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DISK FRICTION LOSS IN CENTRIFUGAL AND MIXED FLOW PUMPS

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1. ABSTRACT

Disk friction represents a high proportion of total loss in pumps, specially the radial type, and hence it has a great effect on pump efficiency. The present work introduces a new approach to deduce a new formula for evaluating disk friction loss in pumps taking into consideration the effect of the specific speed (i.e. impeller shape) and the capacity (i.e. pump size). The used approach relies upon using the available well-known dimensionless, experimental data for the best-known pumps at their design point as function of specific speed, such as the kinematic velocity ratios, the geometric ratios and the overall efficiency. A comparison between all the available data was done in order to select the most common and consistent data to be used in the evaluation. Using these data, a procedure was set to evaluate the other types of losses, mainly hydraulic, volumetric and mechanical losses as function of the specific speed and the capacity, and hence disk friction loss is obtained. The adopted procedure was essential in order to select the most appropriate values of some parameters, such as blade outlet angle, which have a great effect on getting reasonable and consistent values of the different types of losses.

2. INTRODUCTION

Disk friction loss in a pump is caused by the friction between the fluid flow in the gap between the impeller and the casing wall as shown in Fig. 1. Disk friction loss plays an important role in determining its overall efficiency, especially in low specific speed and small capacity machines, particularly in centrifugal and mixed flow pumps and compressors. Evaluation of disk friction was studied by many researchers, theoretically and experimentally and there are many empirical relations introduced to evaluate this type of loss. However, there are great discrepancies between the values of disk friction loss calculated by these relations for a given pump where differences may reach as much as three times the least values.

The classical experimental method of determining the disk friction loss is by using a thin plain disk with constant thickness rotating in a close fitting closed casing, while in most cases of actual pumps, the outer hub and shroud surfaces are neither radial nor complete. In case of high specific speed pumps, the shroud is limited to small radial width and the clearances between impeller and casing are rarely constant or uniform.

In the previous research work on the subject of disk friction loss, some of the disk friction loss formulae included the effect of both the outside surface roughness of the hub & shroud and the axial spacing between the shroud and the hub and the casing. These parameters were considered the two major factors that affect the disk friction power loss value. In previous work none of the researchers investigated whether the specific speed and the size of the pump have any effect on disk friction loss, although the specific speed is considered one of the most important factors which affect the pump performance and characteristics.

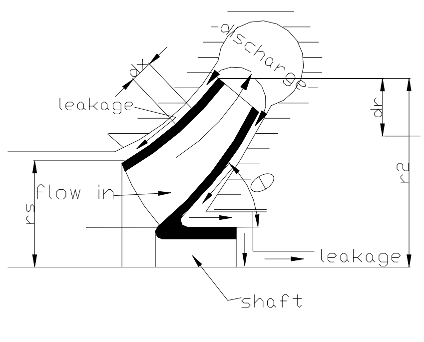

Fig. 1 disk friction loss in centrifugal pumps

So, a new approach for estimating the disk friction loss is adopted is this paper. This approach is based on the one-dimensional theory of turbo-machines, together with the available experimental data published by several well-known authors.

The available experimental data are those of the best overall efficiency of pumps of various specific speed values and capacity (rate of flow through the pump at its b.e.p.), together with optimum values of the various kinematic velocity and geometrical ratios of the tested machines.

The target of this work is to obtain the optimum values of pump hydraulic, volumetric, mechanical and disk friction efficiencies, at different values of specific speed and capacity of water pumps (for which the available published data is known). Consequently from these results a semi-empirical formula for estimating the coefficient of disk friction is reached as function of the value of specific speed and capacity of the pump, together with the roughness ratio of the outer surfaces of the hub and shroud, spacing ratio between impeller and casing and the Reynolds number.

3. CALCULATION PROCEDURE AND FORMULAE USED

In general $\eta_o = \eta_h \times \eta_v \times \eta_m$

also $\eta_m = (SHP - (P_B + P_{d.f.}))/SHP$ (1)

Where P_B is the power losses in bearing and stuffing box

$P_{d.f.}$ is the power loss due to disk friction

Stepanoff [11] and others stated that P_B is nearly equal to 1% from the shaft power.

Disk friction loss is often expressed in terms of the coeff. of disk friction Kd, where $P_{d.f.} = Kd \times \rho \times N^3 \times D^5$ (2)

Where the coefficient Kd is dimensionless.

From the one-dimensional theory of turbo-machines where the flow conditions on the mean meridional streamline represent an average of the complete flow conditions, and in terms of Ku_{2m}, Kcm_2, σ and β_2, and from the outlet meridional velocity diagram as shown in Fig. 2,

$$\eta_h = \frac{g \times H_m}{Cu_2 \times U_2}$$

$$\eta_h = \frac{1}{2 \times \sigma \times Ku_{2m}^2 (1 - \frac{Kcm_2}{Ku_{2m}} \times \cot \beta_2)} \quad (3)$$

And the volumetric efficiency

$$\eta_v = \frac{\dot{Q}_{act}}{\dot{Q}_{th}}$$

Where $\dot{Q}_{th} = \pi \times D_{2m} \times b_2 \times Cm_2 \times B$, $Ns = \frac{2\pi N \times \sqrt{\dot{Q}_{act}}}{(gH_m)^{\frac{3}{4}}}$

$$\eta_v = \frac{Ns^2}{8\pi\sqrt{2} \times B \times Kcm_2 \times Ku_{2m} \times Ku_{2o} \times Kb_2} \quad (4)$$

And the mechanical efficiency

$$\eta_m = \frac{\eta_o}{\eta_v \times \eta_h},$$

$$\eta_m = 1 - P_{d.f.}\% - P_B\% \cong 0.99 - P_{d.f.}\% \quad (5)$$

$$\therefore Kd = \frac{1.37\,(0.99 - \eta_m) \times Ns^2}{Ku_{2o}^{5} \times \eta_o} \quad (6)$$

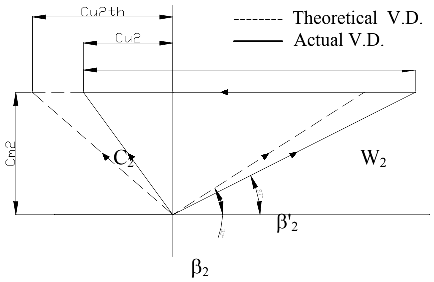

Fig. 2 outlet velocity diagram of centrifugal pump

4. EMPIRICAL ESTIMATION OF VARIOUS TYPES OF LOSS INCLUDING DISK FRICTION

The main published data which will be used in the present evaluation are the kinematic velocity ratios (Ku_{2m}, Ku_{2o}, Kcm_2), the geometric ratios (Kb_2, β_2 and σ) and the overall efficiency (η_o). These data are published by many researchers such as Stepanoff, 1957 [11], Karassik, 1986 [6], Daily and Nece, 1960 [3] and Bohl, 1980 [1].

Overall efficiency

Wislicenus, 1947 [13], Stepanoff, 1948 [11], Karassik, 1986 [6] and Bohl, 1980 [1] published results of tests on hundreds of well designed and good executed (specially castings) pumps.

Results of Stepanoff [11] and Bohl [1] in the form of η_o versus Ns and Q/Qr are shown in Fig. 3.

The results of Wislicenus [13] are exactly equal to those of Stepanoff [11], and the results of Karassik [6] cover a limited range of specific speed (0.2 – 0.6) only.

In order to express the capacity as a dimensionless ratio, Q has been devided by the maximum value of Qr recorded by Stepanoff [11] and the capacity ratio is Q/Qr, where Qr is equal to 0.63 m3/s.

Kinematic velocity ratios Ku_{2m}, Ku_{2o} and Kcm_2

Both Stepanoff [11] and Bohl [1] gave values of Kinematic velocity ratios as function of specific speed only (and not of the capacity). The Kinematic velocity ratios Ku_{2m}, Kcm_2 and Ku_{2o} are shown in Figs. 4 & 5 & 6 respectively.

Outlet Blade angle β_2

Stepanoff [11] did not show any specific pattern of variation of β_2 with Ns, and mentioned only that for best overall efficiency, outlet blade angle value should be between 17 degrees and 30 degrees and suggested to use a value of 22.5 degrees as a mean value for preliminary calculations.

Bohl [1] gave a certain narrow band of exit blade angle variation with Ns but Karassik [6] gave a rather very wide band range. In our calculations attempts were done to find the effect of the exit blade angle on the values of efficiencies with different values of Q/Qr. It was found that β_2 is the only parameter which could have wide range of values at different values of Q/Qr, specially at low Ns pumps. This wide range of exit blade angle at low values of Ns are higher than the proposed values for β_2 of Bohl [1] but within the range indicated by Karassik [6], as shown in Fig.7. This different values of β_2, for a given capacity ratio, are used in order to yield reasonable values of pump hydraulic efficiency. For relatively high specific speed, limited variation of β_2 is only possible and close to Bohl range, as shown in Fig. 7.

Outlet width ratio Kb_2

While Bohl [1] gave a rather wide band of its variation with Ns, Daily as mensioned in Mikhail [8] gave only single mean curve, Both are plotted versus Ns in Fig. 8. It is clear that the outlet width ratio values of Daily are close to the mean values of Bohl's band. None of them showed the effect of the value of Q/Qr on impeller width ratio. We have used the values of Mikhail [8] in the present calculations.

Number of blades Z

Fuchslocher & Schulz [5] suggested the following formula for calculating the number of impeller blades Z, as function of the values of β_1, β_2 and diameter ratios of the impeller.

$$Z = \frac{2\pi \sin\left(\frac{\beta_1 + \beta_2}{2}\right) \times \left(1 + \frac{(\overline{D_{1i}} + \overline{D_s})}{2 \times \overline{D_{2m}}}\right)}{1 - \frac{(\overline{D_{1i}} + \overline{D_s})}{2 \times \overline{D_{2m}}}}$$

where $\overline{D_{1i}} = \frac{D_{1i}}{D_{2o}}$, $\overline{Ds} = \frac{Ds}{D_{2o}}$, $\overline{D_{2m}} = \frac{D_{2m}}{D_{2o}}$

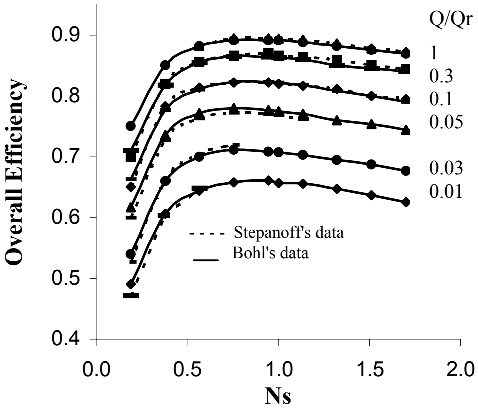

Fig. 3 Overall efficiency versus Ns at b.e.p.

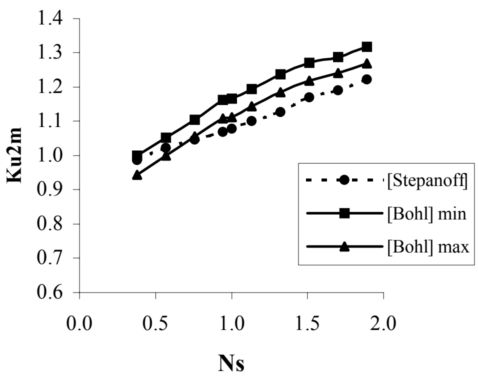

Fig. 4 mean kinematic velocity ratio versus Ns

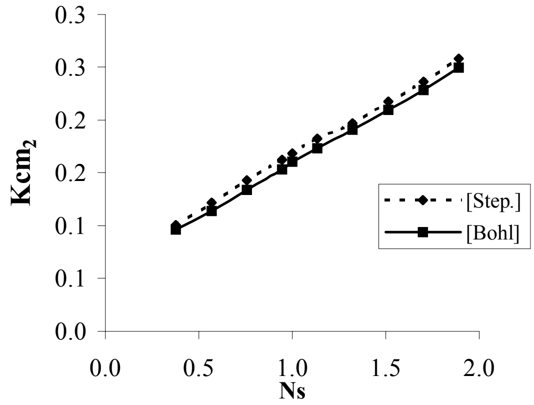

Fig. 5 outlet meridional velocity versus Ns

Slip Ratio σ

It is well known that the slip ratio which is equal to $\frac{Cu_2}{Cu_{2th}}$ is a function of β_2 and number of blades Z. Among the many several empirical relations for expressing σ, the empirical formula prepared by Wiesner [12] and the theoretical deduced curves suggested by Busemann [2] are in very close agreement.

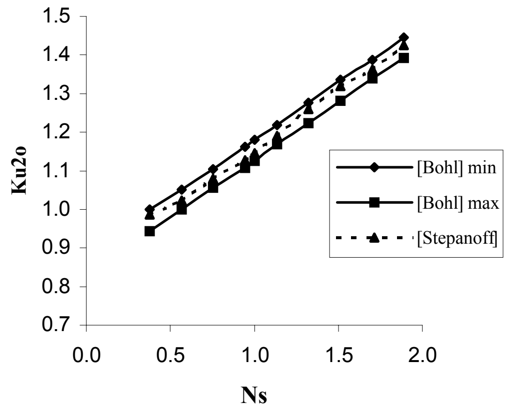

Fig. 6 outlet kinematic velocity ratio versus Ns

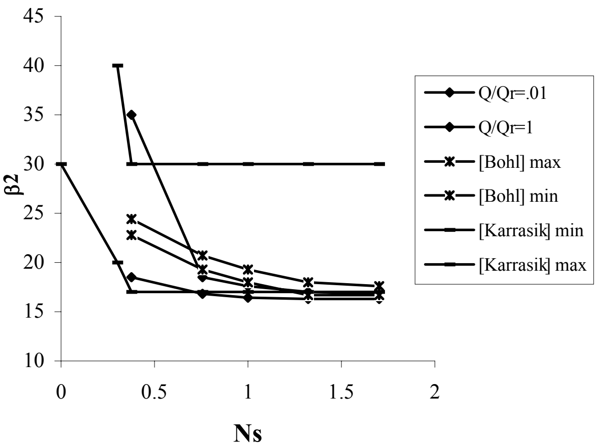

Fig. 7 outlet balde angle versus Ns

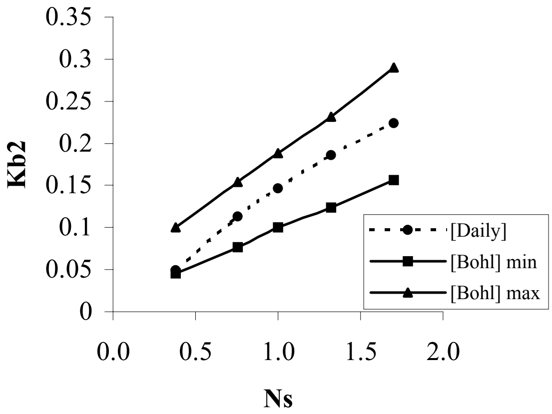

Fig. 8 outlet width ratio versus Ns

Wiesner's formula is as follows

$$\text{slip ratio } (\sigma) = 1 - \frac{\sqrt{\sin \beta_2}}{Z^{0.7}}$$

5. SEMI-EMPIRICAL RESULTS AND VERIFICATION

Using equations (5) for $P_{d.f.}$ % and (6) for Kd and the above mentioned data, calculations were carried out and the results are given in Figs 9 and 10.

	Coefficient = $\frac{Kcm_2}{\sqrt{2gHm}}$	---
Kd	disk friction coefficient	---
Ku_{2m}	mean impeller peripheral speed kinematic ratio = $\frac{U_{2m}}{\sqrt{2gHm}}$	---
Ku_{2o}	exit impeller peripheral speed kinematic ratio = $\frac{U_{2o}}{\sqrt{2gHm}}$	---
g	gravitational acceleration	m/s^2
Hm	actual head	m
N	rotational speed	rps
Ns	dimensionless specific speed = $\frac{(rad/s)\sqrt{m^3/s}}{(m)^{3/4}}$	
$P_{d.f.}$	disk friction power loss	kw

By curve fitting method, we reached a general relation of Kd with Ns, $\frac{Q}{Q_r}, \frac{K}{D}, \frac{S}{D}$ and Re which will be shown later.

The exponents for K/D, S/D and Re were obtained previously by Daily and Nece [3] and Poullikkas [10].

In order to verify experimently, at least partially the above figures and equations, we have carried out tests on a dummy impeller (with blocked impeller passages) similar to our available impeller and casing and let it rotate inside the volute casing of the actual pump (replacing the real impeller).

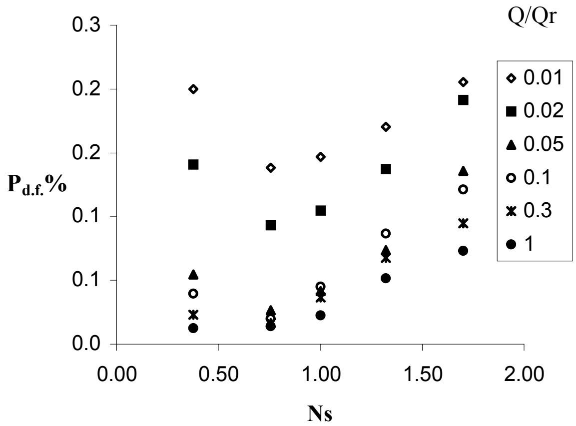

Fig. 9 disk friction efficiency versus Ns at different capacity ratio

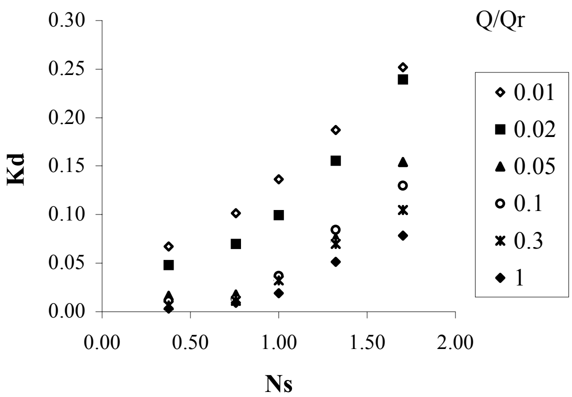

Fig. 10 disk friction coefficient versus Ns at different capacity ratio

NOMENCLATURE

B	blockage effect	---
b_2	impeller width at exit	---
Cu_{act}	actual whirl velocity at impeller exit	m/s
Cu_{th}	theoretical whirl velocity at impl. exit	m/s
D_{2m}	mean exit diameter of impeller	m
D_{2o}	outer diameter of impeller	m
D_{1i}	wearing ring diameter	m
Ds	suction diameter	m
K	surface roughness	m
Kb_2	outlet impeller width ratio $=b_2/D_{2m}$	---
Kcm_2	outlet meridianal flow velocity	
$P_{d.f.}$	disk friction power loss	kw
P_{th}	total theoretical power input	kw
P_M	all mechanical power losses	kw
P_w	water power	kw
Q	flow rate	m^3/s
Q_r	reference flow rate (10,000 gpm = 0.63 m3/s)	
$\frac{Q}{Q_r}$	capacity ratio (flow rate ratio)	---
Re	Reynolds number	---
S	axial spacing between the impeller and casing	m
SHP	input shaft power	
T	torque input	N.m
U_2	peripheral velocity at exit of impeller	m/s

Z	number of blades	---
β_1	inlet blade angle	degrees
β_2	exit blade angle	degrees
η_o	overall efficiency	---
η_h	hydraulic efficiency	---
η_v	volumetric efficiency	---
η_m	mechanical efficiency	---
η_{DF}	disk friction efficiency	---
σ	slip coefficient = Cu_{2act}/Cu_{2th}	---
ω	angular velocity	rad/s
θ	impeller shroud angle	

Abbreviations

b.e.p. best efficiency point

6. EXPERIMENTAL WORK

The experimental work aims to measure the effect of both the surface roughness and the axial spacing on the disk friction power loss in centrifugal pumps.

In this experiment a dummy impeller was manufactured (D_2 = 320mm & Ns = .287) with the same profile and dimensions of the actual impeller of the pump to be tested for the evaluation of the disk friction power loss, and run inside the actual volute casing of the pump as a simulation for the actual impeller and casing.

The test facility is a closed loop test rig, where the suction of the pump is connected to the tank through a flexible connection and the pump delivery is connected through adelivery line to a main header which delivered pumped water back to the tank, test rig is shown in El-Nady [4].

The pump and solid disk to be tested is coupled to a constant speed A.C electric motor (1500 rpm) through a torque meter which facilitates the measuring of the torque transmitted and a special speed sensor for measuring the speed of rotation. The motor control panel is equipped by frequency control system for smooth starting of the pump.

The experimental test was based on 3 groups and each group includes 4 runs. The target was studying the surface roughness effect and spacing ratio on the disk friction power loss by changing the rotating disk surface roughness and axial spacing.

In each group the spacing is constant and the surface roughness is changing. The surface roughness of the rotating disk is varied via sticking emery papers having different surface roughness on both faces of the disk. In the three groups the spacing was changed by reducing the disk thickness by machining. Taylor-Hobson digital device with diamond pin was used to measure the surface roughness of the impeller, solid disk and all the emery paper used.
The surface roughness ratio and the spacing ratio used are indicated in Fig. 11.

In each run the net power loss due to the disk friction for the solid disk is measured by taking the difference between the power measured while the pump casing was filled with water and the power measured while the pump casing was empty of water. So the bearing loss will be excluded from the net power loss.
Fig. 11 is the graphical presentation of the results obtained during the test for the relation between the disk friction power loss and both the surface roughness ratio K/D and the axial spacing ratio S/D.

From curve fitting for the scattered points of Fig. 11, it gives the following relations:

- Disk friction power loss α (surface roughness)$^{0.25}$

The previous result is the same as the relation estimated by Andreas Poullikkas [10] (1995).

- Disk friction power loss α (axial spacing)$^{0.1}$

The previous result is the same as the relation estimated by Daily and Nece (1960) [3], whereas most of the researchers agreed that the Disk friction power loss α (Re)$^{-0.2}$.

Thus the exponents of K/D and S/D in the general formula are taken as obtained in the experimental work and the exponent of Re is taken as obtained before from previous researches. Further since the geometric ratios of the dummy impeller is the same as that of the actual impeller with Ns=.287 and Q/Qr = .035, we have substituted these values in eqn. (7) for K/D = 10x10-6, S/D = 0.017, Ns = 0.287 and Q/Qr = 0.035, it was found that a very close agreement between the values of Kd calculated from the relation (7) and that obtained experimentally.

The general formula for Kd was obtained by curve fitting of the semi-empirical results (Fig. 11) including the effect of K/D, S/D and Re.

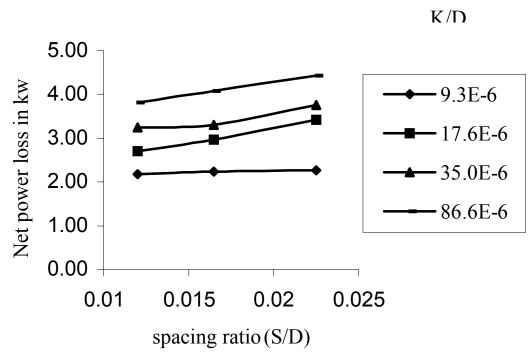

Fig. 11 Experimental results of net power loss versus the disk axial spacing ratio at different values of surface roughness

7. DISCUSSION

The general formulae obtained for the disk friction coefficient are as follows:

$$Kd = 10.6 \times Ns^{1.3} \times \left(\frac{Q}{Q_r}\right)^{-0.47} \times \left(\frac{K}{D}\right)^{0.25} \times \left(\frac{S}{D}\right)^{0.1} \times Re^{-0.2}$$

for Q/Qr < .035 (7)

$$Kd = 15.9 \times Ns^{1.9} \times \left(\frac{Q}{Q_r}\right)^{-0.32} \times \left(\frac{K}{D}\right)^{0.25} \times \left(\frac{S}{D}\right)^{0.1} \times Re^{-0.2}$$

for Q/Qr > .035 (8)

Study of Fig. 10, which shows that Kd increases continuously with Ns and inversely with Q/Qr, an attempt to explain this trend is carried out:

Since by definition $Kd = \dfrac{P_{d.f.}}{\rho \times N^3 \times D^5}$

Where $P_{d.f.}$, in case of radial disk (for low Ns pumps) =

$$= const \times 2\pi\omega^3 \int_{r_s}^{r_2} r^4 dr$$

and Kd = $const \times \left[1 - \left(\frac{r_s}{r_2}\right)^5\right]$, see Fig. 1

and in case of conical surface (for medium and high Ns) where $dx = \dfrac{dr}{\sin\theta}$

$$\Theta\ P_{d.f.} = const \times 2\pi\omega^3 \frac{r^4 dr}{\sin\theta}$$

$$\therefore Kd = \frac{const}{\sin\theta} \times \left[1 - \left(\frac{r_s}{r_2}\right)^5\right]$$, see Fig. 1

The higher the specific speed of the pump, the higher is r_s/r_2, the smaller θ then the smaller the value of sin θ, hence the higher the value of Kd (for certain value of r_s/r_2). Further while r_s/r_2 increases with Ns but since in the formula
$(1- (r_s/r_2)^5)$ its value decreases only slight in the numerator where sin θ decreases in the denominator, then the whole result will increase the value of Kd.
As for decrease of Kd with Q/Qr, it is easy to find that D_2 is function of Q, hence the higher Q/Qr the smaller will be K/D, if the absolute roughness K does not change appreciably, and as mentioned before that Kd is proportional to K/D, hence it will decrease, the higher the value of Q/Qr.

8. CONCLUSIONS

From the computational work carried out and the results which are mainly applicable for single-stage or single suction pump within specific speed range of 0.35 – 1.7 and capacity of 0.0063 – 0.63 m^3/s (100 – 10000 gpm), the following are the main conclusions:

1 – The disk friction power loss percent increases continuously with Ns at large capacity. At other values of capacity (small and moderate values), the disk friction power loss percent decreases slightly with Ns up till a peak value at Ns = 0.7, and increases appreciably at higher values of Ns. It decreases with the capacity continuously at any values of Ns.

2 – The disk friction coefficient Kd increase parabolicaly with the increase in the specific speed and decreases continuously with the increase of the capacity of pumps. From the experimental and computational works, a new formula for Kd is proposed which gives dependence on the specific speed and capacity of pumps in addition to the Reynolds number, the value of the relative surface roughness of the hub and shroud surfaces of the pump and the axial clearance between the pump casing and the hub & shroud of the impeller.

3 – Experimental work carried out on model of radial impeller (but closed passages) with the same dimensions of a real pump and running inside the real casing of the pump, gave a value of disk friction loss which is equal to the value obtained by the existing formula that proving the validity of our results specially as regard disk friction losses and efficiency.

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