EXPERIMENTAL INVESTIGATION OF THE INFLUENCE OF DRAFT TUBE FLOW RESTRICTIONS ON THE AXIAL HYDRAULIC FORCE ON FRANCIS TURBINES

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Abstract. *Environmental regulations have required from Brazilian hydroelectric power companies stronger and more efficient means of restricting the access of fish and other aquatic species into the draft tube of hydraulic turbines. Decrease of fish populations on rivers used for electricity generation has motivated the installation of flow restrictors on the tailrace pipe of hydroelectric power plants as a technical solution to preserve aquatic wild life. However, variations on the flow conditions through the turbine runner modify the performance characteristics and efficiency rate of hydro turbines, jeopardizing their capacity of electricity production. This paper presents the experimental results of an investigation carried out to evaluate the influence of draft tube flow restrictions on the hydraulic forces acting on Francis turbines. A Francis turbine test bench, based on the dimensions of an operational hydrogenerator, is specially designed to allow this investigation. Flow rate and hydraulic pressure on the water intake pipe and the spiral case, the hydrogenerator unit speed, the generator electric frequency and shaft torque and the thrust bearing axial force are monitored by using several mechanical and electrical transducers mounted on the test rig. Performance characteristics are experimentally determined for different operating conditions permitting the obtaining of power and efficiency rate versus operating rotational speed curves. Finally, curves of axial hydraulic thrust versus speed are obtained at different flow restriction conditions, which are included into the test bench by using a movable gate at the draft tube exit.*

Keywords*: Draft Tubes, Hydraulic Turbines, Mechanical Barrier, Fish Barrier.*

1. INTRODUCTION

The decrease of fish and other aquatic populations on rivers used for hydroelectric power plants has caused strong concerns on government environmental agencies (Agostinho *et al.*, 2005). Mechanisms have been devised to allow fish species swim upstream or bypass the turbine runners without injuries or any physical accident along the way (Schowalter and Kuehlert, 2005). However, fish populations have entered the turbine draft tube, mainly during operational flow speed reduction, trying to swim upstream through the hydraulic machine (Abernethy *et al.*, 2003). Preliminary *in-situ* investigations indicate that the flow changes on the river streams provoked by the hydraulic turbine circuit may offer ideal flow conditions for fish attraction. Hence, the access of aquatic animals into the draft tubes of hydraulic turbines has caused numerous ecological casualties, which have generated frequent fish killings (Garrison *et al.*, 2002).

In order to impede fish to swim upstream through turbine runners, different repelling mechanisms and barriers have been thought as feasible solution for the problem (Martinez, 2007). Among all feasible impeding and repelling mechanisms for fish entrance into the turbine draft tubes, the mechanical gate is the simplest, since it can be easily mounted and operated. Although the mechanical gate can be considered the simplest solution for impeding the fish entrance into the hydraulic turbine, its impact on the flow conditions through the runner and on the hydraulic loading generated on the runner shaft is not completely understood.

This paper presents an experimental study carried out to investigate the influence of mechanical barriers installed at the turbine tailrace pipe on the mechanical behavior of the turbine runner shaft. A hydraulic Francis turbine model is specially built to permit this experimental investigation (Paulino *et al*., 2005). The model comprises of a Francis turbine, a draft tube with guides for a movable gate, a water pumping system for simulating the turbine headwater and flow rate conditions, a load cell installed on the shaft thrust bearing, and an instrumentation system for measuring speed, torque, pressure, distributor vane opening, and flow velocities along the turbine hydraulic circuit. Firstly, the turbine efficiency curves are determined to estimate the machine optimal operating points. Secondly, several experiments are conducted to measure the axial load on the shaft thrust bearing caused by different draft tube flow restrictions, which are generated by the movable gate installed at the tailrace pipe. Efficiency curves for the Francis turbine model and curves of hydraulic force versus gate opening are obtained at several operating conditions. The experimental results rendered in this investigation indicate that mechanical gates, installed as a fish barrier at the turbine draft tube, must be operated very carefully because they can lead to an expressive hydraulic force increase on the runner shaft bearing and, consequently, can compromise the turbine mechanical integrity.

2. FRANCIS TURBINE TEST BENCH

Figure 1 depicts a frontal picture of the Francis turbine model developed for this work. At the foreground plane, they are shown the turbine spiral case, made of cast aluminum, the runner shaft and its bearing housing, made of stainless steel, and the display for the axial load cell (mounted on the transverse bar over the turbine). At the right side of the picture, it can be seen the red flexible pipe used for water intake. Part of the turbine draft tube, made of fiberglass and painted with blue ink, is shown at the lower part of the picture. At the background plane, a special water reservoir, made of commercial steel and painted with grey ink, is installed to simulate the downstream afterbay. A side view of the turbine test bench is shown on Fig. 2. In Fig. 2, the movable gate can be viewed at the bench tailrace pipe. At the upper side of the turbine spiral case it is shown a circular wheel used to move the runner vanes.

Figure 1. Frontal picture of the vertical Francis turbine installed at its test bench.

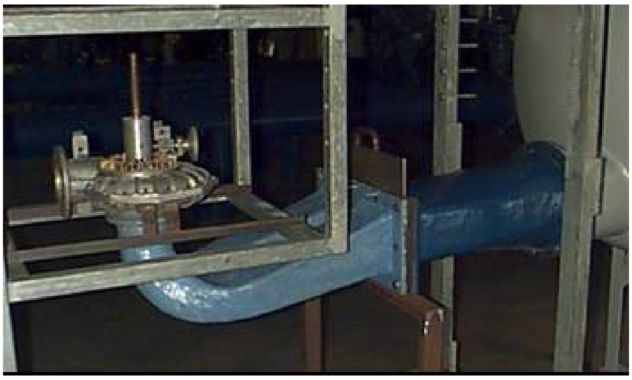

Figure 2. Side picture of the test bench showing the gate installed at the draft tube exit.

A manual motorcycle brake is used to reduce the runner shaft speed during the tests. Moreover, an axial disk load cell is installed under the thrust bearing collar in order to render the experimental values of axial force acting on the turbine runner during the gate movement. Figure 3 presents two pictures that show more details of the braking system used on the test bench and the assembly of the disk load cell on the turbine spiral case. A ruler is used to indicate the level of flow restriction at the tailrace pipe or the position of the mechanical gate, which is shown on the background plane of the left-hand side part of Fig. 3.

Figure 3. Pictures of the braking system (left-hand side) and the axial force transducer (right-hand side).

The turbine runner is designed specially for this investigation. Its dimensions are based on the geometric characteristics of a full-scale turbine runner installed at the power station of Nova Ponte, operated by CEMIG. Figure 4 shows frontal and side pictures of the Francis runner, which is made of commercial steel. The runner height, trailing diameter and leading diameter are equal to 0.029 m, 0.084 m and 0.102 m, respectively.

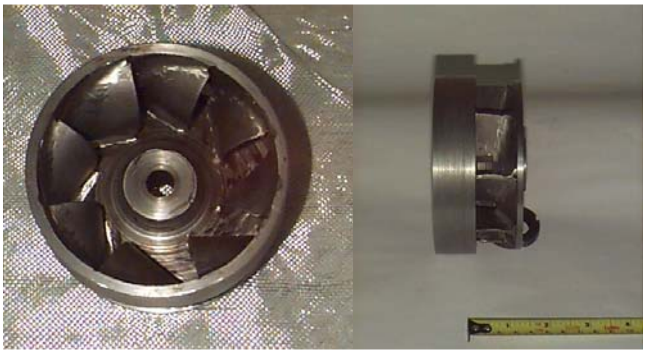

Figure 4. Photos of the turbine runner installed at the test bench.

The water intake on the Francis turbine, shown on Fig. 1, is provided by a pumping system, which works as a dam simulator, capable of simulating several operating pressure and flow rate conditions. Figure 5 shows a picture of this pumping system, which consists of two sets of 10 hp hydraulic pumps, whose electrical motors are connected to two frequency inverters. These inverters allow varying controllably the pump rotating speed. Furthermore, the pumping system has several pressure and flow rate transducers to permit the monitoring of the turbine operating conditions. This system is capable of providing constant intake pressure for any turbine vane openings (Barbosa *et al.*, 2006).

Figure 5. Pumping system developed to provide several turbine pressure and flow conditions.

3. PERFORMANCE AND EFFICIENCY TESTS

Before performing the experimental evaluation of the influence of the mechanical gate, which is installed at the turbine tailrace pipe, on the runner hydraulic forces, the performance and efficiency curves for the turbine model must be determined. These preliminary tests are needed to assess the nominal operating conditions for the turbine model. The turbine optimal performance is expected to occur at these nominal operating conditions. Naturally, the influence of the flow restrictions on the turbine mechanical behavior has to be investigated at the turbine ideal conditions, at which the hydraulic machine should be operated to provide maximum efficiency.

For experimental convenience and feasibility, a second experimental setup is assembled to perform the turbine performance preliminary tests. The turbine shaft is connected to a DC electrical generator and several transducers are employed to measure the runner speed, the shaft torque, the distributor vane opening, and the intake pressure on the turbine model. Figure 6 depicts the second turbine assembly used to render the performance characteristics under several operating conditions. The turbine runner is set at the horizontal position because of the electric generator available for these preliminary tests, which can not be mounted on the vertical position. The performance tests are performed at constant pressure, which is estimated at 11 m of intake headwater from the hydrodynamic equations for reduced models. From these equations, the turbine model water mass flow rate and specific speed are equal to 0.026 m^3/s and 174 rpm, respectively.

Figure 6. Experimental setup employed to determine the turbine performance characteristics.

The first step for these preliminary tests is to select an opening angle for the runner vanes. Then, the pumping system is started up and the pump speed is varied through the frequency inverters. The flowmeters installed on the pumping system allows estimating the water flow rate passing through the turbine runner. Next step is to apply a load on the generator and, thus, to measure the runner speed and shaft torque. The tests are repeated for ten values of the distributor vane opening.

The curves of turbine power versus the runner speed, at ten vane opening levels, are shown on Fig. 7. It is shown that maximum turbine power is obtained for 70% or more of vane opening at the speed range from 1,600 rpm until 1,900 rpm.

The turbine efficiency contour plots are shown on Fig. 8, which also depicts the curves of turbine power versus speed. The numbers shown on the grids of Fig. 8 are related to the turbine efficiency. It can be noticed that the maximum efficiency for the turbine reduced model is about 50 %, which occurs at speed range from 1,500 rpm to 1,800 rpm, at vane opening of 70%.

4. MEASUREMENTS OF THE RUNNER HYDRAULIC THRUST

From the preliminary tests performed on the Francis turbine model, the maximum efficiency is reached for vane opening of 70 % at the speed range from 1,600 rpm to 1,800 rpm. Then, the investigation of the influence of draft tube flow restrictions on the hydraulic axial forces acting on the runner shaft bearing is performed in two stages. Firstly, the turbine runner mechanical behavior is evaluated at nominal operating conditions. In this stage, the influence of the mechanical gate on the runner shaft thrust is evaluated at an instant when the turbine is fully operational. Secondly, the runner shaft thrust is evaluated at the moment that the hydrogenerator unit is approaching an operational stop. At the machine pre-stop condition, the runner shaft rotates freely, since the shaft is unloaded (that is, the braking system is disconnected), and the vanes are almost closed.

The first stage measurement of the hydraulic thrust on the turbine runner, caused by flow restrictions on the tailrace pipe, has the following steps: 1. Set the turbine vane opening at 70 % (maximum efficiency); 2. Select the mechanical gate restriction at 0 % (100 % of tailrace pipe cross section open); 3. Start the pumping system up to render the intake water pressure of 11 m. The system is kept working for 10 minutes before the measurements are taken; 4. Use the braking system to slow down the runner shaft until reaching a desired speed within the range of maximum efficiency; 5.

Read the value of axial load on the load cell display. This procedure is repeated for four levels of restriction generated by the mechanical gate: 18.75 %; 37.5 %; 56.25 % and 75 %. The mechanical gate movement is performed from 100 % of gate opening until 25 % of gate opening holding the pumping system on. To illustrate the operation of the mechanical gate installed at the turbine tailrace pipe, Figure 9 shows a picture of the mechanical gate generating a level of 43.75 % of flow restriction at the draft tube exit.

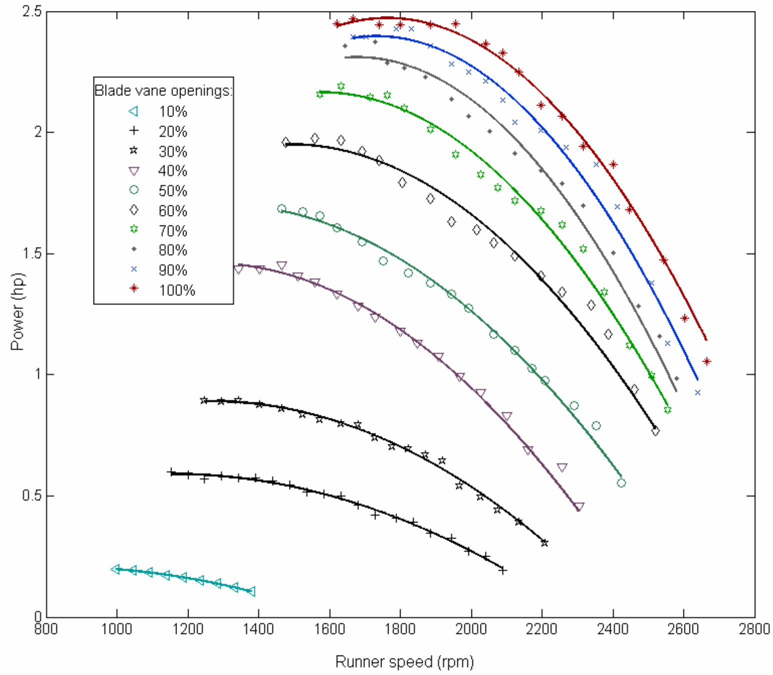

Figure 7. Curves of power versus speed for the turbine model at different blade vane openings (given in percentage).

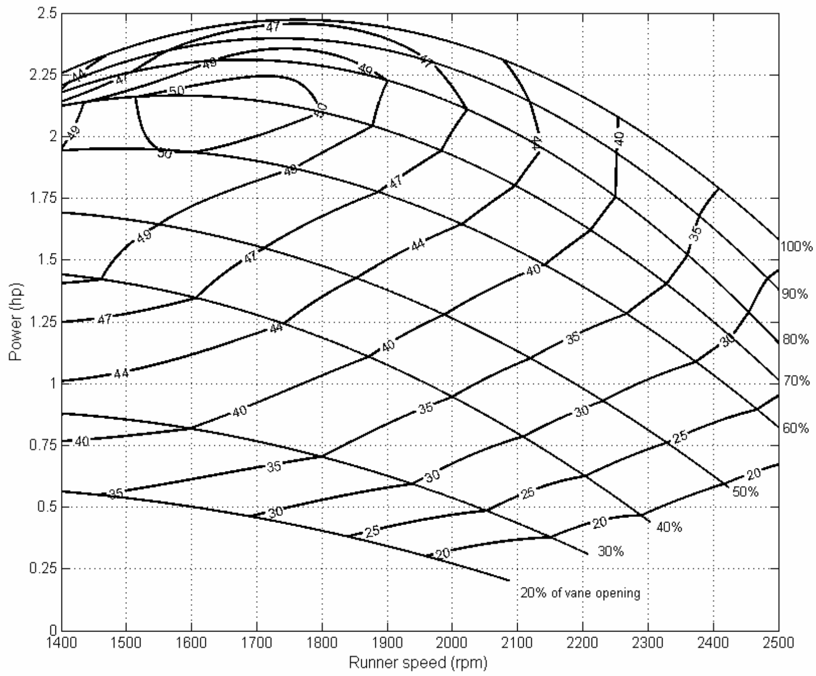

Figure 8. Efficiency contour plots for the turbine reduced model.

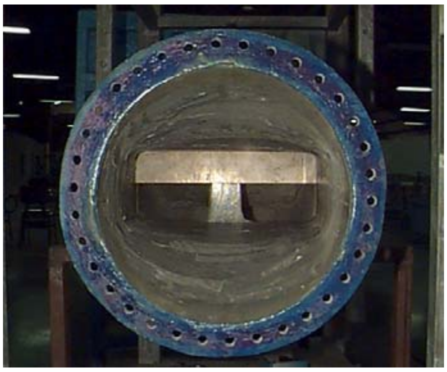

Figure 9. Picture of the mechanical gate offering 43.75 % of restriction for the water flowing out from the tailrace pipe.

The second stage measurement of the hydraulic thrust on the runner shaft is based on the following steps: 1. Restrict the turbine water intake by reducing the vane opening to 20% or 30%; 2. Set the mechanical gate position to a desired level of flow restriction at the tailrace pipe. Four levels of gate opening are employed in this second stage of the experimental procedure: 20%, 40%, 60% and 80%; 3. Start the pumping system up to render the turbine intake headwater at 11 m. Let the system works for 10 minutes before starting the measurements; 4. Measure the runner rotational speed and read the axial force on the transducer display. The mechanical gate is moved with the pumping system on.

5. EXPERIMENTAL RESULTS

In this item, the experimental results obtained through the several tests performed to measure the runner hydraulic thrust are presented. Primarily, the force transducer calibration is performed to assess the reliability of the force measurements. Then, the runner hydraulic thrust measurements are organized on curves that show clearly the strong influence of the draft tube flow restrictions on the mechanical behavior of the turbine model. In order to facilitate the presentation of these experimental results, this item is subdivided into three sections: i. Force transducer calibration; ii. Hydraulic thrust measurements at nominal operating conditions; iii. Hydraulic thrust measurements at pre-stop conditions.

5.1. Force transducer calibration

Standardized weights are used to perform the static calibration of the axial force transducer. Figure 10 depicts the calibration curve for the axial cell load used in the hydraulic thrust measurements. Three voltage readings on the transducer display, which are obtained in three different static calibration tests, are averaged to render the curve shown on Fig. 10. It is evident that the force transducer presents a linear behavior within the measurement range. The maximum difference among these readings does not reach 5% in the calibration tests.

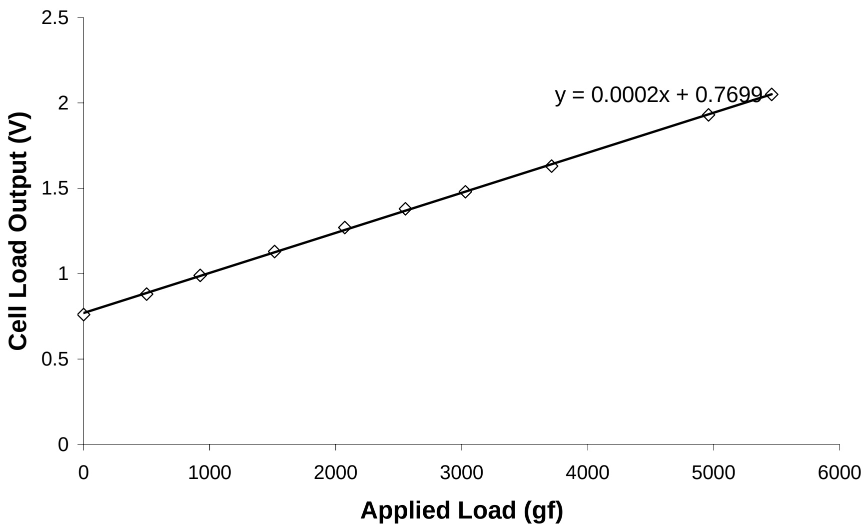

Figure 10. Calibration curve for the axial force transducer.

5.2. Hydraulic thrust measurements at nominal operating conditions

The first part of the axial force measurement tests are performed on nominal operating conditions at four runner rotating speeds: 1,600 rpm; 1,700 rpm; 1,800 rpm; 1,900 rpm. The turbine vane opening is held at 70%, which represents a condition of maximum turbine efficiency as shown on Fig. 8.

At each selected rotating speed, the axial hydraulic force acting on the runner thrust bearing is measured at five levels of flow restriction, which is caused by the mechanical gate installed at the draft tube exit (see Fig. 2). The measurement process is repeated five times at each speed and the averaged values of axial force are computed. Figure 11 depicts the four curves of axial hydraulic force versus the gate closing level (in percentage) at four rotating speeds. The curves present the same increasing trend with the gate closing. From the experimental results obtained in these tests, it can be noticed that he rotating speed does not affect significantly the hydraulic thrust of the turbine running on nominal operating conditions. On the other hand, the variation of the hydraulic axial force is almost 30% from the condition of free flow (without flow restriction) to the condition of 75 % of gate closing level. These experimental results show that the installation of mechanical gates at the turbine tailrace pipe can jeopardize the satisfactory operation of the turbine thrust bearing. The increase on the turbine runner hydraulic thrust due to flow restrictions at normal operating conditions can indicate that the use of mechanical gates as fish barriers must be carefully evaluated to avoid unexpected mechanical problems on the hydro-generator unit.

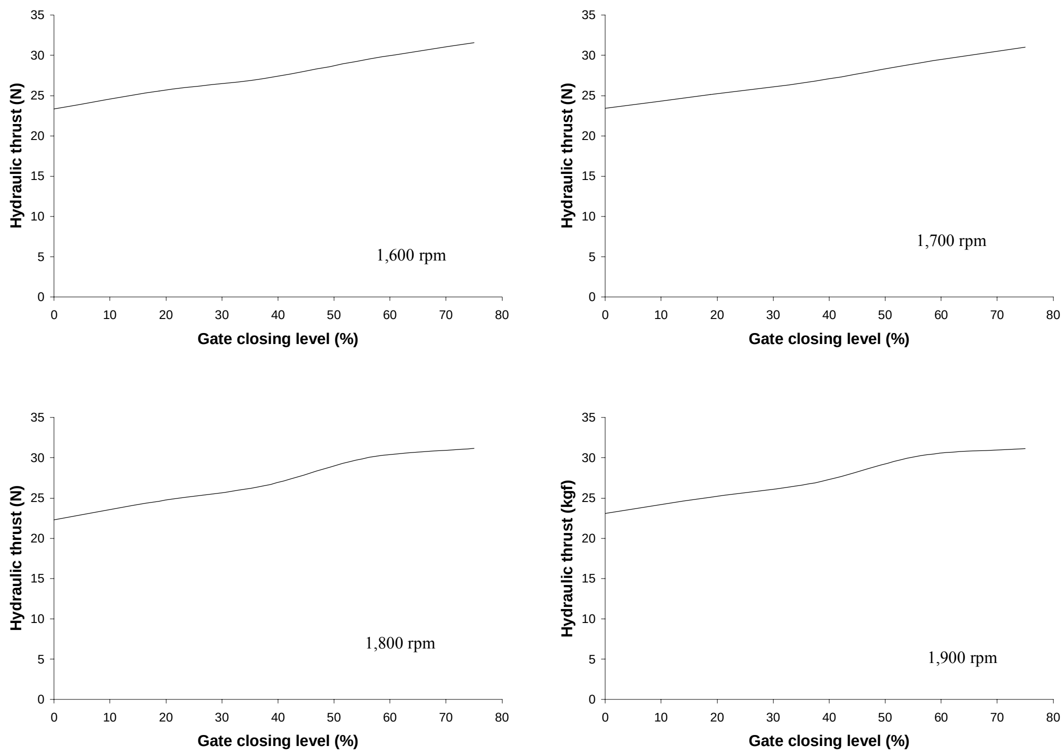

Figure 11. Experimental curves of hydraulic thrust in function of the flow restriction associated with the mechanical gate closing level.

5.3. Hydraulic thrust measurements at pre-stop conditions

The second part of the measurement tests simulate an operating condition at which the turbine runner is disconnected from the generator and the runner shaft rotates freely without any load. Then, the mechanical gate is displaced vertically in order to close the tailrace pipe flow section. Four measures of axial force are taken at five levels of gate closing (0%; 20%; 40%; 60%; 80%). Two turbine vane openings are employed in this part of the investigation: 20% and 30%. Figure 12 depicts the experimental curves of hydraulic axial force in function of the gate closing level at two values of vane opening. The average runner rotating speed is about 3,220 rpm during the tests with 30% of vane opening and 2,950 rpm with 20 % of vane opening. The experimental results show that the axial force generated by the flow restriction on the turbine thrust bearing increases as the mechanical gate closing level increases. Moreover, as expected, the smaller is the blade vane opening the smaller is the hydraulic thrust. The hydraulic thrust variation along these tests reaches 14.0 % and 10.3 % at vane openings of 20% and 30%, respectively, as the mechanical gate closing level varies from 0% until 80%. The variation of the axial hydraulic forces acting on the turbine shaft thrust bearing is smaller in the tests at pre-stop conditions than in those at nominal conditions. The experimental results indicate that the design, installation and operation of mechanical gates as fish barriers at the turbine draft tubes must undergo a meticulous scrutiny in order to avoid mechanical failures on the hydrogenerator caused by an increase on the hydraulic flow restrictions.

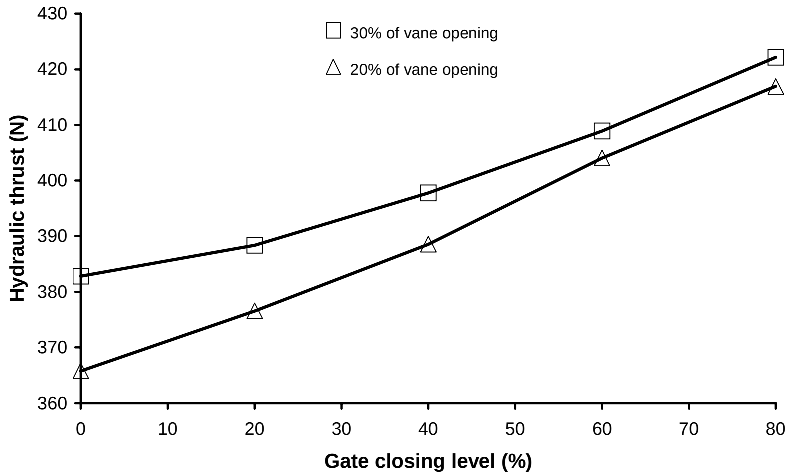

Figure 12. Experimental curves of hydraulic thrust at pre-stop conditions for two turbine vane openings.

6. CONCLUSIONS

The experimental results obtained in this investigation bring some insights into the influence of the draft tube flow restrictions on the mechanical behavior of Francis turbine. The use of mechanical gates as fish barriers at the draft tube exit must be carefully evaluated to avoid jeopardizing the hydrogenerator unit mechanical integrity.

7. ACKNOWLEDGEMENTS

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