Phi_{ML} = \varphi_0 + arctg\frac{M\omega_I}{\gamma_{ML}}; \quad (19)$$

- for the systems with rotative motor

$$\omega_{MR} = \frac{V_{oM}}{2\pi}\sqrt{\frac{2}{J.C_h}}; \quad (20)$$

$$\xi_{MR} = \frac{\omega_{MR}}{2.}\left(\frac{2\pi}{V_{oM}}\right)^2\left(\alpha_{SIS}J+\gamma_{MR}C_h\right); \quad (21)$$

$$H_{MR} = \frac{V_{oM}.M_E}{\pi.J.C_h}; \quad (22)$$

$$\Pi_{MR} = \frac{q_0}{\pi C_h}\sqrt{1+\left(\frac{\gamma_{MR}}{J.\omega_I}\right)^2}.\omega_I^2; \quad (23)$$

$$\Phi_{MR} = \varphi_0 + arctg\frac{J\omega_I}{\gamma_{MR}}; \quad (24)$$

where $C_h = 2.\beta_{SIS}$ – denotes the hydraulic capacity of the hydrostatic system and it is $= 2.V_{SIS}/E$.

5. DYNAMIC MODEL OF THE HYDROSTATIC SYSTEM

The solution of the homogenous equation representing the system behaviour without excitation has the next expression

$$p_1 = p_0.e^{-\xi.\omega.t}\sin(\omega\sqrt{1-\xi}.t); \quad (25)$$

The proprietary solution of the non-homogenous equation verifying the has the folloving expression

$$p_2 = p_S + p_R \sin(\omega_I . t + \phi_R); \quad (26)$$

where

$p_S = \dfrac{F_E}{A}$ denotes the static pressure for the linear motor systems; (27)

$p_S = \dfrac{2\pi M_E}{V_{0M}}$ denotes the static pressure for the rotative motor systems; (28)

$p_R = \dfrac{q_0}{\pi C_h} . \dfrac{\sqrt{1+\varepsilon^2} . \mu^2}{\sqrt{(1-\mu^2)^2 + (2\xi)^2 . \mu^2}}$; denote dynamic pressure magnitude; (29)

$$\phi_R = arctg \frac{a_o[\varepsilon(1-\mu^2) + 2\xi\ \mu] + (1-\mu^2) - 2\xi\ \varepsilon\mu}{\varepsilon(1-\mu^2) + 2\xi . \mu a_o[(1-\mu^2) - 2\xi\ \varepsilon\mu]} \quad (30)$$

- initial phase of the pressure variation

$$a_o = tg\varphi_0; \quad (31)$$

$$\varepsilon = \gamma_{ML} / M.\omega_I \text{ ;or } \varepsilon = \gamma_{MR} / J.\omega_I; \quad (32)$$

$\mu = \omega_I / \omega_{MR}$ denotes the relative pulsation of the pressure. (33)

6. DYNAMIC RESPONSE OF THE HYDROSTATIC SYSTEM ON THE EXCITATION BY FLOWS PULSATION

The resonance phenomenon well known from the mechanical systems can be also arisen into the hydrostatical systems. This phenomenon appears on the pressure factor such as it was previously presented for the linear or rotative motor driving systems. The resonance influences the absolute pressure magnitude into the driving system and its phase, and appears when the excitation pulsation - in this case the angular speed at the pump shaft ω_I, becomes equal to the natural pulsation of the system - ω_{ML} for linear motor systems and ω_{MR} for rotative motor systems. Let we note this fact as follows

$$\omega_I = \omega_{SIS}; \Rightarrow \mu = 1; \quad (34)$$

where ω_{SIS} – denotes ω_{ML} or ω_{MR} in respect with the analyzed system.

The resonance pressure magnitude factor defined as a ratio between p_R and $p_o = q_o / \pi C_h$ will be

$$P_1 = \frac{p_R}{p_o} = \frac{\mu^2 \sqrt{1+\varepsilon^2}}{\sqrt{(1-\mu^2)^2 + (2\xi_{SIS})^2 . \mu^2}}; \quad (35)$$

and the specific diagram is depicted in Fig.1 for different values of damping $\xi_{SIS} - (\xi_{ML}$, or $\xi_{MR})$ and $\varepsilon = 1$.

The magnitude increasing at the resonance appears for the case of

$$\omega_I = \omega_{SIS} \frac{1}{\sqrt{1-2\zeta_{SIS}^2}} \text{ ;for } \zeta < \frac{1}{\sqrt{2}}; \quad (36)$$

and the pressure value at resonance becomes

$$p_{1rezonanta} = \frac{1}{2\zeta_{SIS}} \sqrt{\frac{1+\varepsilon^2}{1-2\zeta_{SIS}^2}}; \quad (37)$$

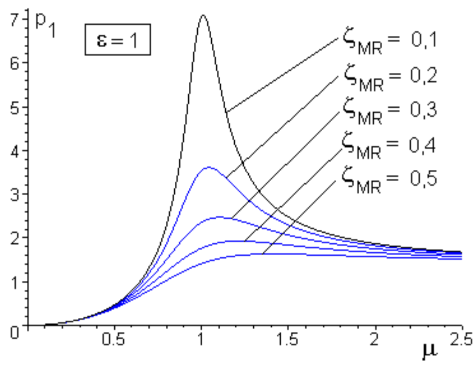

Fig. 2. The magnitude factor variation for static pressure as a function of relative pulsation μ

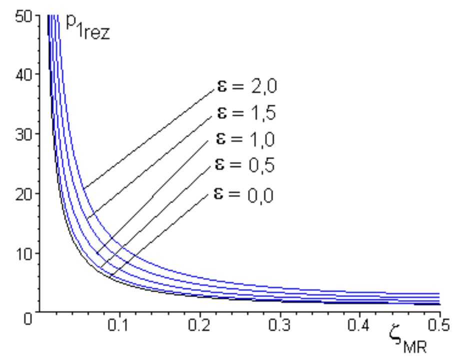

Fig. 3. The magnitude factor of the resonance

7. CONCLUSIONS

The analysis proposed and briefly presented in this work reveals the possibility to obtain useful characteristics directly by analytic and theoretical approaches. The evaluation of the dynamic behaviour character of hydrostatic driving systems can be described by the folloving essential parameters

- The natural pulsation of a hydraulic driving system ω_{SIS} depends on the folloving parameters:
  - o geometrical characteristics of the system through the piston area (A) - for the linear motor systems, and through the motor

specific capacity (V_{oM}) - for the rotative motor systems,
- o working body mass distribution reduced at the motor shaft,
- o hydraulic capacity of the driving system.

- Damping factor of driving system ξ_{SIS} which depends both on the same parameters previously presented, and on specific factors of the hydraulic fluid used in the system.
- The magnitude of dynamic component of the pressure response - this parameter effectively generates the dynamic behaviour of the system; its evaluation even from the design phase of the driving system will lead to avoiding a possible pressure resonance phenomenon.

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