Waterjets Group

Final Report and Recommendations to the 21st ITTC

1 GENERAL

1.1 Membership and Meetings

The Specialist Committee on Waterjets appointed by the 20th ITTC consisted of the following members:

Professor Dr. Gilbert Dyne
SSPA Maritime Consulting AB, Göteborg,

Professor Dr.-Ing. Claus Kruppa (Chairman)
Technische Universität Berlin,

Mr. Bertrand Lamberti
Bassin d'Essais des Carènes, Paris, and

Mr. Knut J. Minsaas
MARINTEK, Trondheim.

The committee did not appoint a Secretary.

The following committee meetings were held:

- 7 - 8 April 1994
at MARINTEK, Trondheim,
- 17 - 18 October 1994
at KaMeWa, Kristinehamn,
- 20 - 21 April 1995
at Bassin d'Essais des Carènes, Paris, and
- 11 - 12 December 1995
at SSPA Maritime Consulting AB, Göteborg.

1.2 Task Assignments

The 20th ITTC did not assign any specific tasks to the committee. From recommendations formulated for and adapted by the 20th ITTC one could deduce that the following two subject areas might have an element of overlap with possible tasks of the Specialist Committee on Waterjets.

For the Propulsor Committee of the 21st ITTC the task „to investigate high speed propulsors and interaction effects for high speed marine vehicles“ undoubtedly should have included waterjets. With the existence of a Specialist Committee on Waterjets, it was not disputed that the Propulsor Committee should drop waterjets from its list of high speed propulsors to be dealt with. The Chairmen of the Propulsor Committee and the Specialist Committee on Waterjets came to this conclusion early in 1994.

For the Powering Performance Committee of the 21st ITTC the task to „develop a method for predicting the propulsive performance of craft propelled by waterjets“ was also regarded as to overlap with the work of the committee. The subject was consequently dropped and transferred, with the consent of the 21st ITTC Executive Committee.

Apart from those indications of a general interest in waterjet propulsion, the committee deduced the need for its existence from the subjects of and the discussions in the Workshop on Waterjets held on 20 September during the 20th ITTC. The workshop was organised by Mr. Minsaas of MARINTEK and consisted in the presentation of 10 contributions and a number of discussions. In a prepared statement Mr. Minsaas drew a number of conclusions and made some recommendations. Unfortunately, these were not discussed at the conference and not included in Volume 2 of the Proceedings. In fact, Volume 2 of the 20th ITTC Proceedings only contains 2 of the 10 workshop contributions. As a result it was suggested to cover the workshop more comprehensively in a special document to be available at the 21st ITTC (see The Waterjets Session, 20th ITTC, 1993).

In view of the above events the committee decided at its first meeting to concentrate its efforts on the formulation of **a power prediction procedure** and the associated model testing techniques. In this context it was realised that the need for **a standard nomenclature** is an important one.

Further issues which could have been dealt with had emerged from the 20th ITTC Workshop on Waterjets. Examples of these are testing techniques for assessing **inlet ventilation susceptibility** and **validation of CFD based flow predictions**. However, these topics were regarded as being of lower priority.

The Report of the Specialist Committee on Waterjets will start with a brief **literature survey**, followed by a discussion of the proposal for a **power prediction method,** the main topic of this report. In this context **sensitivity aspects** are touched upon. The report will be concluded with **Recommendations to the Conference**.

2 LITERATURE SURVEY

Major sources of contributions to the subject of waterjet propulsion are the proceedings of FAST'93 in Yokohama and FAST'95 in Lübeck-Travemünde. The RINA also organised a special symposium on waterjets in London in December 1994. A very comprehensive survey paper on waterjet propulsion was presented by Allison (1993).

Where the papers published since the last ITTC relate to general definitions and concepts, to performance prediction and model testing techniques they will be briefly listed in the following paragraphs. If the material presented relates to the main topic of this report more detailed discussions will follow in the relevant sections.

Allison (1993) defines the momentum flux from the nozzle as „gross thrust“. If the ingested momentum flux is subtracted from the „gross thrust“ he obtains the „net thrust“. This is in obvious contrast to the definitions adopted in the Report of the High Speed Marine Vehicles Committee (18th ITTC, 1987). Matsumoto et al. (1993) discuss some interesting aspects of free jet characteristics under off-design conditions. They are stressing the necessity for proper matching of nozzle diameter and pump. As in other Japanese literature, specific speed and suction specific speed are based on a volume flow rate in m³/min, thus yielding $\sqrt{60}$ higher values than in most of the European literature. Coop and Bowen (1993) are dealing with interaction effects between hull and waterjet. So do van Terwisga (1993), Alexander (1994), Alexander et al. (1994) and van Terwisga and Alexander (1995). Dyne and Lindell (1994) discuss aspects of model self-propulsion experiments which are largely related to the main topic of this report. Component testing is addressed by Steen and Minsaas (1995). Masilge (1991) analyses the model test results obtained for a body of revolution with waterjet propulsion and a boundary layer intake.

More than one paper is concerned with flush inlet design and validation of CFD based numerical analysis of the inlet flow (e.g. Okamoto et al., 1993; Szantyr and Bugalski, 1995; Dai et al., 1995; Seil et al., 1995; Yang et al., 1995, Latorre and Kawamura, 1995). Pylkkänen (1994a, 1994b, 1994c) has dealt with two-dimensional CFD calculations of waterjet inlets and compared pressure and velocity distributions with experimental data. Svensson (1994) discusses experience with high-powered installations, and novel pump design features are presented by Kawakami et al. (1993). Allison and Goubault (1995) show the results of a design study with a systems approach, concentrating on weight versus efficiency aspects.

Very little is published with regard to full scale trial results that permits a thorough analysis. From what little information is available for detailed analysis it seems that the higher the design speed the lower the ratio of change of momentum flux (ΔM) over bare hull resistance (R_T). Since ΔM used in these analyses is different from the quantity defined in this report below, no general conclusions about „negative thrust deduction" at high speed should be drawn at this stage.

Iannone and Rocchi (1993) propose the use of model self-propulsion tests with progressively varying tow rope force to predict waterjet-hull interaction effects, without relying on resistance tests. However, this method requires that the thrust of the waterjet unit is measured separately. The subject is addressed further by Iannone (1994, 1995) as well as Di Ciò and Iannone (1994).

3 DISCUSSION OF POSSIBLE POWER PREDICTION METHODS FOR WATERJET PROPULSION SYSTEMS

3.1 Introduction

In June 1995 the 21st ITTC Specialist Committee on Waterjets sent out descriptions of two possible power prediction methods for waterjet systems, asking more than 30 institutes and companies interested in the subject for comments. In one of the methods thrust was computed from the change of momentum flux, in the other thrust was measured directly.

The description of the **momentum flux method** is, with minor alterations, reproduced in Appendix A followed by a **proposed list of symbols** as Appendix B. The alterations are mainly a result of a discussion about appropriate definitions and symbols with Prof. Schmiechen, in his capacity as member of the 21st ITTC Symbols and Terminology Group. The most important change is that the frictional wake fraction w_f is replaced by the energy velocity V_E, the two being related by

$$V_E = V(1 - w_f) .$$

Furthermore, the influence of jet rotation has been considered as demanded by NSWC.

The ideas behind the **direct thrust measurement method** are reproduced in Appendix C.

3.2 Contributors

Replies were received from the following institutions:

DGA/DCN	Mr. B. Lamberti, France
VWS	Prof. M. Schmiechen, Germany
KaMeWa	Mr. R. Aartojärvi, Sweden
MHI	Mr. T. Hoshino and Dr. E. Baba, Japan
VTT	Dr. J.V. Pylkkänen, Finland
MARIN	Mr. T.J.C. van Terwisga, The Netherlands
NSWC	David Taylor Model Basin, USA
SSPA	Mr. P. Lindell, Sweden
KTH	Prof. O. Rutgersson, Sweden
INSEAN	Dr. L. Iannone, Italy
CERG	Dr. P. Chantrel, France
KSRI	Dr. O. Orlov, Russian Federation
MARINTEK	Mr. K. Minsaas, Norway

As a result of these contributions a detailed discussion of the momentum flux method is presented in Section 3.3. The method has been modified, unclear details have been clarified and the importance of the different elements of the method for the accuracy of the power prediction is revealed. The individual contributions will be mentioned in this section, but they will not be summarised individually.

Section 3.4 is summarising the comments dealing with the direct thrust measurement method (Appendix C). Since no detailed description, comparable with that of Appendix A, exists for this method, only those comments have been compiled which contribute to a list of advantages and disadvantages of this type of prediction method.

3.3 Momentum Flux Method

The main idea behind the momentum flux method, as described in Appendix A, was to formulate a method which, in principle, agreed with procedures used by most towing tanks and manufacturers involved in waterjet testing alike. Deliberately, no assumptions simplifying the prediction work were introduced and no detailed information on how to determine several of the important parameters in the method was given. In the following, a more user oriented version of the prediction method will, as requested by many contributors, be discussed.

The prediction is based on self-propulsion tests in combination with special pump tests not described in Appendix A. No resistance tests are needed. As pointed out by VTT, MARIN, NSWC and KTH resistance tests with closed waterjet intakes will probably be carried out nevertheless, to get a starting point for the selection of the waterjet system. Many customers who can not afford to carry out self-propulsion tests are eager to know if there is some general relation between the bare hull resistance R_T and the change of momentum flux ΔM computed from the self-propulsion tests. In the committee's opinion there is no such general relation, the main reason being that the flow around the vehicle in the two tests is quite different in many respects and that this difference varies with vehicle shape and speed.

Thus, the waterjet system is inducing forces and moments changing the running trim and sinkage considerably. The flow at the transom at lower speeds will be disturbed by the jet in an unpredictable way when the outlet nozzle is partly below the water surface. However, if the resistance tests are carried out with the same running trim and sinkage as the self-propulsion tests, and if the waterjet is ejected above the water surface, it can be of some interest to compare the measured resistance R_T with the computed change of momentum flux ΔM (see Item No. 6 below). Theoretically, this resistance minus the tow rope force should be equal to ΔM.

KaMeWa and NSWC raise the question if the method can also be used in the frequent cases when, for practical reasons, the model pump is not to scale. The answer is yes, but the procedure will then be applied in a slightly different way, as will be shown at the end of Section 3.3.

In the following the comments made by different contributors will be considered in the same sequence as the different items of the prediction method of Appendix A.

Item No. 1. The tow rope force is computed in much the same way as for a vehicle fitted with propellers. For planing craft this means that it has to be determined after the self-propulsion tests have been carried out and the real varying wetted surface is known. It is therefore suggested that these tests are carried out with at least two different tow rope forces to permit an interpolation to the correct value.

As pointed out by NSWC the tow rope force should include a contribution from the difference in energy velocity at Station 1 between model and full scale, if estimation of the waterjet shaft speed is to be derived directly from the model tests. This is correct if one desires to scale shaft speed as well as flow rate and jet velocity determined at model scale simply by using powers of the scale factor λ. In this case the influence on the tow rope force can be computed from the difference between model momentum flux M_{1M} at Station 1 and the corresponding $\rho_M/\rho_S \cdot M_{1S}/\lambda^3$ estimated for the full scale vehicle. Due to the lower Reynolds number at model scale the non-dimensional energy velocity at Station 1 will be lower than at full scale. This means that the adjustment to the tow rope force will be negative, i.e. the total tow rope force will be smaller. If this procedure is followed the scale effect considerations regarding the boundary layer, discussed under Item No. 13, have already been accounted for and can be omitted in that step.

Item No. 2. Most comments on the measurements taken during self-propulsion tests are concerned with the flow rate Q_J and the energy velocity at Station 1, V_{E1}.

Many different systems for measuring the flow rate have been used in the past. These can be distinguished as follows:

- MHI (Hoshino and Baba, 1993) and MARINTEK have collected and weighed the water ejected from the waterjet system in a tank following the model. The procedure has the advantage that it can be used at speed, but the jet outlet has to be well above the water surface.

- MHI and SSPA determine the relation between pressure drop over the jet nozzle and flow rate either by collecting water in a tank at zero speed or by force measurements, under bollard pull conditions. Since the intake momentum flux is zero under these conditions, the force measurements in the second case give the jet momentum flux. This permits one to calibrate the flow rate, provided the jet area is known and the jet velocity is uniform. The flow rate at the self-propulsion test is then derived from the measured pressure drop over the nozzle and the calibration curve.

- KaMeWa has recently tested the idea of deflecting the jet at 90 degrees by placing a T-tube shortly behind the nozzle. Since the momentum flux leaving the T-tube has no component in the direction of thrust, force measurements permit to determine the jet momentum flux and from it the flow rate as described above. Experience with this method in free-surface water tunnels is reported to be satisfactory. However, difficulties are expected during self-propulsion tests where waves around the model must not be disturbed by the space required for the T-tube arrangement. MARINTEK has also proposed this method.

- Pitot tube measurements in the jet have the advantage of yielding possible velocity variations over the cross section. But difficulties in measuring close to the edge of the jet make an integration for the flow rate uncertain. In this respect, LDV measurements are more promising, although short measuring times and unsteady conditions during self-propulsion tests are problems to be resolved.

- In the case of air ingestion at the waterjet inlet, as has been experienced in the case of SES running in a seaway, no method for flow rate assessment seems to exist. Both DGA/DCN and SSPA recommend that under such conditions correlation with torque measurements be sought.

The importance of correctly assessing the flow rate is stressed in Appendix D of this report, where it is shown that an error of 1 % in Q_J results in an error of 3-4 % in the predicted power. Considerable efforts must therefore be made to find a reliable method to determine and check Q_J.

One way of checking if there is something wrong with the flow rate Q_J could be to study the outlet loss coefficient ζ_{57} computed from the energy flux at Station 5 and 7, E_5 and E_7. This coefficient should generally be small. Since E_5 is made up by the static pressure and the velocity behind the pump it is far less sensitive to errors in the flow rate than the energy flux E_7 in the jet. If the computed ζ_{57} gets an abnornal value or is varying strongly with speed the reason can be that the flow rate is wrong. It must be stressed, however, that a determination of E_5 from mean pressure and speed values at Station 5 also involves errors (see Item No. 9).

Most of the contributors are concerned about the location of Station 1 (see Appendix A). Both when calculating the change of momentum flux and the power, the energy velocity at Station 1, V_{E1}, is used and obtained by different ways of integration. As explained in Appendix A, V_E is a measure of the local energy of the flow, computed from the absolute velocity and the static pressure. Assume that V_{E1} is measured at a station ahead of the inlet where there is no influence from the waterjet system. If the measuring plane is moved backwards the absolute velocity is increased due to flow acceleration. However, the static pressure is decreased at the same time and the energy velocity remains almost unchanged. Although the higher velocity makes the height h_1 of the cross section smaller, the different integrated velocities at Station 1 will be the same. The location of Station 1 is therefore arbitrary, within reasonable limits. One could probably define the location of Station 1 as the station just before intake losses occur, as suggested by VWS. However, due to the strong three-dimensional flow close to the intake a certain difference between the absolute velocity u_1 and the corresponding axial velocity u_{1x} will occur (see Appendix A). It is therefore recommended that Station 1 be placed at some distance ahead of the intake. To check the three-dimensional flow at this location it could be wise to perform some oil paint streak tests, in order to determine the direction of the limiting streamlines of the shear flow around the intake.

If the boundary layer is thin it can be difficult to use Prandtl tubes to determine the energy velocity distribution at Station 1, as pointed out by several contributors. An alternative is to use a Pitot tube rake combined with a static pressure tap at the hull. The pressure is then assumed to be constant across the boundary layer. A good idea could be to match the measured velocities by a power law profile, as suggested by MARIN as well as Steen and Minsaas (1995). It is proposed to compare the velocity profile determined in this way with corresponding results for a turbulent boundary layer along a flat plate calculated according to boundary layer theory.

As pointed out by NSWC the original version of Appendix A, without specifically mentioning it, did not allow for jet rotation. If the model waterjet system is working far from its design point jet rotation must no longer be neglected (e.g. Matsumoto et al., 1993). In this case, it may be necessary to measure both the axial and tangential velocity distributions in the jet by some kind of multi-hole probe. When the rotational velocity is known the corresponding static pressure reduction caused by the rotation may be computed together with the energy velocity. However, these measurements are difficult to carry out, especially near the edge of the jet where air entering the pressure tubes may lead to erroneous results.

Item No. 3. As mentioned above, many contributors are concerned with the determination of the flow quantities at Station 1. The location of Station 1 has already been discussed. Now size and shape of the cross section will be considered. Knowing the flow rate Q_J, the size of the intake area is computed from the law of continuity, assuming the width and the shape of the cross section to be known. If the height of the cross sectional area is less than the thickness of the boundary layer the formula given under Item No. 3 in Appendix A can be used directly. If the boundary layer is thinner, the calculation of the integrated energy velocities used for thrust and power calculations can be simplified since the local energy velocity is constant outside the boundary layer ($V_{E1} = V$). In this case the shape of the cross section outside the boundary layer is of no interest - only the amount of fluid passing through it. This amount is obtained by integrating the flow rate at Station 1 from the hull to the edge of the boundary layer. If this inner flow rate is Q_{bl} the outer rate will be $Q_J - Q_{bl}$.

The test example mentioned in Appendix D shows that an error of 20% in the selected width of the rectangular cross section, b_1, only means an error of about 1% in predicted power. In addition, the influence of section shape is small.

If no information to the contrary is available it is recommended to use a rectangular cross section with a width 30 % larger than the width of the intake. The confidence of the selected width b_1 and cross section shape can be tested by a sensitivity analysis.

Items No. 4 and 5. Momentum and energy flux are calculated according to the formulas given in Appendix A. If the thickness of the boundary layer is smaller than the height of cross section at Station 1 the calculation of M_1 and E_1 can be simplified as follows:

$$M_1 = \rho \cdot b_1 \cdot \int_0^{\delta} V_{E1}(z) \cdot u_{1x} \cdot dz + \rho \cdot V \cdot (Q_J - Q_{bl})$$

and

$$E_1 = \frac{1}{2} \cdot \rho \cdot b_1 \cdot \int_0^{\delta} V_{E1}^2(z) \cdot u_{1x} \cdot dz + \\ + \frac{1}{2} \cdot \rho \cdot V^2 \cdot (Q_J - Q_{bl}) ,$$

where δ is the thickness of the boundary layer and Q_{bl} is the boundary layer flow rate.

Corresponding formulas for the jet (Station 7) were originally different from those given in Appendix A, the difference being that the energy velocity concept was meanwhile introduced. Now also the jet rotation, as requested by NSWC, can be taken into account. If the jet at a radius r has a tangential velocity $u_{7\phi}$, the local energy velocity V_{E7} becomes

$$V_{E7} = \sqrt{u_{7x}^2 + u_{7\phi}^2 + \frac{2}{\rho} \cdot (p_7 - p_0)} ,$$

where the pressure reduction p_7 - p_0 in the jet can be computed from

$$p_7 - p_0 = -\rho \cdot \int_r^{R_J} \frac{u_{7\phi}^2}{r} \cdot dr ,$$

in which R_J is the radius of the jet.

The sensitivity analysis given in Appendix D indicates that the influence of a moderate jet rotation on the power is small but that the change of momentum flux is slightly affected due to the reduced static pressure in the jet. If it is difficult to determine the velocity components of the jet accurately it is, in any case, wise to make a rough experimental check of how strong the jet rotation is and how large the axial jet velocity deviations are, to ensure that any simplifications are justified.

Items No. 6 and 7. When calculating the change of momentum flux ΔM the variation of the energy velocity in the jet can be ignored if this variation is the same in model and full scale. The reason is that ΔM, as used in the present prediction method, will have no direct influence upon the predicted power and an error in model scale will be counteracted by a corresponding error in full scale. This is not the case for the calculation of the effective jet system power, where large velocity variations in the jet can be important.

Item No. 8. Although none of the contributors has criticized the simple formulation of the elevation power in Appendix A it is clear that in cases where only part of the jet is above the undisturbed free water surface it must be calculated from

$$\rho \cdot g \cdot \int_{Q_J} z \cdot dQ_J ,$$

where the integration is performed only above the undisturbed free surface ($z \geq 0$).

Item No. 9. The internal losses are very important for the power prediction as pointed out by KaMeWa, SSPA, INSEAN and CERG. The inlet loss coefficient ζ_{13} is determined from the energy flux at Stations 1, 3 and 0 as follows

$$\zeta_{13} = \frac{E_1 - E_3}{E_0} .$$

If the velocity distribution just ahead of the pump were to be uniform, E_3 could easily be determined from the mean velocity Q_J/A_3 and a mean pressure p_3 - p_0 measured at the self-propulsion test, i.e. from

$$E_3 = Q_J \cdot \left[\frac{1}{2} \cdot \rho \left(\frac{Q_J}{A_3} \right)^2 + (p_3 - p_0) \right] .$$

This is, however, almost never the case. In reality the velocity is quite non-uniform with large velocity variations.

For a typical velocity distribution just ahead of the pump, the inlet loss coefficient ζ_{13} becomes as much as 0.2 lower if the calculations are carried out in a correct way rather than using the mean value of the velocity at Station 3. This means an error of 6 - 7 % in predicted power.

Since in many practical cases it is impossible to determine the velocity distributions at any station inside the waterjet system in self-propulsion tests, the conclusion is that internal loss coefficients must be determined in special tests with large models, permitting detailed velocity and pressure measurements to be performed. In these tests, which can be carried out either in a special test rig in a towing tank or in a cavitation tunnel, the modelling of the boundary layer ahead of the intake is important.

Items No. 10 and 11. While the determination of the effective pump power, as described in Appendix A, is straight-forward, provided the internal losses are known, the complicated transmission of energy to the fluid by the waterjet pump (CERG) makes the calculated model shaft power much more uncertain. Both the pump efficiency η_P and the installation efficiency η_{inst} must be determined in special test rigs.

One way to solve this problem could be first to test the pump in a conventional pump test rig to get the pump efficiency and then repeat the tests with a special inlet, modelling the flow at Station 3 ahead of the waterjet pump. The difference in the results will then give the installation efficiency.

A more direct way would be to test the pump in the same rig as used for testing the internal losses. The flow into the pump will then be modelled in a natural way and the scale effects caused by different boundary layers ahead of the intake in model and full scale could easily be clarified. The tests can be used either to determine the internal losses and the product of pump and installation efficiencies separately or to give an effective jet system efficiency relating the shaft power to the effective jet system power plus the elevation power:

$$\eta_{WJ} = \frac{P_{JSE} + \rho \cdot g \cdot Q_J \cdot h_J}{P_{DM}} \; .$$

A practical problem is that this procedure probably means that two waterjet systems must be manufactured, a smaller one for the self-propulsion tests and a bigger one for the special tests. It is, however, difficult to see that there is any alternative, if the towing tank is to give reliable power predictions.

Item No. 12. A check of the model shaft power can, as pointed out by INSEAN, be used only to estimate of the total error in efficiencies and loss coefficients - not the details. As a matter of fact, it can only be used to determine the effective jet system efficiency at model scale.

Item No. 13. One of the problems when determining the scale effects is the boundary layer in full scale. Since also the roughness must be taken into account it is recommended that the thickness of the turbulent boundary layer be calculated from boundary layer theory, assuming a Reynolds number giving the same skin friction coefficient as computed for the full scale vehicle where also the roughness allowance has been included.

The procedure to determine full scale values of Q_J, M_1, h_1 and M_7 is exactly the same as described under Items No. 3-5 for the model. A condition is, of course, that the inlet and outlet shapes are geometrically similar (VTT).

If the special tests mentioned above have been carried out with a large waterjet system, it would be possible to convert the results to full scale with some confidence, especially if the tests have been performed with a full scale boundary layer at the inlet.

Item No. 14. The predicted full scale power is computed as shown under Item No. 14. The rate of revolutions, requested by MHI, is obtained from the pump characteristics, knowing the flow rate and the mean total head across the pump. If these characteristics are not known the results from the special test rig mentioned above can be utilized.

Self-propulsion tests with stock waterjet pumps. As mentioned above the method described in Appendix A will now be reconsidered for waterjet model pumps which are not to scale.

For different reasons the self-propulsion tests are often carried out with stock pumps rather than with geometrically similar models of the full scale pumps. If this is the case the prediction procedure will be as follows:

- Although the pump is not to scale the inlet and outlet configurations must be.

- The steps described under Items No. 1-8 of Appendix A will be the same as before. However, it should be avoided that the model pump is working too far from its optimum so that the jet has a strong rotation and large axial velocity variations.

- The internal losses and the efficiency values in model scale are of no interest so the procedure continues with Item No. 13 (scale effect considerations).

- To be able to determine the internal losses and the different pump efficiencies in full scale it is necessary to carry out the special tests mentioned above with a large waterjet system with scaled pump and internal ductings.

- If results from such tests are not available the predictions will have to depend on the accuracy of estimating internal losses and pump efficiencies.

3.4 Direct Thrust Measurement Method

In Appendix C Mr. Lamberti of DGA/DCN, Bassin d'Essais des Carènes, Paris has summarized his opinion on waterjet power predictions. His main points are:

- Towing tanks and waterjet manufacturers must work closer together, the tank being responsible for the self-propulsion test, the manufacturer for reference tests with the final waterjet system and for the final prediction.

- Many uncertainties in using the momentum flux method described in Appendix A remain.

- An alternative is to measure the waterjet thrust directly, both in a towing tank and in the reference tests mentioned.

Several comments on the direct thrust measurement method were obtained and will briefly be summarized as follows:

- All contributors commenting on the aspect of cooperation, i.e. SSPA, KTH, INSEAN and CERG, agree that a close cooperation between towing tanks and manufacturers is an important condition for a good result. However, as pointed out by KTH, there is a problem when only a part of the complete testing is carried out by the towing tank. The manufacturer has to guarantee the performance of the waterjet system and it is natural that he is not willing to give away data that is important in the competition with other manufacturers. For the tank people, this is of course a distressing handicap which they do not have when making power predictions with conventional propellers.

- Drawbacks of the momentum flux method mentioned by the contributors are difficulties to determine the flow rate (SSPA, KTH, CERG), the momentum and energy flux at Station 1 (NSWC), internal losses (INSEAN, CERG) as well as pump and installation efficiencies (INSEAN, CERG). In spite of this, VTT, MARIN, KaMeWa, SSPA, MARINTEK and KTH seem to prefer this approach, the main reason being that it is easier and cheaper to perform. An additional advantage is that stock pumps can be used for the self-propulsion tests and that a model-ship correlation of the flow rate can be obtained by measuring the pressure drop in the outlet nozzle also in full scale.

- The main drawbacks of the direct thrust measurement method mentioned by the contributors are measuring errors caused by the large perimeter sealing (KaMeWa, SSPA, NSWC and INSEAN), dynamic problems (MARIN) and the impossibility to measure the direct thrust at full scale, hampering model-ship correlation studies

(SSPA, KaMeWa). The principles of this procedure are preferred by KSRI and INSEAN. They are accepted as good as the momentum flux method by CERG. Arguments are that a direct measurement of axial force, vertical force and moment will increase the understanding of the waterjet-hull interaction and that the use of the reference test makes the procedure similar to that for a conventional propeller.

4 RECOMMENDATIONS TO THE CONFERENCE

As a result of the discussions and contributions reviewed in Chapter 3 of this report the following recommendations to the conference are made by the Specialist Committee on Waterjets:

- Reliable performance predictions for waterjet propelled craft require that **model self-propulsion tests** are carried out. Performance predictions, based on resistance tests only, may lead to serious errors.

- For successful performance predictions the **cooperation** between towing tanks and waterjet manufacturers must be enhanced, largely with regard to decisions on the geometry of inlet, ducting, pump and nozzle, as well as in relation to pump performance, both at model and full scale.

- Experience obtained with model self-propulsion tests evaluated by **the momentum flux method** should be collected and reported.

- **Direct thrust measurements** in model self-propulsion tests represent an alternative to the momentum flux method and should be evaluated further.

- Full scale **trial data** with waterjets should be collected and published.

5 REFERENCES

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APPENDIX A

POSSIBLE POWER PREDICTION METHOD FOR WATERJET SYSTEMS

The prediction method for waterjet systems presented below is based upon momentum flux considerations and has many elements in common with earlier methods, among them the one discussed in the Report of the High Speed Marine Vehicles Committee (18th ITTC, 1987). There are, however, some differences, the following ones being the most important:

- The gross thrust concept as defined at the 18th ITTC (1987) has been replaced by a change in momentum flux ΔM, balancing the thrust from the pump and the internal ducting force, plus the change of hull resistance caused by the action of the propulsion unit, including trim effects.

- The change in momentum flux ΔM and the shaft power P_D are calculated using the local energy velocity V_E defined by

$$\frac{V_E}{V} = \sqrt{\left(\frac{u}{V}\right)^2 + C_p} \ ,$$

where u and C_p are the local total velocity and the static pressure coefficient

$$C_p = \frac{p - p_0}{\frac{1}{2} \cdot \rho \cdot V^2}$$

measured at some distance in front of the intake. The energy velocity is a measure of the reduced local total head inside the boundary layer caused by frictional forces along the hull. Therefore, it takes both the kinetic and the potential energy into account. Outside the boundary layer one obtains $V_E = V$. The relation between V_E and the local total head H is given by

$$H = \frac{V_E^2}{2 \cdot g} .$$

- The torque Q and the rate of revolutions n, measured in self-propulsion tests, are monitored to check some of the uncertain quantities in the prediction procedure.

- Since the prediction method for waterjet systems differs so much from that for conventional propellers the use of the same nomenclature in the two cases has been abandoned wherever this may lead to confusion.

In the description given below the different stations of the streamtube through the waterjet system (see Fig. A1) are numbered as given in Table A1. The prediction method is developed for flush intakes but can also be used for Pitot intakes by setting $V_{E1} = V$. It will be described for a body-fixed coordinate system x-z in the following.

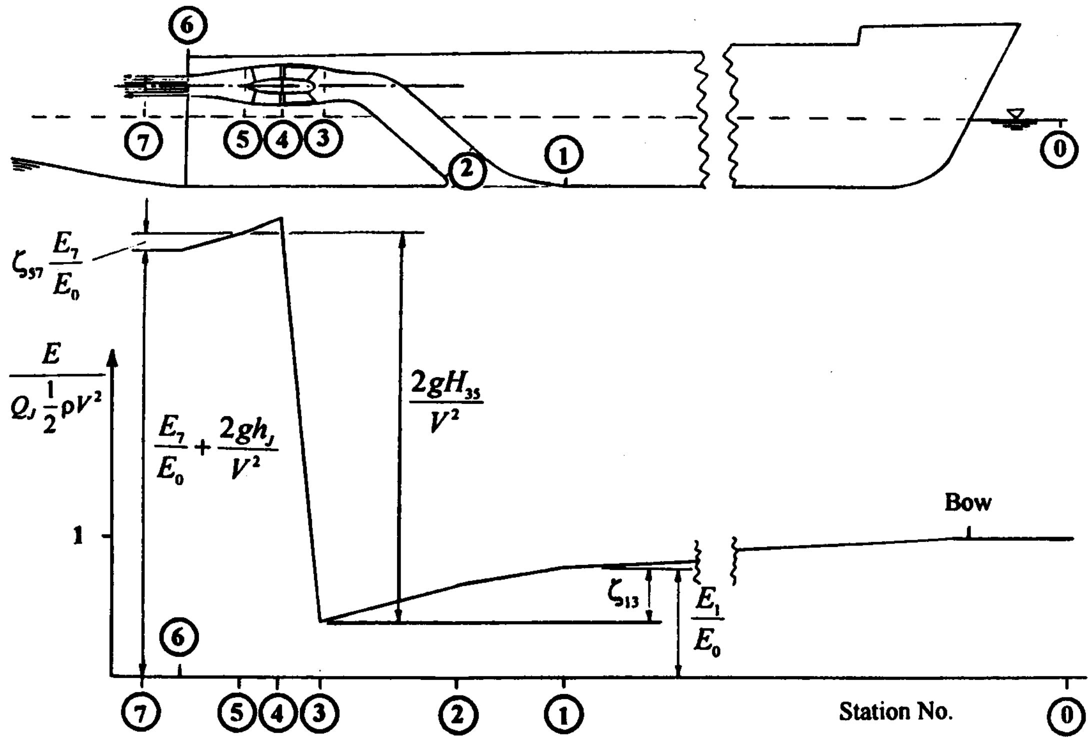

Figure A1. Definition of Station Numbers and Normalised Energy Flux

Table A1

Station No.	Location
0	in undisturbed flow far ahead of the vehicle
1	far enough in front of the intake ramp tangency point, before inlet losses occur
2	normal to the internal flow at the aft lip of the intake
3	just ahead of the pump
4	between pump and stator or between stages
5	behind stator
6	at the nozzle outlet plane
7	behind the nozzle outlet plane where the static pressure coefficient in the jet is zero (vena contracta)

1 Tow Rope Force

A tow rope force F_D is computed from the estimated difference in resistance coefficient C_T at model and full scale. Thus F_D includes differences in skin friction C_F, hull roughness and air drag. It also allows for the drag of appendages not present at model scale, etc.

2 Measurements during Self-Propulsion Tests

At the self-propulsion test the following quantities are determined apart from running trim and sinkage:

- the volume flow rate Q_J,
- the torque Q and the rate of revolutions n,
- the velocity distribution $u_1(z)$ at the intake using the difference between the total pressure and the static pressure of a Prandtl tube located

at Station 1 (observe that a Prandtl tube is rather insensitive to flow inclinations which means that it is measuring the total velocity $u_1(z)$ and not the component $u_{1x}(z)$!),

- the static pressure coefficient $C_{p1}(z)$, obtained from the difference between the static pressure p_1 at Station 1 and the static pressure p_0 in undisturbed flow, measured by a second Prandtl tube located ahead of the vehicle, at Station 0,
- the local energy velocity $V_{E1}(z)$, calculated from $u_1(z)$ and $C_{p1}(z)$, as defined above,
- the velocity component $u_{1x}(z)$ in the x-direction, which in many cases can be assumed to be equal to $u_1(z)$,
- the velocity distributions $u_{7x}(A_7)$ and $u_{7\phi}(A_7)$ in the jet, using a Pitot tube or multi-hole probe at Station 7, if the jet velocity distribution is irregular (the static pressure coefficient is computed).

3 Size of Intake Area

The maximum height h_1 of the intake area A_1 is determined from the law of continuity using the flow rate Q_J, the velocity profile $u_{1x}(z)$ and assuming the maximum width b_1 and the intake area to be known. The shape of the intake area is assumed either to be rectangular or half-elliptic, as proposed by Alexander et al. (1994). Thus h_1 and A_1 are determined implicitly from

$$Q_J = \int_{A_1} u_{1x}(z) \cdot dA_1 \quad .$$

The velocity $u_{1x}(z)$ is assumed to vary only in the direction of z until a better knowledge of three-dimensional inflow effects may permit more realistic approaches. The need for a better knowledge was stressed in the Report of the High Speed Marine Vehicles Committee (20th ITTC, 1993), largely with regard to quality control aspects of any power prediction method.

4 Momentum Flux

The momentum flux at Station 1 used for force calculations, see Item No. 6, is obtained by integrating the local energy velocity V_{E1} as follows:

$$M_1 = \rho \cdot \int_{Q_J} V_{E1} \cdot dQ_J \,,$$

where

$$dQ_J = u_{jx} \cdot dA_j .$$

At Station 7, the „momentum flux" is instead computed from

$$M_7 = \rho \cdot \int_{Q_J} u_{7x} \cdot dQ_J + \int_{A_7} (p_7 - p_0) \cdot dA_7 \,,$$

where p_7 - p_0 is the pressure reduction caused by tangential velocities $u_{7\phi}$ of the jet

$$p_7 - p_0 = -\rho \cdot \int_r^{R_J} \frac{u_{7\phi}^2}{r} \cdot dr .$$

A_7 is the jet cross sectional area and R_J is the radius of the jet. A_7 may be smaller than the geometrical nozzle area (A_6) due to jet contraction.

The name „momentum flux" for M_7 is strictly not correct in cases when the jet is rotating, since M_7 will then contain a pressure term.

5 Energy Flux

The energy flux used to calculate the power and the internal losses is obtained by integrating the local energy velocity V_{Ej} at Station j as follows:

$$E_j = \frac{1}{2} \cdot \rho \cdot \int_{Q_J} V_{Ej}^2 \cdot dQ_J .$$

In the undisturbed flow ahead of the vehicle, Station 0, the energy flux is:

$$E_0 = Q_J \cdot \frac{1}{2} \cdot \rho \cdot V^2 .$$

6 Change of Momentum Flux

The change of momentum flux ΔM, for corresponding streamtubes in resistance and self-propulsion tests, can be written as

$$\Delta M = M_7 \cdot \cos\alpha - M_1,$$

where α is the angle between the centreline of the jet and the horizontal plane. Since ΔM, as mentioned, is supposed to be equal to the sum of the horizontal forces on the pump and the internal ducting plus the change of hull resistance due to the action of the waterjet system, it is equal to the effective model resistance minus the tow rope force. The effective full scale resistance can therefore be computed from

$$R_S = \frac{\rho_S}{\rho_M} \cdot \Delta M \cdot \lambda^3,$$

where λ is the scale factor and ρ_M and ρ_S are the water densities at model and full scale.

7 Effective Jet System Power

It is now possible to compute the effective jet system power P_{JSE}, from the increase of energy between Station 1 and Station 7:

$$P_{JSE} = E_7 - E_1.$$

8 Elevation Power

The power needed to lift the water to a height h_J above the undisturbed water surface is computed from

$$\rho \cdot g \cdot Q_J \cdot h_J.$$

9 Internal Losses

Power is also needed to overcome the inlet and outlet losses. If ζ_{13} and ζ_{57} are the loss coefficients for the intake and diffusor on one hand and for the outlet nozzle on the other hand, one obtains

$$\zeta_{13} \cdot E_0 + \zeta_{57} \cdot E_7.$$

These coefficients can also be expressed as follows:

$$\zeta_{13} = \frac{E_1 - E_3}{E_0},$$

$$\zeta_{57} = \frac{E_5 - E_7}{E_7}.$$

10 Effective Pump Power

The effective power P_{PE} is the sum of the contributions described in Items No. 7-9 (see also Fig. A1), i.e.

$$P_{PE} = P_{JSE} + \rho \cdot g \cdot Q_J \cdot h_J + \zeta_{13} \cdot E_0 + \zeta_{57} \cdot E_7$$

or

$$P_{PE} = \rho \cdot g \cdot Q_J \cdot H_{35},$$

where H_{35} is the increase of the mean total head across the pump:

$$H_{35} = \frac{1}{\rho \cdot g \cdot Q_J} \cdot \left[E_7 \cdot (1 + \zeta_{57}) - E_1 + E_0 \cdot \zeta_{13}\right] + h_J.$$

11 Model Shaft Power

If one knows the pump efficiency η_P, determined in a conventional pump test rig and the pump installation efficiency η_{inst}, taking care of the inflow non-uniformities to the pump in the waterjet system, one can determine the power P_{DM} needed to propel the model :

$$P_{DM} = \frac{P_{PE}}{\eta_P \cdot \eta_{inst}}.$$

12 Check of Model Shaft Power

The shaft power can also be determined from the torque measurements. If P_{DM} is not equal to $2 \cdot \pi \cdot Q \cdot n$, then the estimate of internal loss coefficients (ζ_{13}, ζ_{57}) or efficiency values (η_P, η_{inst}) should be reconsidered.

13 Scale Effect Considerations

To be able to compute the full scale power one must determine full scale values of volume flow rate, size of intake area and energy velocities at Stations 1 and 7. Due to the scale effects of the boundary layer profile at the intake these quantities can not directly be converted from corresponding model values but the following procedure must be followed:

- First, the full scale boundary layer thickness and velocity profile at the inlet are predicted, also considering that the full scale vehicle has a certain hull roughness. The static pressure coefficient is assumed to be the same as in the model test.

- Second, full scale values of Q_J, M_1, h_1 and M_7 are computed from the momentum theorem, using the full scale velocity profile and maintaining the change of momentum flux

$$\Delta M_S = R_S = \frac{\rho_S}{\rho_M} \cdot \Delta M_M \cdot \lambda^3 .$$

- Third, full scale values of E_1 and E_7, appropriate internal loss coefficients ζ_{13S} and ζ_{57S}, pump and installation efficiency figures η_{PS} and η_{instS} are estimated.

14 Predicted Full Scale Power

Using the figures estimated in Item No. 13, the full scale effective pump power P_{PES} is computed as described in Item No. 7 - 10 together with the increase of mean total head across the pump H_{35S} as shown in Item No. 10. The pump shaft power, finally, will be:

$$P_{DS} = \frac{P_{PES}}{\eta_{PS} \cdot \eta_{instS}} .$$

APPENDIX B

PROPOSED LIST OF ITTC SYMBOLS FOR WATERJETS

ITTC Symbol	Computer Symbol	Name	Definition or Explanation	SI-Unit
A_j	AJ	Cross sectional area at Station j		m^2
b_1	B1	Maximum width of cross sectional area at Station 1		m
C_p	CP	Local pressure coefficient	$(p-p_0)/(\rho V^2/2)$	1
E_j	EJ	Energy flux at Station j	$1/2 \cdot \rho \cdot \int_{Q_J} V_{Ej}^2 \cdot dQ_J$	W
h_1	H1	Maximum height of cross sectional area at Station 1		m
h_J	HJ	Height of jet centreline above undisturbed water surface		m
H_{35}	H35	Mean increase of total head across pump and stator or several pump stages		m
IVR		Intake velocity ratio	V_2/V	1
JVR		Jet velocity ratio	V_7/V	1

ITTC Symbol	Computer Symbol	Name	Definition or Explanation	SI-Unit
M_1	MF1	Momentum flux at Station 1	$M_1 = \rho \int_{Q_J} V_{E1} dQ_J$	N
M_7	MF7	„Momentum flux" at Station 7	$M_7 = \rho \int_{Q_J} u_{7x} dQ_J + \int_{A_7} (p_7 - p_0)\, dA_7$	N
ΔM	DMF	Change of momentum flux	M_7 - M_1	N
p_j	PRJ	Local static pressure at Station j		N/m^2
p_0	PR0	Ambient pressure in undisturbed flow		N/m^2
P_{JSE}	PJSE	Effective jet system power		W
P_{PE}	PPE	Effective pump power		W
Q_{bl}		Volume flow rate inside boundary layer		m^3/s
Q_J	QJ	Volume flow rate of jet		m^3/s
u_j	UJ	Local total velocity at Station j		m/s
u_{jx}	UJX	Local axial velocity at Station j		m/s
$u_{7\phi}$	UJFI	Local tangential velocity at Station 7		m/s
V_j	VJ	Mean velocity at Station j		m/s
V_{Ej}	VEJ	Local energy velocity at Station j	$\sqrt{u_j^2 + \frac{2}{\rho}\cdot(p_j - p_0)}$	m/s
α	ALFA	Angle between centreline of jet and horizontal plane		1
η_{inst}	ETAIN	Pump installation efficiency		1
η_P	ETAP	Pump efficiency		1
η_{WJ}	ETAWJ	Effective jet system efficiency		
ζ_{13}	ZETA13	Inlet and diffusor loss coefficient Station 1-3, based on E_0		1
ζ_{57}	ZETA57	Duct and nozzle loss coefficient Station 5-7, based on E_7		1

APPENDIX C

DIRECT THRUST MEASUREMENT METHOD

Together with the „Possible Power Prediction Method for Waterjet Systems", sent out to about 30 establishments in June 1995, a document was distributed which had been prepared by Mr. B. Lamberti of DGA/DCN, Bassin d'Essais des Carènes entitled „Power Prediction Method: Possible Variations on the same Theme". The content of this latter document is reproduced in a somewhat condensed version in the remainder of this appendix.

As in the Momentum Flux Method any alternative prediction method must be based on a number of essential requirements. The following are regarded as of particular importance:

- Inlet and outlet of the waterjet system must be geometrically scaled, whereas the model pump does not. As a result, pump efficiency measured at model scale is not representative.

• Cooperation between waterjet manufacturer and towing tank is an unavoidable necessity, both with regard to defining inlet, duct and nozzle geometry and with reference to the pump characteristics needed for prototype performance prediction. This cooperation could comprise tests for inlet optimisation by the towing tank.

• Sea trials evaluation is an essential requirement for the validation of prediction methods. Interests in the trial results will differ beween manufacturer and towing tank. The former may be interested in effects of cavitation, air ingestion or pump performance in a seaway, whereas the latter is likely to be concerned with the scaling of flow rates and possibly resistance, in particular in the case of high speed marine vehicles. If, as a result of sea trials, the required power is noticeably less than predicted (this is where „negative thrust deduction" comes in as an explanation!) the reason for this phenomenon must be brought to light. It must be clarified if the manufacturer has incorporated too large a safety margin in the design or if the towing tank has poorly predicted resistance or interaction effects between hull and propulsor. Thus, the cooperation between waterjet manufacturer and towing tank may perhaps be the most important issue to be addressed if performance prediction methods are to be improved.

With reference to the direct thrust measurement method it may be helpful, in the first instance, to look at a question which is not masked by scaling problems. Let us assume that one and the same waterjet propelled model, with identical inlet and nozzle, has been tested, by the model basin with the model pump „A", by the manufacturer with model pump „B". Would it be possible for the towing tank to predict model performance with regard to power and rate of revolutions, if pump „A" was replaced by pump „B"? The answer may appear simple, as one is supposed to know the pump efficiency η_P from a conventional pump test rig and the installation efficiency η_{inst} allowing for inflow non-uniformities which originate from upstream conditions in the waterjet installation. In this case the ratio

$$(\eta_P\ \eta_{inst})_A\ /\ (\eta_P\ \eta_{inst})_B$$

can be used to convert powers if pump „A" is replaced by pump „B". Yet the question remains how the four efficiences involved were obtained and how to predict the rate of revolutions.

Considering the definitions under Item No. 10 and 11 of Appendix A one gets the impression that η_P is obtained from some reference test, equivalent to an open water test for conventional propellers. On the other hand η_{inst} is the ratio of shaft powers measured during the reference test on one side and the self-propulsion test in the towing tank on the other side. In this case η_{inst} is equivalent to the relative rotative efficiency η_r for conventional propellers. However, in order to determine its magnitude, one has to define a quantity on which η_r is dependent and which has to be kept constant. Should this be the flow rate or, as with conventional propulsors, the thrust? If the latter is understood to be the total thrust, not only the thrust of the impeller, it is difficult to measure. Yet it can be measured if the whole propulsion unit, including inlet, nozzle and immediate surroundings, is separated from the rest of the model, e.g. by using a labyrinth-type seal. In this case the resulting force on inlet, nozzle and its immediate surroundings, on ducting, impeller and stator can be assessed and interpreted as effective thrust.

On the other hand, one may try to describe, in the most sophisticated way, the thrust by a change of momentum flux and may or may not agree on the various simplifications involved. In any case, the formula will include the product of two terms: the flow rate Q_J and a velocity change between Station 7 and 1. For commonly encountered values of the jet velocity ratio small errors in the jet velocity lead to much larger errors in the above mentioned product. For instance, for a pitot intake and a jet velocity ration JVR = 1.25, 1.5, 1.75 the error in momentum is 3.3, 4, 6 times the error in V_{E7}. Even if the jet velocity could be measured with an accuracy of ± 1 % the associated accuracy for the thrust would be unacceptably large. It is argued that the direct thrust measurement could avoid this degree of uncertainty.

If a direct thrust measurement is used in the reference test and in the self-propulsion test one finds oneself in exactly the same situation as with conventional propellers, i.e. the propulsion characteristics K_T, K_Q - J are known at model scale in „open water" (reference test) and in the „behind" condition. The remaining problems could then be identified as pure scaling effects.

Turning to scaling effects it is felt that an important part relates to the differences between the reference test at model scale and the large scale test in a conventional pump test rig. It may be advisable to leave this problem to the pump manufacturer who is usually more experienced in scaling from model dummy pump to the prototype.

Model shaft power (Item No. 12 of Appendix A) has to be checked very carefully, i.e. internal losses and efficiencies should be reconsidered. However, the flow rate Q_J should not be excluded since each contribution to the effective pump power is proportional to Q_J. The most common way of determining Q_J, i.e. calibrating the pressure drop over the nozzle, is usually accurate enough for providing a sound extrapolation basis. When air ingestion is involved, it usually shows up in the pressure drop over the nozzle and helps to identify this phenomenon.

It would be the responsibility of the towing tank, in cooperation with the manufacturer, to determine what should be called „apparent wake" (i.e. the ratio of advance coefficients for thrust identity between reference and self-propulsion test), relative rotative efficiency and thrust deduction. In this context two statements may be justified:

- The three quantities mentioned may or may not depend on the thrust itself.
- There are no reasons why we should scale relative rotative efficiency and thrust deduction if we do not do it in conventional propulsion.

Obviously, the „apparent wake" is not identical with the actual wake. If the reference test is carried out with local Reynolds numbers at the inlet corresponding to those at the self-propulsion test it simply accounts for three-dimensional inflow. As a result, it seems justified not to scale this term.

As a final goal one has to look at the thrust very carefully. Due to the boundary layer the velocity profiles will not be similar at model and full scale, i.e. for equivalent flow rates the internal flow at Stations 1, 2, 3 and the velocity distribution across the pump inlet mouth will be different. Hence, the flow rates can not be similar and one should consider determining the interaction coefficients from a scaled model flow rate, e.g. following one of the three principles:

- no scaling at all,
- following the momentum flux method with comprehensive measurements,
- by introducing an „equivalent" wake coefficient.

In the last case the equivalent wake is derived from

$$w_{equiv.} = \frac{T}{\rho Q_J V} + 1 - JVR .$$

In any case, the scaling has to be matched to the scaling methods of the pump manufacturer and the obvious uncertainty remains that scaling is always somewhat hazardous. Reliable sea trials may improve this situation if there is a true cooperation between pump manufacturer and towing tank.

APPENDIX D

SENSITIVITY ASPECTS

1 Introduction

The following paragraphs will attempt to throw some light on the influence that a number of assumptions may exert on the estimate of change of momentum flux ΔM and effective jet system power P_{JSE}. Such assumptions may have to be made in connection with the 3-dimensional inlet flow geometry, the characteristics of the boundary layer ingested and the velocity distribution in the free jet, if no direct measurements are available.

The quantitative analysis described below is based on a case study and a physical model which is largely in line with the definitions of Appendix A. The case study assumes jet velocity ratios of $1.4 \leq JVR \leq 2.0$ and a height of the rectangular intake cross sectional area which is twice the boundary layer thickness. The velocity distribution in the boundary layer is supposed to follow a seventh root law. For other assumptions the results may be different but the order of magnitude and the relation between different errors will remain largely unchanged.

2 Results

Flow Rate. An error of 1% in the flow rate will result in errors of 1% in ΔM and P_{JSE}, if the jet area is misjudged. However, if the jet velocity is erroneous the errors are 2-3% in ΔM and 3-4% in P_{JSE}.

Inlet Flow. An error of 20% in the width of the inflow cross sectional area will result in an error of 1% in ΔM and 0.6% in P_{JSE}, if the cross section is rectangular. For equal width an elliptical shape of the inflow streamtube will increase ΔM 0.9% and P_{JSE} 0.5%, compared with the rectangular shape. Differences in the power law characterising the boundary layer profile exhibit effects of similar magnitude.

Jet Flow. Measurements of the axial velocity distribution in free jets (e.g. Masilge, 1991) clearly show that retarded flow exists both in the centre of the jet and at the outer regimes. From numerous analyses it has been found that reality is well described by

$$u_{7x} = u_{7x\,max}\left\{1-\left(\frac{r}{R}\right)^{n} - k\left[1-\left(\frac{r}{R}\right)^{3/2}\right]^{12}\right\}$$

where high values of n mean „full“ velocity profiles and r/R is the non-dimensional jet radius. k is a measure of the speed reduction towards the centreline of the jet.

For n = 1000 and $0.2 \leq k \leq 0.4$ the non-uniformity effect on change of momentum flux and effective jet system power was investigated. For two jet velocity ratios the results of Table D1 and Table D2 were obtained.

Table D1

JVR		1.4	1.7
	0.2	0.54 %	0.38 %
k	0.3	0.80 %	0.56 %
	0.4	1.19 %	0.83 %
Effect on ΔM			

The results indicate the percentage increase in change of momentum flux for deviations in the jet velocity distribution from uniformity.

Table D2

JVR		1.4	1.7
	0.2	0.63 %	0.53 %
k	0.3	0.97 %	0.81 %
	0.4	1.42 %	1.20 %
Effect on P_{JSE}			

From Table D2 it can be seen that for moderately loaded waterjet systems (JVR=1.4) larger deviations in the jet velocity distribution (k=0.4) may lead to a power increase of 1.42%.

The influence of n (fullness of jet profile near the jet surface) was evaluated and documented in Table D3 for k=0.3 and JVR=1.4.

Table D3

		$\delta\Delta M/\Delta M$	$\delta P_{JSE}/P_{JSE}$
	50	7.20 %	7.85 %
n	100	3.92 %	4.25 %
	1000	0.80 %	0.97 %

The results show that deviations from a constant velocity profile may significantly affect change of momentum flux and power demand. Maintaining „full" velocity profiles near the jet surface will help to improve waterjet system performance.

Jet Rotation. Various calculations have shown that the influence on the predicted power is small. However, the reduced static pressure in the jet has detrimental effects on the change of „momentum flux" due to the pressure term $\int_{A7}(p_7 - p_0)\, dA_7$ in the momentum equation.

3 Conclusions

From the investigation carried out for a particular case study the following general conclusions can be drawn:

• Power predictions based on the momentum flux method, where the flow field at Station 1 is analysed by integration, are little affected by erroneous assumptions with regard to the width of the ingested streamtube and the boundary layer velocity profile.

• Deviations from a uniform velocity distribution in the jet may lead to noticeable changes in momentum flux and power. They can not be neglected. As a result, the velocity variations and the jet rotation should be carefully checked.

• A high accuracy in measuring the flow rate should be attempted.