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AN OPTIMUM SET OF LOSS MODELS FOR CENTRIFUGAL PUMP

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ABSTRACT

Present work is aimed to calculate optimum set of loss models for centrifugal pumps. These optimum set of loss models are directly correlated with the geometrical and hydraulic parameters of centrifugal pumps. These allow the study of the variation of performance with geometry. For the simulation computer program code has been developed which permits wide range of variables to be investigated.

Keywords: Centrifugal pumps, loss models, geometrical and hydraulic parameters.

Symbols:

a = constant used in determination of the shaft power

A_{th}= Throat area at volute in m^2

B_1= impeller width at inlet in m

B2= impeller width at outlet in m

B_3=inlet width of volute in m

D= diameter in m

D_e= impeller eye diameter in m

D_h=hub diameter in m

D_{sh}= shaft diameter in m

dp/r = Mean blade loading

f = weighting factor

f_s = shear stress N/m^2

g = gravitational acceleration in m/s^2

H = head in m

h_1 =suction head loss in m

h_2= incidence loss in m

h_3=blade loading loss in m

h_4= skin friction loss in m

h_5 =mixing loss in m

h_6=disk friction loss in m

h_7=recirculation loss in m

h_8=expansion loss in volute

h_9=internal loss in m

h_{10} = enlargement volute loss in m

V_m = absolute velocity in m/s

h_L=Total loss in head in m

H_d = depression head in m

K_m=capacity constant

$NPSH_R$=net positive suction required head in m

N_s = specific speed of pump in rpm

N=speed in rpm

P_{in}= input power in HP

P_{sh}= shaft power in HP

Z = Number of blades

r = radius in m

R_t = tongue radius in m

R_e= Reynolds number

t = blade thickness in m

U= peripheral velocity in m/s

U_e= peripheral velocity at the eye diameter velocity in m

V_r= relative velocity in m/s

V_{rui}= whirl component of relative velocity in m/s

V_u= whirl component of Velocity in m/s

V_{th}= throat velocity in m/s

Q=flow rate in m3/s

Greek-

α=angle at which the water leaves the impeller

θ = volute angle

β = blade angle

θ_t= tongue angle

η = overall efficiency

θ_A= maximum total angle between the side of the volute

ν = kinematic viscosity m^2/s

τ = permissible stress N/m^2

σ = slip factor

Φ = flow coefficient

σ_b= blade cavitation coefficient

Ψ= head coefficient

σ_{th}=Thoma cavitation coefficient

ω= angular velocity in rad/s

Subscripts:

0 = Eye of impeller
1 = Inlet to the impeller

2 = Outlet to the impeller

3 = Inlet to the volute

S. No.	Loss mechanical	Loss model
1.	Suction loss	$$\Delta h_{Suction} = f_1 \frac{v_1^2}{2g}$$ Where f1 = 0.2 - 0.3
2	Incidence loss	$$\Delta h_{Incidence} = f_2 \frac{v_{rui}^2}{2g}$$ Where f2= 0.5 - 0.7
3	Disc friction loss-	$$H_3 = \frac{f_{dR} D_2^2 u_2^3}{16Qg}$$ Where $f_{dR} = \frac{2.67}{Re^{0.5}}$ $Re<3x10^5$ $f_{dR} = \frac{0.0622}{Re^{0.2}}$ $Re \geq 3x10^5$ $Re = \left(\frac{u_2 r_2}{\vartheta}\right)$
4	Mixing loss-	$$h_u = \frac{1}{(1+\tan^2\alpha_2)}\left(\frac{v_2^2}{2g}\right)$$
5	Recirculation loss-	$$(\Delta h)_{Re} = \frac{8x10^{-5}\sinh(3.5\alpha_2^3)D_f^2 u_2^2}{g}$$
6	Blade loading loss	$$(\Delta h)_{Blade} = \frac{0.05 D_f^2 u_2^2}{g}$$
7	Skin friction loss	$$\Delta h = \frac{2CL_b V_r^2}{D_{hyd} g}$$
8	Expansion loss	$$(\Delta h)_{Expansion} = \frac{0.75(V_{u2} - V_{th})^2 + V_m^2}{2g}$$
9	Volute skin friction loss	$$(\Delta h)_{volute-skin-friction} = \frac{0.5 V_{th}^2}{2g}$$
10	Enlargement loss	$$(\Delta h)_{Enlargement} = \frac{(V_{th} - V_d)^2}{2g}$$

Table 1: Optimum set of loss models for centrifugal pumps

I.INTRODUCTION

In conventional design method of centrifugal pump, efficiency is the function of specific speed, which is available in form of graph, empirical correlation in various text books and references .But in practical, efficiency has direct influence due to change of flow pattern, Renoldnumber, relative eddies in the impeller blade passage .The empirical loss correlation method has been developed, which is well documented in standard references [11, 13, 15]. Recent studies using the same method have been carried out by [4, 5] and many others. Takagietal [6] formulated loss models for leakage flow and disc friction based on their measurement of centrifugal pumps at three different specific speeds .Jhanapandi and Prasad [5] surveyed a number of available loss correlations and found satisfactory set of models for almost the full range of operating conditions of low specific speed submersible pumps. Present work is aimed to calculate optimum set of loss models for centrifugal pumps. These optimum set of loss models are directly correlated with the geometrical and hydraulic parameters of centrifugal pumps.

II.SIMULATION OF LOSS MODEL

Present work is carried out under the following assumptions, the flow comes in through the inlet without any pre-swirl, the flow in the van less space is of a free-vortex type, and the volute casing is constructed of gradually increasing circular cross-sections with a constant average velocity. For calculation of optimum set of loss models for centrifugal pumps input data are design specifications and geometrical and hydraulic variables, given below. Empirical loss models from the open literature [4, 5, 13, and 15] collected and listed below in Table 1. Geometrical and hydraulic parameters are calculated with the help of conventional design method given in [10, 12, and 14].For simulation purpose computer program has been generated in C [7, 8] which permits wide range of variables to be investigated in a short interval of time

2.1 Design Specification

Design of the centrifugal pump input data: Volume flow rate, Pump total head, specific speed, Density of liquid, Operating fluid viscosity.

2.2 Geometric parameter

Vane angle, Number of vanes Impeller discharge width, Hub/Tip ratio, Inclination of the mean stream line to axial direction, Blade cavitations factor.

2.3 Hydraulic parameter:

Flow coefficient, Head coefficient, Blade velocity, Relative velocity and other hydraulic parameter needed to describe the flow direction and magnitudes become direct function of geometry**.**

III.RESULTS & DISCUSSIONS

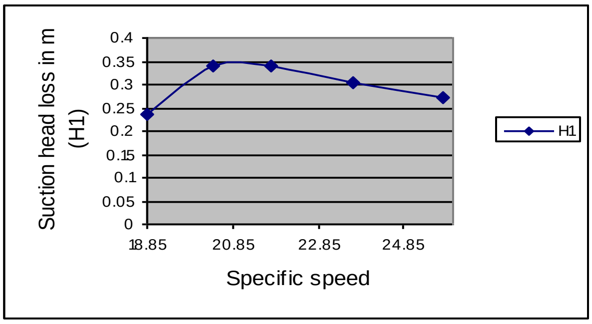

Fig 2 specific speed-suction head loss

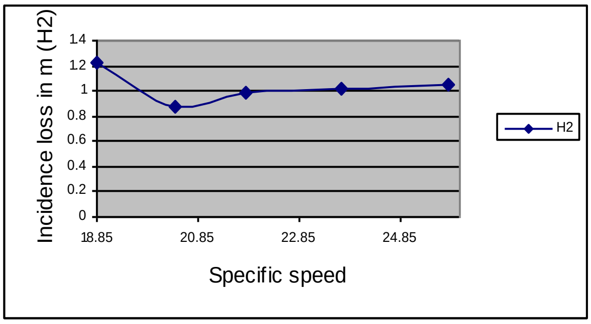

Fig 3 Specific speed-incidence loss

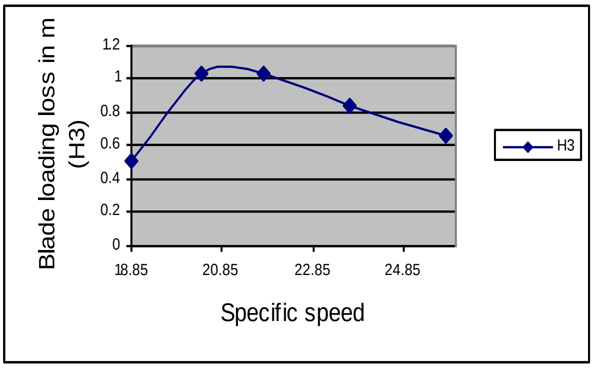

Fig 4 Specific speed-blade loading loss

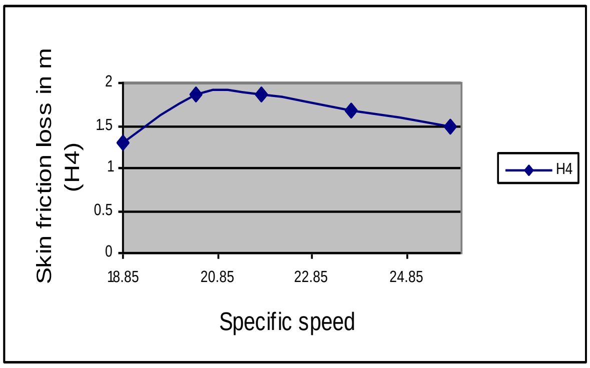

Fig 5 Specific speed-skin friction loss

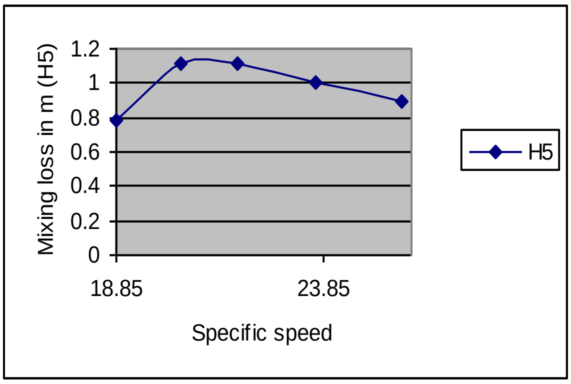

Fig 6 Specific speed-mixing loss

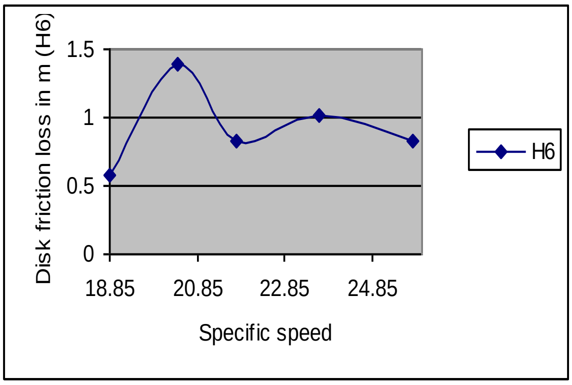

Fig7 Specific speed-disk friction loss

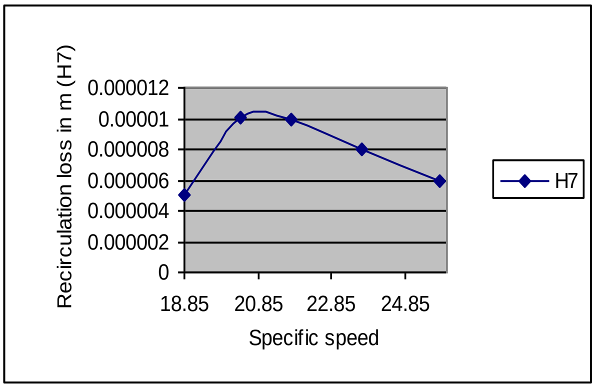

Fig8 Specific speed-recirculation loss

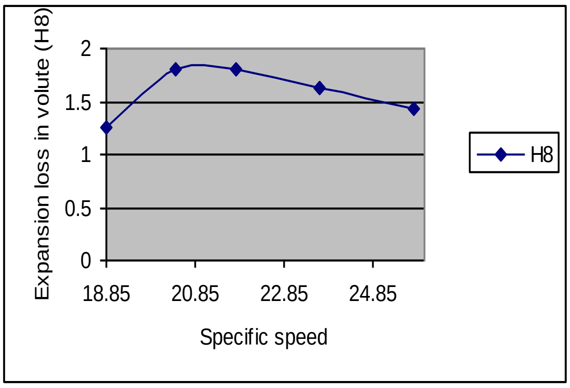

Fig 9 Specific speed-expansion loss

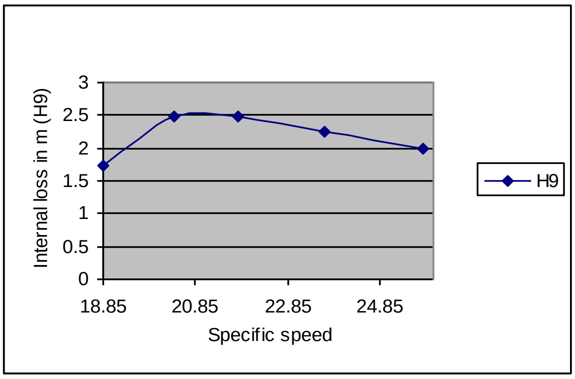

Fig 10 Specific speed - internal loss

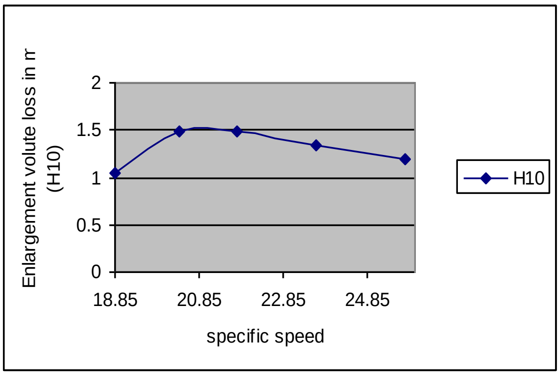

Fig 11 Specific speed enlargement volute loss

Fig. 2 to 11 show variation of different internal hydraulic losses with variation of specific speed. These figures show that the various hydraulic losses first increase with specific speed and after some value start decreasing except the incidence loss which decreases first and after some value becomes almost stable. Thus the total internal hydraulic efficiency first decreases and then increases after some value. Fig. 12-14 show the variation of efficiency with different non dimensional fluid parameters. With increase in slip factor and head coefficient, efficiency increases but with increase in flow coefficient, efficiency decreases. Thus efficiency is stable for the given range of non-dimensional parameters.

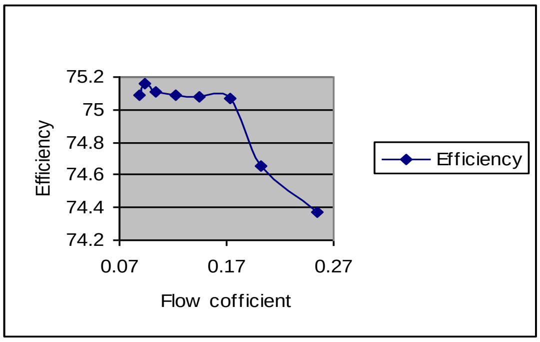

Fig 12 Flow coefficient - efficiency

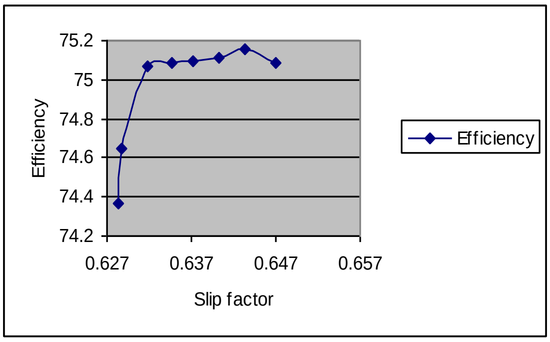

Fig 13 Slip factor - efficiency

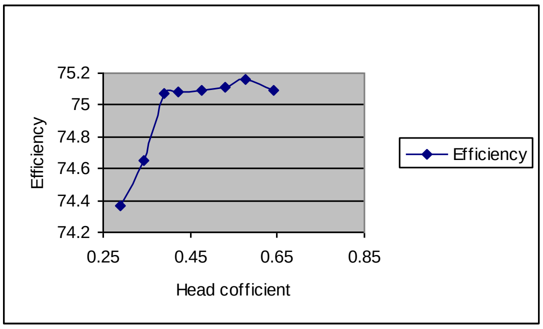

Fig 14 Head coefficient-efficiency

IV.CONCLUSIONS

Practical performance of centrifugal pump has direct influence due to change of flow pattern, Renoldnumber; relative eddies in the impeller blade passage, roughness of the surface etc. The present work can easily calculate the actual hydraulic losses of centrifugal pumps at given operating condition. As these loss models directly depend on geometrical and hydraulic parameters of centrifugal pumps, loss models can also be utilized for prediction of performance for different types of turbines and pumps. In the Fig 12-14 stable range of non dimensional number can be chosen for design purpose.

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