Cavitation scale-effects in pumps

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Das weiterentwickelte Ausführungsprojekt des Hochwasserschutzbeckens Orden wird vom Ingenieurbüro Maggia in [14] beschrieben. Als Talabschluss dient eine Bogenstaumauer von 40 m Höhe, 180 m Kronenlänge und 10 m Basisstärke (Bild 6b). Der Grunddurchlass weist einen Durchmesser von 1,60 m auf und wird auf der Einlaufseite durch einen vom Mauerfuss bis zur Krone hochgezogenen Grobrechen derart abgeschirmt, dass das Geschiebe durchgeht, dagegen Baumstämme, Wurzelstöcke, Zweige usw. aufgehalten werden. Dieses bemerkenswerte Bauwerk soll in diesem Jahre vollendet werden.

Abschliessend danke ich dipl. Ing. *Ch. Bischoff*, Chef der Abteilung Fluss- und Wildbachverbauungen des Tiefbauamtes Graubünden, für seine Angaben über Orden und dipl. Ing. *R. Sigg*, Assistent im Institut für Hydromechanik und Wasserwirtschaft der ETHZ, für die Durchführung der entsprechenden graphischen Retentionsrechnung.

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Cavitation Scale-effects in Pumps

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Notation

C_{m2}	Radial velocity at impeller exit
C_p	Specific heat of liquid at constant pressure
g	Gravitational constant
H	Pump developed head
H_e	Ideal Euler head
h_{fg}	Latent heat of vapourization
J	Mechanical equivalent of heat
Kr	Non-dimensional parameter defined as $\dfrac{J\,h^2_{fg}}{v_g^2\,C_p}\,\dfrac{v_f}{T\,P_s}$
n	Reynolds number exponent
$NPIH$	Net positive inlet head; i. e. upstream total head minus vapour pressure head
$NPIH_i$	$NPIH$ at inception
$NPIH_b$	$NPIH$ at breakdown
$\triangle NPIH$	$NPIH_b - NPIH_i$
Pr	Prandtl number
P_{at}	Atmospheric pressure
P_s	Equilibrium vapour pressure
Q	Volumetric flowrate
Re	Reynolds number
r_f	Reciprocal of liquid specific gravity
r_v	Vapour to liquid volume ratio
T	Temperature
u_2	Circumferential velocity at impeller exit
v_f	specific volume of liquid
v_g	specific volume of vapour
Φ_2	Impeller blade angle at exit
ϕ	Flow coefficient
ψ	Head coefficient
σ	Thoma Cavitation coefficient $\sigma = NPIH/H$
σ_c	Critical Cavitation coefficient $\sigma_c = NPIH_b/H$ (at breakdown)

Introduction

For many years the Thoma cavitation coefficient σ has been widely used for correlating the cavitation performance of water turbines and pumps. For pumps σ is a dimensionless index of the margin between the vapour pressure and the pressure at inlet to the pump. This margin is often called *net positive suction head (NPSH)* or as we prefer to call it the *net positive inlet head (NPIH)* and is made dimensionless by dividing by the *cavitation free pump head, H,* to give $\sigma = NPIH/H$. The critical value, σ_c, may be defined in several ways such as the value of σ when the pump head (or efficiency) breaks down completely ($\sigma_c = NPIH_b/H$) or when the head (or efficiency) decreases say 3 % from its cavitation free value. For convenience it is often easiest to use the breakdown point for head as giving the most clearly defined value of σ_c.

However it has become increasingly obvious that the applicability of σ_c as a scale parameter is restricted to water over a narrow range of temperatures. For example when compared on a σ_c basis, a pump delivering water at 150 °C will have a better cavitation performance than the same pump handling water at 15 °C. Similarly the same pump when pumping kerosene will have a better cavitation performance in terms of σ_c than when it is pumping cold water. Generally speaking apart from mercury and the liquid metals, water gives the worst cavitation performance of all liquids whereas liquid hydrogen is particularly good. Such differences in σ_c can possibly be explained by the different thermal properties of fluids and their influence upon cavitation bubble growth. Although such effects on σ_c can be allowed for empirically it is desirable to find a basic theory which will explain such behaviour and some new cavitation parameter which will take account of thermal influences. At the moment there is no generally accepted explanation nor any generally applicable method of correlating pump performance for various liquids.

In practice the changes in σ_c are often of economic importance when deciding upon *N P I H* and the siting of pumps in a system. A particularly difficult problem is the siting of liquid metal pumps in a nuclear power-station cooling system because the correct choice of *N P I H* has an important influence upon the excavation required for the pumps and thus upon the civil engineering costs. It may also be of importance in deciding upon the running speed and inlet conditions for a boiler feed pump.

This paper reviews research into the basic problem directed by the senior author over recent years. In addition to theoretical studies and comparisons with published literature tests have been made on a small 25 H. P. pump [N_s (British) $= 1000$] over a range of speeds from 900 to 2500 r.p.m. These tests and the specially designed cavitation circuit have been described by *T. C. Chivers* [1]. The original object of the research was to analyse the validity of making pump tests with cold water and applying the results to other conditions. It was also hoped to compare the various methods of correlation suggested so far and if possible to establish a new parameter for general application to all pumps and liquids. So far experimental results have only been obtained using water, but its thermal properties have been varied by testing at temperatures from 0—150 °C. A theoretical cavitation model [1], [2] has been produced by *T. C. Chivers,* which was successful in predicting the pump's cavitating behaviour over the temperature range tested. This theoretical model was then developed to cover the problem of a general pump correlation [1], [3] and has been successfully applied to test data obtained by other people.

There are however, two reservations which must be made:

1st) although the predictions were often successful for one particular pump, the investigation showed that conclusions regarding correlation for pumps in general cannot be made from one pump.

2nd) whilst the developed parameter correlates the results examined its evaluation requires extensive data on fluid properties, which at present restricts it to pure liquids.

Theories of Cavitation

In recent years two review papers [4], [5] have appeared on the general subject of cavitation correlation in pumps. It is obvious from these papers that the majority of published work on this subject has been based on the "thermal approach", and this will be discussed later.

Another approach to the problem is to consider acoustic phenomena. If small quantities of vapour are present in a liquid then the sonic velocity in the fluid can be very low, and since the mass flow rate through a converging diverging nozzle becomes choked when sonic velocity is reached at the throat, then an analogy can be made between this phenomenon and that of breakdown

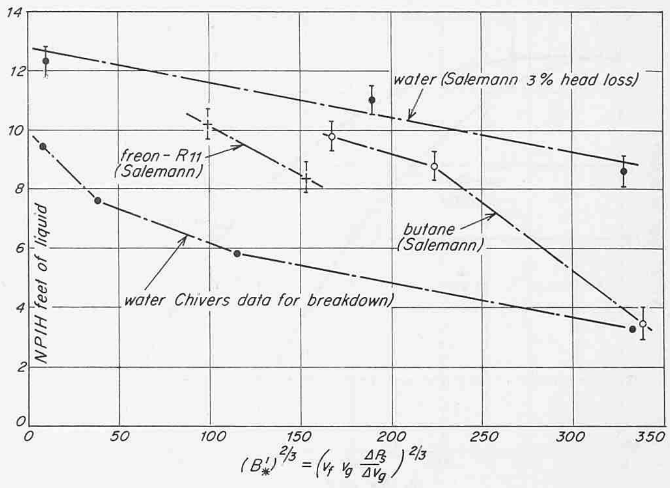

Fig. 1. Experimental results plotted against Jacobsen's parameter
The vertical lines represent the experimental scatter of the test results

(i. e. $dH/dQ = -\infty$) in a centrifugal pump under cavitating conditions.

This idea of sonic velocities influencing cavitating performance in machines is not new; reference [6] considers this aspect, but more recently *J. K. Jakobsen* [7] derived a correlation parameter,

$$B_*' = v_f\, v_g \frac{\Delta P_s}{\Delta v_g}$$

whilst *W. A. Spraker* [8] showed statistically that acoustic phenomena could be influential in the correlation of pump performance.

J. K. Jakobsen derived a linear relationship between *N P I H* and his parameter, and showed excellent correlation at design point between water and liquid oxygen data. However, if *Salemann's* pump data [9] for various liquids are used then correlation is not obtained (Fig. 1). Further, the various authors' data for breakdown conditions do not result in a linear relationship between *N P I H* and *Jakobsen*'s parameter.

The calculation of acoustic velocities in reference [7] is based on thermal equilibrium between the vapour and liquid phases and a uniform distribution of bubbles in the liquid. Following this approach *T. C. Chivers* calculated the critical vapour to liquid volume ratio which would result in sonic velocity being reached at the inlet to his test pump. It was found that large vapour-to-liquid volume ratios (5:1) are required for sonic velocities at higher temperatures (150 °C). This increasing trend with temperature can be interpreted as giving qualitative correlation with actual results in that at low temperatures (15 °C) breakdown occurs close (*N P I H* wise) to "inception" and could be associated with small vapour-to-liquid volume ratios (0,03). As temperature increases so the change in *N P I H* between "inception" and breakdown increases, and this could be associated with the development of increased vapour volumes before sonic choking was obtained. When breakdown is initially reached, cavitation is expected to be localized at impeller inlet, and to spread through the pump as the developed head reduces; when this head is zero then the flowrate is a maximum. Under such maximum flow conditions it may not be unreasonable to assume that the cavitating mixture does not change materially whilst passing through the pump, in which case the maximum r_v-value can be estimated.

The ideal Euler characteristic for *Chivers*' pump gives $\Phi = \text{tg}\, \beta_2$ for $\psi = 0$ and for the pump in question Φ_2 is 31,4 °. Therefore $C_{m2}/U_2 = 0{,}61$.

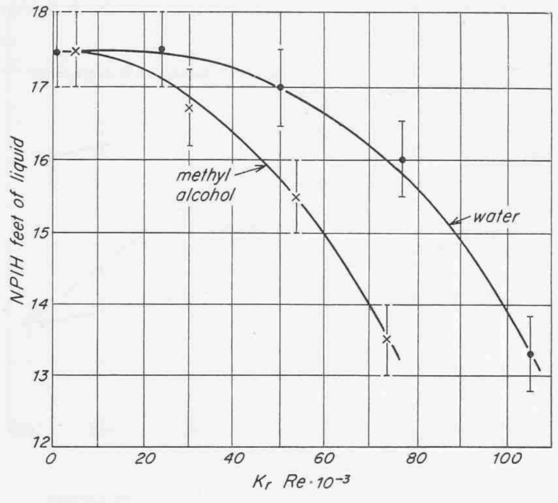

Fig. 2. Data from Spraker's pump 5, plotted against Barenboim's parameter

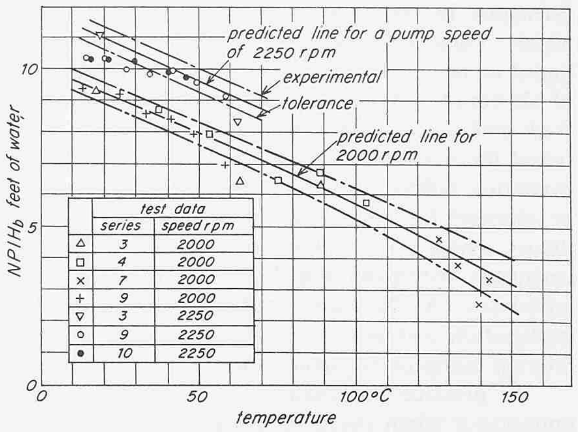

Fig. 3. Comparison between the predicted variation of $NPIH_b$ with temperature and test data

Considering a speed of 2000 r.p.m. it is found that the maximum flowrate is 76 l/s, and for the best efficiency flowrate in the non-cavitating section it is found that $r_{v\ max}$ $= 4{,}95$. This value shows that, according to the assumptions made, if the cavitation mixture is homogeneous, breakdown should not occur in the pump characteristic above 135 ° C.

In practice, the pump characteristic is below the Euler line, hence the limiting r_v-value would be less than that calculated. However, beakdown has been obtained in the pump characteristic at temperatures of 150 °C, and this fact would discredit the acoustic theory as presented.

A basic assumption in the argument was that the vapour and liquid phases were in thermal equilibrium; this is certainly not so, as the liquid phase must be superheated relative to the vapour phase. If now the sonic velocity which coincides with the beakdown condition at 150 °C is calculated, then 15 °C superheat of the liquid phase is required. This represents a difference of 16,8 m of water in equilibrium pressures, and demonstrates that thermal equilibrium between the two phases cannot be assumed.

The information available is insufficient to form conclusions on the role of acoustic phenomena in the cavitation process, but it may be concluded that calculations based on thermal equilibrium are not valid.

Another approach to the problem of correlation was made by *A. B. Barenboim* [10], who made a complex dimensional analysis of the problem. Then using a process of elimination found the best correlation of *Salemann*'s data when:

(1) $NPIH = F\,(Kr \cdot Re)$

where *Barenboim*'s parameter is

(2) $$Kr = \frac{J\ h_{fg}^2\ v_f}{v_g^2\ C_p\ T\ P_s}$$

However if the data obtained by *W. A. Spraker* [11] are plotted against *Barenboim*'s parameter Kr then correlation is not obtained: The test points of water and methyl alcohol in fig. 2 do not lie on one unique curve. Hence, both *Jakobsen*'s and *Barenboim*'s parameters would appear not to be generally applicable, and this demonstrates that general conclusions cannot be drawn from test data obtained from one pump only.

A new approach to the problem of cavitation has been made by *T. C. Chivers,* and his proposed model, described in the next section, has been successful in predicting the behaviour of the particular pump tested. However, this success must be tempered by the comment made above regarding the drawing of general conclusions from tests on a single pump.

A Model for Cavitation in a Pump

When considering the effect of cavitation on the performance of centrifugal pumps the general assumption is made that similar performance should result for similar vapour to liquid volume ratios (r_v) irrespective of the fluid or its temperature. Further when examining correlation, a universal parameter is sought which considers fluid properties but ignores geometric parameters (for example reference [11]. It would seem that these assumptions cannot be substantiated. A centrifugal pump is essentially a volumetric flow/head generating device, and is capable of handling gas or liquid and producing substantially the same characteristics (there are of course Reynolds number effects). Hence it would appear reasonable to expect a centrifugal pump to behave normally when handling a uniform mixture of gas and liquid, that is when cavitating. This means that one has to consider fluid density in the cavitating region as the primary variable, and this can only be evaluated from energy considerations in their widest sense and not simply from thermal aspects. Accurate determination of the density of a two-phase fluid is extremely difficult and such a measurement in a centrifugal pump impeller has not yet been attempted. Hence assumptions have to be made regarding the cavitation process. Fellowing the photographic work of *E. Grist* [12], it appeared that, for the pump in question, when operating at the best efficiency point the cavitation process could be regarded as one of bubble growth. However, the use of bubble growth theory is restricted in that the initial radius and the number of bubbles are required before fluid density can be calculated.

However from energy considerations, and simply obtained pump test data, it is possible (making assumptions [2]) to arrive at a vapour to liquid volume ratio at a particular temperature. If now it is assumed that bubble nuclei are usually present in the liquid stream; that their initial radius does not change with temperature (this is probably reasonable in a de-aerated liquid); and that the number of nuclei remains constant, then it is possible to estimate the vapour to liquid volume ratio at any other temperature. The actual calculations involved require that some assumptions be made regarding bubble growth, because the actual bubbles are growing and collapsing in an undefined pressure (or superheat) field. This problem can be sidestepped by considering a mean bubble growth rate which

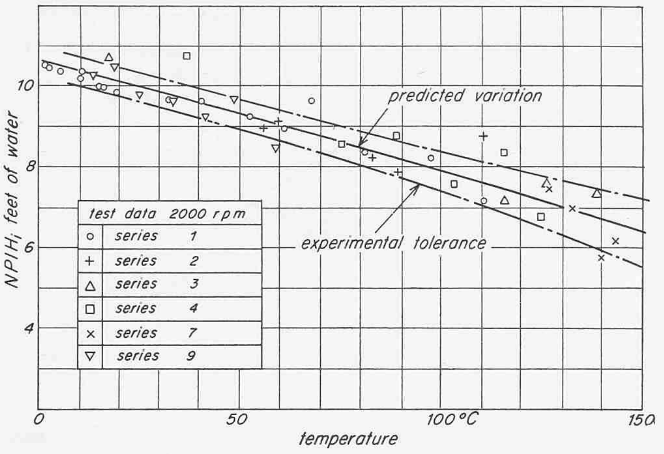

Fig. 4. Inception data compared with the predicted variation with temperature

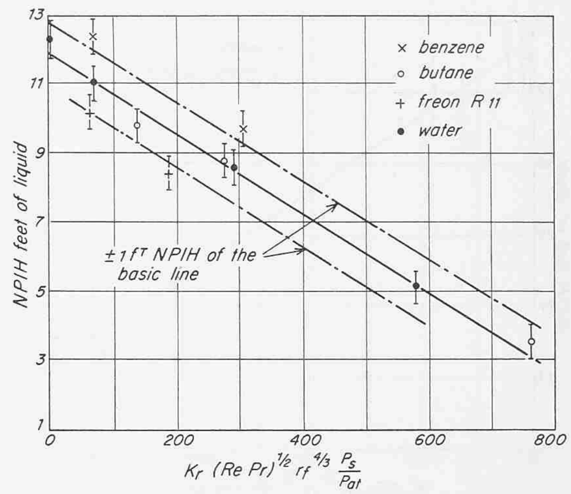

Fig. 5. Salemann's data plotted against the derived parameter [formula (3)]

is simply proportional to the *Rayleigh* [13] solution, which is proportional to time. The assumption is then made that the proportionality between the actual mean growth rate and the Rayleigh solutions does not vary with temperature. This assumption cannot really be substantiated, other than by *Chivers*' final result that the test data agreed with his predictions (Figs. 3 and 8).

Figure 3 shows the predicted variation of $NPIH_b$ together with the experimental tolerance which can be reasonably expected from the test data, using temperature as abscissa. Two solid lines are drawn, both based on test results obtained at 2000 r.p.m. and 16 °C. It is seen that at 2000 r.p.m. excellent agreement exists between the predicted results and the actual experimental data; whereas at 2250 r.p.m. the agreement is not so good below 40 °C. The upper limit of 60 °C at 2250 r.p.m. was dictated by instrumentation problems, and these would have to be overcome before experiments could be conducted to see if the agreement existing between 40 °C and 60 °C was continued at the higher temperatures. One anomaly which does arise is that at 2000 r.p.m. the results indicate no "plateau" (zero slope) in the $NPIH_b$ – Temperature curve as found by others (Figs. 1 and 2 for example) (neither is one predicted), whereas at 2250 r.p.m. the experiments show such a "plateau". It is thought that this is a genuine effect, since the result at 2000 r.p.m. and 2250 r.p.m. were taken concurrently in test series nine, hence the instrumentation cannot be blamed.

The general arguments and assumptions which *T. C. Chivers* has used [1], [2] enable $\Delta NPIH$ (the change in $NPIH$ between inception and breakdown conditions) to be calculated $(NPIH_i = NPIH_b + \Delta NPIH)$. Hence $NPIH_i$ can be calculated from $\Delta NPIH$ and $NPIH_b$. The calculated line is shown in Figure 4, again with the experimental tolerance which can be expected from instrumentation errors; here the data scatter is quite large, but this can be reasonably expected since cavitation inception is not clearly defined from pump test data.

It should be noted that Fig. 4 contains test data from series one and two which has been included by adding the measured $\Delta NPIH$ to the calculated $\Delta NPIH_b$, and it should be regarded as verifying the $\Delta NPIH$ calculations rather than the calculation of inception conditions.

Correlation

The general success of this cavitation model was encouraging, and the problem of correlation seemed an obvious extension. As developed the model assumed a proportionality between actual bubble growth and the Rayleigh solution and this constant will certainly vary with different liquids. Hence the problem was re-examined using the *Plesset* and *Zwick* [13] solution for the growth of vapour bubbles in superheated liquids. This led to the conclusion that cavitation performance for a particular pump could be expressed as:

$$NPIH = f\left\{Kr\,(Pr\,Re)^{1/2}\,r_f^{4/3}\,P_s/P_{at}\right\}$$

This parameter was developed considering breakdown conditions, whereas the only experimental data available for various liquids is for a relatively small head decrease compared with cavitation free conditions. However *Salemann*'s data [9] was correlated to 1 ft (0,3 m) on $NPIH$, Fig. 5 is typical but it is debatable whether it is any better than that using the thermal parameter $1/\beta$, Fig. 6. However when plotting *Sprakers* [11] data neither the derived parameter nor the thermal parameter give correlation although the latter is more successful. It is argued [3] that for the small decrease in head (3 %) due to the onset of cavitation, Reynolds number could possibly have a large influence on correlation, but with the present state of knowledge this is not calculable, in which case the parameter can be written as:

$$NPIH = f\left\{Kr\,(Pr\,Re)^{1/2}\,r_f^{4/3}\,P_s/P_{at}\right\}Re^n \quad (4)$$

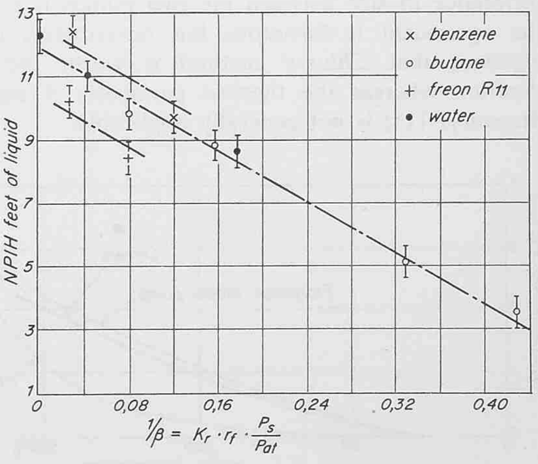

Fig. 6. Salemann's data plotted against the reciprocal of β, the thermal parameter

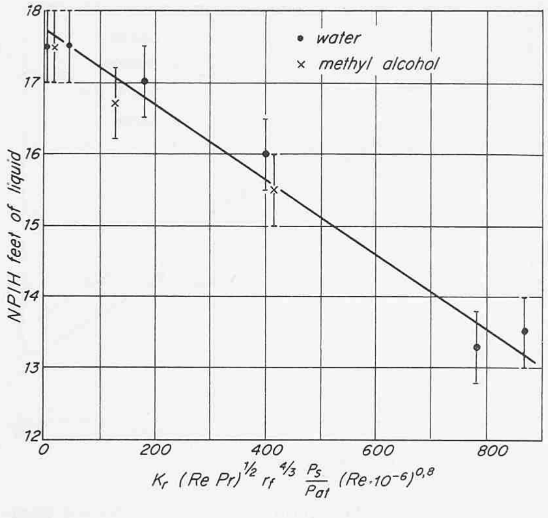

Fig. 7. Data from Spraker's pump 5, plotted against the derived parameter multiplied by Reynolds number to the power of 0.8

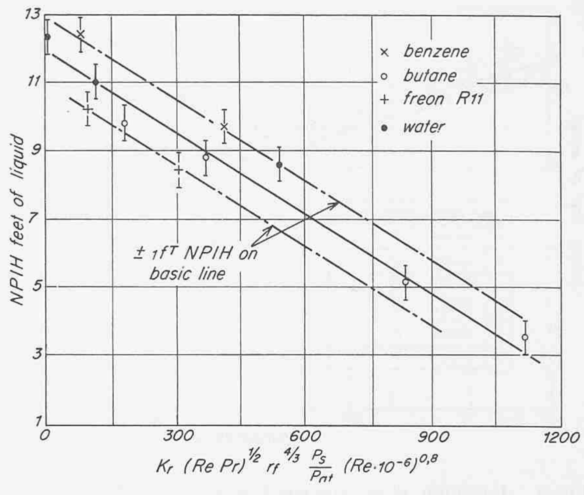

Fig. 8. Data from Salemann's pump 5, plotted against the derived parameter multiplied by Reynolds number to the power of 0.8

Here n will vary with both the degree of cavitation and the actual pump. Using this equation it is found that the correlation of *Salemann*'s data is empirically improved if $n = 0{,}4$, whereas for both of *Spraker*'s pumps if $n = 0{,}8$ complete correlation results (Fig. 7). If $n = 0{,}8$ is used with *Salemann*'s data then the correlation is still an improvement over that obtained with $n = 0$ (Fig. 8). Hence it is proposed that to give the best general correlation of the test data published so far the derived parameter should be multiplied by $Re^{0,8}$.

Chivers has not attempted a universal correlation in which all data, irrespective of pump, reduce to one line, for it is his opinion that cavitation cannot be divorced from the geometry of that which is cavitating. As yet the pertinent geometry is not defined, and the universal plotting of cavitation data is believed to be some way off.

It seems therefore that *Chivers* derived parameter

$$Kr\ (Re\ P_r)^{1/2}\ r_f^{4/3}\ P_s/P_{at} \cdot (Re\ 10^{-6})^{0,8} \tag{5}$$

is worth further investigation as a correlation parameter for predicting changes in cavitation performance for a pump provided its duty at one temperature is known.

It is interesting to note that recent research by *D. Florjancic* [14] on a model boiler feed pump tends to confirm *Chivers*' method of prediction. As seen in Fig. 9 the agreement between $NPIH$ calculated by *Chivers*' results and the experimental results is good. Because of the large difference in size between the two pumps it may be that the agreement is fortuitous but nevertheless it provides evidence that *Chivers*' method is worth further consideration whereas the thermal parameter β suggested by *Stepanoff* [15] is not generally applicable.

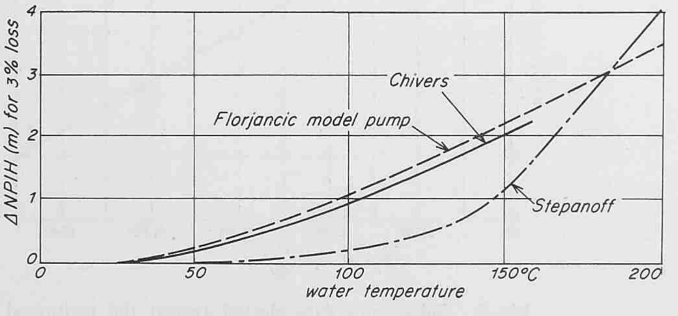

Fig. 9. Comparison of pump test results

Further research

The cavitation test circuit originally at Cardiff University has now been moved to Southampton University where after considerable improvement and modification it has been reassembled. Because of the difficulty of observing cavitation effects in a rotating impeller it was decided to include in the circuit a convergent-divergent test nozzle (two-dimensional) where cavitation could be more easily observed.

The area grading of the nozzle from the throat downstream was similar to that between the inlet and outlet of an impeller blade passage. The test section is liberally equipped with pressure and temperature measuring points and the cavitation from the throat downstream is easily visible.

Experiments so far by *R. A. Furness* show that the change with temperature of $NPIH$, based either on breakdown to 50 % diffuser efficiency or on 3 % loss compared with cavitation free conditions is similar to that found in the pump impeller and experiments are proceeding to establish the applicability of *Chivers*' method prediction applied to this idealized case.

It is clear that we must still await the development of a generally applicable cavitation parameter for pumps but that *Chivers*' method gives a way of calculating $\Delta NPIH$ for any particular case.

Acknowledgements

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Le nouveau stand d'essai universel pour machines hydrauliques de l'École Polytechnique Fédérale de Lausanne

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Préface

Voici bientôt sept ans – c'était au début du mois d'avril 1964 – que nous avons eu, ma femme et moi-même accompagnés de l'un de mes collaborateurs, la grande joie d'avoir été accueillis si chaleureusement au sein de la famille de Monsieur *Charles-Louis Jaeger*, à Rugby, en Angleterre. Notre voyage n'avait cependant pas pour seul but cette aimable rencontre; nous avions encore au programme la visite du laboratoire d'essais de machines hydrauliques de l'English Electric Company (EEC) où C.-L. Jaeger était ingénieur-conseil. Cette visite devait nous renseigner sur ce qui se faisait en dehors de notre pays, en prévision de l'étude et de la réalisation d'un nouveau stand d'essai à l'Institut de Machines Hydrauliques de l'Ecole Polytechnique de l'Université de Lausanne devenue, depuis le début de l'année 1969, l'Ecole Polytechnique Fédérale de Lausanne. Monsieur Jaeger a eu la grande amabilité de nous introduire auprès des responsables du département de recherches et d'essais en laboratoire de machines hydrauliques de l'EEC. Que la dette de grande reconnaissance que j'ai contractée envers mon cher et distingué collègue trouve aujourd'hui son acquittement par ce modeste hommage à son adresse.

1. Généralités

La désignation de *stand d'essai universel pour machines hydrauliques à réaction* se justifie du fait que, d'une part, ce stand unique permet d'essayer *n'importe quel type de machine hydraulique à réaction*, à savoir les turbines radiales *Francis*, turbines diagonales *Dériaz*, turbines axiales *Hélice* et *Kaplan* classiques, turbines axiales *Hélice* et *Kaplan* bulbes, pompes radiales, pompes diagonales *Dériaz*, pompes axiales *Hélice* et *Kaplan*, machines réversibles (pompes-turbines) et que d'autre part, ces huit types de machines hydrauliques peuvent être essayés aussi bien en *circuit ouvert* qu'en *circuit fermé*.

Pourquoi un pareil stand d'essai à l'Institut de Machines Hydrauliques (IMH) de l'Ecole Polytechnique Fédérale de Lausanne (EPFL)? Il y a plus de quinze ans, le soussigné a commencé une étude de *recherche sur la systématisation du tracé d'aubages de turbines Francis*, recherche qu'il entreprit d'abord seul puis, dès l'année 1964, avec l'aide financière du «Fonds pour l'encouragement par la Confédération des recherches scientifiques et techniques», secondé de l'aide technique, sous forme d'essais en laboratoires, des cinq principaux constructeurs suisses de machines hydrauliques (Bell, Charmilles, Escher Wyss, Sulzer et Vevey). Cette collaboration entre le Fonds, les constructeurs et le soussigné a fait l'objet d'une convention signée le 9 février 1966 à l'OFIAMT à Berne. En plus de la recherche sur les turbines Francis, la convention s'applique également à celle de toutes les autres machines hydrauliques à réaction, tant turbines que pompes.

L'IMH ne disposant pas, à ce moment-là, d'un stand d'essai adéquat, les essais effectués sur les différentes plates-formes des dits constructeurs étaient une précieuse contribution à la recherche puisqu'il fallait, le plus tôt possible, pouvoir vérifier par des résultats pratiques la théorie émise à la base du calcul des tracés d'aubage.

Mais en plus, en prévision d'un résultat final aussi précis et fidèle que possible, les ultimes essais devaient pouvoir se faire sur un seul et même stand donnant toute garantie quant à la précision des résultats, donc des mesures. C'est ainsi que fut décidée la création d'un nouveau stand d'essai à l'IMH répondant entièrement aux exigences requises.

L'effort financier considérable de 4,43 Mio fr. que demandait ce projet, tant en ce qui concerne le gros œuvre que le stand d'essai proprement dit – l'équipement du stand d'essai était devisé à 2 Mio fr., montant qui sera atteint mais pas dépassé – nécessitait l'aide de plusieurs bailleurs de fonds, dont la Confédération, le canton de Vaud, la commune de Lausanne, la Société d'aide aux laboratoires de l'EPUL et un certain nombre d'exploitants de centrales hydro-électriques de la Suisse romande et de la Suisse alémanique.

Cependant, la recherche à elle seule ne pouvait justifier un pareil investissement. Il fut décidé d'utiliser ce stand également pour des essais pour des tiers, à savoir pour des constructeurs de machines hydrauliques, pour des mandataires ou bureaux d'ingénieurs-conseils d'exploitants de centrales hydro-électriques, etc.

Il n'est pas possible, dans un cadre aussi restreint que celui du présent article, d'entrer dans les détails de tous les éléments constituant le nouveau stand d'essai. Une description très détaillée fait l'objet de la Publication No 6 de l'IMH qui vient de sortir de presse, en français et en anglais. Notre description se limitera donc aux éléments principaux du stand d'essai.

2. Description du stand d'essai

L'installation complète du stand d'essai est reproduite sur la figure 1. Elle comprend essentiellement les *circuits hydrauliques* proprement dits du stand, les *éléments principaux* de ces circuits, les *éléments auxiliaires*, la *plate-forme d'essai*, élément principal du stand. Nous traiterons dans cet ordre ces différentes parties de l'installation.