THE MUTUAL INFLUENCE AND BLADE-ROW INTERACTION BETWEEN PUMP AND TURBINE IN A HYDRODYNAMIC TORQUE CONVERTER

MAREK WOŹNIAK[1], ZBIGNIEW PAWELSKI[2], PABLO DELA FUENTE[3], GUSTAVO OZUNA[4]

Lodz University of Technology, Ruhr Universität Bochum (Germany),
University of Sonora (Mexico)

Summary

In this paper the results of a numerical calculation of the unsteady flow inside a one-stage two-phase automotive torque converter will be presented.

For the investigation the finite volume method has been employed. The commercial 3D Navier-Stokes Software CFX of ANSYS Inc. was used for the flow simulation. Here the incompressible Reynolds-Averaged-Navier-Stokes (RANS) equations will be solved using the k-ε turbulence model.

The flow field is determined by the blade position of both rotors, which have different rotating velocities. Whenever two adjacent blade rows at different speed, unsteady interactions occurs in the flow. The unsteady flow at the pump exit and turbine inlet will be analyzed through instantaneous flow fields in a period so that the rotor-rotor interaction can be in detail understood. The inlet flow of the turbine was markedly periodic and influenced by the pump jet/wake. In contras the pump outlet flow showed a little dependence on the turbine relative position.

Keywords: hydrodynamic torque converter, CFD, unsteady state, flow simulation.

1. Introduction

Many vehicles on the road today have automatic transmissions. A hydrodynamic torque converter is essentially integral part of automatic transmissions. Hydrodynamic drives, namely hydraulic torque converters are widely used in vehicle power transmission systems. They connect the drive side with the driving engine and substantially determine

[1] Lodz University of Technology, Department of Vehicles and Fundamentals of Machine Design, 116 Żeromskiego Street, 90-924 Lodz, e-mail: mamwroz@poczta.fm

[2] Lodz University of Technology, Department of Vehicles and Fundamentals of Machine Design, 116 Żeromskiego Street, 90-924 Lodz, e-mail: zbigniew.pawelski@p.lodz.pl, ph.: +48 42 631-23-91

[3] Ruhr Universität Bochum, Germany

[4] Universidad de Sonora, Departamento de Ingenieria Industrial, Rosales s/n Colonia Centro, cp. 83000, ph.: +52 662 259 22 83

the stationary and dynamic behaviour of the whole system. So it is necessary to optimise its operating work through understanding its internal flow behaviour.

The fluid in a hydrodynamic torque converter is responsible for the power transfer from the engine to the gearbox. Torque converters like this model are commonly used in vehicle power transmission systems such as cars and buses. The typical parts of this machine are the pump, connected to the engine, a turbine connected to the transmission, a stator, which makes possible the torque conversion through the redirection of the flow to the pump and most importantly the working fluid, without which the torque converter could not work. The internal flow inside the torque converter is highly three-dimensional, turbulent, viscous, complex and highly unsteady. Because of these characteristics and the three elements rotating at different velocities is difficult to analyze.

One of the principal problems concern unsteadiness the blade-row interaction between the components where the geometry of the flow changes due to the proximity of the components with the rotation of the row. The trend of increase in aerodynamic loading of the blades and decrease in machine size and weight makes it important to understand fully these unsteady blade row interactions. With increasing aerodynamic loading of the blades the considerations of the blade-rows interactions gain importance. The blades and vanes moving relatively to each other interact due to viscous wakes and potential effects [3].

The torque converter here presented was simulated and validated in the papers [1-2]. Here the geometry definition, the mesh study, the validation with measurements was presented and discussed. The torque converter was designed by ZF Sachs AG and its geometry was published in [4]. The one-stage two-phase W240 H.T.C. has an outer diameter of 240 [mm]. The pump contains Z_p =31, turbine Z_t =29 and stator Z_s =11 blades respectively. The operating fluid is automatic transmission fluid ATF LT 71141 whose density is ρ =802 [kg/m^3] and the viscosity is ν =0,00653 [Pa s] at 95 [°C] [5]. All calculations were performed at a constant temperature of the fluid.

In the present work, a previously presented paper [2] is usedastle basis to calculate the 3D unsteady blade rows interactions. Firstly, efforts were made to achieve a reasonable relation between computational time-computational resources and satisfactory results through the validation with measurements. Secondly, the effects of unsteady flow were examined with particular attention to the difference between steady and unsteady results. Finally, in this paper a detailed examination of the internal flow in the gap between pump/turbine will be analyzed.

2. Unsteady flow in torque converters

Because of the axial proximity of the components in the torque converter, all elements are influences through an interaction called blade/row interaction that causes periodically unsteadiness. The flow in this region is very specific because the time-dependent flow field behaves unevenly so that a strongly oscillating stream at the turbine inlet takes place.

The small distance between pump and turbine leads to an increase of the mechanical flow interactions between the components and causes repercussions of the turbine obstruction on the downstream flow in the turbine, while the pump causes a downstream influence on the turbine flow passage.

The interactions are on viscous and lead to the generation of wakes at the trailing edge of the blades and the potential effect due to the flow obstruction at the leading edge of the next blade-row with the stagnation point. Affecting interaction is also the internal flow highly periodically at the contact surfaces (interfaces). This will be verified through the calculation results. This will particularly focus the flow field behaviour in the interaction area between pump and turbine.

Researches in the field of computational simulation of the blade/row interactions in the torque converter has its origin at the beginning of the 90s. Browarzik [6] measured the relative flow field between pump and turbine using hot-film anemometry. The results showed that the flow at the trailing edge of the pump is strong characterized by a wake/zone. Achtelik [7, 8] measured the unsteady flow of a torque converter. For this purpose, a semiconductor-based-high-frequency-response probe was developed and used to reproduce the flow field in the interaction region between the rotors. The flow field measurements were analysed and compared at two different operating points and showed a dependence of the interaction effects in function of the time dependant rotor positions and that the flow is extremely unsteady due to blade frequency of the rotors. Besides will also wakes/area of both impellers caused. Flack [9] investigated by measurements the interaction area between pump and turbine at two different operating points in twin different torque converters with different geometries. He observed that the speed fluctuations at the pump exit are smaller than that at the turbine inlet and that these were independent of the operating point. These fluctuations are caused by the pump-turbine interactions. The smaller torque converter showed larger fluctuations than the bigger at the turbine inlet due to the "jet-wake" generated from the pump. Kraus also [10] investigated the unsteady flow in the interaction area between pump and turbine. Measurements at four different relative positions at four operating points have been made. He came to the conclusion that the flow field at the turbine inlet is very uneven due to the periodic jet/wake area of the pump. The highest unsteadiness was found at high operating points. In fact, in this context can be the following author be referenced [11-16].

3. Results

To illustrate the relative interactions in the flow field between pump and turbine will be following the time-dependant meridional velocity presented at the pump exit and turbine inlet by different rotor positions through axial contours and charts, in figure 1. The vector sum of the axial and radial vector components of velocity results in the meridional velocity. It lies in the meridional plane. For the unsteady simulation at design point v=0.7, which corresponds to the highest value of the efficiency. This means the turbine speed n_p=2000 [rpm] amounts to 1400 [rpm].

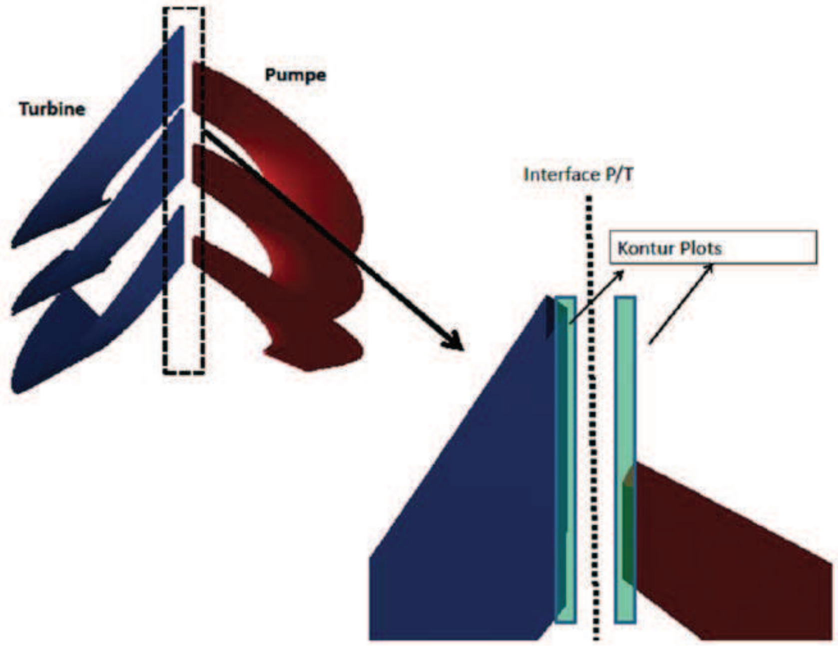


Figure 1. Position of the planes at the pump exit and turbine inlet to describe the blade/row interaction.

3.1 Upstream influence of the turbine

The figure 2 shows the time evolution of the meridional velocity (Cm) at pump exit plane (left) and the curves on the right represent the meridional velocity curve course in 50% of the hub-shroud position (center cut) in the plane. To interpret the figures in the relative frame of reference stay still the pump blades = 4, 16 and 28 [°] (red line in the charts), while the turbine blades move due to their smaller relative velocity from the right to the left side.

The pictures by different positions show a similar behavior between the over canals at the pump exit and it can clearly be seen that a time-dependent velocity distribution takes place. The position of the generation of the wakes can clearly be identified in the diagrams, while the shock position of these and the formation of the potential effect are characterized through a velocity deficit. Here the pump exit flow will be influenced by the potential effect of the downstream situated turbine. The reflected shock causes pressure disturbances on the turbine blades, which will be transported and affect the pump exit flow. While in the lower channel area is not noticeable a time-dependant change of the velocity, in the middle of the passage can be observed, due to the aerodynamic blade/row interactions. Besides the interaction between the rotors is in the middle of the passage and close to the shroud stronger. The closer the pressure sides of both blades, the higher the velocity maxima in the middle of the passage, the smaller the region of high velocity and the stronger the upstream influence of the turbine, which is characterized by a reduction of the speed in the turbine passage. As shown in the picture by 0.6 T, the maxima of the velocity are close to the pressure side and are approximately 9 [m/s], while the minima almost 2 [m/s] at the hub-suction side. The influence of the turbine decreases in the direction of the pump pressure side, therefore uneven forces and torques will be generated on the pump blade. The main cause of these phenomenal are the downstream located turbine blade row, the pitch-ratio between the two components, that rotate at different speeds, the axial gap between the components and the complex geometry. The Cm charts showed maximal difference between higher and lower peak of approximately 7.5 [m/s]. The lowest points of the curves correspond to the position of the trailing edge of the pump blade, it means the of generation of the jet/wake.

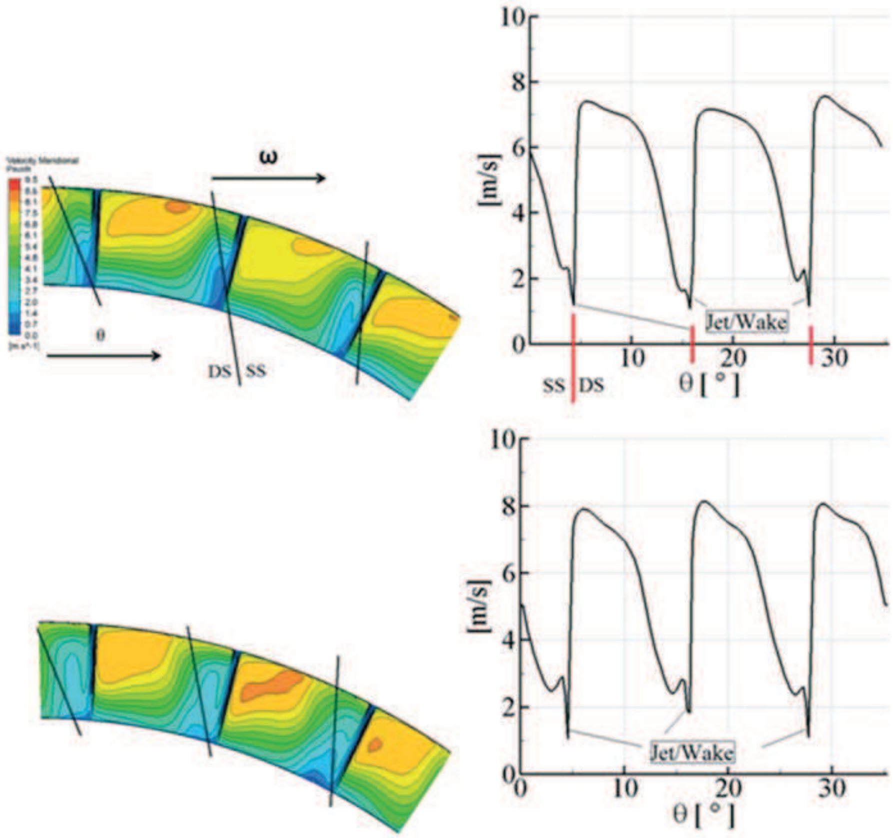


Figure 2. Meridional velocity (Cm) contour plot for a period T at the pump exit and turbine relative position (indicated by the black lines). Rotor relative positions 0 T, 0.2 T, 0.4 T, 0.6 T and 0.8 T.

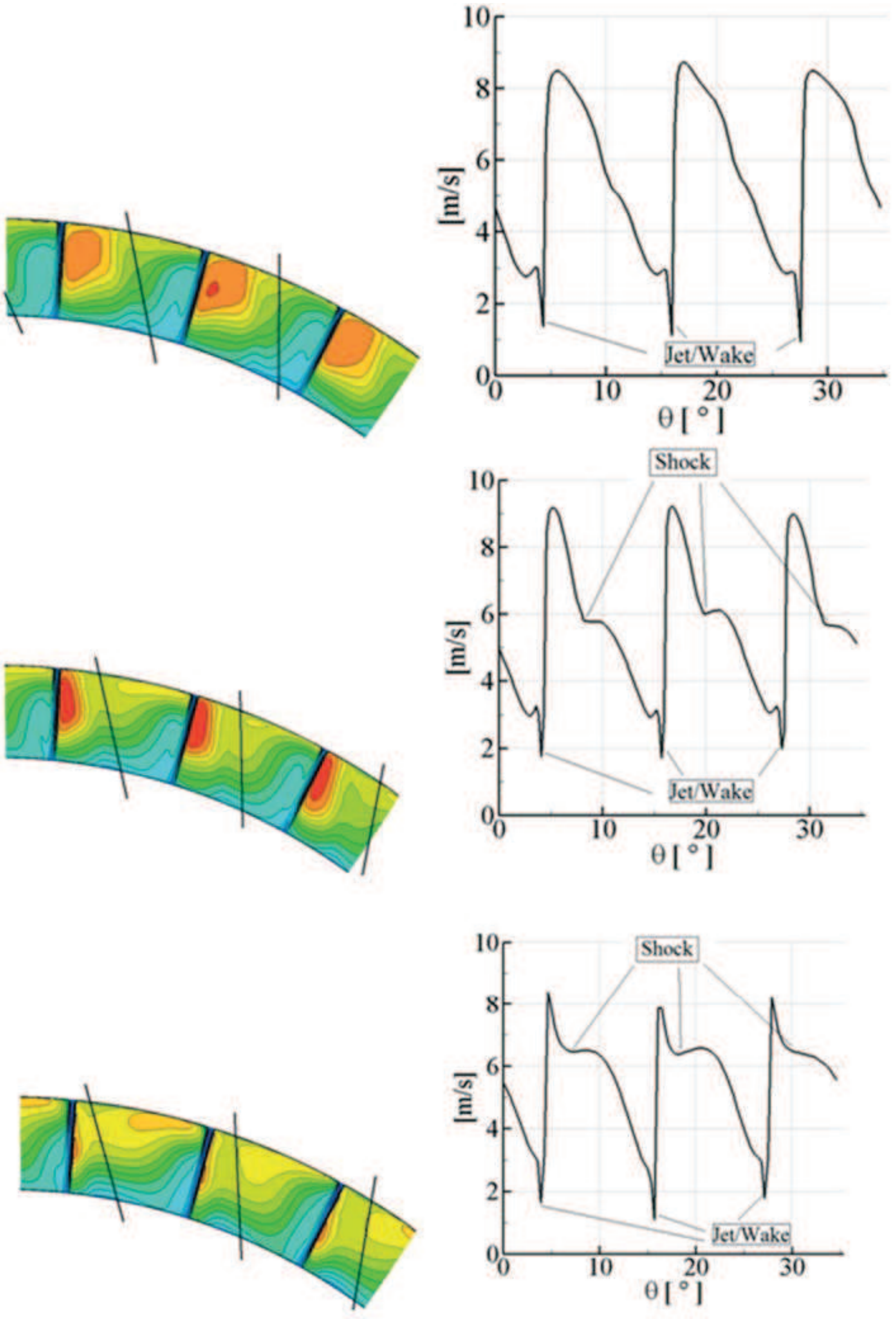


Figure 2. Meridional velocity (Cm) contour plot for a period T at the pump exit and turbine relative position (indicated by the black lines). Rotor relative positions 0 T, 0.2 T, 0.4 T, 0.6 T and 0.8 T.

3.2 Downstream influence of the pump

The figure 3 shows the calculated distribution of the meridional velocity in a constant axial contour plot at the turbine inlet and the charts represent the Cm course in function of the angle. The movement of the turbine in a circumferential direction relative to the pump is in the positive counterclockwise movement. Here again five representative positions of the pump/turbines blades will be analyzed and presented. The aim is to determine the speed distribution in the relative reference system of the turbine inlet for different relative positions of the pump. To interpret the figures in the relative frame of reference the turbine blade stay still at = 6, 18 and 30 [°] while the pump blade move due to its higher rotation speed and relative movement, to the right.

At first sight can a clearly dependence speed be seen (figure 3). This dependence is cause by the different position of the pump and turbine blades. The different velocity changes resulting from the time-dependant flow field cause unsteady oscillating effects here at the turbine inlet. At the inlet of the turbine close to the pressure side velocity disturbances take place, exactly where the jet/wake spread. The rotor-rotor interaction is influenced here by the flow in all the turbine passage. The speed at the turbine pressure side is strongly influenced by the relative blade position. The wake at the trailing edge of the turbine causes periodic unwanted flow angles, this involves a speed reduction. The wakes generate at the pressure side pressure variations cause pressure fluctuations, force fluctuations, oscillating torques on the turbine blades and perturbations at the leading edge of the turbine, due to the periodically varying inflow. Looking at the pictures on the left side of figure 2, then the position of the pump blade can easily be founded by the velocity deficit (wake-area) in the turbine passage. This low velocity region between two large areas of high speed is easy to see at 0 and 0.2 T, due to the proximity of both blades, and corresponds to the pump and its suction side blade position. The loss fields generated by the pump are localized on the proximity of the hub of the turbine pressure side, as can be seen clearly in figures that correspond to the time 0 T, 0.2 T and 0.4 T. These three areas of low velocity (loss areas), which correspond exactly to the relative blade positions of the pump, can be seen clearly in the figure 2. The smaller the distance between the trailing edge of the pump and the leading edge of the turbine, the more forced the pump exit flow in the turbine passages, opposite to the pump pressure side, and with this will be stronger interaction between the blade/rows.

The high-speed area starts at the turbine suction side at the shroud-corner and moves to the pressure side and forming two high-speed jets at the turbine plane before the high-speed the pressure side of hub-corner scattered, and the period begins again. This high-speed area covers at 0 T and 0.2 T approx 70% of the passage, this way the turbine flow accelerates again quickly after the pump blade is passed. Then the flow on the suction side of the blade will be gradually accelerated again. In this case, the perturbation of the turbine run direction pump against the main incoming flow. At the positions 0.4 T, 0.6 T and 0.8 T splits the large area of high speed (free turbine passage) in two areas through a zone of lower speed, that corresponds to the position of the pump blades. Here, the superposition of the pump and turbine wakes take place. The zone of high speed of the turbine in the passage has a movement from right to left and reached 9.8 [m/s] at 0.6 T.

The right charts of figure 3 show the meridional velocity at midspan between the three turbine blades of a period. The illustrations show clearly the dof the velocity that corresponds to the jet/wake of the pump and its trailing edge. These travel time from right to left, because the view is in the relative frame of reference of the turbine. The three speed maxima correspond to the location of the pressure side at the turbine inlet area. At 0 T and 0.2 T is the superposition of fast and slow fluid flow (wake) clearly. Here the wake will be with the high speed area compensated. At 0 T amounts to the speed inside the wake between 1-3 [m/s]. Outside of the wake increases the speed very quickly again to a maximum until reaching a maximum of 9 [m/s]. Moreover that the local minima in the passage move to the left side, towards the pump pressure side is conspicuous.

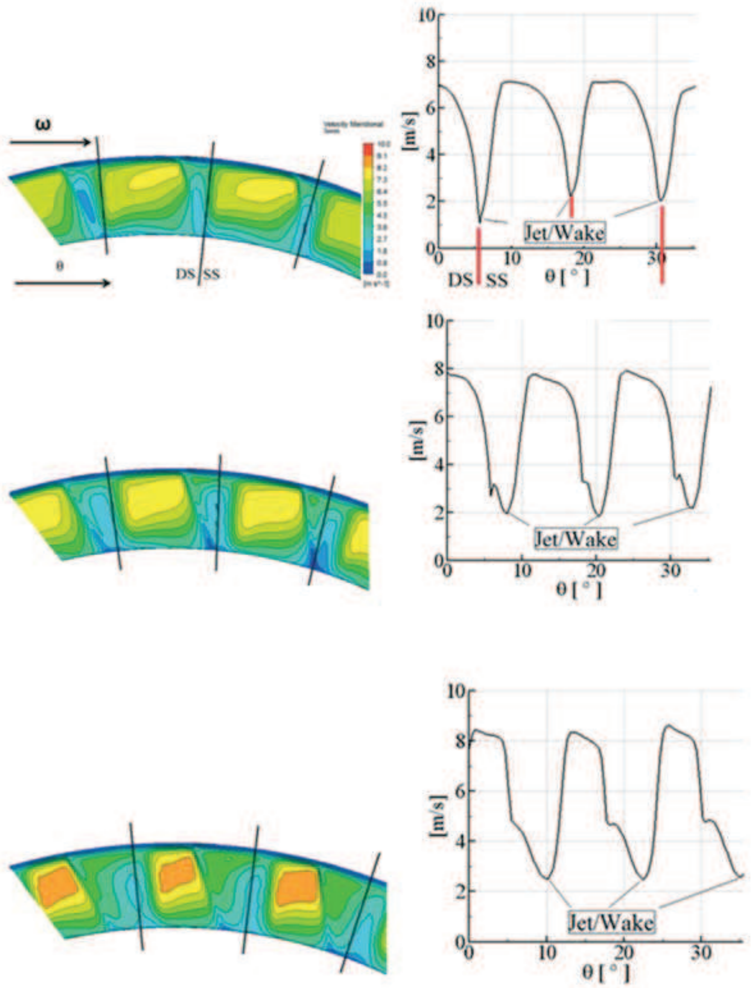
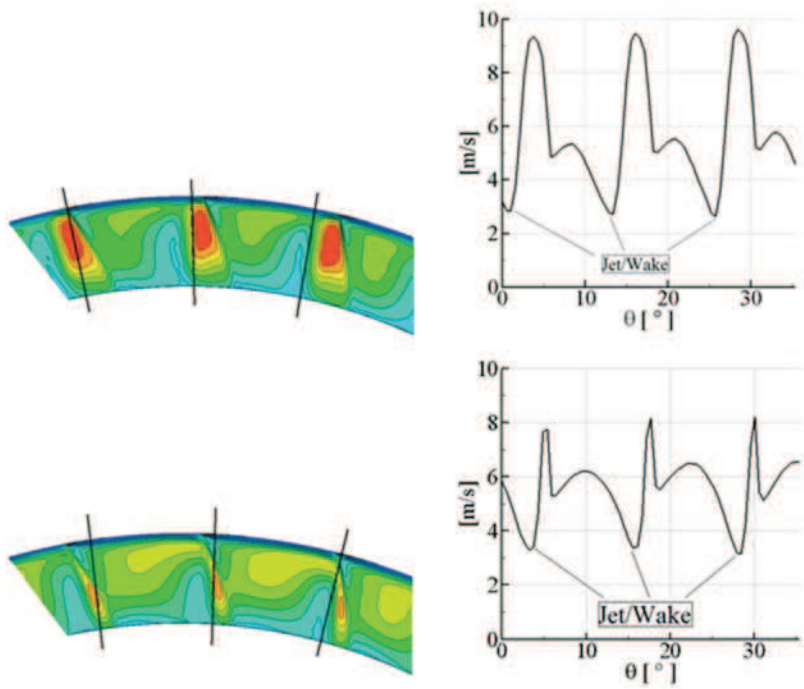


Figure 3. Meridional velocity (Cm) contour plot for a period T at the turbine inlet and pump relative position (indicated by the black lines). Rotor relative positions 0 T, 0.2 T, 0.4 T, 0.6 T and 0.8 T.



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3.3 Flow between pump and turbine

In assessing the circulating flow field in the interaction area, three identical local velocity minima can be localized, which correspond to the wake of the pump. By the comparison of the meridional velocity for a specific time (0.8 T) at the pump exit and turbine inlet plane of figure 4 can clearly be seen, that upstream influence caused by the turbine decreases rapidly and it is expected here that this is barely visible in the pump passage. In the figure 4, the variation of the upstream interaction decreases from 7.8 to 5.3 [m/s] (turbine inlet). In the pump exit plane the potential effect causes a decrease from 8.3 to 6.5 [m/s].

4. Conclusion

The present paper shows the flow investigation of a torque converter. The results showed a 3D unsteady behavior in the pump/turbine interaction area between incoming wakes and turbine passage structure. The wake area shows a wake structure in the region at turbine inlet.

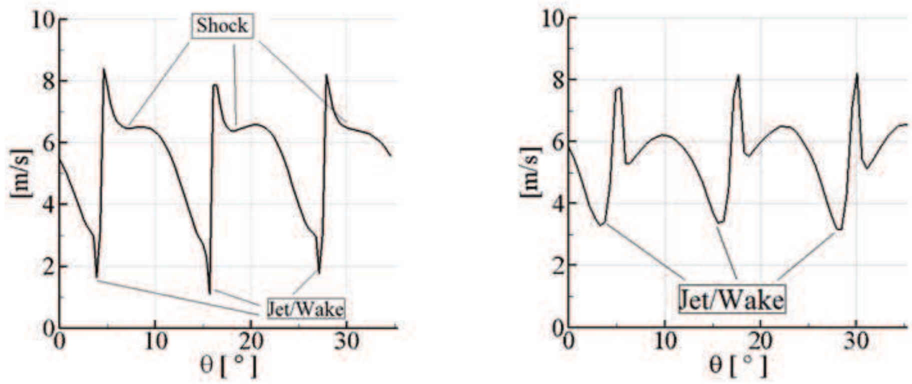

Figure 4. Comparison of the meridional velocity Cm at the pump exit plane (left) and the turbine inlet plane (right) at 0.8 T

The turbine blade position have little influence on the pump exit flow field, whereas the turbine inlet flow shows a significant periodic dependence on the relative pump blade positions. It was noted that the rotor wakes at turbine inlet are stronger than those at pump exit.

It can be concluded, that the upstream transported reflected shock on the turbine blade influences the pump perturbations of exit flow. The simulation shows clearly that a slightupstream interaction or speed variation takes place and especially in the area of the pressure side of the turbine can clearly be recognized. In addition, the studies showed no special characteristics with the exception of the shock (potential effect) and the jet/wakes. The upstream disturbances cause a speed reduction at the turbine pressure side, which leads to a periodic pushing back of the high velocity zone to the pump pressure side, followed by a slightly acceleration on the suction side in the turbine passage.

In summary, the pump has a significant influence on the turbine inlet flow field, because the non uniform pump exit flow will be directly forced into the turbine passage. The influence of the pump blade position on the inlet turbine contour is due to the non uniform flow will be forced inside the turbine passage independent of the relative position between the pump and turbine blades.

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