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MEAN CIRCULATION IN THE TORQUE CONVERTERS OF THE FIRST-CLASS

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

On the basis of this observation, in this paper, the hydrodynamic conditions of mean circulation in the torque- converters of the first-category/class are analyzed. Also, the geometrical and constructive-functional shapes, what establish an optimal mean circulation and, implicitly, a high operation ratio of torque-converters are analyzed.

Keywords: torque converter, mean circulation, first-class, overall efficiency.

1. Introduction

The mean circulation of liquid, in the working place of torque-converters, is responsible of the efficient conduction of mechanical power input rating from the pump impeller to the runner of hydraulic turbine and, therefore, of the operating performances of hydraulic transmissions.

It results that, the very efficient working of torque-converters requires the existence of an optimal mean circulation in the hydrokinetic's working place, [3].

Thus, unlike the fluid coupling, in the instance of torque-converter, the mean circulation there is into all workings. Consequently, here, it appears directly the question of mean circulation's optimization and, implicitly, the torque converter's optimization.

Further on, it proves that, the fixed coil has a significant purpose at the development of optimal mean circulation in the torque-converter.

The fixed coil quite possibly to be located at the entry on the pump-runner vanes-(first-class torque converter)-, or to the exit off the pump-runner vanes-(second-class torque converter), [1], [2], [4], [5], [6].

2. Generalities

The torque converters are hydraulic machinery with a complex structure.

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As well as we have asserted before, the analysis of mean circulation presents a different importance and for the case of torque converters. Thus, in the torque- converter, the mean circulation there is always, $(\varphi_e \neq 0)$, at all workings.

Therefore, here, it appears directly the problem of mean circulation's optimization and the torque converter's optimization, implicitly. Further on, the analysis of mean circulation in the torque converter is made clearly for the first-class torque converter, *(P→ T →S →P),* [1], [2], [6], and others.

3. Basic relations. Notations in use

The geometrical overall dimensions of the first-class torque converter, *(P→ T →S →P)*, had an only step for the runner of a hydraulic turbine and for fixed coil, are presented in the Figure nr. 1.

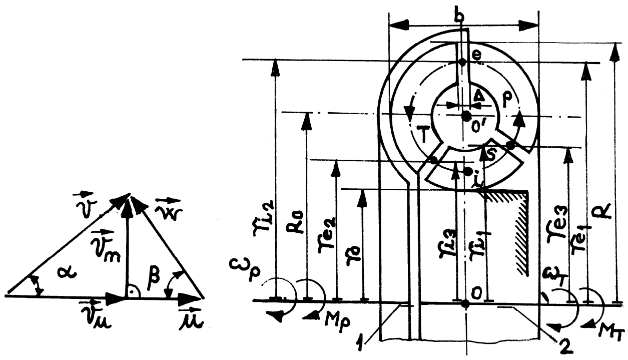

Fig. 1. The geometrical overall dimensions of the first-class torque converter. The velocity triangle.

For concordance, we mention that, the notations in use, in the Fig. 1 and further on, have the significances universally accepted, [1], [2], [3], [6], and others.

Kept into account a the expression of the velocity of conveying, $u = \omega \cdot r$, and with the fundamental in facts equation of the momentum, $\vec{M}_{S\to L}$, [3], applied to the torque converter of the first-class, [1], [2], [3], the momentums at the pump shaft, M_P, at the turbine shaft, M_T, and to fixed coil, M_S, are obtained.

Thus, with the notations from the Fig. 1, it results:

$$\begin{aligned} M_P &= \rho \cdot Q_P \cdot \left(r_{e_1}^2 \cdot \omega_P - r_{i_1}^2 \cdot \omega_T\right); \\ M_T &= \rho \cdot Q_T \cdot \left(r_{i_2}^2 \cdot \omega_T - r_{e_2}^2 \cdot \omega_P\right); \\ M_S &= \rho \cdot Q_S \cdot \left(r_{i_3}^2 \cdot \omega_T - r_{e_3}^2 \cdot \omega_P\right); \end{aligned} \tag{1}$$

The hydraulic circuit been a closed-circuit arrangement, it results that:

$$Q_P = Q_T = Q_S \equiv Q = const.; \tag{2}$$

Then, the relations (1) become:

$$\begin{aligned} M_P &= \rho \cdot Q \cdot \left(r_{e_1}^2 \cdot \omega_P - r_{i_1}^2 \cdot \omega_T\right); \\ M_T &= \rho \cdot Q \cdot \left(r_{i_2}^2 \cdot \omega_T - r_{e_2}^2 \cdot \omega_P\right); \\ M_S &= \rho \cdot Q \cdot \left(r_{i_3}^2 \cdot \omega_T - r_{e_3}^2 \cdot \omega_P\right); \end{aligned} \tag{3}$$

With the same notations, [3], for:
the reduction ratio, i:

$$i = \frac{\omega_T}{\omega_P} = \frac{n_T}{n_P}; \tag{4}$$

or, *the slip of torque converter, s*:

$$s = 1 - i = \frac{\omega_P - \omega_T}{\omega_P}; \tag{5}$$

the radii ratio, δ:

$$\begin{aligned} \delta_1 &= \frac{r_{i_1}}{r_{e_1}}; \\ \delta_2 &= \frac{r_{i_2}}{r_{e_2}} = \frac{r_{e_1}}{r_{e_2}} = \frac{r_{i_1}}{r_{e_2}} \cdot \frac{1}{\delta_1}; \\ \delta_3 &= \frac{r_{i_3}}{r_{e_3}} = \frac{r_{e_2}}{r_{e_3}} = \frac{r_{e_2}}{r_{i_1}} = \frac{1}{\delta_1 \cdot \delta_2}; \end{aligned} \tag{6}$$

the dimensionless coefficient of correlation of the speeds, φ_e:

$$\varphi_e = \frac{v_m}{u_e} = \frac{v_m}{r_e \cdot \omega_P}; \tag{7}$$

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

$$Q = v_m \cdot A \cong v_m \cdot \pi \cdot R^2 \cdot \left[1 - \left(\frac{r_0}{R}\right)^2\right] = \varphi_e \cdot r_e \cdot \omega_P \cdot A; \tag{8}$$

where: $A \cong \pi \cdot R^2 \cdot \left[1 - \left(\frac{r_0}{R}\right)^2\right]$,

then, the relations of momentums (3) become:

$$M_p = \rho \cdot Q \cdot r_{e_1}^2 \cdot \omega_P \cdot \left[1 - \left(\frac{r_{i_1}}{r_{e_1}}\right)^2 \cdot \left(\frac{\omega_T}{\omega_P}\right)\right] = \rho \cdot A \cdot r_{e_1}^3 \cdot \omega_P^2 \cdot \left[1 - \delta_1^2 \cdot i\right] \cdot \varphi_e; \tag{9}$$

$$M_T = \rho \cdot Q \cdot r_{i_2}^2 \cdot \omega_T \cdot \left[1 - \left(\frac{r_{e_2}}{r_{i_2}}\right)^2 \cdot \left(\frac{\omega_P}{\omega_T}\right)\right] = \rho \cdot A \cdot r_{e_1}^3 \cdot \omega_P^2 \cdot \left[i - \frac{1}{\delta_2^2}\right] \cdot \varphi_e ; \quad (10)$$

$$M_S = \rho \cdot Q \cdot r_{e_3}^2 \cdot \omega_P \cdot \left[\left(\frac{r_{i_3}}{r_{e_3}}\right)^2 \cdot \left(\frac{\omega_T}{\omega_P}\right) - 1\right] = \rho \cdot A \cdot r_{e_3}^3 \cdot \omega_P^2 \cdot \left[\delta_3^2 \cdot i - 1\right] \cdot \varphi_e ; \quad (11)$$

4. The analysis of power transfer

The torque converters of the first-class quite possibly to have different performance data, [1], [2], [4], [5], [6], and others, depending on the shape of hydraulic circuit, the degree of filling, χ_f, the number of blade steps of the vane wheel and of fixed coil, the existence an of adjustable blade step, and others.

Because of existence of fixed coil in the torque converter's structure, quite possibly to be achieved a variation of momentums to the turbine shaft, 2, (Fig. 1), given the momentums from the pump shaft, 1, through the adequate alteration of the running direction of motive fluid.

These differences of momentum depend on the shape of the blades, (especially, the blades of fixed coil), and of the angles theirs.

On the other hand, these differences of momentum, for the same blades of fixed coil, depend on the reduction ratio $i = \frac{n_T}{n_P}$. Thus, it results amplifications of the pump's momentum, M_P, for the majority values of reduction ratio, $\frac{n_T}{n_P}$, *because, it is known, [1], [2], [5], [6], the this is the basic part of torque converter.*

The maximum amplification of pump's momentum is achieved when the turbine's speed is $n_T = 0$.

Supposing, now, there is a reduction ratio $i^* = \frac{n_T}{n_P}$, for that, the fixed coil's momentum is $M_S = 0$, then, it results that $M_P = M_T$. This operating point of torque converter quite possibly to be named *power operating point*, because the momentum of fixed coil, M_S, oneself alters the sense of rotation, [5], he passed through the point of zero moment, $(M_S = 0)$.Therefore, at the power operating point, $\eta_{CHC} \cong \eta_{CH}$, thus that the torque converter of the first-class, (CHC), works, approximately, similar with the fluid coupling, (CH).

If, however, the reduction ratio $\frac{n_T}{n_P} < i^*$, then, the momentum $M_T > M_P$, and $M_S < 0$, because the fixed coil tends to circulate in reverse direction given the runner of a turbine.

When, the reduction ratio is $\frac{n_T}{n_P} > i^*$, the momentum $M_P > M_T$, while the momentum $M_S > 0$, because the fixed coil tends to circulate in the same sense with the vane wheel.

On the basis of those who were presented, succinct, till now, it results that, a torque converter of the first-class quite possibly to amplify the momentum of pump impeller, for the bulk reductions ratio $\frac{n_T}{n_P} = i$, $\left(i = \frac{n_T}{n_P} \in [0{,}0,....,0{,}80) \right)$, but, even the maintaining constant of momentum, $M_T = M_P = M$, when $M_S = 0$ and $i^* = \frac{n_T}{n_P} \in (0{,}80,..,0{,}85)$, respectively, the reduction of pump's momentum, M_P, when $i = \frac{n_T}{n_P} \in (0{,}80,......,1{,}0]$.

On the other hand, from the relation (11), it results that, always, from the viewpoint of working and projecting geometry, the momentum $M_S < 0$, if the radii ratio $\delta_3 < 1{,}0$, (Fig. 1), and the reduction ratio $i \in [0{,}0,...,1{,}0]$, because, at these conditions, the term $\left(\delta_3^2 \cdot i - 1\right) < 0$.

Thus, on the basis of relations (9), (10) and (11), it results that, the momentum $M_T > M_P$, if the reduction ratio is $\frac{n_T}{n_P} = i < i^*$, while the momentum $M_S < 0$, therefore, when $\left(\delta_3^2 \cdot i - 1\right) < 0$ and $\left(1 - \delta_1^2 \cdot i\right) > \left(i - \frac{1}{\delta_2^2} \right)$.

As, from the relations (9) and (10), it has resulted that $M_T > M_P$, for the reduction ratio $i = \frac{n_T}{n_P} \in (0{,}95,...,1{,}0]$, and $M_T < M_P$, for $i = \frac{n_T}{n_P} \in [0{,}0,...,0{,}95)$, while, from the relation (11), it has resulted that, always, $M_S < 0$, for $i \in [0{,}0,.....,1{,}0]$, quite possibly to write:

$$M_T = M_P \mp M_S ; \tag{12}$$

But, the momentum $M_S < 0$; therefore:

$$M_T = M_P + M_S\,; \tag{13}$$

From the relations (12) and (13), it results that, *the torque converter of the first- class is a torque converter-amplifier, if the values of reduction ratio are included in the interval* $i \in [0{,}0,....,0{,}85)$.

From the relation (13), it results that, for to obtain the greatest increases of momentum, $M_T >> M_P$, you must needs like the momentum M_S to be as great as possible, respectively, the bracket $\left(\delta_3^2 \cdot i - 1\right) < 0$ to be as great as possible. This presupposes like the fixed coil's blades to be arranged, from the viewpoint of projecting geometry, [1], [6], thus that this desideratum to be fulfilled.

Therefore, further on, interest ourselves that conditions are necessary for like the momentums M_T=*maximum*, respectively M_S =*maximum*, or the following expression:

$$\left(\delta_3^2 \cdot i - 1\right) \cdot \varphi_e = maximum; \tag{14}$$

In accordance with the relation (14), it results that, this obtains the optimum, if the bracket has the absolute optimum and the dimensionless coefficient of correlation of the speeds, φ_e has, also, the optimum. Thus, the bracket has the absolute optimum, when the radii ratio $\delta_3 = \frac{r_{i_3}}{r_{e_3}} < 1{,}0$ has the minimum and, also, the reduction ratio, $i \in [0{,}0,...,1{,}0]$ has the minimum, but, that to provide the regular of torque converter of the first-class.

The radii ratio δ_3 characterizes the torque converter's versatility of the first-class and, implicitly, the radial extension of fixed coil, [1], [2].

From the viewpoint of projecting geometry and proper operation, the radii ratio δ_3 not quite possibly to be however small, (usually, $0{,}5 < \delta_3 < 1{,}0$, Fig. 1, [1], [2],[6]), while the reduction ratio, i, is necessary to be more great than zero, (usually, $i = 0{,}60,....,0{,}80$, [1], [5], [6]).

The above-mentioned values for the radii ratio, δ_3, and the reduction ratio, i, are necessary with a view to working of torque converter of the first-class with as great as possible overall efficiency, ($\eta_{CHC} \cong 0{,}85,...,0{,}90$,where, $\eta_{CHC} = \frac{M_T \cdot \omega_T}{M_P \cdot \omega_P} = \frac{M_T \cdot n_T}{M_P \cdot n_P} \equiv \frac{M_2 \cdot n_2}{M_1 \cdot n_1} = K \cdot i$, $K = \frac{M_2}{M_1}$, $i = \frac{n_2}{n_1}$),[5],[6],and others.

It results that, $M_T = maximum$, respectively $M_S = maximum$ quite possibly to be achieved, first of all, *if the dimensionless coefficient of correlation of the speeds has the optimum,* $\varphi_e = maximum$*, respectively, if it is provided/achieved an optimal mean circulation in the torque converter of the first-class.*

The running and geometrical conditions, in order that $\varphi_e = maximum$, it results from the velocity triangle specific to the torque converter, (Fig. 1), that is:

$$tg\beta = \frac{v_m}{u} = \frac{v \cdot \sin\alpha}{u} = \frac{v}{u} \equiv \frac{v_m}{u_e} = \varphi_e \,; \left(\alpha = 90^\circ\right); \qquad (15)$$

On the basis of the relation (15), quite possibly to achieve a quantitative evaluation of inlet angle of impeller β, if we take into the usage values of the dimensionless coefficient of correlation of the speeds, φ_e, $\varphi_e = 0{,}02,..,0{,}12\ (0{,}17)$, [1], [2], and others.

Thus, for the inlet angle of impeller β, it obtains the following values:

$$\beta = (1^\circ, \ldots\ldots, 10^\circ)\,; \qquad (16)$$

If, at once, we spread the analysis and we consider the torque converter of the first-class as well as a sequence of three-blades, (two revolving blades,-P, T, and one fixed blades, -S), and we observe the their sequence, P→ T→ S→ P, (Fig. 1), then, the fixed coil quite possibly to be considered like an ante-fixed coil, [7], and others.

In the event of absence of the ante-fixed coil, the angle $\alpha_1 = 90^\circ$, (normal entry in the pump impeller),[7], and others.

Then, if it is thought that we have ante-fixed coil in the torque converter of the first-class and that the pre-revolution effect is present into the his working,- the angle $\alpha_1 \neq 90^\circ$, $\left(\alpha_1 < 90^\circ\right)$.

Consequently, it results that, if the angle $\alpha_1 < 90^\circ$, then, the angle β_1, (Fig. 1), evidently, increases over the optimum indicated of the relation (16). At the same time, the curvature of blades increases.

At the reference material, [1], [2], [4], [7], and others, it is recommended like the inlet angle of impeller, $\beta_1 = \left(8^\circ, \ldots, 30^\circ\right)$, while, the outlet angle of impeller , $\beta_2 = \left(30^\circ, \ldots, 90^\circ\right)$.

Thus,-the relations (7), (15),-the dimensionless coefficient of correlation of the speeds, φ_e, increases and, implicitly, *it is ensured an optimal mean circulation in the torque converter of the first-class*.

5. Conclusions

a).The theoretical study analyses the hydrodynamic conditions as for the mean circulation and the power transmission from the torque converter of the first-class, as well as the geometrical and functional features that optimize the torque converter's working of the first-class.

b).The torque converter of the first-class is a torque converter-amplifier, if the values of reduction ratio are included in the interval $i = \frac{n_T}{n_P} \in [0,0,...,0,85)$. The torque converter of the first-class has a more great curvature of blades and, consequently, it achieves a more favorably mean circulation.

c).The theoretical results obtained in this paper are applicable at pure research, in the design, running and the manufacturing of torque converters of the first-class with high operating performances.

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