CHELSEA POWER STATION PERFORMANCE TESTS

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Summary

Chelsea power station is equipped with five vertical axis Francis runners. These runners operate under a net head of 29.26 meters and each develops 30.3 MW. The tests performed on the fully homologous model, included both the new runner and a reproduction of the existing runner. This permits the prediction of the relative gain in efficiency and power of the newly designed runner to a precision of ±0.25%. Using the method of current-meter measurement, two efficiency tests are needed to obtain the performance of both the old and new runners with an accuracy of ±2.10%, evaluated in accordance with ISO 3354 standards. The field tests clearly demonstrated that the model tests accurately predicted the difference in efficiency between the old and new runners. The absolute values of the peak efficiencies were however, significantly lower than the stepped-up model values for both the old and new runners (3.4% for the new runner, 2.8% for old runner). Model tests remain a solid base of comparison for large scale hydroelectric projects. This is especially true when it is possible to remodel the old existing runners for rehabilitation projects.

Résumé

La centrale Chelsea est équipée de cinq roues Francis à axe vertical. Ces roues fonctionnent sous une chute de 29.26 mètres et développent chacune 30.3 MW. Des essais sur modèle réduit entièrement homologue, incluant la reproduction d'un modèle de la roue existante en centrale, ont permis d'anticiper avec précision de ±0.25% les gains relatifs de rendement et de puissance pour la nouvelle conception de roue. Deux essais de mesure de rendement in situ, à l'aide de la méthode de jaugeage aux moulinets, ont permis de mesurer les performances de la vieille et de la nouvelle roue à une précision de ±2.10%, évalué à l'aide de la norme ISO 3354. Les essais prototypes ont mesuré un rendement sommet inférieur aux anticipations du modèle de près de 3.4% pour la nouvelle roue et de 2.8% pour l'ancienne. Toutefois, la différence de rendement anticipé par le modèle entre la nouvelle conception et l'ancienne roue s'est retrouvée intégralement avec les essais prototype. Les essais sur modèle demeurent une base solide de comparaison de rendement avec précision pour les projets hydroélectriques de grandes envergures. Cela est d'autant plus valable lorsqu'il est possible de modéliser l'ancienne roue prototype pour les projets de réfection.

1.0 Introduction

In 1988, GE Hydro, at the request of Hydro-Quebec, was asked to develop a new improved runner to replace the existing one at the Chelsea power station in the Gatineau region. In order to obtain a precise comparison of the relative gains in efficiency and power between the new and the old runners, GE Hydro proceeded to fabricate a reduced scale model of the existing runner as well.

The conditions for measuring the efficiency at site are not very favorable (a low head Francis turbine in a semi-spiral casing). Because of this, the current-meter method was used.

Presented in this article are the results obtained from the model tests, the process of manufacturing these prototypes and the manner in which the efficiency was measured on the prototype.

A detailed analysis of the test procedures, as well as the effects of the roughness of the surfaces on the efficiency, lead certain conclusions to be drawn about the accuracy of the results.

2.0 Description of the power station

Chelsea hydroelectric power station, situated on the Gatineau river in the Gatineau region, is the property of Hydro-Quebec. Each of the five vertical axis Francis units develop a turbine power of 30.3 MW under a rated net head of 29.26 meters.

The hydraulic design of this power station dates back to the 1920's. The casing is of the semi-spiral type with two piers that divide the oncoming flow into three equal passages. One of these piers ends close to the stay ring which is composed of ten stay vanes which are followed by twenty wicket gates. The old runner had seventeen blades whereas the new runner has thirteen. The draft tube is of the Moody type with a central cone that begins just under the runner cap. The hydraulic profile of the machine is well illustrated in Fig. 1.

3.0 Efficiency tests on the reduced model

3.1 Homology details

The tests done on the fully homologous model allowed the quantitative evaluation of the anticipated performances of the newly designed replacement runner for the Chelsea power station.

The model accurately reproduces the penstock up to the upstream gates. The scale of the model to the prototype is 1:13. The upstream gate side slots were not modeled. The upstream part of the penstock was simplified with the use of an elbow on the model as shown in Fig. 2.

An opening, 1.7m by 1.2m, along the left pier of the casing at its base was not modeled either (illustrated in Fig. 1). In this opening, a gate allows the drainage of the casing during the dewatering of the machine.

The pressure balancing holes in the runner crown are not reproduced on the model for either of the new or old runners. The profile of the top of the crown is not reproduced either. The top of the crown on the prototype runners is downward sloping from the periphery to the center. On the model, the top of the crown is horizontal.

The profile of the draft tube is accurately reproduced, up to the tailgate entrance. The tailgate side slots are not reproduced on the model. Fig. 3 shows the detail of the tailgate slots.

3.2 A model runner duplicating the existing runner of the power station

The geometry of the existing runner at the Chelsea power station was available in the form of a coordinate file on computer. This greatly facilitates the manufacturing of the model runner. The throat diameter of the model runner was set at 300 mm.

In order to make sure that the model runner accurately reproduced the prototype, several measurements of the existing runner in the station were taken and used for the manufacturing of the model. For example, the nose profiles of the blades were measured using templates. Furthermore, the opening between the blades at the outflow of the runner, the thickness of the blades at the inflow and outflow and the distance of the inflow edge at different heights with respect to the diameter of the band seal was also measured on the prototype.

The seal diameters at the band and crown are homologous. The length of these seals is slightly larger on the model than on the prototype.

Fig. 4 illustrates the difference between the blades of the two manufactured runners; the newly designed and existing runners.

3.3 Test conditions

All tests were performed in accordance with the IEC 193 test code for hydraulic turbine models. The accuracy of the measured efficiency was evaluated to be ±0.25%.

An upstream tank installed on test stand #2 of the GE Hydro hydraulic laboratory allowed the attachment of the entrance of the rectangular casing of the Chelsea reduced model. This tank ensures an even flow distribution at the entry of the model allowing the stations conditions to be reproduced.

The exit of the draft tube is attached to a downstream tank in which the tailwater level at the station is simulated. Fig. 5 shows the arrangement of the model when mounted on test stand #2.

During the efficiency and cavitation tests, the head was maintained at a level of 15.25 meters. The effect of the variation of the prototype tailwater level on the performance of the machine was analyzed by varying the pressure in the downstream tank.

The temperature of the water circulating in the test stand was maintained at 20 °C with the aid of a cooling unit. The water temperature under normal operating conditions at the power station is observed to be 15 °C.

3.4 Anticipated performances of the prototype from the model tests

The test results on the model, as well as the step-up formulas from the IEC 995 test code were used to calculate the anticipated performances of the prototype. The step-up in efficiency was estimated to be 1.55% for the new runner and 1.60% for the old runner from a model test data corrected to a Reynolds number equal to 7×10^6.

Since the construction of the power station dates back to the 1920's, the surface finish of the passages is not hydraulically smooth. This roughness has a negative effect on the performance step-up.

To this date, no code exists to estimate the losses in performance caused by these rough surface finishes. Therefore, a simplified method was used to predict the headloss, and thus the efficiency loss caused by the rough inner surfaces (see ref. [1]). This method involves geometrically reducing the passages to simple pipes. Then, the Moody diagram is used to calculate the losses. The roughness of the hydraulic surfaces of the Chelsea turbines was of the order of 10mm in the casing, 3mm in the stay ring, 2mm on the wicket gates and 5mm in the draft tube. This method allows the head loss to be evaluated at 0.9% of the net head, and consequently, the efficiency loss is 0.9% at the point having a flowrate equal to 99.11 m^3/s. This loss varies as a function of the square of velocity or flowrate of water through the turbine. At full power, the efficiency loss is calculated to be 1.4%. These corrections lead to a net step-up of about 0.75% at peak efficiency and 0.4% at full power. Fig. 6 illustrates, as presented in the model test report (ref.[2]), the efficiency measured on the two modeled runners, the anticipated performance as per the IEC 995 code and the anticipated performance after having accounted for the surface finish.

4.0 Manufacturing of the new prototype runner

The first prototype runner, delivered for the contract, was a welded fabrication with cast blades close to finished dimensions and ground by hand using five templates. These templates ensured that the hydraulic profile would be identical to that of the model. The blades of the four other prototype runners were numerically machined. The reference geometry, used for casting and fabrication of the templates for the blades of the prototype, was taken from the coordinate information reference file used for the fabrication of the model runner.

Upon verification of the dimensions of the prototype, it was seen that the outflow opening between the blades was slightly larger than those measured on the model, but were still within the allowed tolerance of ±2%.

All of the remaining dimensional verifications done on the prototype demonstrated an excellent homology between the prototype and the model runner. The outflow thickness of the blades as well as the thickness at other locations on the blade were all verified. Several templates were used to position the finished blades, as well as to verify their profiles.

5.0 On site efficiency tests

Efficiency tests using current-meters were performed on two different units; Unit #2 with the new runner in 1992 and Unit #3 with the old runner in 1993. The Service d'Essais et Expertises Techniques (SEET) of Hydro-Quebec were responsible for the test done at the power station. Therefore, their reports are used as reference (see ref. [3&4]).

In Unit #2 only the wicket gates were cleaned and repainted. Fig. 7 clearly shows the distributors of both units before testing.

5.1 Description of test installation

Both the upstream and downstream levels were measured using pressure transducers. Transducers were also installed downstream of the penstock in order to measure the headloss in the penstock. The transducers were installed in the same measurement plane as that used to measure the net head when testing the model.

The current-meters were installed in the penstock on a structure, perpendicular to the flow, which was able to move vertically on cables. In each of the three sections in the casing, there were 33 uniformly distributed current-meters. Smaller current-meters were used on the inner surface and larger ones in the center. Fig. 8 shows the location of the pressure pickups as well as the structures supporting the current-meters.

The power was measured at the generator and the mechanical and electrical losses were attributed to this. The power at the generator was measured using the three wattmeter method.

5.2 Test procedures

The determination of the efficiency of the turbine was done, as much as possible, in accordance with the IEC 41 and international ISO-3354 test standards. The accuracy of the two tests was found by SEET to be about ±0.65% in terms of efficiency measurement. The relative uncertainty between the tests of the two units was evaluated to be ±0.2%.

Nearly twenty operating conditions were measured starting from 50% of the servomotor, right up to full power. The test measurements were taken over the course of two days. The first day, the three current-meter structures were in their low position. The second day, the structures were in their high positions (rose about 0.6m). This procedure had the advantage of doubling the number of current-meters to measure the flowrate. Fig. 9 illustrates the two positions of the current-meter structures.

A few minutes between all of these operating conditions were enough to allow the flow to reach a steady-state. The measurements were taken simultaneously for a period of 180 seconds.

5.3 Test results

During these tests, the head remained in the allowed range of ±3% of the nominal head. This permitted a precise transposition of the results of the flow and of the power atthe nominal head of 29.26m.

Fig. 10 shows the performances measured for both units of the power station. Also shown are the anticipated performances of the model.

6.0 Analysis

In summary, using the tests done on the model, a step-up in efficiency of the order of 0.75% was calculated at operating conditions giving the best efficiency. It was in the order of 0.40% when operating at maximum power. The surface roughness of the prototype and the step-up formulas of the IEC 995 code were accounted for in the calculations.

Apart from this, the prototype tests measured values well below those that were anticipated. Essentially, peak efficiency was of the order of 3.4% less than the anticipated value for the two runners. In the case of efficiency at maximum power, the step-up was almost achieved with the new runner; 0.4%

less than anticipated. With the old runner, the step-up was measured to be 1.4% less than what was anticipated for maximum power.

The measured efficiency difference between the old and new prototype runners, at the peak efficiency point, was exactly as anticipated from the model tests, 1.7%. The measured efficiency difference at full load was 1% higher than anticipated from the model tests.

6.1 On site test results for the old runner

The inferior performance of the old runner can be explained by the fact that the roughness of its surface had not been considered when stepping-up the performances from the model. The surface roughness of the old runner is in the order of 2mm. Using the same method of calculation as in section 3.4, it is possible to anticipate a deterioration in peak efficiency of the order of 0.6%, and 1.2% when at maximum power

This calculation reduces the difference between the anticipated efficiency of the model and that measured with the old prototype runner. The difference becomes 2.8% at the peak and 0.2% at maximum power.

6.2 Fabrication of the new runner

It is also noted that the openings between blades on the new runner were fabricated slightly larger than those of the model, but still within specified tolerances. This causes a slight increase in the measured power of the prototype when compared with the power obtained if the openings of the prototype would be identical to the stepped-up model. It is in fact noticed that the peak shifts slightly to the right for the new runner, which is not the case for the old one. However, this does not explain the margin of difference between the prototype and the model because, from experience, it is known that this does not affect the peak efficiency (ref. [5]).

6.3 Tests on the reduced model

The efficiency tests conducted by the laboratory were done using a tailwater level that was relatively high (producing a high cavitation coefficient). This is quite normal during model testing and is also acceptable by the test code. GE Hydro's test stand #2 has been compared to other independent laboratories on several occasions and the efficiency levels have always been confirmed (ref. [6]).

Cavitation tests permit verification of the influence of the tailwater level on the performance of the machines. These tests have shown that when operating at the tailwater level of the power station, the performance of the two runners remain unchanged with respect to the measurements taken during the performance tests. Fig. 11 shows the effect of the variation of the tailwater level on the performance of the two runners.

6.4 Evaluation of the error on discharge during field test

Throughout this sequence of testing, several measurements have been taken and the errors in these measurements have also been determined. Among all of these errors, the one incurred when measuring the discharge is the most notable. The ISO 3354 standard is used in the determination of the error when estimating the discharge.

6.4.1 Errors in the evaluation of local velocity

The local velocity is evaluated with the aid of current-meters that are installed on a movable structure. This measurement is not without uncertainty and bears errors coming from several sources.

The measurement of the rotational speed of the current-meter contains and error. The error definitely decreases as the duration of the measurement increases. In the case of Chelsea, a measurement period of 180 seconds was used. A comparison of the measurements taken for the same operating conditions allowed the error to be ±0.5%.

The calibration of these current-meters can also lead to an uncertainty in the measurement of the velocity. SEET used 210 current-meters, which is in agreement with the IEC 41 code. Essentially, this code requires 81 current-meters under favorable conditions and 228 current-meters under less favorable conditions such as those at Chelsea. The current-meters are calibrated at independent laboratories. Calibration defines a curve that allows the calculation of the velocity to a precision of within ±0.25%.

The response time of a current-meter to the variation in velocity caused by turbulent flow can also create an error in the measurement. However, this error is quite difficult to predict precisely. Therefore, the value of ±0.5%, proposed by the code, can be taken as a reference value.

It should also be noted that the blockage due to the structure holding the current-meters can have an influence on the flowrate measurement. This is well covered in the ISO 3354 standard for measuring clean water flow in closed circuits. In the case of Chelsea, the structure had a surface equal to 4.4% of the surface measurement and this lead to an additional error of ±0.37%.

The addition of all of these errors gives the total error in local velocity measurement, which was found to be ±0.84%. This is found by taking the square root of the sum of the squares of the individual errors.

6.4.2 Errors in estimation of the flow

The previously calculated error in local velocity of ±0.84% will be used to evaluate the error in the flowrate.

To calculate the flow using the ISO 3354 standard, a roughness factor "m" which defines the velocity profile between the inner surface and the current-meter closest to the inner surface must be calculated. The velocity distributions were not uniform in the case of Chelsea. In this case, factor generally varies between 10 and 100 whereas normally, the roughness factor "m" varies between 4 and 10 for uniform distributions. Several velocity calculations were made using a roughness factor varying between 10 and 100 and the variation of the flowrate was ±0.4%.

The error in measuring the surface of the plane where the current-meters are installed directly influences the error in the flow measurement. In the case of Chelsea, the surface was divided into three conduits of 16.353 m^2 (4.877m X 3.353m). An error of ±10mm in measurement of height and length on the conduit contribute to a combined error of ±0.5% in the area. Seeing as the surface roughness of the casing was found to be ±10mm, the value found here is not unrealistic. Therefore, it is always necessary to keep this measurement in mind when evaluating the system error for the flow. An error of ±0.25 % is proposed to be used in the case of Chelsea.

For the interpolation of the velocities, SEET used two different equations; one for defective current-meters inside the structure and the other for defective current-meters along the inner surface. These equations use the velocity measured around the defective current-meter to make an estimate.

During the tests, there were six current-meters inside the structure that did not function. The error can be estimated by interpolating for any six current-meters inside the structure using the appropriate interpolation formula. The error for this type of interpolation is established to be ±0.1%.

It was also necessary to interpolate twelve current-meters on the inside surface that were missing and three that were defective. Therefore, interpolating for these fifteen current-meters using the appropriate interpolation formula gives an error of ±0.3%.

Hence, the total error for the interpolation of the velocities can be evaluated to be the sum of the two previous errors, ±0.4%.

Finally, the total error in flow measurement for the tests done at the Chelsea power station can be estimated as follows:

error in local velocity measurement	±0.84%
error due to roughness factor measurement	±0.40%
error in conduit area measurement	±0.25%
interpolation error	±0.40%

The standard deviation on the measurement of the flowrate can be found by taking the square root of the sum of the squares of the individual errors. It is found to be ±1.04%. According to the ISO 3354 standard, the error in flow measurement can now be determined with a level of confidence of 95%, by doubling the calculated standard deviation (±2.1%).

Since the error in head and power were evaluated to be ±0.10% and ±0.20% respectively, the expected total error in measurement of efficiency on site was of the order of ±2.1%.

7.0 Conclusion

The tests, both in the laboratory or at site, were done in accordance with the standards that are presently in use. Moreover, the fabrication of the new prototype runner was done respecting all specified tolerances. Finally, the surface finish of the prototype was considered in the calculation of the expected performance of the prototype.

The model tests were used to quantify, within an accuracy of ±0.25%, the anticipated gain in efficiency and power of the prototype with respect to the existing machine.

The on site efficiency tests cannot anticipate a better accuracy than ±2.1% as evaluated with the ISO 3354 standards. Essentially, several factors, such as the large dimensions of the machine, the non-uniform velocity flows and the large number of instruments, make it very difficult to control the flow measurement.

Despite all of these precautions, it is still not possible to explain the absolute loss of 3.4% (2.8% for the old runner) between the anticipated efficiencies found from the model tests and those measured on the prototype at peak conditions.

Therefore, model tests remain an extremely precise comparison base for large scale hydroelectric projects.

Guarantees imposed on the absolute efficiency of the prototype must be made with precaution, as well as with a good understanding of the difficulty controlling step-up phenomena between model and prototype, and on site tests.

8.0 References

[1] Grenier, Roger, **Methods of predicting the effect of rough surfaces on the efficiency of hydraulic turbines,** I.A.H.R., Symposium 1994, Beijing, China, 1994.

[2] Desbiens, Eric, **Rapport final d'essais sur modèle réduit pour la Centrale de Chelsea,** Rapport d'essais GE Hydro, 1989.

[3] Caron, Nicolas, **Essai de rendement turbine par Jaugeage aux Moulinets du groupe No. 3 à la Centrale de Chelsea avant remplacement de la roue,** Rapport d'essais SEET, Hydro-Québec, 1993.

[4] Caron, Nicolas, **Essai de rendement turbine par Jaugeage aux Moulinets du groupe No. 2 à la Centrale de Chelsea avant remplacement de la roue,** Rapport d'essais SEET, Hydro-Québec, 1993.

[5] Ramchandani, Haresh, **Influence of Manufacturing Tolerances on Hydraulic Turbine Performance,**, Sixth International Symposium on Hydro Power Fluid Machinery, ASME, Anaheim, California, USA, 1992.

[6] Desbiens, Eric, **Model Testing: A Strategic Choice in Hydraulic Machines Development,** Waterpower '95, San Francisco, USA, 1995.

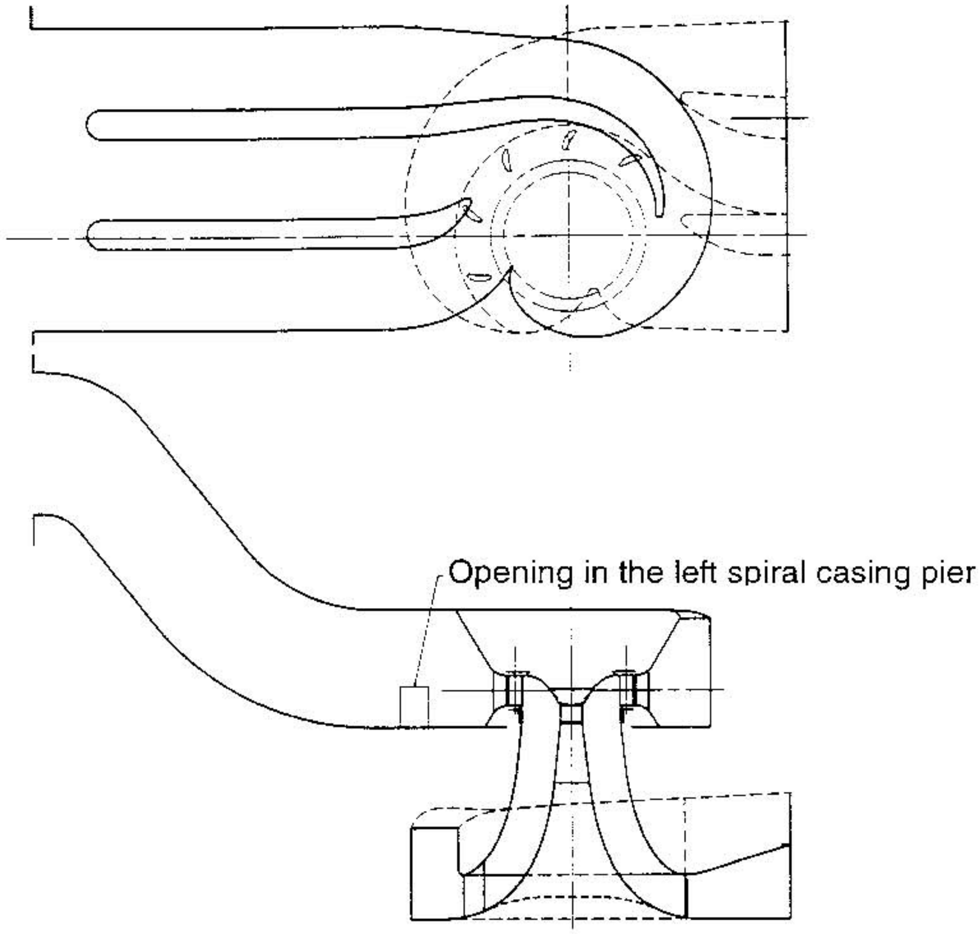

Fig. 1 : Hydraulic passageways of Chelsea power station

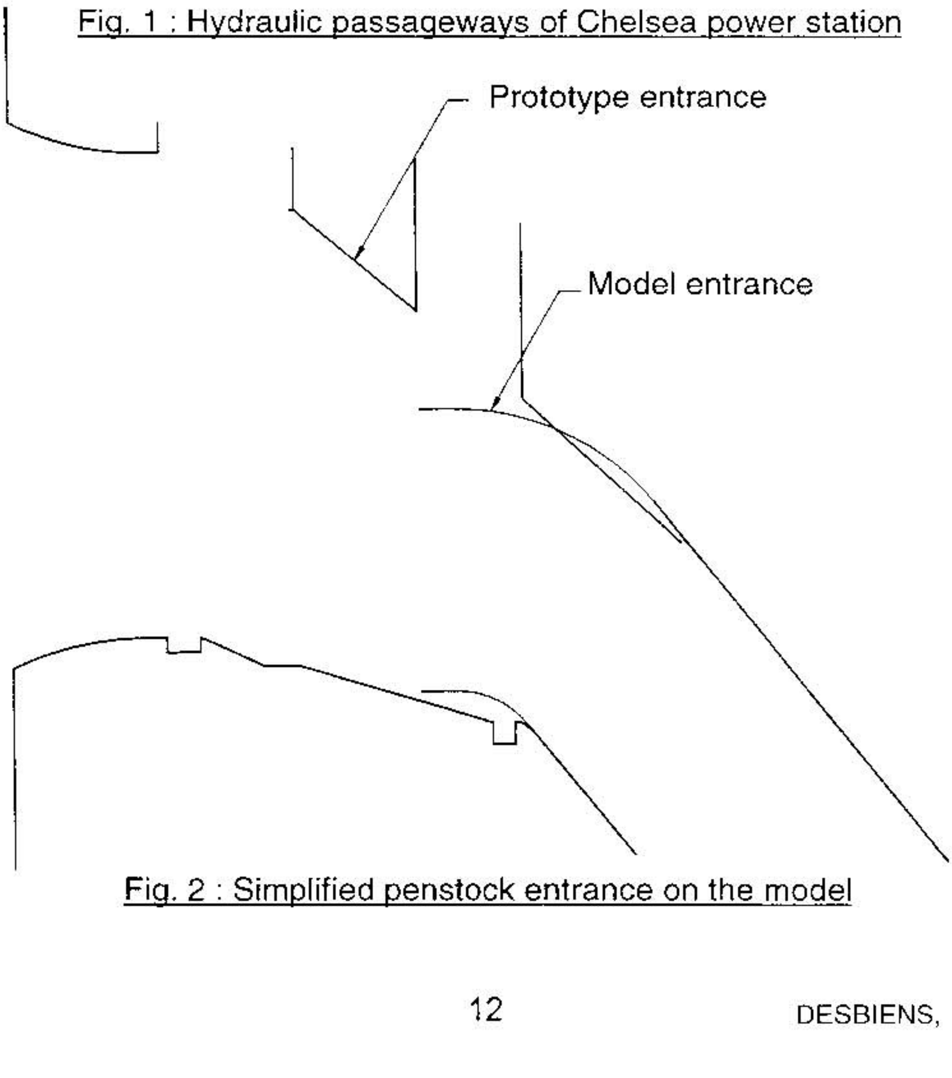

Fig. 2 : Simplified penstock entrance on the model

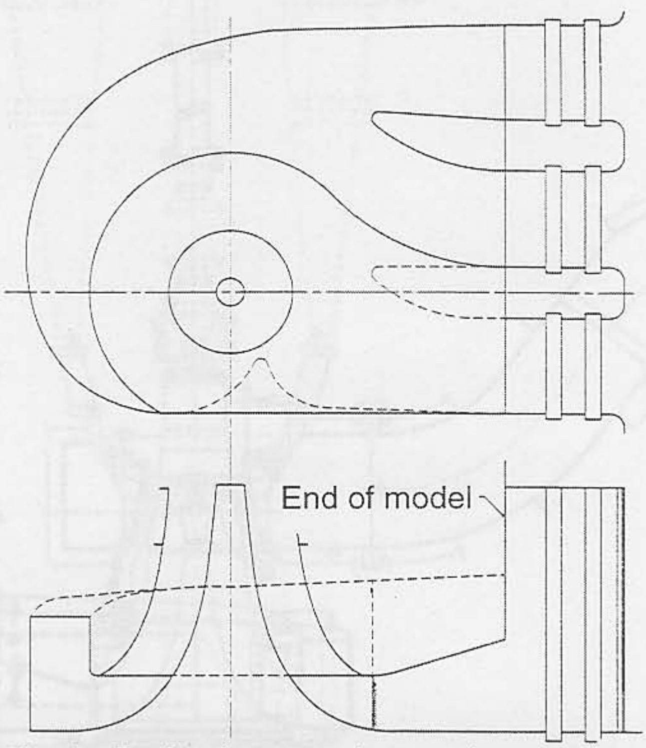

Fig. 3 : Draft tube gate slots on the prototype

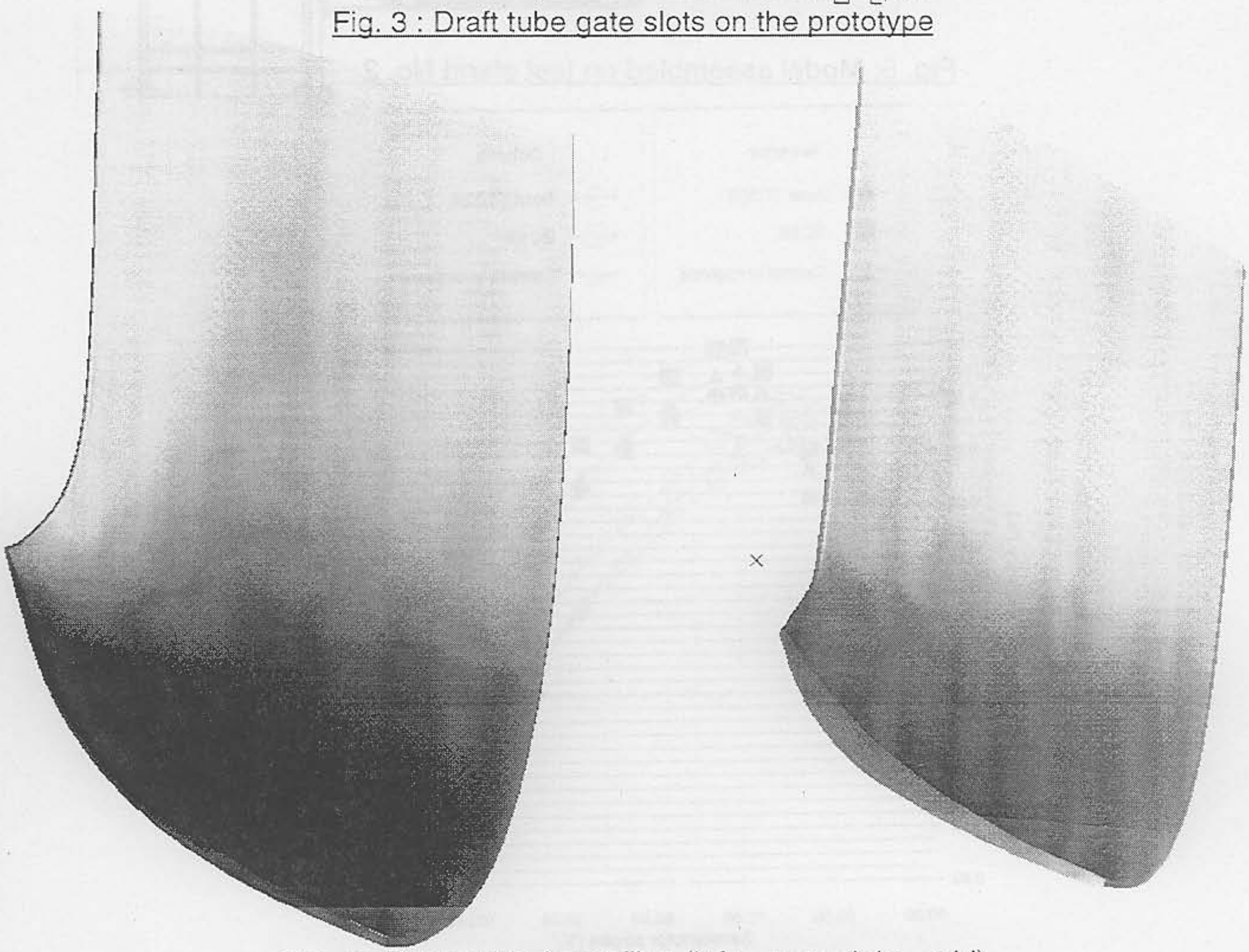

Fig. 4 : Runner blade profiles (left = new, right = old)

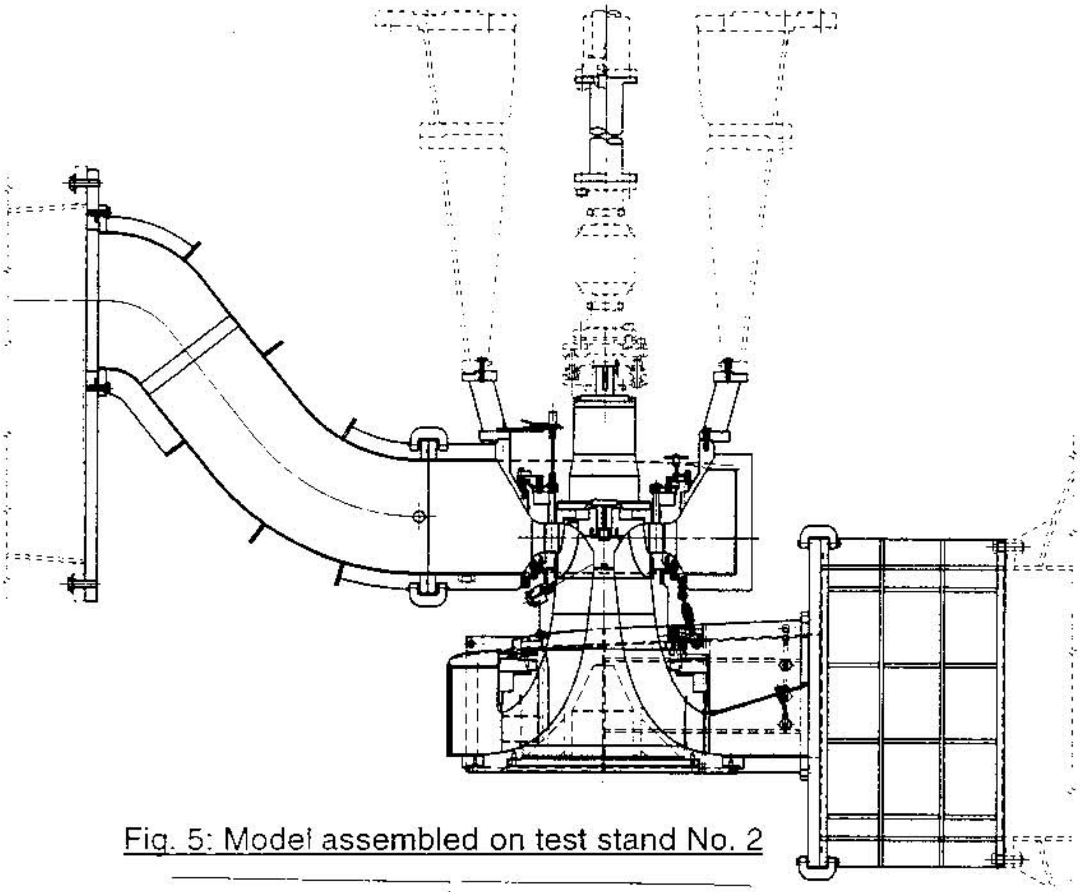

Fig. 5: Model assembled on test stand No. 2

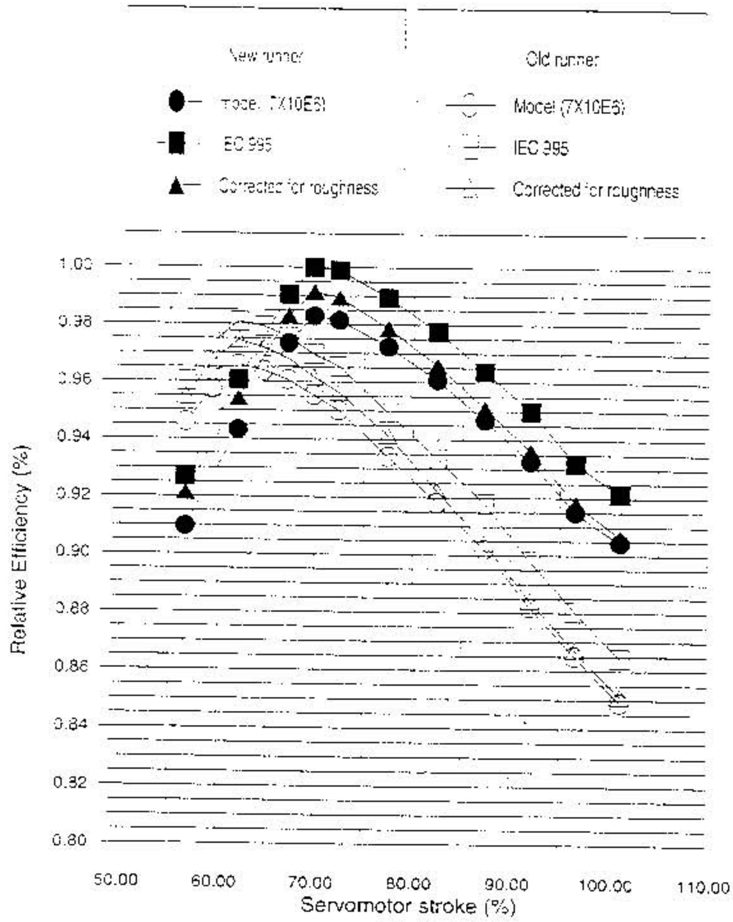

Fig. 6 : Expected efficiencies from the model

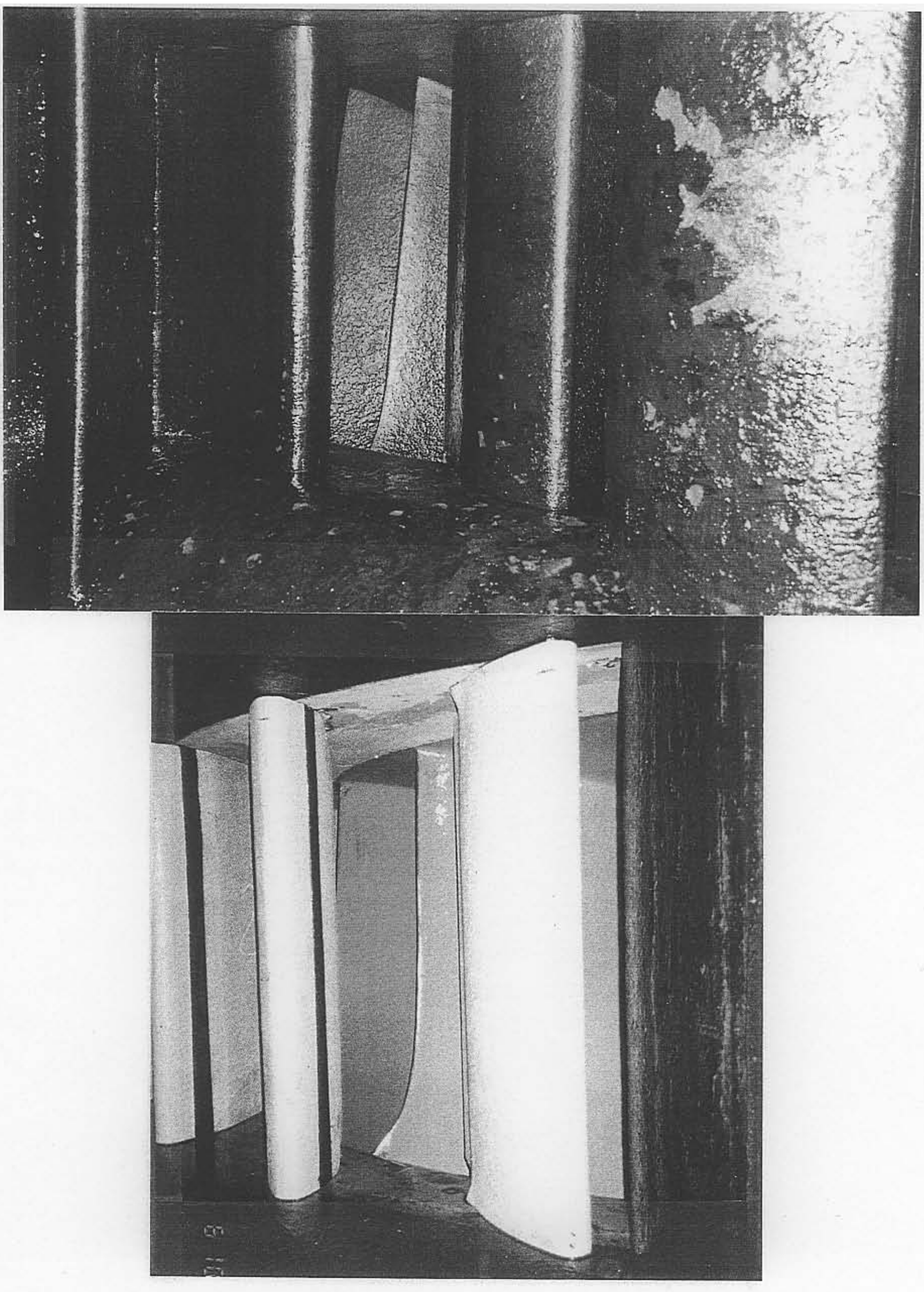

Fig. 7 : Distributors comparison (top = old, bottom = repeinted)

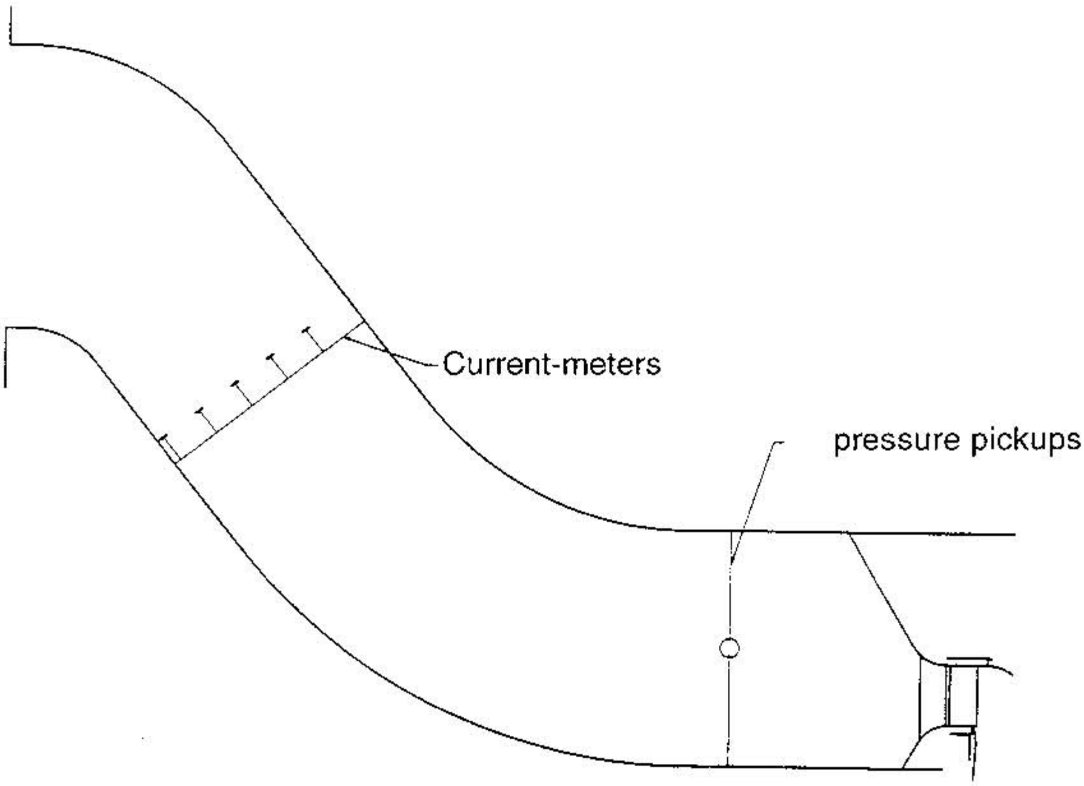

Fig. 8 : Location of current-meters and pressure pickups

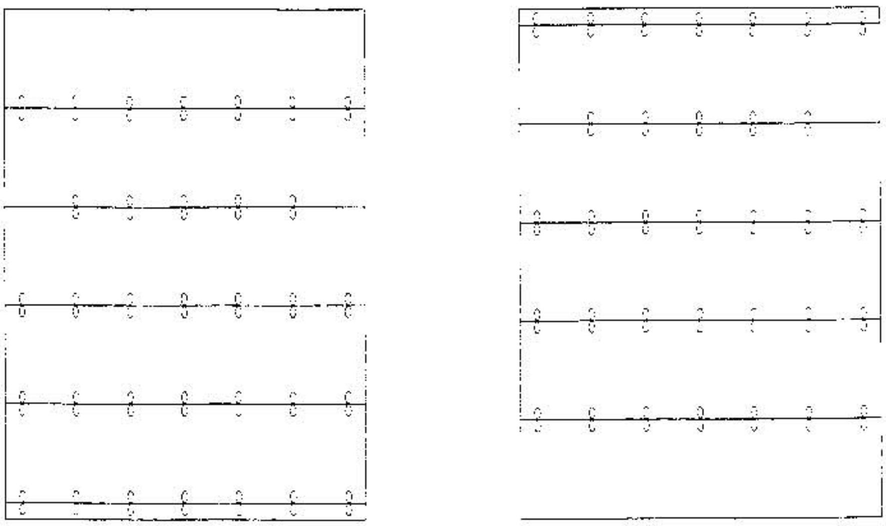

Fig. 9 : Bottom and top positions of current-meters structure

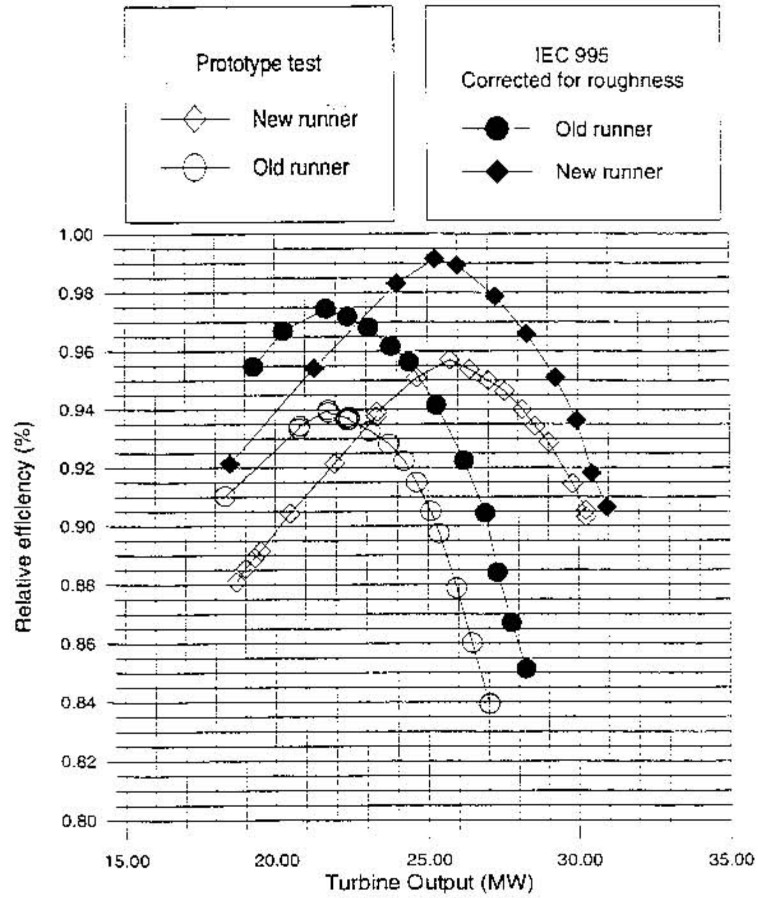

Fig. 10 : Measured efficiency on the prototype and expected values from the model

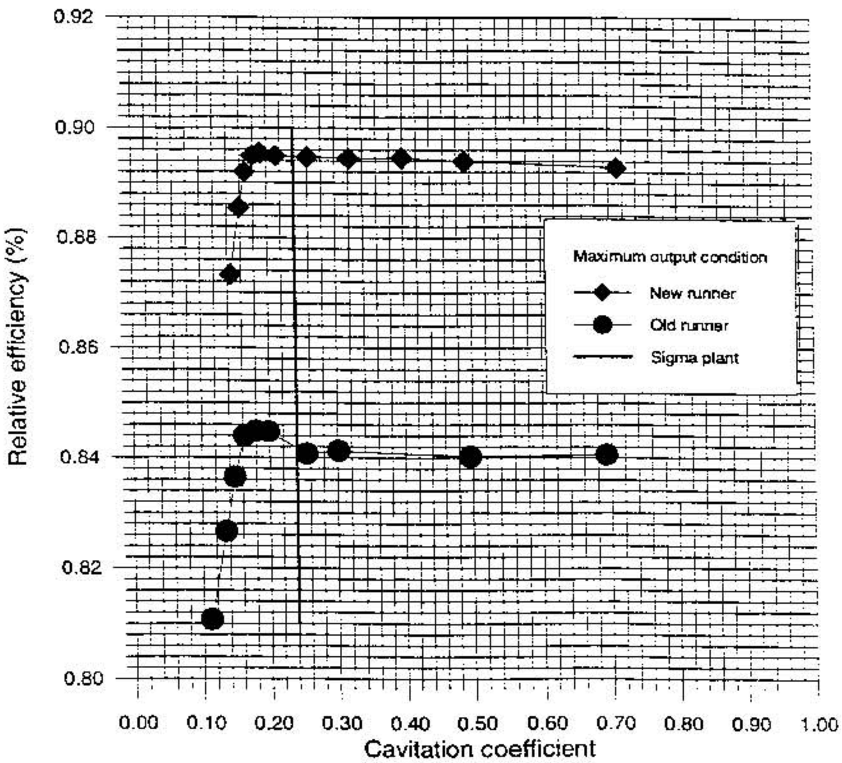

Fig. 11 : Cavitation test for new and old runner