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ANALYSIS OF MEAN CIRCULATION IN THE FLUID COUPLINGS

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Abstract: The mean circulation in the working place of hydraulic transmissions,(fluid coupling and torque converters) is responsible of the conduction driving moment from the pump impeller, P, to the runner of a turbine, T, and of energetically performances of the hydrokinetics, implicitly.

On the basis of this observation, in the work, the hydrodynamic conditions of mean circulation in the fluid coupling and the hydraulic transmissions, generally, are analyzed. Also, the geometrical and constructive-functional shapes, what establish an optimal mean circulation and, implicitly, a high operation ratio of hydraulic transmissions are analyzed.

The theoretical results established in the work are applicable at the unceasing activity of research, design and promotion of high-level technologies, as well as and at the achievement of hydraulic transmissions with high energetically performances.

Keywords: fluid couplings, mean circulation, radii ratio, reduction ratio.

1. Introduction

The mean circulation in the working place of hydraulic transmissions,-fluid couplings and torque converters,-is responsible of the conduction driving moment from the pump impeller, P, to the runner of a turbine, T, and of operating performances of the hydrokinetics, implicitly. Therefore, the regular of hydraulic transmissions, generally, requires the existence of an optimal mean circulation in the working place of hydrokinetics.

Thus, in the instance of fluid coupling, (P→ T→ P), the existence of mean circulation conditions, first of all, the conduction driving moment from the pump impeller to the runner of a turbine. Afterwards, the problem of optimization of the mean circulation is raised.

In the instance of torque converter, the mean circulation exists in any operating conditions. Consequently, here, it appeases, first of all, the problem of optimization of the mean circulation and, implicitly, of best working of the torque converter. Further on, the existence of mean circulation in the fluid couplings and the best operating conditions are analysed.

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2. Generalities

It is known,[1], [2], [4], [5], [7], [10], [12], [13], that, in the structure of fluid coupling, it exists two revolving blades, respectively, the pump impeller, P, and the runner of a turbine, T, (Fig. 1).

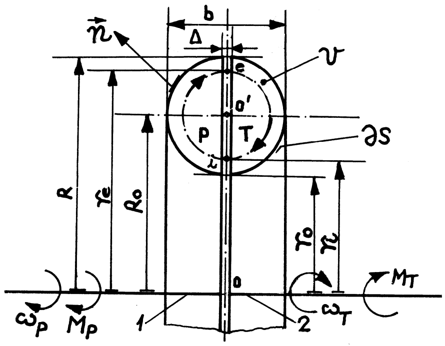

Fig. 1. The fluid coupling. Schematic diagram.

The motive fluid covers the revolving blades in the sequence P→T→P, (Fig. 1). Also, it is known that, from the viewpoint of the hydrodynamics, not are dissipations of energy in the fluid coupling. Therefore, the moments of momentum are equal, M_P=M_T, at the taking out off the pump-runner vanes P, and to the entry on the impeller blades, T. The case is likewise and at the taking out off the impeller blades and to the entry on the pump-runner vanes.

Because of the same reasons, it results that and the vector's scalars are equal, $\Gamma_{e_P} = \Gamma_{i_T}, \Gamma_{e_T} = \Gamma_{i_P}$ at the exit off the pump-runner vanes, P, and to the entry on the impeller blades, T, and conversely.

3. Basic relations. Notations in use

The law of moment of momentum, appliquéd for the volume-control, v, (Fig. 1), offers the momentum $\vec{M}_{S\to L}$, respectively the result of interaction between the solid area and the motive fluid, [1], [2], [3], [5], [7], [10], and others.

The relation of momentum $\overline{M}_{S\to L}$, written for the pump impeller and the runner of a turbine, [1], [2], [3], [7], becomes:

$$M_P = \rho \cdot Q_p \cdot \left(r_e^2 \cdot \omega_P - r_i^2 \cdot \omega_T\right); \tag{1}$$

$$M_T = \rho \cdot Q_T \cdot \left(r_e^2 \cdot \omega_T - r_i^2 \cdot \omega_P\right); \tag{2}$$

Introducing, further on, the notations, [1], [2]:

the reduction ratio, i : $\quad i = \dfrac{\omega_T}{\omega_P} = \dfrac{n_T}{n_P}$; (3)

or, *the slip of fluid coupling ,s:* $s = 1 - i = \dfrac{\omega_P - \omega_T}{\omega_P}$; (3.1)

the radii ratio, δ : $\delta = \dfrac{r_i}{r_e}$; (4)

the dimensionless coefficient of correlation of speeds, φ_e :

$$\varphi_e = \frac{v_m}{u_e} = \frac{v_m}{r_e \cdot \omega_P}; \qquad (5)$$

and, *the volumetric flow*, $Q, (Q_P = Q_T \equiv Q)$:

$$Q = v_m \cdot \frac{\pi \cdot R^2}{2} \cdot \left[1 - \left(\frac{r_0}{R}\right)^2\right] = \varphi_e \cdot r_e \cdot \omega_P \cdot A; \quad A = \frac{\pi \cdot R^2}{2} \cdot \left[1 - \left(\frac{r_0}{R}\right)^2\right]; \qquad (6)$$

Then, the relations (1) and (2) become:

$$M_p = \rho \cdot Q \cdot r_e^2 \cdot \omega_P \cdot \left[1 - \left(\frac{r_i}{r_e}\right)^2 \cdot \left(\frac{\omega_T}{\omega_P}\right)\right] = \rho \cdot A \cdot r_e^3 \cdot \omega_P^2 \cdot \left[1 - \delta^2 \cdot i\right] \cdot \varphi_e; \qquad (7)$$

$$M_T = \rho \cdot Q \cdot r_e^2 \cdot \omega_T \cdot \left[1 - \left(\frac{r_i}{r_e}\right)^2 \cdot \left(\frac{\omega_P}{\omega_T}\right)\right] = \rho \cdot A \cdot r_e^3 \cdot \omega_P^2 \cdot \left[i - \delta^2\right] \cdot \varphi_e; \qquad (8)$$

4. Operating conditions of the mean circulation

It is known, [1], [2], [4], [10], that, at a fluid coupling, the moments of momentum are equal, $M_P = M_T$, if the friction losses are neglected. Then, out of the relations (7) and (8), it results that $M_P \equiv M_T$ if and only if there is the following equality:

$$\left(1 - \delta^2 \cdot i\right) = \left(i - \delta^2\right); \qquad (9)$$

The relation (9) is very if and only if $i \equiv 1$, therefore, if the friction losses are neglected, (the slip $s \cong 0$). In fact, into the fluid coupling, there is friction losses, (the slip $s \cong 0{,}03, \ldots, 0{,}02$, [1],[2],[3]), so that $M_P > M_T$. Therefore, the relation (9) becomes:

$$\left(1 - \delta^2 \cdot i\right) > \left(i - \delta^2\right); \qquad (10)$$

Also, the speed of runner of a turbine, n_T, is smaller than the speed of pump impeller, n_P ,because of the friction losses in the fluid coupling. On the other hand, from the relations (7)and(8), it results that the moments of momentum $M_P = M_T = 0$ when the dimensionless coefficient of correlation of speeds $\varphi_e = 0$, that is, when the mean velocity $v_m = 0$. This takes place when the speeds of runner of a turbine and of pump impeller are equal, $n_P = n_T$, (therefore, $i = 1,0$).

For to be possible conduct a moment of rotation, you must needs to exist a small slip, $(s \neq 0)$, between the pump impeller and the runner of a turbine. Thus, is effected a mean circulation of motive fluid from the pump impeller towards the runner of a turbine and reverse. The same conclusion results, evidently, and from the relations (7) and (8). Then, with the relation (9), the relations (7) and (8) quite possibly to be written under the following shape:

$$|M_P| = |M_T| = \rho \cdot A \cdot r_e^3 \cdot \omega_P^2 \cdot \left[1 - \delta^2 \cdot i\right] \cdot \varphi_e = \rho \cdot A \cdot r_e^3 \cdot \omega_P^2 \cdot \left[i - \delta^2\right] \cdot \varphi_e \; ; \quad (11)$$

The relation (11), where $M = |M_P| = |M_T|$, indicates that, *the momentum M is conducted in the fluid coupling only when* $\varphi_e \neq 0$*, that is, when, in the working area of fluid coupling, there is mean circulation.* The operating conditions of the mean circulation result from the study of pressure distribution in the working area of fluid coupling, respectively, of hydraulic transmissions, generally.

5. The study of pressure distribution

The analysis presented further on is valid, generally, for any kind of hydraulic transmission, [1], [2], [3], [4], [10], and others. Thus, for the determination of pressure distribution, we consider a work space shaped ring surface, filler with motive fluid. The work space circulates with the angular frequency $\vec{\omega}$ all round of the symmetry axis, z, (Fig. 2).

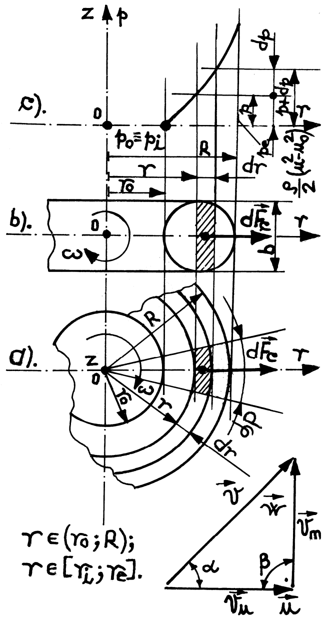

Fig. 2. The pressure distribution on the external wall of a revolving ring surface.

At an absolute steady position, $(\vec{\omega}=0)$, the motive fluid, in the work space, observes *the Pascal's principle*. Thus, the pressure head p_0 is uniformly distributed in all the motive fluid.

At a state of motion with the angular frequency $\vec{\omega}$, $(\vec{\omega}>0, \vec{\omega}=const.)$, over the pressure head p_0, the dynamic pressure *p* is superposed. The dynamic pressure is determined of the centrifugal force, $\vec{F}_c$, and it increases once with the radius, *r*, $r \in [r_0,...,R]$, (Fig. 2.c).

On the basis of Fig. 2, quite possibly to write the elementary centrifugal force, $d\vec{F}_c$, thus:

$$dF_c = r \cdot \omega^2 \cdot dm\,, \tag{12}$$

where, *dm* is the elementary mass of motive fluid in the ring surface.

The elementary mass of motive fluid, *dm*, is given through:

$$dm = \rho \cdot dVol = \rho \cdot d\varphi \cdot r \cdot b \cdot dr\,, \tag{13}$$

thus, that, the relation (12) becomes:

$$dF_c = \rho \cdot d\varphi \cdot r^2 \cdot \omega^2 \cdot b \cdot dr\,; \tag{14}$$

If the element of area *dA* is given through the following relation:

$$dA = r \cdot d\varphi \cdot b\,, \tag{15}$$

then, the relation (14) becomes:

$$dF_c = dp \cdot dA = dm \cdot r \cdot \omega^2\,; \tag{16}$$

The relation (16) offers the elementary centrifugal force, $d\vec{F}_c$, corresponding to a rise of pressure *dp*, from the value *p* to the value *p+dp*, (Fig. 2.c). Thus, from the relations (12), (13), (14), (15), (16), it results:

$$dp = \frac{dF_c}{dA} = \rho \cdot r \cdot \omega^2 \cdot dr\,; \tag{17}$$

Through the integration of separable differential equation (17), we obtain:

$$p = \int_{r_0}^{r} dp(r) = \frac{\rho}{2} \cdot \omega^2 \cdot \left(r^2 - r_0^2\right) = \frac{\rho}{2} \cdot \left(u^2 - u_0^2\right); \tag{18}$$

The overall pressure, in the ring surface of hydraulic transmission, is equal with the sum between the pressure head, p_0 and the mean dynamic pressure, *p*. In a fluid coupling, the mean dynamic pressures have various values within the pump impeller and in the runner of a turbine, because the speed decreases from entry towards the taking out of hydrokinetic, $n_p > n_T$, respectively, $\omega_P > \omega_T$.

On the contrary, at the torque converter, for the duty cycle rating, the speed n_T quite possibly to differ much given the speed from entry, n_P, $n_P >> n_T$, [1].

At these conditions, the relation (18) becomes:

$$p_P = \frac{\rho}{2}\cdot\omega_P^2\cdot\left(r^2 - r_0^2\right);\quad p_T = \frac{\rho}{2}\cdot\omega_T^2\cdot\left(r^2 - r_0^2\right);\quad p_P > p_T; \tag{19}$$

The differential pressure, Δp, between the pump impeller and the runner of a turbine, is:

$$\Delta p = p_P - p_T = \frac{\rho}{2}\cdot\left(\omega_P^2 - \omega_T^2\right)\cdot\left(r^2 - r_0^2\right) > 0; \tag{20}$$

The relation (20), written at the outside radius, $r = r_e$, offers:

$$\Delta p_2 = \Delta p\big|_{r=r_e} = \frac{\rho}{2}\cdot\left(\omega_P^2 - \omega_T^2\right)\cdot\left(r_e^2 - r_0^2\right) > 0; \tag{21}$$

The relation (20), written, now, at the inner radius, $r = r_i$, offers the differential pressure Δp_1, thus:

$$\Delta p_1 = \Delta p\big|_{r=r_i} = \left(p_P - p_T\right)\big|_{r=r_i} = \frac{\rho}{2}\cdot\left(\omega_P^2 - \omega_T^2\right)\cdot\left(r_i^2 - r_0^2\right) > 0; \tag{22}$$

On the basis of relations (19), the relations (20), (21) and (22) are strictly positive. It results that, at the both cases, $(r = r_e; r = r_i)$, the dynamic pressure, *p*, is greatest in the pump impeller, (Fig. 3).

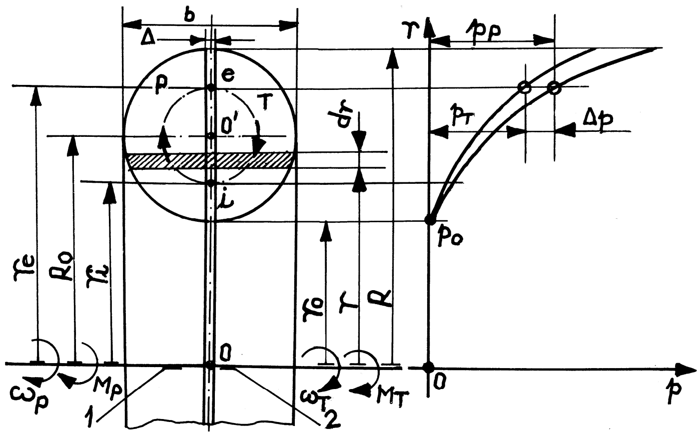

Fig. 3. The pressure distributions in the ring surface of fluid coupling.

Because of differential pressure, Δp, $p_P > p_T$, at any radii $r \in [r_i, ..., r_e]$, $r \in (r_0, ..., R)$, *it appears, therefore, a mean circulation of fluid*, from the pump impeller towards the runner of a turbine, (Fig. 3). The direction of mean circulation is established of the differential pressure $\Delta p_2 = \Delta p\big|_{r=r_e}$, this been more great than the differential pressure $\Delta p_1 = \Delta p\big|_{r=r_i}$, because $r_e > r_i$. Consequently, the motive fluid pass out off the pump impeller, *P*, and it enters on the runner of a turbine, *T*, all round of radius r_e. Into reverse direction, the motive fluid flows all round of radius r_i, (Fig. 3).

In the working area included between the radii r_i and r_e, then, when $\Delta r = r_e - r_i \to 0$, $r_e \to R_0$, $r_i \to R_0$, the differential pressure $\Delta p_1, (\Delta p_1 < \Delta p_2)$, tends to brake the mean circulation of motive fluid. Therefore, the resultant differential pressure, Δp, is:

$$\Delta p = \Delta p_2 - \Delta p_1 = \frac{\rho}{2} \cdot \left(\omega_P^2 - \omega_T^2\right) \cdot \left(r_e^2 - r_i^2\right); \tag{23}$$

Consequently, it results that, the mean circulation of liquid is maintained of the resultant differential pressure, Δp. On the other hand, from the relation (23), it results, that, the mean circulation is braked, $(\Delta p = 0)$, when either $\omega_P = \omega_T$, or $r_e = r_i = R_0$, (Fig. 3).

The discharge head, ΔH, corresponding to the differential pressure Δp, is:

$$\Delta H = \frac{\Delta p}{\rho \cdot g} = \frac{1}{2 \cdot g} \cdot \left(\omega_P^2 - \omega_T^2\right) \cdot \left(r_e^2 - r_i^2\right), \tag{24}$$

and *this balances, to the extreme,* the discharge head corresponding to the friction losses, $\sum h_{p-friction}$, and corresponding to the loss by shock, $\sum h_{p-shock}$.

Therefore, we have:

$$\Delta H \geq \sum h_{p-friction} + \sum h_{p-shock}, \tag{25}$$

where: $$\sum h_{p-friction} = \varsigma \cdot \frac{v_{m_e}^2}{2 \cdot g} = \varsigma \cdot \varphi_e^2 \cdot \frac{u_e^2}{2 \cdot g} = \varsigma \cdot \varphi_e^2 \cdot \frac{r_e^2 \cdot \omega_P^2}{2 \cdot g}, \tag{26}$$

where, ς is the dimensionless coefficient of friction losses in the fluid coupling; again, $\sum h_{p-shock} \cong \psi \cdot \frac{\Delta v_u^2}{2 \cdot g} \cong 0$, because $\psi \cong 1{,}0$, while the rate change $\Delta v_u \cong 0$, for the optimal working of fluid couplings.

Therefore, the relation (25) becomes:

$$\Delta H \geq \sum h_{p-friction} = \varsigma \cdot \varphi_e^2 \cdot \frac{r_e^2 \cdot \omega_P^2}{2 \cdot g}; \tag{27}$$

Then, the relations of energy balance (24) and (27), *to the extreme*, offer:

$$\Delta H \cong \sum h_{p-friction}, \tag{28}$$

or, explicitly, it results:

$$\left(r_e^2 - r_i^2\right) \cdot \left(\omega_P^2 - \omega_T^2\right) - \varsigma \cdot \varphi_e^2 \cdot r_e^2 \cdot \omega_P^2 = 0; \tag{29}$$

With the relations (3) and (4), the relation (29) becomes:

$$\left(1-\delta^2\right)\cdot\left(1-i^2\right)-\varsigma\cdot\varphi_e^2=0\,; \tag{30}$$

Or, with the expression of the dimensionless coefficient of correlation of speeds, φ_e , [1], [2], [3],

$$\varphi_e=\sqrt{\frac{2\cdot s\cdot(2-s)}{\varsigma}}\,, \tag{31}$$

then, the relation (30) becomes:

$$\left(1-\delta^2\right)\cdot\left(1-i^2\right)-2\cdot s\cdot(2-s)=0\,; \tag{32}$$

Thus, from the relation (11), it has resulted that, the momentum *M* is conducted in the fluid coupling only, then, when $\varphi_e \neq 0$, $(v_m \neq 0)$.

On the other hand, from the relation (30) or (32), it results that, the mean circulation in the fluid coupling appears only, then, when the reduction ratio $i<1$, that is, when $\omega_T < \omega_P$.

If we observe, now, the expression of the overall efficiency of fluid coupling, η_{CH} , [1], [2], [4], [10], and others,

$$\eta_{CH}=\frac{M_T\cdot\omega_T}{M_P\cdot\omega_P}=\frac{M_T\cdot n_T}{M_P\cdot n_P}\,, \tag{33}$$

and we agree *the theoretical relation of momentums*, $M_P = M_T$, it results that:

$$\eta_{CH}\cong\frac{\omega_T}{\omega_P}=\frac{n_T}{n_P}=i\,; \tag{34}$$

Then, with the relation (34), the previous relations (30) or (32) become:

$$\left(1-\delta^2\right)\cdot\left(1-\eta_{CH}^2\right)-2\cdot s\cdot(2-s)=0\,; \tag{35}$$

As, always, the overall efficiency of fluid coupling is sub-unitary, $\eta_{Ch} < 1{,}0$, $(\eta_{CH\,\max}\cong 0{,}97)$, [1],[2],[3], it results that, indeed, the mean circulation in the fluid coupling appears only, then, when the reduction ratio $i=\eta_{CH}\leq 0{,}97, (i<1{,}0; \omega_T<\omega_P)$.

If we spread the analysis of relations (11), (30), (31), (32), (33), (34) and (35), it comes out that, at least under mathematical aspect, it exists the following cases:

$$\begin{aligned}&\varphi_e=0\text{, then, when: } i=-1;+1;\\ &\varphi_e\neq 0\text{, then, when: } i\neq -1;+1;\\ &\varphi_e>0\text{, then, when: } -1<i<+1\text{, or } i\in(-1,\ldots,+1);\end{aligned} \tag{36}$$

In conclusion, it results that, the fluid coupling conducts a momentum *M*, only, then, when $i<+1$, respectively $i=\eta_{CH}\leq 0{,}97, (0<i<+1)$, that is, at speeds $n_T < n_P$.

6. Conclusions

The paper has a theoretical essence and it refers to the fluid couplings.

The study analyses, under theoretical aspect, the hydrodynamic conditions of existence of the mean circulation in the fluid couplings and in the hydraulic transmissions, generally.

Also, the power transmission as well as the constructive, geometrical and functional shapers, they that lead to an optimal mean circulation, respectively a high operating efficiency of hydrokinetics are analyzed.

Essentially, on the basis of theoretical results obtained, quite possibly to be formulated the following conclusions:

a). Always, in the working of fluid couplings and of the hydraulic transmissions, generally, the pressure is more great into the pump impeller, *P*, $p_P > p_T$. As a result, it appears a mean circulation of liquid from the pump impeller towards the runner of a turbine.

b).The momentum, *M*, is conducted in the fluid coupling only, then, when the dimensionless coefficient of correlation of speeds, $\varphi_e \neq 0$, respectively the mean speed, $v_m \neq 0$, $(i < 1{,}0; 0 < i < +1{,}0; i = \eta_{CH} \leq 0{,}97; n_T < n_P)$.

c).The theoretical results obtained in this study are applicable at the pure research, the design, the working and the production of fluid couplings with high operating performances.

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