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Transient behavior of a centrifugal pump during starting period

Issa Chalghoum *, Sami Elaoud, Mohsen Akrout, Ezzeddine Hadj Taieb

Research Laboratory "Applied Fluid Mechanics, Process and Environment Engineering", National Engineering School of Sfax, BP'W'3038 Sfax, Tunisia

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ABSTRACT

A theoretical analysis on transient flow inside a centrifugal pump was carried out, building on unsteady and incompressible fluid equations applied to the impeller during the starting periods. The transient flow behavior is governed by two hyperbolic partial differential equations; namely continuity and motion equations. The characteristics method of the specified time intervals was used to analyze the dynamic characteristic of the pump. To follow the dynamic behavior of the pump during startup periods, a numerical study was performed on the pump impeller with different openings of the discharge valve. The comparison between the numerical and experimental results of the pump characteristic curve has shown a good concordance. The results have also revealed that the pressure increase is important in the case of a short starting period of the pump and a large mass of water in the pipelines. In this study, the effect of the impeller diameter and number of blades on the pressure evolution was also analyzed.

1. Introduction

Centrifugal pumps are fundamental elements that are commonly encountered in turbomachinery applications. Transient flow behavior through centrifugal pump during the starting and stopping periods was object of several experimental and theoretical studies. Experimental studies have shown that the impulsive pressure and lag in circulation formation around the impeller vanes are the main reasons for the difference between dynamic and quasi-steady characteristics of turbo pump during its starting period [1–3]. During this period, it has been shown that three factors are responsible for the departure of the dynamic characteristics of a pump during startup from the normal steady-state performance [4]. These factors are the mass of water in the pipeline, the valve opening percentage and the starting time. The theoretical and predictive models have been carried out on the transient characteristics of a centrifugal pump during starting and stopping periods [5,6]. Both models were experimentally validated. Unfortunately, the authors did not focus on the startup time and pump geometry effects on the pump characteristic and the dynamic pressure curves.

By solving transient angular momentum and energy equations, a theoretical expression of the instantaneous head, including terms of angular acceleration, flow acceleration, and velocity profile variation, is presented [8]. In addition, it has been demonstrated that instantaneous velocity profile variation is another important factor determining transient characteristics. Added to that and through experiments on the hydrodynamic performance of a centrifugal pump during transient operation [7], neat differences between transient and quasi-steady characteristics are found. In fact, quasi-steady assumptions were not valid for transient (accelerating or decelerating) applications. A study on the transient pump startup [9] has shown the effect of the inertia of the pump impeller on the head and the discharge during the starting period. In this study, it has been deduced that for a pump with a high moment of inertia, an extended time for the fluid is needed to reach the steady-state flow.

The transient effect was investigated by performing fast startups of the pump in cavitating and non-cavitating conditions [11,12]. In cavitating conditions, different types of unsteady behaviors were obtained. Besides, the cavitation effects on the pump head evolution at constant rotation speed have been studied. Moreover, to explain the evolution of the pump head during the startup phase, a physical analysis was proposed.

Nevertheless, for a pump having a small impeller moment of inertia, the steady state operating speed will be approached much more rapidly than the fluid which would reach steady-state flow. Building on the characteristic method, the transient flows in quasi-rigid pipelines caused by starting pumps has been investigated [14]. In this work, the authors studied different pump starting times and their effect on the dynamic behavior of the flow. The obtained results have revealed that the transient pressure rise in the case of a slow startup is more significant than that in the case of a rapid startup. Furthermore, the steady-state operating point is

* Corresponding author.
E-mail addresses: chalghoumissa@yahoo.fr (I. Chalghoum), elaoudsa@yahoo.fr (S. Elaoud).

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reached more rapidly during the fast startup case than in the case of a slow startup. An experimental study on the transient behavior of a centrifugal pump with open impeller during startup phase was investigated [16]. Their results shows that the rotational speed characteristic is mainly depends on the startup characteristic of the driving motor. In addition, the flow rate lags behind the rotational speed to rise to final stable value. When increasing the discharge valve opening percentage, the delay becomes more important.

Recently, Computational Fluid Dynamic (CFD) has been used to study the transient flow through centrifugal pumps [13–15]. In the previous work, the author used the numerical Dynamic Slip Region (DSR) method to simulate the transient flow evolutions in a centrifugal pump during a rapid starting period. The results of this method has been confirmed with experimental ones. Besides, a three-dimensional CFD simulation was carried out [17] in order to disclose the difference in transient behavior during the startup period when centrifugal pumps transport a single-phase water and a solid–liquid two-phase flow.

Despite the various numerical and experimental studies cited above, the effects of the pump geometry and the starting time on the pump characteristic curve are not yet well investigated. Added to that, the efficiency of the characteristics method as a mathematical tool to study the transient flow inside centrifugal pump was not put in evidence.

The purpose of this paper is to study the transient flow through a centrifugal pump during startup periods. The transient regime was created by considering a linear variation of the rotational speed. The governing partial differential equations were numerically solved using the methods of characteristics. In this study, the effect of the starting time and the geometric properties of the impeller on the pressure and the characteristic curve evolutions was analyzed and the reliability of the characteristic method was then proved.

2. Mathematical analysis formulations

2.1. Governing equations

The unsteady flow through the impeller passage was assumed as a flow through a rotating duct with a diameter equivalent to the mean hydraulic diameter of the impeller passage.

Under this hypothesis, the basic equations of continuity and motions, in a reference-rotating frame related to the impeller passage and having the same origin as the fixed reference, are the following:

$$\frac{d\rho}{dt} + \rho div\overrightarrow{W} = 0 \tag{1}$$

$$\rho\frac{d\vec{W}}{dt} = \overrightarrow{gradp} + \overrightarrow{div}\bar{\bar{\tau}} + \rho\vec{F} - \rho[2\vec{\Omega}\wedge\vec{W} + \dot{\vec{\Omega}}\wedge\overrightarrow{OM} + \vec{\Omega}\wedge(\vec{\Omega}\wedge\overrightarrow{OM})] \tag{2}$$

where ρ is the density of the fluid and $\bar{\bar{\tau}}$ the stress tensor of viscosity, $\vec{F}$ the external body force, $\vec{W}$ the relative velocity and $\vec{\Omega}$ the rotational speed.

Fig. 1 shows the control volume defined by two successive blades limited by suction and discharge sides of the impeller. The principle of the study of turbo machinery is based on the velocity diagram for determining the absolute velocity at any point of the blade. In particular, point 2 represents the outlet of the impeller; the outlet absolute velocity $\vec{V}_2$ is given by the following equation:

$$\vec{V}_2 = \vec{W}_2 + \vec{U}_2 \tag{3}$$

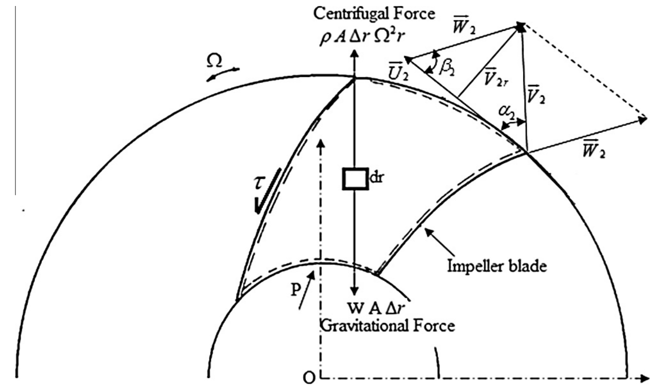

Fig. 1. Forces in rotating impeller in the frame of Refs. [5,18,19].

where $\vec{U}_2$ is the outlet tangential velocity and $\vec{W}_2$ is the relative velocity.

In a centrifugal impeller, only the radial component of the absolute velocity permits the passage of fluid from the impeller to the volute. According to the velocity diagram shown in Fig. 1, this component is:

$$\vec{V}_{2r} = \overrightarrow{V}_2 \times \sin\alpha_2 = \overrightarrow{W}_2 \times \sin\beta_2 \tag{4}$$

where β_2 is the outlet blade angle and α_2 is the angle between velocity vectors $\vec{V}_2$ and $\vec{U}_2$.

This relationship allows the passage from the rotating frame to the absolute frame related to the volute. The moving frame is used as reference to study the flow through the impeller.

In centrifugal pumps, the flows through the impeller are due to the centrifugal force that acts in the radial direction, in which Eqs. (1) and (2) are considered. As a consequence, the 3D model, represented by Eqs. (1) and (2), is reduced to the 1D model, which is an assumption used by many authors [4–6,14].

2.2. Equations for one-dimensional non-compressible flows

By the application of the mass conservation and momentum laws to an element of fluid between two sections of abscissa r and $r + dr$ of the impeller passage, we get the following equations of continuity and motion. Eqs. (1) and (2) are simplified [5,17]:

$$\frac{\partial H}{\partial t} + \frac{C^2}{gA}\frac{\partial Q}{\partial r} = 0 \tag{5}$$

$$\frac{1}{A}\frac{\partial Q}{\partial t} + g\frac{\partial H}{\partial r} + \lambda\frac{Q|Q|}{2D_hA^2} - r\Omega^2 = 0 \tag{6}$$

where A is the impeller passage section, ∂r is the space distance along the rotating passage Q is the fluid discharge, H is the head, λ is the Darcy Weisbach friction coefficient, C is the pressure wave speed and $D_h = 4lb/2(l + b)$ is the hydraulic diameter of the section A where b is the blade height and l is the width of the control volume.

The governing Eqs. (5) and (6), for the unsteady flow through the rotating impeller passage are similar to those for water hammer equations for conduit flow, except for the extra term $(r\Omega^2)$ in the equation of motion Eq. (6), which is due to the centrifugal force of the rotating impeller. Indeed, this is the force causing the dynamic head developed by the impeller.