Impeller performance

Axial flow impeller shapes: part 1

The increasing importance of low head applications necessitates a study of the different axial flow pump impellers used for both pump and turbine operations. In this first of two articles, Punit Singh and Franz Nestmann of Karlsruhe Institute of Technology survey the available impeller shapes and present a theoretical model that reveals the behaviour of the impellers' internal hydraulic variables.

Low head applications of turbomachinery (both pumps and turbines) are gaining increasing importance in modern day engineering. This is particularly true for turbine operation for energy recovery and decentralized power applications[1,2]. While the requirements for pumping have been handled well by the industry, by comparison the technology for low head turbine applications has not yet been optimized.

Definition of parameters and subscripts

c	absolute velocity, m/s
g	acceleration due to gravity, m/s^2
H	head, m
k_a, k_b, etc.	constants of the Euler line
Q	discharge, l/s or m^3/s
u	tangential blade velocity, m/s
w	relative velocity, m/s
α	absolute flow angle, degrees
β	relative flow angle, degrees
ζ	pressure loss coefficient
*	blade direction
f.p	forward curved pump impeller
f.t	forward curved turbine impeller
p	pump mode
t	turbine mode
u	tangential direction
x	axial direction

The pumps for low head applications are mainly of the axial flow type, which are manufactured with either forward or backward curved blades. These two blade shapes have different influences on the performance of the pumps. However, neither experimental nor theoretical investigations on these influences have been effectively compared or presented. In addition, the total pump characteristics (four-quadrant analysis), which have become a norm for the industrial use of pumps under specialized operating conditions after Stepanoff[3], have not been published for all the axial flow pumps available on the market. One of the quadrants of these characteristics pertains to turbine operation. However, it is not clear whether forward or backward curved blades are more suitable for turbine applications. In order to clarify both pump and turbine operations with these impeller shapes, a theoretical study would be beneficial before detailed experimental work is initiated.

The objectives and problem outline of this two-article series are summarized as follows:

a) To survey the different axial flow pumps available with forward and backward curved impeller shapes.

b) To develop a theoretical model based on turbomachinery fundamentals for both pump and turbine operation and to study the behaviour of the internal hydraulic variables in these impellers.

c) To study the implications for turbine operation of both these impeller shapes and recommend suitable impellers for further experimental optimization.

d) To initiate a simulation model for the four-quadrant analysis of both these impeller shapes.

Part 1 analyses the forward curved blades, while part 2 will focus on backward curved blades and compare the pump and turbine operations for the two blade shapes.

Means and methods

The study collates classical and modern-day literature on the development of optimum axial flow impeller shapes. Further, a theoretical model based on the Euler theory of turbomachines is developed to study the internal hydraulic behaviour of the two impeller

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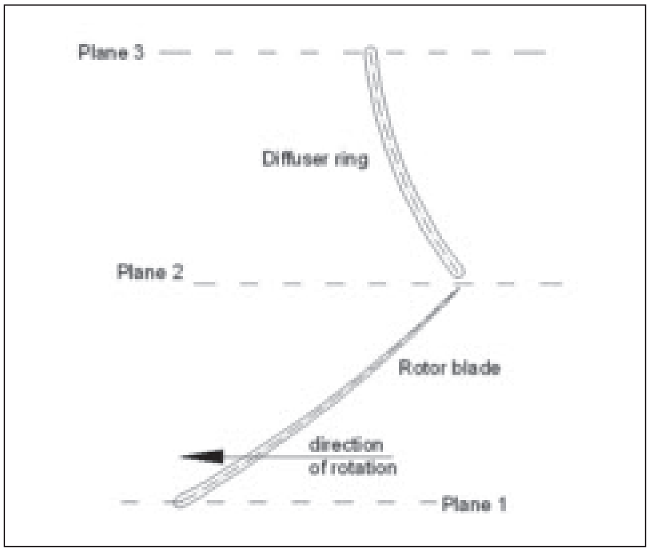

Figure 1. Blade section of a forward curved impeller and diffuser.

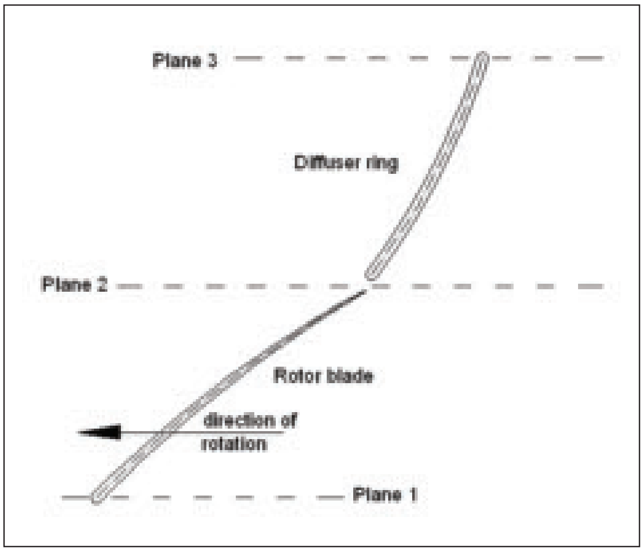

Figure 2. Blade section of a backward curved impeller and diffuser.

shapes for both pump and turbine operations. The model will be useful in evaluating the influences of the blade shapes and also help in studying suitable shapes for dedicated turbine operation.

Survey of axial flow impellers

The basic shape of the axial blade is defined based on the relationship of the curvature with respect to the direction of the blade rotation. The forward curved variety is the most common type of axial flow pump impeller available on the market and is also seen in the blade shapes of fans and compressors. Figure 1 shows a section of a forward curved axial impeller stage (impeller and diffuser ring). This type of impeller has been investigated by Baumgarten *et al.*[4a] and Stark and Siekmann[4b] for axial flow pumps, while Dixon[5] has carried out studies on axial flow fans and compressors with forward curved blade designs. The energy transfer and internal hydraulics for this blade shape in pumps and turbines are discussed in the next section.

The shape of the backward curved impeller along with the diffuser blade is shown in Figure 2. This peculiar shape of axial impeller has been widely used by KSB (for example, in the Amacan P series; www.ksb.com) and has also been reported by Springer[4c]. The energy transfer and behaviour of the internal variables for both pump and turbine modes will be examined in the second article.

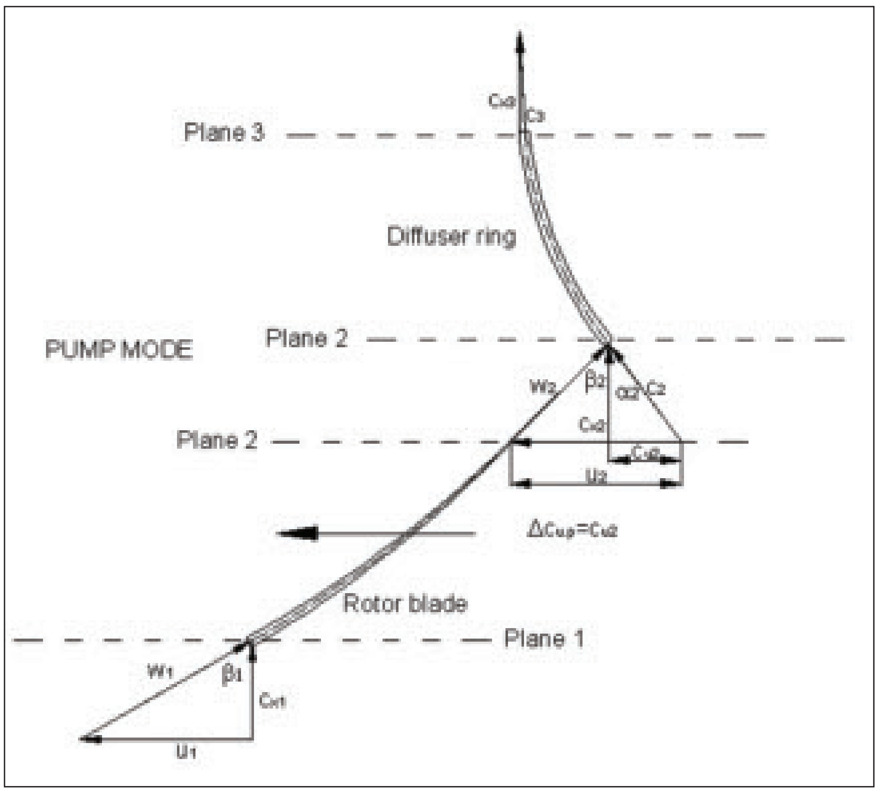

Figure 3. Internal flow analysis of a forward curved blade in pump mode.

Theoretical model and analysis

A theoretical model is derived here for pump and turbine operations using forward curved impellers. The model relies on Euler fundamentals for axial machines as suggested by Singh and Nestmann[6].

Pump operation

The construction of the velocity triangles in pump mode involves certain assumptions, namely, that the absolute flow entry to the impeller is swirl-free, and that there is zero flow incidence at the blade entry. Further, the deflection of the relative velocity at the blade exit is neglected, which is a reasonable assumption provided the entry is incidence free, as pointed out by Dixon for compressor cascade results[5]. In addition, the flow areas at the exit and the entry are identical, making the axial flow velocities identical as the compressibility effects of the working fluid, which is liquid, have been neglected.

The velocity triangles for the complete stage of a forward curved impeller are plotted in Figure 3. The shape of the diffuser is based on the resulting direction of the absolute flow velocity at the rotor exit and the need to make the flow swirl-free (to have c_{u3} as small as possible) at the stator exit. It can also be seen that the relative velocity is reduced across the rotor blade. The hydraulic shaft power or Euler momentum is positive with respect to the blade velocity, as shown in Equation 1:

$$(\Delta c_u \cdot u)_{f.p} = (c_{u2} - c_{u1}) \cdot u = c_{u2} \cdot u \text{ (as } c_{u1} \text{ is zero)}$$

$$(\Delta c_u \cdot u)_{f.p} = k_{a.f.p} - k_{b.f.p} \cdot Q \qquad (1)$$

In addition, internal variables such as the impeller profile loss coefficient result in losses that have to be accounted for before arriving at the actual momentum gained by the fluid (Equation 2). On the whole, the velocity triangles in forward curved pump impellers behave in accordance with the established theory of diffusion through the rotor and positive pressure gradient.

$$(\Delta c_u \cdot u)_{fluid\ f.p} = c_{u2} \cdot u - \zeta_{f.p} \cdot c_x^2/2 \qquad (2)$$

Turbine operation

The internal hydraulic analysis of the two blade shapes in turbine mode also entails some assumptions. Firstly, it is assumed that the axial flow velocity required to operate the turbine at its best efficiency point (BEP) and at the same speed as the pump would be higher (by 1.4 to 1.6 times) than the corresponding pump axial flow velocity[7]. The flow within the static nozzles (or diffusers in pump mode) follows the respective geometries without any deviation. Further, as the first step it is assumed that the relative flow at the rotor exit is deflection-free, which is quite contentious as will be seen as the analysis progresses. Another important assumption is that the direction of blade rotation is in the direction of the inlet swirl to the rotor.

The construction of the velocity triangles in Figure 4 and the analysis in Equation 3 show that all the parameters follow the turbomachinery principles with positive angular momentum and increase of relative velocity across the rotor. The behaviour of the internal flow variables for relative flows with and without deflection at the turbine exit is examined in the discussion section.

$$(\Delta c_u \cdot u)_{f.t} = (c_{u2} - c_{u1}) \cdot u = \text{positive value}$$

$$(\Delta c_u \cdot u)_{f.t} = k_{c.f.t} + k_{d.f.t} \cdot Q \qquad (3)$$

$$(\Delta c_u \cdot u)_{fluid\ f.t} = (\Delta c_u \cdot u)_{f.t} + \zeta_{f.t} \cdot c_x^2/2 \qquad (4)$$

Results

The theoretical Euler line for forward curved pump impellers (Equation 1), along with the actual head gained by the fluid, is plotted in Figure 5 for a wide range of flows. The losses mainly include the profile losses through the rotor given by Equation 2. The magnitude of these losses will depend purely on the profile loss coefficient, which is a function of the angle of incidence at different flow conditions of the pump. There is a zone of instability in the partial-flow region for these impellers.

The turbine mode characteristics for forward curved impellers, comprising the Euler line (Equation 3), actual pressure line (Equation 4), zero speed (N = 0) and zero torque (T = 0) lines, are plotted in Figure 6. The extents of zone C (energy generation zone) and zone D (energy consumption zone) are also shown. The operating point with a positive head is also indicated. The Euler line can extend into the negative head zone, but this is not shown in Figure 6.

Discussion

The main discussion point concerns the turbine operation of forward curved impellers. The results (Figure 4) revealed that, for the present design of the diffuser blades (in pump mode), the flow entry would cause considerable incidence effects and would also probably result in deflection effects at the exit. Following the conventional direction of deflections in turbine blades, it is seen that the natural deflection would cause the net Euler torque to increase, but the profile loss coefficient would also increase, resulting in higher net head across the turbine stage. An experimental study of incidence, deflection and profile losses would clarify this phenomenon, which cannot be ascertained with the present theoretical model. The model also places the operating point of the turbine in the safe operating region (zone C), but the actual efficiency of this point would be a function of incidence

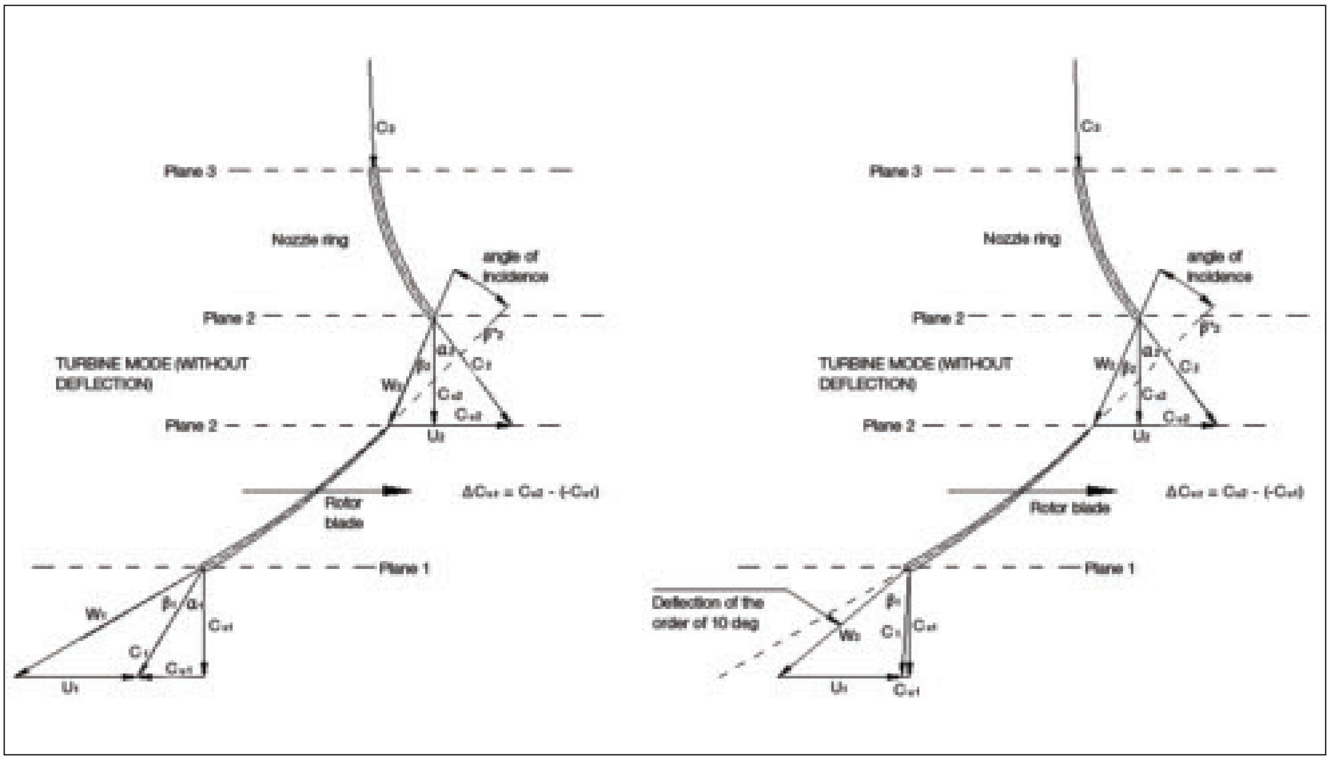

Figure 4. Internal flow analysis of a forward curved blade in turbine mode, with and without deflection.

and deflection effects and the profile loss coefficient.

Conclusions and recommendations

This detailed survey of impellers reveals that manufacturers are using both forward and backward curved impellers primarily intended for pump mode operation. The application of the theoretical model to pump operation did not show any deviation from the expected performance, but the turbine operation of forward curved impellers showed evidence of considerable levels of incidence effects and possible deflection effects at the exit. One of the recommendations for the future study of forward curved impellers would be to carry out elaborate experimental studies at both the cascade and dynamic levels to determine the exact behaviour of incidence, deflection and profile losses.

The remaining objectives of the study, including evaluation of the optimum turbine impeller shape, will be addressed in the second article of this two-part series. ■

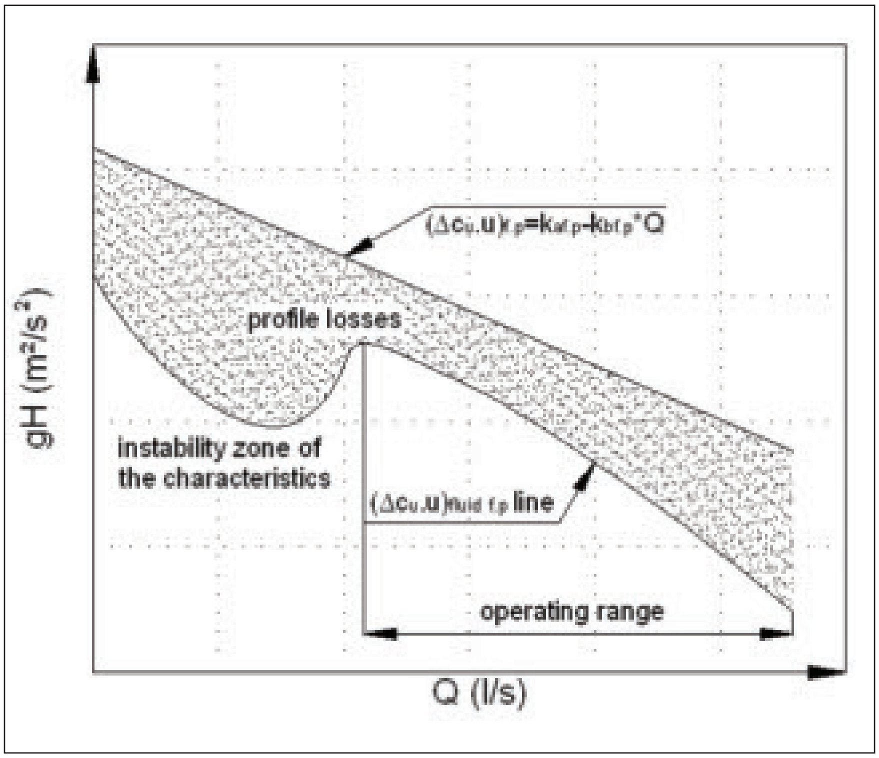

Figure 5. Behaviour of the H-Q line in forward curved pump impellers.

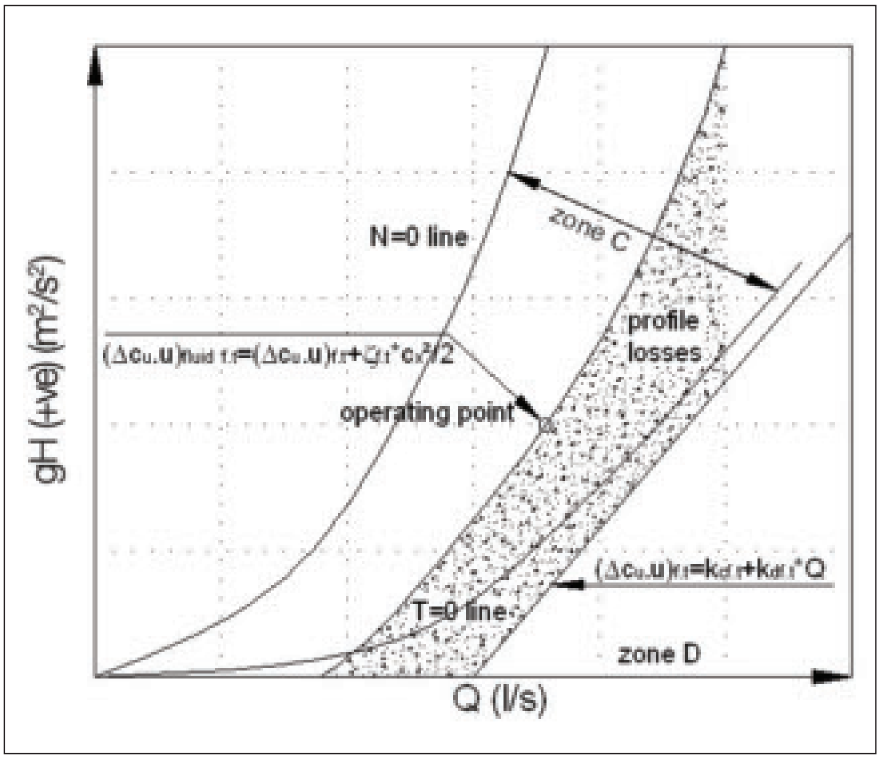

Figure 6. Behaviour of the H-Q line in forward curved turbine impellers.

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