MEAN CIRCULATION IN THE TORQUE CONVERTERS OF THE SECOND-CLASS

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<u>Abstract</u>: The mean circulation presents a different importance and in the case of torque converters of the second-class, just because of the sequence in what the motive fluid pass through the constitutive elements of the torque converters, (P→ S→ T→ P).

In this article, the hydrodynamic conditions of mean circulation in the torque converters of the second-class are studied.

Also, the geometrical, constructive and functional shapes, which establish an optimal mean circulation and, implicitly, a high operation ratio of the torque converters of the second-class are analyzed.

Keywords: torque converter; second-class; mean circulation; overall efficiency.

1. Introduction

The mean circulation of liquid, in the working place of torque converters of the second-class, influences, in a large measure, the efficient conduction of power input rating from the pump impeller, *P*, to the runner of hydraulic turbine, *T*, because of, first of all, of inverted sequence between the vane wheel, *T*, and the fixed coil, $S, (P \to S \to T \to P)$, comparatively with the torque converters of the first-class, $(P \to T \to S \to P)$.

Further on, it proves that, this inverted sequence influences substantially the range of reduction ratio, *i*, where, the torque converter works like a torque converter-amplifier, a torque converter-reducer, or like a fluid coupling, [1], [3], [4], [6], [7].Thus, the existence of the fixed coil at the exit from the pump impeller, *P*, in the torque converter's structure of the second-class, has a reverse influence about the conducted momentum, comparatively with the case when the fixed coil is placed at the inlet in the pump impeller, *P*, in the torque converter's structure of the first-class, [1], [4].

2. Generalities

From the viewpoint of running, the torque converter of the second- class not differs, fundamentally, from the torque converter of the first-class, [1], [2],[4].

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Thus, this from behind has, generally, more great overall efficiency and it is, consequently, far greater frequently used at practical applications, in industry, [1], [2], [6], [7].

Also, in the torque converter of the second-class, the absorbed momentum, M_P, increases at the same time with the increase of hydraulic turbine's speed, n_T, [1], [2], [7].

Further on, the analysis of mean circulation in the torque converter of the second-class, $(P \to S \to T \to P)$, is done, been emphasized the above-mentioned hydrodynamic shapes, [1], [4], [6].

3. Basic relations. Notations in use

The geometrical overall dimensions of the torque converter of the second-class, $(P \to S \to T \to P)$, had an only step for the runner of a turbine and for fixed coil, are presented in the Fig. 1.

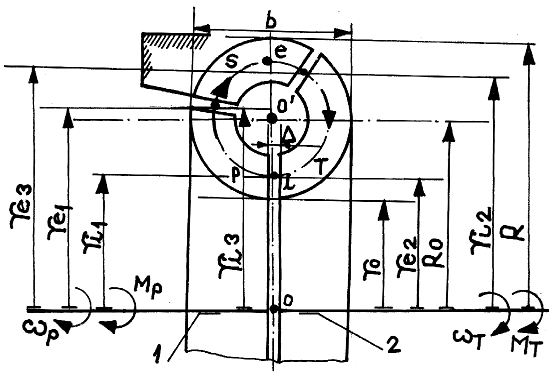

Fig. 1. The geometrical overall dimensions of the second-class torque converter.

We mention that, the notations in use, within the Fig. 1 and further on, have the established significances in literature, [1], [2], [4], [7].

Kept into account a the expression of velocity of conveying, $u = \omega \cdot r$, and with the fundamental in facts equation of the momentum, $\vec{M}_{S \to L}$, [3], [4], applied to the torque converter of the second-class, [3], the momentums at the pump shaft, M_P, at the hydraulic turbine shaft, M_T, and to fixed coil, M_S, are obtained.

Thus, with the notations in use at the Fig. 1, it results:

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

Because, and in this case, [3], [4], the hydraulic circuit of torque converter of the second-class is closed, it results that:

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

With the relation (2), the relations (1) become:

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

Using, further on, the definite notations in the literature, [3], [4], [7], and re-defined:

the radii ratio, δ,

$$\begin{aligned} \delta_1 &= \frac{r_{i_1}}{r_{e_1}}; \\ \delta_2 &= \frac{r_{i_2}}{r_{e_2}} = \frac{r_{e_3}}{r_{e_2}} = \frac{r_{e_3}}{r_{e_1}} \cdot \frac{1}{\delta_1}; \\ \delta_3 &= \frac{r_{i_3}}{r_{e_3}} = \frac{r_{e_1}}{r_{e_3}} = \frac{1}{\delta_1 \cdot \delta_2}; \end{aligned} \tag{4}$$

Then, the relations of momentums (3) become:

$$M_P = \rho \cdot Q \cdot r_{e_1}^2 \cdot \omega_P \cdot \left[1 - \left(\frac{r_{i_1}}{r_{e_1}} \right)^2 \cdot \left(\frac{\omega_T}{\omega_P} \right) \right] = \rho \cdot A \cdot r_{e_1}^3 \cdot \omega_P^2 \cdot \left[1 - \delta_1^2 \cdot i \right] \cdot \varphi_e; \tag{5}$$

$$M_T = \rho \cdot Q \cdot r_{i_2}^2 \cdot \omega_T \cdot \left[1 - \left(\frac{r_{e_2}}{r_{i_2}} \right)^2 \cdot \left(\frac{\omega_P}{\omega_T} \right) \right] = \rho \cdot A \cdot r_{i_2}^3 \cdot \omega_P^2 \cdot \left[i - \frac{1}{\delta_2^2} \right] \cdot \varphi_e; \tag{6}$$

$$M_S = \rho \cdot Q \cdot r_{e_3}^2 \cdot \omega_T \cdot \left[\left(\frac{r_{i_3}}{r_{e_3}} \right)^2 \cdot \left(\frac{\omega_P}{\omega_T} \right) - 1 \right] = \rho \cdot A \cdot r_{e_3}^3 \cdot \omega_P^2 \cdot \left[\delta_3^2 - i \right] \cdot \varphi_e; \tag{7}$$

4. The analysis of power transfer

Without in order to respect, the general shapes, as for the power transfer in the torque converter of the first-class, [4], are valid and for the torque converter of the second-class, but, with some differentiations what, further on, are praised.

Thus, if it is thought that, and in the torque converter of the second-class, there is a reduction ratio i^*, [4], for that $M_S = 0$, then, the momentums $M_T = M_P$.

If, but, the reduction ratio $i = \frac{n_T}{n_P} < i^*$, then, the momentum $M_P > M_T$, while the fixed coil's momentum $M_S > 0$, because the fixed coil tends rotate in the same sense with the vane wheel.

When the reduction ratio is $i > i^*$, the momentum $M_P < M_T$, while the momentum $M_S < 0$, because the fixed coil tends rotate in the reverse direction given the runner of a turbine.

On the basis of those above- mentioned, it results that, a torque converter of the second-class it may be to amplify the pump's momentum, M_P, for a limited range of the reduction ratio $i = \frac{n_T}{n_P}, (i \in (0{,}65,....,1{,}0])$, but, and the holding load $M_P = M_T$, when $M_S = 0$ and $i^* = \frac{n_T}{n_P} \in (0{,}55,...,0{,}65)$, respectively, the reduction of pump's momentum, for the bulk reduction ratio's values, when $i \in [0{,}0,...,0{,}70)$.

On the other hand, from the relation (7), it results that, from the viewpoint of working and design, the fixed coil's momentum $M_S > 0$ and $M_S \leq 0$, if the radii ratio $\delta_3 < 1$, (Fig. 1), and the reduction ratio $i \in [0{,}0,..,1{,}0]$, because, it these conditions, the term $(\delta_3^2 - i) > 0$ and $(\delta_3^2 - i) \leq 0$.

Thus, on the basis of relations (5), (6) and (7), it results that, the momentum $M_T < M_P$, if the reduction ratio is $i < i^*$, while the momentum of fixed coil $M_S > 0$, therefore, when $(\delta_3^2 - i) > 0$ and $(1 - \delta_1^2 \cdot i) > \left(i - \frac{1}{\delta_2^2}\right)$.

As, from the relations (5) and (6), 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 $i \in [0{,}0,...,0{,}95)$, while, from the relation (7), it has resulted that, the momentum $M_S > 0$, for the reduction ratio $i = \frac{n_T}{n_P} \in [0{,}0,...,0{,}70)$ and the momentum $M_S < 0$, for $i = \frac{n_T}{n_P} \in (0{,}65,..,1{,}0]$, it results that, always, there is the following relation:

$$M_T = M_P \pm M_S; \tag{8}$$

But, the momentum of fixed coil $M_S > 0$, for the bulk reduction ratio's values, $i \in [0{,}0,..,0{,}70)$, therefore:

$$M_T = M_P - M_S; \tag{9}$$

The relations (8) and (9), as well as the previous analysis, indicate that, *the torque converter of the second-class is a torque converter-reducer, if the values of the reduction ratio are included in the interval* $i = \frac{n_T}{n_P} \in [0{,}0,....,0{,}70)$, already, known.

On the other hand, if we have with a view that, for a torque converter, generally, the reduction ratio is $i = \frac{n_T}{n_P} \in [0{,}0,....,1{,}0]$, it results that, *the torque converter of the second-class is a torque converter-reducer for the majority reduction ratio's values*.

Also, the torque converter of the second-class achieves a rising characteristic $\lambda(i)$,[1], [2], and others, that is, the taken momentum increases once with the increase of turbine's speed, n_T.

From the relation (9), it results that, for to obtain an extremely great reduction of momentum, $M_P >> M_T$, it is necessary like the momentum of fixed coil, M_S,to be positive and as more great, respectively the value of the bracket $(\delta_3^2 - i) > 0$ to be as more great.

This is achieved provided that the fixed coil's blades to be arranged, from the viewpoint of projecting geometry, [1], [4], [7], thus that this desideratum to be fulfilled.

Therefore, further on, to us interest that conditions are necessary for like the momentums $M_T = minimum$, respectively $M_S = maximum$, or the following expression to obtain the value:

$$(\delta_3^2 - i) \cdot \varphi_e = maximum; \qquad (10)$$

In accordance with the relation (10), 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.

The bracket $(\delta_3^2 - i)$ was analyzed previously, this quite possibly to be $(\delta_3^2 - i) > 0$, respectively $(\delta_3^2 - i) \leq 0$, depending on the values of reduction ratio $i \in [0{,}0,...,1{,}0]$ and of the radii ratio's values δ_3, $\delta_3 = \frac{r_{i_3}}{r_{e_3}} < 1{,}0$, that it is sub-unitary always, (Fig. 1).

The conclusions as for the influence of radii ratio δ_3 and of reduction ratio, i, respectively the efficiency of torque converter of the second-class, η_{CHC},are, somehow, similar with those from the torque converter of the first-class.

That in why, further on, *the working conditions in order that* $M_S = maximum$*, respectively* $\varphi_e = maximum$*, that is, the conditions of existence of an optimal mean circulation in the torque converter of the second-class, succinct, are analyzed.*

The geometrical and functional conditions, in order that the dimensionless coefficient of correlation of the speeds $\varphi_e = maximum$, result from the velocity triangle typical of the torque converters,[4], respectively they result of the relation established in the reference [4].

Thus, quite possibly to do a quantitative evaluation of inlet angle of impeller β, similar with the torque converter of the first-class, been obtained the values indicated of the relation established in the reference [4].

If, at once, we spread the analysis and we consider the torque converter of the second-class as well as a sequence of three-blades, (two revolving blades,-*P, T*, and one fixed blade,-*S*) and we observe the their sequence, $P \rightarrow S \rightarrow T \rightarrow P$, (Fig. 1), then, the fixed coil quite possibly to be considered like an after-fixed coil,[8], and others.

Thus, it is known that, [8], and others, for the rotary pumps and for hydrodynamic pumps, generally, *the after-fixed coil ensures a normal exit to the circumferential speed* $\vec{u}_2$, $\left(\alpha_2 = 90^\circ\right)$*, from the pump impeller.*

Therefore, in the event of torque converter of the second-class, *it may be to consider that not there is ante-fixed coil.*

If the ante-fixed coil not there is, then it is ensured a normal entry to the circumferential speed $\vec{u}_1$, $\left(\alpha_1 = 90^\circ\right)$, in the pump impeller,[8], and others.

It results that, in the event of torque converter of the second-class, the angle $\alpha_1 = 90^\circ$, that the inlet angle of impeller β_1 has the value included in the range recommended in literature, [4], and it determines the adequate curvature of blades.

As we have mentioned, and, in the reference [3], [4], in literature, [1], [2], [8], and others, it is indicated like the inlet angle of impeller, $\beta_1 \in \left(8^\circ,.....,30^\circ\right)$, while the outlet angle of impeller, $\beta_2 \in \left(30^\circ,.....,90^\circ\right)$.

From this analysis, it results that, in the event of torque converter of the second-class, the curvature of blades (the angle β) is much small than in the event of torque converter of the first-class.

Also, in accordance with the relation definite in the reference [4], it results that and the dimensionless coefficient of correlation of the speeds, φ_e, has a more small value. To put it differently, in the event of torque converter of the second-class, it is ensured a more small mean circulation than in the torque converter of the first-class.

These forms, entirely, operate thus like *the efficiency of torque converter of the first-class to be more great than the efficiency of torque converter of the second-class, for the same power and the overall dimensions.*

5. Conclusions

The principal conclusions of the theoretical study presented in this paper are the following:

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

b).The torque converter of the second-class is, unpractical, a torque converter-reducer, if and only if the values of reduction ratio are included in the interval $i = \frac{n_T}{n_P} \in [0{,}0,.......,0{,}70)$. This presents a more small curvature of blades and, consequently, it achieves a more small mean circulation.

c). The theoretical results established so far indicate that, the torque converter of the second-class achieves, unpractical, inferior efficiency given the torque converter pf the first-class.

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

The future concerns, in the domain of the torque converters, have in view to confirm the theoretical results obtained in this article, through ample and rigorous experimental studies.

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