Applied Energy 187 (2017) 10–26

Contents lists available at ScienceDirect

Applied Energy

journal homepage: www.elsevier.com/locate/apenergy

Optimal design of multistage centrifugal pump based on the combined energy loss model and computational fluid dynamics

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HIGHLIGHTS

- Combined method of energy loss model and CFD is proposed for pump's optimal design.
- All the kinds of energy losses in a multistage centrifugal pump are calculated.
- The relationships between the calculation models and the energy losses are obtained.

ARTICLE INFO

Article history:
Received 23 July 2016
Received in revised form 30 October 2016
Accepted 13 November 2016

Keywords:
Energy loss model
Pump optimization
Multistage centrifugal pump
Computational fluid dynamics

ABSTRACT

This paper proposes a method to optimize the design of a typical multistage centrifugal pump based on energy loss model and Computational Fluid Dynamics (ELM/CFD). Different grid numbers, turbulence models, convergence precisions, and surface roughness are calculated for a typical multistage centrifugal pump. External characteristic experiments are also conducted to benchmark the numerical simulation. Based on the results, the ELM/CFD method was established including various kinds of energy loss in the pump, such as disk friction loss, volumetric leakage loss, interstage leakage loss as well as the hydraulic loss, which occurred at inlet section, outlet section, impeller, diffuser and pump cavity, respectively. The interactive relationships among the different types of energy losses were systematically assessed. Applying suitable setting methods for numerical calculation renders more credible results, and ensuring the integrity of the calculation model is the key contributor to the accuracy of the results. The interstage leakage loss is converted by the disk friction loss; thus, they are positively correlated, that is, the disk friction loss can be reduced by decreasing the interstage leakage loss. Concurrently, the volumetric leakage loss is negatively correlated with the disk friction loss; thus, increasing the volumetric leakage loss can effectively reduce the disk friction loss. The increment of the volumetric leakage loss is greater than the decrement of the disk friction loss for general centrifugal pumps. This relationship between these types of losses, however, does not apply to pumps with significantly low specific speed. Therefore, reducing the volumetric leakage and interstage leakage losses is the most effective technique to increase the efficiency of general centrifugal pumps. The impeller should be designed according to the maximum flow design method, because the inevitable volumetric leakage loss will improve the pump efficiency under rated flow condition. Several methods have been proposed to improve the pump efficiency.

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1. Introduction

Pumps are a kind of general machinery with varied applications. According to statistics, pumps' energy consumption accounts for nearly 22% of the world's energy used by electric motors, so pumps have huge energy consumption and great energy saving potential [1–3]. More and more energy saving strategies and end-users push industries and researchers to concentrate on improving the pump efficiency [4,5]. Moreover, multistage centrifugal pumps, as fundamental elements for providing high-energy liquid, have attracted increasing attention in the past several years [6–10].

At present, there are several methods to improve the pump efficiency, namely, test optimization, velocity coefficient optimization, Computational Fluid Dynamics (CFD) optimization and energy loss model (ELM) optimization. Being semi-theoretical and semi-empirical, test optimization plays an important role in the pump design, and orthogonal tests are widely used in the industry. For

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http://dx.doi.org/10.1016/j.apenergy.2016.11.046
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Nomenclature

Symbols

b_2	outlet width of the impeller blade
D_1	inlet diameter of the impeller (mm)
D_2	outlet diameter of the impeller (mm)
D_3	inlet diameter of the positive diffuser (mm)
D_h	hub diameter of the impeller (mm)
G	grid size (mm)
H	total head of the multistage pump (m)
H_1	head of the first stage pump (m)
H_2	head of the second stage pump (m)
H_t	theoretical head of the multistage pump (m)
H_{in}	loss head of the inlet section (m)
H_{out}	loss head of the outlet section (m)
h	hydraulic loss head of the unit fluid through the pump (m)
h_{ip}	hydraulic loss head of the unit fluid through the impeller (m)
h_{df}	hydraulic loss head of the unit fluid through the diffuser (m)
h_{ca}	hydraulic loss head of the unit fluid through the pump cavity (m)
$M1$	calculation model without pump cavity (–)
$M2$	calculation model without ring (–)
$M3$	calculation model without front ring (–)
$M4$	calculation model with 0.25 mm front ring (–)
$M5$	calculation model with 0.5 mm front ring (–)
$M6$	calculation model with 1 mm front ring (–)
$m1$	real pump model with 1 mm front ring (–)
$m2$	real pump model through placing a plastic seal ring on the front ring of $m1$ (–)
$m3$	real pump model with 0.5 mm front ring (–)
N	Stage number of the multistage pump (–)
P	shaft power of the multistage pump (W)
P_1	shaft power of the first stage pump (W)
P_2	shaft power of the second stage pump (W)
P_{1h}	hydraulic power of the first stage pump (W)
P_{2h}	hydraulic power of the second stage pump (W)
P_{1m}	loss of disk friction power of the first stage pump (W)
P_{2m}	loss of disk friction power of the first second pump (W)
P_m	loss of disk friction power of the multistage pump (W)
P_{M2}	loss of disk friction power of $M2$ (W)
P_{M3}	loss of disk friction power of $M3$ (W)
P_{M6}	loss of disk friction power of $M6$ (W)
P_h	hydraulic power of the multistage pump (W)
P_v	loss of volumetric leakage power of the multistage pump (W)
P_u	working power of the multistage pump (W)
Q	flow rate (m^3/h)
Q_r	rated flow (m^3/h)
q	average ring leakage amount of the multistage pump (m^3/h)
q_1	ring leakage amount of the first stage pump (m^3/h)
q_2	ring leakage amount of the second stage pump (m^3/h)
q_b	interstage leakage amount of the multistage pump (m^3/h)
q_b	outlet width of the impeller blade
R	radii of the front shroud or rear hub (mm)
r	radii of any locations on the front shroud or rear hub (mm)
Z	number of the impeller blades (–)
Z_p	number of positive diffuser blades (–)
Z_n	number of negative diffuser blades (–)
α_3	inlet angle of the positive diffuser blade (°)
α_6	outlet angle of the negative diffuser blade (°)
β_1	inlet angle of the impeller blade (°)
β_2	outlet angle of the impeller blade (°)
η	efficiency of the multistage pump (–)
η_h	hydraulic efficiency of the multistage pump (–)
η_m	mechanical efficiency of the multistage pump (–)
η_v	volumetric efficiency of the multistage pump (–)
θ_w	wrap angle of the impeller blade (°)
μ	surface roughness (μm)
τ	shear stresses (Pa)
ΔP_h	hydraulic loss power of the multistage pump (W)
ΔP_{in}	hydraulic loss power of the inlet section (W)
ΔP_{ip}	hydraulic loss power of the impeller (W)
ΔP_{df}	hydraulic loss power of the diffuser (W)
ΔP_{ca}	hydraulic loss power of the pump cavity (W)
ΔP_{out}	hydraulic loss power of the outlet section (W)
$\Delta\gamma_1$	leakage coefficient with interstage leakage (–)
$\Delta\gamma_2$	leakage coefficient with volumetric leakage (–)
$\Delta\zeta_1$	increasing coefficient of the loss of disk friction power with interstage leakage (–)
$\Delta\zeta_2$	decreasing coefficient of the loss of disk friction power with volumetric leakage (–)

Subscripts

ca	pump cavity
h	hydraulic
df	diffuser
hb	rear hub
in	Inlet section
ip	impeller
m	mechanical
n	negative diffuser
out	outlet section
p	positive diffuser
r	rated
t	theoretical
v	volumetric
w	wrap

example, Zhou et al. [11] designed 16 impellers with the same diffuser base on the orthogonal table, and the suitable parameter combination for the highest pump efficiency was captured by employing the orthogonal tests optimization. In order to improve the efficiency of stainless steel stamping multistage pump, Wang et al. [12] established the function relationship between the efficiency and three factors of impeller through the quadratic regression orthogonal test. To realize the multiobjective optimization of pump efficiency and cavitation performance, Xu [13] used orthogonal method to carry out the range analysis and studied the influence order of each parameter. Although orthogonal test is very credible and effective for the pump optimization, however, it consumes too much time and material because of a series of testing schemes.

Velocity coefficient optimization is a kind of similar conversion method based on lots of excellent hydraulic models, and suitable velocity coefficient should be selected as the basis of pump size according to the specific speed. This method is simple to use, but it's very difficult to design new excellent hydraulic model due to the limitation of existing models and experience. In 1948, Stepan-

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