Cavitation in Water Jet Pumps

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Based on the similarity law of jet pump, a new cavitation parameter \( \sigma \) is proposed which has the similar meaning to Thomas's cavitation parameter in turbo pumps. The \( \sigma \) values are determined theoretically and presented as the function of specific speed \( n_{s} \) so that we may predict cavitation directly by using it in the designing. Experiments are carried out and \( \sigma \) values are confirmed. Comparing these theoretical values with the data previously reported, it is recognized that these are reliable.

1. Introduction

In designing a centrifugal pump, whether the performance of the pump will be affected by cavitation or not can be predicted by the use of a diagram or formulas expressing the relationship between the cavitation parameter and the specific speed.

A number of studies have been made on the cavitation in jet pumps.\(^1\)\(^-\)\(^7\) Several formulas of cavitation parameter can also be shown, and particularly Cunningham et al.\(^6\) explained the relationships among the eight cavitation parameters each differently defined.

No theoretical treatment, however, has been made of the mechanism of the generation of cavitation, nor are there any data that can be widely used for designing jet pumps. The aim of this study has been to obtain both theoretically and experimentally the coefficient of cavitation which affects the performance of a pump, and by plotting this against the specific speed \( n_{s} \), establish design data which will be generally applicable to jet pumps in the same way as the characteristic diagram of the centrifugal pumps.

2. Notations

\[
\begin{align*}
A \quad & = \text{Cross section} \\
A_0 \quad & = \text{Cross section of the shear layer at the inlet of the throat} \\
D \quad & = \text{Diameter} \\
\eta \quad & = \text{Specific speed} = Q_0/\left(\rho H_0^{1.5} \right) \\
\rho_0/\rho_1 \quad & = \text{Caliber ratio} \\
K \quad & = \text{Cavitation parameter in the shear layer} \\
K_1 \quad & = \text{Time-average pressure drop coefficient} \\
M \quad & = \text{Flow ratio} = Q_2/Q_1 \\
N \quad & = \text{Head ratio} = (H_1-H_2)/(H_1-H_3) \\
Q \quad & = \text{Volumetric flow} \\
p \quad & = \text{Pressure (static)} \\
p_0 \quad & = \text{Pressure outside the shear layer}
\end{align*}
\]

\( \text{(} p_0 = p_{atm}, \text{ at the throat inlet ascertained by the flow model) } \)

\( \text{p}_s \quad = \text{Suction chamber pressure (the velocity at this point is negligible.) } \)

\( \text{p}_\infty \quad = \text{Minimum local pressure in the throat-inlet cross section} \)

\( r \quad = \text{Radius} \)

\( r_c \quad = \text{Distance between the point of the minimum local pressure and the center of axis} \)

\( u \quad = \text{Velocity} \)

\( u_\theta \quad = \text{Speed along the axis} \)

\( u_{2c} = Q_2/(A_t-A_0) \)

\( u_m = (Q_1+Q_2)/A_0 \)

\( y \quad = \text{Radial distance from the center of the shear layer whose flow rate is \( (u_\theta+u_2)/2 \); positive in the outward direction} \)

\( \alpha = (u_{1b}/u_\theta)^2 \)

\( \gamma \quad = \text{Specific weight of a fluid} \)

\( \zeta = 2(z_2-P_{sw})/(P_\infty) \)

\( \eta = M-N \)

\( \delta \quad = \text{Half of the thickness of the shear layer} \)

\( \rho \quad = \text{Fluid density} \)

\( \sigma \quad = \text{Cavitation parameter} = (H_1-(p_0/\gamma))/(H_1-H_2) \)

Subscripts

1 \( = \text{First fluid (Driving fluid)} \)
2 \( = \text{Second fluid (Pumped fluid)} \)
3 \( = \text{Mixture of fluids or jet pump outlet} \)
\( m \quad = \text{Muzzle or nozzle-outlet cross section} \)
\( t \quad = \text{Throat or throat-inlet cross section} \)
\( w \quad = \text{Wall} \)
\( \bar{m} \quad = \text{Mean value} \)

3. The law of similarity and general performance

The authors established the law of similarity applicable to the jet pumps by dimensional analysis,\(^8\) and experimentally proved that the general performances of identical pumps mutually meet well when expressed using the established similarity variables\(^8\) \((M, N, \eta \text{ and } n_0)\). In order to collect data which can be used for designing efficient jet pumps, we also carried out experiments on jet pumps, and found that the throat length, the nozzle-
to-throat distance, diffusion angle, and caliper ratio $D_0/D_1$ all affect jet pump performance and have optimum values.\(^{(a)}\) It was also shown that, although a change in the caliper ratio results in a significant alteration in the specific speed $n_{s1}$ at the maximum efficiency, no change in any other dimensions alters $n_{s1}$ to a large extent.\(^{(a)}\) A design constant diagram was also prepared by plotting $M$, $N$, $\eta$, and $D_0/D_1$ versus $n_{s1}$. Fig. 1 shows a design constant diagram revised by adding some more data obtained later.

![Fig. 1. Design constant diagram](image)

4. Definition of the cavitation parameter

In the study of cavitation in the centrifugal pump, Thoma showed that the head drop $\Delta H$ in the impeller can be expressed as the total lift multiplied by some coefficient and named it cavitation parameter, which is based on the concept that under similar operating conditions the pressure distribution in pumps becomes also similar. Based on the same concept, the velocity head, instead of $H$, is used for the external flow.

The same law of similarity of pressure distribution as above is certainly applicable to the jet pumps in view of the similarity rule applicable to them. The jet pump differs from others in that, since the pumping effect is generated by mixing of two fluids, various heads can be used as the divisor $H$ for an equation $\sigma \equiv \Delta H/H$, for example, the jet velocity head of the first fluid, the velocity head of the second fluid, or the velocity differential head of these fluids.\(^{(a)}\)

These parameters, however, cannot be used as readily as the cavitation parameter of the centrifugal pump because they do not express direct relationship with the lift $H_1-H_2$ of the second fluid which receives the pumping effect. Moreover, the relationships among these parameters are somewhat uncertain because they vary according to whether the nozzle-outlet static pressure is assumed to be equal to the suction chamber pressure $p_1$ or it is assumed to be equal to the throat inlet pressure $p_t$.

The authors, instead, propose a cavitation parameter $\sigma$, expressed in the following equation, which can be directly used and whose physical significance is easily understood.

$$\sigma = \frac{H_1 - p_1}{T}$$

where $H_2$ is the total head of the second fluid at the inlet of the jet pump; $H_1-H_2$ is the total lifting head of the second fluid in the pump; and $p_1$ is the saturated vapour pressure of the second fluid. The equation shown above can be regarded as of the same type as that of Thoma's cavitation parameter $\sigma$.

5. The mechanism of the generation of cavitation

Bonington pointed out that cavitation tends to occur at two places in the water jet pump,\(^{(b)}\) i.e., one at the throat inlet and the other at the diffuser inlet. (Refer to Fig.6.) It was ascertained that cavitation occurs at the diffuser inlet only when the flow ratio is greater than that at the design point. It was also confirmed by the observation of flow that cavitation occurs at the throat inlet when the pump is operated at a high performance rate adopted in its design. This paper will, therefore, discuss the cavitation generated at the throat inlet only.

For the flow pattern produced in the course of mixing in the throat by the two fluids after coming out of the nozzle and the suction chamber, Barchion et al.\(^{(c)}\) and Razinsky et al.\(^{(d)}\) devised flow models for the case where the axial position of the nozzle outlet and that of the throat inlet concur, and described their experiment.\(^{(e)}\)

Although on the basis of their descriptions a slight pressure drop in the axial direction can be explained, no explanation of the pressure change in the radial direction was given in detail. Rouse made public a number of valuable reports on the cavitation generated in a turbulent shear layer occurring between two parallel flows of a large velocity difference.\(^{(f)}\)

According to these reports, the average initial cavitation parameter in a free jet in water can be $K=0.55$, where $K$ is defined as $K=[p_0-p_1]/(\rho_0 u_0^2/2)$, (where $p_0$=absolute pressure around the jet; $p_1$=saturated vapour pressure; and $u_0$=velocity of the jet). It was also reported that a slight cavitation occurred even when the cavitation parameter $K$ was as high as approximately 0.7. The reports also said that as the $K$ value decreased, the time and intensity of cavitation increased.

Although Rouse's reports were related to free jet, it can be considered that even with a jet in the confined region such
as in the jet pump the vortex distribution in the shear layer cannot differ greatly from that reported by Rouse. Particularly, the above-given value can be generally applied to the region taken up in this paper, i.e., between the nozzle outlet and the throat inlet, because the outer edge of the jet shear layer is not in contact with the solid wall.

5.1 Pressure drop in the shear layer explained as due to turbulence

As was explained by Townsend, (12) who introduced a variable component of velocity into Navier-Stokes' equation, and was referred to by Razinsky et al., (11) there is a local static-pressure drop expressed as \( p_0 - p = v^2 \), corresponding to the radial turbulence \( v^2 \). The effect of this local static-pressure drop, however, is very small, and the values based on numerous experimental data are all too small to be compared with Rouse's cavitation parameter mentioned earlier.

5.2 Pressure drop in the shear layer explained as due to vortex ring

Since this study purports to clarify the condition under which the pump performance starts to decline on account of cavitation, it was necessary to obtain the \( K \) value under this condition. The authors considered that intermittent cavitation occurs, as described above, at \( K = 0.55 \), but as \( K \) becomes smaller the cavitation is continuously generated in the center part of the shear layer, and the cavitation in the case of a circular jet develops over a cylindrical surface, causing the slipage between the first and the second fluids to increase. Thus, because the transmission of momentum via this cylindrical surface is extremely reduced, and, at the same time, the entropy of water increases due to the cavitation, the pump performance lowers.

The authors defined such pressure drop coefficient as \( K_1 = (p_0 - p)/\gamma(u_1 - u_2)^2/2 \) in accordance with that for free jet, and obtained the average vortex center pressure which corresponds to the time-average velocity fluctuation, as follows. Namely, by referring to Fig. 2, the thickness of the shear layer was designated as \( 2 \delta \), and the radial distance from the center of the shear layer measured outward was made the positive \( y \). Let us explain this assuming a 2-dimensional shear layer, as an example.

\[
\begin{align*}
  u &= \frac{u_1 + u_2}{2} - \frac{u_1 - u_2}{2} \left( \frac{3}{\gamma} \right) y + \frac{1}{2} \frac{y}{\gamma} \\
  \end{align*}
\]

(1)

and a vortex ring is assumed to be moving at a speed \( (u_1 + u_2)/2 \), around the middle of the shear layer as the center with the rotational angular velocity expressed as \( (u_1 - u_2)(1.5(y/\delta) - 0.5(y/\delta)^3)/2y \), then the pressure \( p \) at a radius \( r \) is given as

\[
\begin{align*}
  p_0 - p &= \frac{\rho}{4} (u_1 - u_2)^2 \\
  \int_0^\infty (1.5(y/\delta) - 0.5(y/\delta)^3) dy \\
  &= 0.396 \frac{\rho}{2} (u_1 - u_2)^2 \\
  \end{align*}
\]

(2)

Hence

(3)

and \( K_1 = 0.396 \) can be considered appropriate as compared to the intermittent cavitation parameter \( K = 0.55 \).

Thus, the pressure drop in the cross section of the flow is ascertainment, and the amount of time-average pressure drop is clarified. The following is a discussion on how the amount of static pressure drop caused by a rise in the dynamic pressure of the second fluid can be obtained on the basis of this radial pressure distribution using a flow model described below.

5.3 Flow pattern between the nozzle outlet and the throat

The configuration of a jet pump between the nozzle outlet and the throat is generally as shown in Fig. 2 or Fig. 3. The outer edge of the shear layer is in contact with the wall at the throat (the parallel portion). The reason for this is, as clear from the engineering consideration given by Ueda, if the outer edge of the shear layer contacts the funnel-shaped portion (called the diffuser guide by Ueda) located before the throat, the axial resistance to the flow increases and hence
it lowers the overall efficiency of the pump. Conversely, if the outer edge of the shear layer goes deep into the throat, the distance between the nozzle outlet and the throat inlet becomes short. As a result, the passage of the second fluid formed by the outside surface of the nozzle and the funnel guide is narrowed; the resistance to the second fluid increases; and the overall efficiency lowers. In addition, as pointed out by Sanger, if the distance between the nozzle outlet and the throat inlet is small, the cavitation performance of the pump is reduced.

Since we are now discussing the cavitation in the jet pumps of high performance, here we assume, for reasons given above, that the outer edge of the shear layer is in contact with the wall at the throat inlet.

5.4 Equation of momentum

In order to obtain the wall pressure \( P_{\text{wall}} \) at the throat inlet using the above mentioned flow model, the law of momentum was applied to the space in the suction chamber (the hatched section) shown in Fig. 3. In this case, because the momentum of the inflowing second fluid is negligible, the pressure on the nozzle outlet plane is put as \( p_2 \). The flow near the wall of the funnel guide is a potential flow, whose velocity distribution can be considered almost the same as that of a single flow (the second fluid flow alone) determined by \( p_2 \) and \( P_{\text{wall}} \). Hence, the pressure distribution on the funnel plane can be approximated using the pressure distribution of a single flow determined by \( p_2 \) and \( P_{\text{wall}} \). The following momentum equation may be obtained.

\[
A_2(2p_2-p_{\text{wall}})+p_{\text{wall}}A_2 = \int_{A_1} p\,dA + \rho \int_{A_1} u^2\,dA \tag{4}
\]

5.5 Cavitation parameters of jet pumps with different caliber ratios

5.5.1 Jet pumps with large caliber ratio

The equation of continuity is

\[
Q_1+Q_2=\int_{A_1} u^2\,dA=\int_0^{r_1-2\delta} u_1^2\,2\pi rd\theta
+\int_{r_1-2\delta}^{r_1} \left[ \frac{u_1+u_2}{2} - \frac{u_1-u_2}{2}\left[ 1 + \frac{r_1^2 - r_i^2}{2} \delta \right] \right] \frac{1}{\delta} \frac{1}{2} \pi r^2 dr
- \frac{1}{2} \left( \frac{r_1^2 - r_i^2}{\delta} \right)^{1/2} \pi r_1^2 \tag{5}
\]

Since in the case of a large caliber ratio, \( \delta \) is small as compared to \( r_1 \), the above equation can be approximated as

\[
Q_1+Q_2=(A_1-A_2)u_1+A_2(u_1-u_2)/2
\]

Because such approximation as above is advantageous from the engineering point of view — it simplifies the calculations — a similar approximation can also be made for the integration of momentum and pressure.

The above equation is rewritten as

\[
A_2=\frac{u_1A_1-(Q_1+Q_2)}{(u_1-u_2)/2} \tag{5}
\]

The second term of the right member of the equation (4) is calculated using the equation (1) of velocity distribution,

\[
\rho \int_{A_1} u^2\,dA = \rho \left[ \frac{u_1^2}{2} (A_1-A_2) + \frac{(u_1+u_2)^2}{2} \right] + 0.486 \left( \frac{u_1-u_2}{2} \right)^2 A_2 \tag{6}
\]

Likewise, the first term of the right member of the equation (4) is calculated by use of the equation (2) of pressure distribution,

\[
\int_{A_1} p\,dA = p_2A_2-0.243 \frac{\rho}{2} (u_1-u_2)^2 A_2 \tag{7}
\]

If we substitute the equations (5), (6) and (7) for the equation (4), rewrite the equation (4) using the equation of continuity, put the nozzle outlet static pressure as \( p_2 \), and apply Bernoulli's equation to both potential regions between the nozzle-outlet cross section and the throat-inlet cross section, then the following equation is obtained.

\[
\zeta + a \frac{A_2}{A_1} + 0.243 \left( \sqrt{\zeta + a} - \sqrt{\zeta} \right) = 0 \tag{8}
\]

where \( \zeta = 2(p_2-p_{\text{wall}})/(p_{\text{wall}}^2) \), and \( a = (u_{1R}/u_{2R})^2 \).

As stated in Section 4, the cavitation parameter \( \sigma \) is based on the law of similarity of pressure distribution, and when \( p_2 \) becomes \( P_{\text{wall}} \), the reduction in performance on account of cavitation can be considered to start. Also, because the dynamic pressure of the second fluid in the suction chamber is negligible as compared to \( p_2-P_{\text{wall}} \), the following equation holds.

\[
\sigma = \left( (p_2-p_{\text{wall}}) + (p_{\text{wall}}-p_1)/(\gamma(H_1-H_2)) \right) \tag{9}
\]
If we substitute the equation (3) for the equation (9) by applying $\xi$ obtained by the equation (8) and $a$ used in the same equation, and use $n_{s1} = Q_2/\left[D_1^2 (H_1/H_2)^{1/2}\right]$ (Unit: $k/W/m^2$), then we have

$$\sigma = 8 \frac{9.81 r_t^2}{\sqrt{\sigma + (\xi - \sqrt{\xi})}} \left(\frac{D_1}{D_2}\right)^4 \left(1 + \frac{1}{M}\right)^{10^4}$$

Since $n_{s1}$ and $M$ at the design point of jet pumps with different caliber ratios can be determined by the use of the design constant diagram of Fig. 1, discussed in Section 3, the cavitation parameter at the design point of the jet pump with different caliber ratios can be obtained.

### 5.5.2 Jet pumps of small caliber ratio

If calculations are carried out by the process shown above, $A_4$, $A_5$, $A_8$ at $D_2/D_2$ of 0.6, namely, the potential region of the first fluid ceases to exist at the throat inlet. Although various approximation formulas can be used for the velocity distribution of the shear layer whose potential region of the first fluid has ceased to exist, here the 3/2 power velocity distribution, used by Razinsky et al., (11) is adopted. According to this, the velocity at the radius $r$ of the throat inlet can be expressed as

$$u = u_0 + (u_0 - u_2) \left(1 - \frac{r}{r_0}\right)^{3/2}$$

and the equation of continuity, the equation (6) of the term of momentum, and the equation (7) of the term of pressure can be rewritten, respectively, as

$$Q_1 + Q_2 = \int_{u_i}^{u_0} w dA = \int_{A_i}^{A_0} \left[\frac{u_i}{u_i - u_2}\right] \left[1 - \left(\frac{r}{r_0}\right)^{3/2}\right] 2\pi r dr$$

$$\rho = \rho_0 + \int_{A_i}^{A_0} \left[\frac{u_i}{u_i - u_2}\right] \left[1 - \left(\frac{r}{r_0}\right)^{3/2}\right] 2\pi r dr$$

$$= \rho_0 + \frac{1}{0.5} \left[\frac{u_0}{u_0 - u_2}\right] \left[1 - \left(\frac{r}{r_0}\right)^{3/2}\right] 2\pi r dr$$

$$= 0.2571 \left[\frac{u_0}{u_0 - u_2}\right] \left[0.5 - 0.8382\right]$$

The pressure distribution in the throat-inlet cross section can be calculated, if the pressure is assumed to be equal both at the center and on the wall, and an average vortex corresponding to the velocity distribution is assumed to exist as stated before. The equation (14) is the outcome of this calculation. By Substitution of the equations (12), (13) and (14) for the equation (4), $\xi \leq 2(\rho_2 - \rho)\left(P_2 - P_{1}\right)/(\rho_2 u_2)$, used in the equation (8), becomes

$$0.864 \xi - 0.780 \xi + 1/\left[(1 + M)(D_1/D_2)\right]$$

$$= 0.0832$$

If the pressure distribution used for obtaining the equation (14) is applied, the equation to replace the equation (3) will be

$$\rho w = \frac{\rho_0}{2} (u_0 - u_2)$$

and the equation to replace the equation (10) will be

$$\sigma = \frac{8}{9.81 r_t^2} \left[\xi + 0.3652 \left(\frac{1}{1 + M} \frac{D_2}{D_1}\right)^4 \left(1 + \frac{1}{M}\right)^{10^4}\right]$$

The solid line in Fig. 1 is obtained by either of the equation (10) or the equation (17).

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### 6. Experiment

A schematic representation of the apparatus used for the experiment is shown in Fig. 5. The driving water pressure was measured by a triple U-tube connected to a tap on the side wall of the lead-in pipe located immediately before the jet pump inlet, and, when the pressure exceeded the working range of the U-tube, by a Bourdon tube pressure gauge. The pressure in the suction chamber and the discharge pipe was determined by the pressure difference from the atmospheric pressure measured by the U-tube. For the atmospheric pressure the mean value of the readings on a standard barometer taken before and after the experiment was adopted. For measuring both the driving-water flow and the driven-water flow, the pressure readings on a U-tube before and after an orifice were taken to obtain the differential pressure. Also, to ensure the accuracy of measurement, different orifices were used for various flows. The water discharged from the jet pump was returned to the main tank of approximately 30 m$^3$.

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Fig. 5. Flow-chart of the apparatus

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based on those of water measured in this tank. According to the measurements made by the method used by Hoshimi, (17) the air content was at all times an approximately 0-4% supersaturation by volumetric ratio.

The jet pump proper was provided with a constant throat diameter of 20 mm, and its caliber ratio $D_p/D_d$ was altered by changing nozzles. Fig. 6 shows an example for a caliber ratio of 12/20. As is clear from the discussion of the flow pattern in 5.3, when the caliber ratio is changed, the optimum nozzle-to-throat distance changes. Therefore, in the experiment an appropriate (high performance) profile was maintained by changing blocks cut perpendicular to the axis. Because all the blocks were made of transparent methacylate, resin, cavitation could be observed.

![Fig. 6. An example of jet pump proper](image)

6.2 Cavitation in jet pumps of different caliber ratios

Let us change the caliber ratio $D_p/D_d$ (hence change the specific velocity $u_d$), determine the capacity ratio corresponding to this specific velocity by the use of Fig. 1, set it as the design point of the jet pump, make $M$ constant for identical pumps, repeat the experiment, described in 6.1 above, and show the results with $D_p/D_d$ as a parameter, then a graph as shown in Fig. 7 can be drawn.

![Fig. 7. N-sigma curves of jet pumps with different caliber ratios](image)

7. Comparisons with reported test results

By the use of the cavitation parameters reported in the past, it is difficult, as discussed in Section 4, to precisely calculate the sigma propounded in this paper. Let us though strict accuracy cannot be expected compare them, making some comments on them. In his experiment Sanger (14) used a completely deaerated water, changed the nozzle-to-throat inlet distance of a jet pump of $(D_p/D_d)^2 = 0.197$, altered the flow ratio $M$, and ascertained the cavitation parameter, defined as $u_d(\eta_{H_2-p_v}/\sqrt{g})$, to be 0.11. The values of $M$ and $N$ at the design point were used in the calculation of the sigma, and the plotting, marked with a square in Fig. 4, was obtained. The value, however, was found excessively...
small even if the water used was completely deaerated.

Hansen and Na\(^{(5)}\) carried out an experiment on a jet pump of \((D_0/D_0)^2=0.25\), and showed that the cavitation parameter, defined as \(\sigma^*_2(H_2-p_2)/\nu_{2c}^2/(2g)\) (\(\nu_{2c}\) is the velocity calculated by \(\nu_{2c}(A_1-A_2) = q_2\), was 1.36. No clear indication, however, of the profile from the nozzle outlet to the throat inlet was given by them. If it can be assumed that the nozzle-outlet plane and the throat-inlet plane agree, \(\sigma^*_2=1.18\) can be obtained using the values of \(M\) and \(N\) at the design point. On the assumption that the nozzle outlet plane is well upstream of the throat inlet plane and the static pressure at the nozzle outlet is equal to \(p_2\), \(\sigma=0.995\), the plotting marked with a \(\bigcirc\) mark is obtained which shows good agreement between the theoretical and the experimental values given in this paper.

Mueller\(^{(2)}\) altered the velocity ratio of a jet pump of \((D_0/D_0)^2=0.48\) in his experiment, and obtained \(\sigma^*_2H_2/\nu_{2c}^2/(2g)=0.38\) at \(\nu_{2c}/\nu_1=0.36\). Using the values of \(M\) and \(N\) at the design point, the plotting marked with a triangle was obtained (in this case, the saturated vapour pressure was very small and therefore was neglected). It can be considered that the excessively large sigma value was due to the profile of the passage of the second fluid between the nozzle’s outside surface and the funnel-shaped guide, as shown in Fig.8.

**Fig. 8. Mueller’s nozzle**\(^{2}\)

Cunningham et al.\(^{(6)}\) obtained \(\sigma^*_2(H_2-p_2)/\nu_{2c}^2/(2g)=1.10-1.33\) by experiment on jet pumps of three area ratios, i.e., \((D_0/D_0)^2=0.295, 0.39\) and 0.50. If the conversion similar to those made above is carried out, it will be found that the plotting falls within the hatched area in the figure.

Bonnington\(^{(1)}\) conducted an experiment on two jet pumps of different caliber ratios, i.e., \((D_0/D_0)^2=0.5\) and 0.75, and obtained the cavitation ratio \(\sigma_2^*\) over a wide range of velocity ratios. The readings were 0.19 and 0.36 for jet pumps of these caliber ratios at their design points. Conversion was made as in the case of Mueller’s by neglecting the saturated vapour pressure, and the result was plotted and marked with a rectangle in the figure. The case where the static pressure at the nozzle outlet is equal to \(p_2\) was included, just the same as in the conversion of the data by Hansen.

Except Sanger’s low value data and Muller’s excessively high value data for which the cause is clear, all data can be considered to agree well.

8. Conclusions

The conclusions can be summarized as follows.

1) The authors have presented a cavitation parameter, i.e., \(\sigma^*_2(H_2-p_2)/H_2\), which is readily usable and similar in significance to the cavitation parameter for the centrifugal pumps.

2) The mechanism of static pressure drop in the region from the suction chamber to the center of the shear layer in the throat-inlet cross section has been theoretically analyzed, and the sigma-\(n_{q3}\) curve is obtained using the values of the flow ratio, the head ratio and the caliber ratio corresponding to the specific speed \(n_{q3}\), as has been proposed by the authors. This sigma-\(n_{q3}\) curve, as is the sigma-\(n_2\) curve of the centrifugal pumps, is handy and useful for directly judging at the designing stage whether or not a performance decline due to cavitation will occur.

3) The accuracy of the abovementioned sigma-\(n_{q3}\) curve has been verified by experiment.

4) As a result of comparison with the data reported in the past, the present sigma-\(n_{q3}\) curve has been shown to agree with them except some.

References

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