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# HYDRODYNAMICS CHARACTERIZATION OF ROTOR-STATOR MIXER WITH VISCOUS FLUIDS

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The hydrodynamics generated by a rotor-stator mixer has been characterized experimentally in terms of power draw and flow patterns with viscous Newtonian and shear-thinning fluids. The power consumption was correlated to the impeller speed and represented by means of the dimensionless power number versus Reynolds number curve. The impeller shear rate constant, determined from a macroscopic balance as well as with the Metzner–Otto method, was found to be dependent on the fluid rheological properties. An attempt was also made to determine the pseudo-cavern (well-mixed region surrounded by a stagnant fluid) dimension as a function of the hydrodynamic parameters. Results show that the shape and magnitude of the pseudo-caverns in the vicinity of the rotor-stator mixer are central to the understanding of the relationship between the power draw and the ability of the impeller to mix the fluid into the vessel.

*Keywords: rotor-stator; power consumption; shear-thinning fluids; mixing behaviour; pseudo-cavern.*

## INTRODUCTION

Rotor-stator mixers are extensively used in many industrial applications, especially when high shear rates or high energy dissipation rates are required. For instance, they are used for their brute force ability to homogenize highly viscous materials, incorporate powders into liquids or in emulsification processes. Scientific literature on rotor-stator mixers is very scarce and information about their fundamental behaviour, including flow patterns, ability to mix and/or power consumption is seldom found. There is a significant lack of knowledge to design or predict the performance of such impellers. The primary function of a rotor-stator mixer is to impart shear to the medium. However, energy is also converted into stirring and recirculation of the product in batch applications, or pumping when used in-line. Since the energy dissipation rate is high and located in a small volume close to the mixer, the ability to recirculate may be limited by the viscosity of the product. Therefore, in large volume tanks, additional impeller (turbines, anchor, planetary mixers, and so on) promoting bulk motion is often required, which results in high capital/operating costs and also in long and tedious development time due to the trial and error nature of the design and the uncertainty of the scale-up procedure.

In industry, rotor-stator mixers are commonly used with non-Newtonian fluids. When the medium exhibits strong shear-thinning properties, flow segregations are often observed with open impellers (Doucet *et al.*, 2003). Cavern effects were first described by Wichterle and Wein (1975) as a well-mixed region around the impeller surrounded by a stagnant region in the case of yield stress fluids. Moving away from the well-mixed region where turbulence often prevails, the shear rates decrease, the viscosity increases and the fluid motion becomes laminar and nearly stagnant close to the vessel wall. Although the term 'cavern' was coined for yield stress fluids (considering such fluids exist—see Macosko, 1994 and Watson, 2004), the same phenomenon can be observed with shear-thinning fluids. In this work, we will use the term 'pseudo-cavern' to describe the cavern effect with such fluids to avoid confusion. The cavern dimension is a key process parameter to perform mixer scale-up and several models have been proposed to predict the caverns diameter and height. Wichterle and Wein (1975) first determined the cavern size by moving the agitator toward the liquid surface or vessel wall and then observing the appearance of a dye that had been added into the cavern. They proposed an empirical correlation to determine cavern size:

$$\frac{\rho N^{2-n} D^2}{m} = \left(\frac{d_C}{eD}\right)^2 \quad (1)$$

where $e$ is an experimentally determined constant function of the impeller type.

Solomon *et al.* (1981) proposed a generalized model based on torque balance, considering an idealized spherical cavern centered on the impeller (Rushton turbine). Their model assumes that the flow within the cavern is tangential,

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as in an unbaffled tank. This assumption is valid until the cavern reaches the baffles. The proposed model reads as

$$\left(\frac{d_C}{D}\right)^3 = \left(\frac{4Po}{\pi^3}\right)Re_y \quad (2)$$

where $Re_y$ is the ratio of the inertial forces due to the fluid motion to the fluid yield stress, namely

$$Re_y = \frac{\rho N^2 D^2}{\tau_y} \quad (3)$$

Using the same assumptions, Elson *et al.* (1986) modified equation (2) by considering the cavern shape as a right circular cylinder of height $h_C$ yielding

$$\left(\frac{d_C}{D}\right)^3 = \left(\frac{Po}{\pi^2((h_C/d_C) + (1/3))}\right)Re_y \quad (4)$$

These two models are valid when $D \leq d_C \leq T$. Elson *et al.* (1986) found good agreements with cavern measurements for two non-Newtonian solutions and four impeller sizes. They also found that in laminar flow, when $Re \leq 30$, the cavern diameter is constant and equal to the impeller diameter.

Hirata and Aoshima (1996) investigated cavern shapes and boundaries with laser Doppler anemometry. They assumed that the cavern border was defined by the locus of points where the local tangential mean velocity was equal to 1% of the impeller tip speed. They carefully described the cavern growth scheme versus the Reynolds number. They confirmed the validity of the Elson *et al.* model (1986) as long as $d_C \leq T$, and proposed an extended model based on viscous dissipation to deal with situations when the caverns grow up to the vessel wall under baffled conditions. Their model assumes that the representative shear stress $\tau_d$ in the mixing zone is proportional to the average shear stress $\tau_{av}$ responsible for the dissipation rate in a cavern, namely

$$\tau_d = \alpha\tau_{av} \quad (5)$$

where $\alpha$ is a constant and

$$\tau_{av} = \left(\frac{P}{V_C}\mu_a\right)^{1/2} \quad (6)$$

They finally obtained for a cylindrical cavern, assuming $\tau_d = \tau_y$ on the cavern boundary

$$\left(\frac{h_C}{d_C}\right)\left(\frac{d_C}{D}\right)^3 = \frac{4\alpha^2}{\pi}\frac{Po}{Re}Re_y^2 \quad (7)$$

Recently, a new mathematical model has been introduced by Amanullah *et al.* (1998) in order to predict torus-shaped cavern diameters using axial flow impellers. The model best applicable for extremely shear-thinning fluids needs the knowledge of the fluid velocity at the cavern boundary, a piece of information seldom known in practice.

Although the power consumption and the knowledge of the energy dissipation mechanisms in rotor-stator mixers are essential for process design, they have not received much attention in the literature. Bourne and Studer (1992) found that the energy dissipation rate was uniform within the annulus between the rotor and the stator. They identified three types of shear: the planar shear which occurs when fluid velocities change with respect to directions normal to the flow and is generated by the narrow rotor-stator gap, the elongational shear due to the rapid acceleration of the fluid, and the shear associated with the smallest eddies in turbulent flow.

Myers *et al.* (2001) studied the power draw with a high-shear homogenizer in the laminar, transitional and turbulent regimes. They obtained power curves for the up-pumping and down-pumping operating modes. In the turbulent regime, the effect of off-bottom clearance and baffling were studied. They also made an attempt to correlate a single non-Newtonian experiment with the Newtonian curve using a Metzner and Otto approach (1957).

Padron (2001) seems to be the first to have produced consistent work on the power draw of rotor-stator mixers. His thesis describes the influence of the rotor, stator and tank geometry, and flow regime with Newtonian fluids for rotational speed ranging from $25\,s^{-1}$ to $150\,s^{-1}$. When establishing the power curves, he introduced several definitions for the Reynolds numbers in order to determine the best possible correlation. The first one is the common Reynolds number for impellers in tanks, that is:

$$Re = \frac{\rho N D^2}{\mu_a} \quad (8)$$

The second definition introduces the rotor-stator gap width and the tip speed because the dissipation of power in the gap could be related to the planar shear:

$$Re = \frac{\rho V_{tip}\delta_{gap}}{\mu_a} \quad (9)$$

Another definition was also proposed, which uses for the characteristic length the hydraulic radius of the stator defined as the area of the slot divided by its wetted perimeter, namely

$$Re = \frac{4\rho V_{tip} R_h}{\mu_a} \quad (10)$$

The definition of the power number

$$Po = \frac{P}{\rho N^3 D^5} \quad (11)$$

was also subject to several variations introducing the gap width or the hydraulic radius as the characteristic length instead of the classical rotor diameter.

Padron (2001) found that the transition regime occurred at different Reynolds numbers depending on the rotor-stator type. In the turbulent regime, he suggested that the main source of power dissipation was not the viscous shear in the stator gap, but rather the dissipation in the turbulent jets discharged through the stator grid. It is also suggested