HYDRODYNAMICS OF A DUAL SHAFT MIXER WITH NEWTONIAN AND NON-NEWTONIAN FLUIDS

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Abstract: The hydrodynamic performance of a multi-shaft stirred vessel comprising a Paravisc impeller mounted on central shaft and an off-centred rotor-stator was investigated in the laminar regime with Newtonian and non-Newtonian fluids. The power consumption and the mixing time were determined using torque measurements and a discolouration method, respectively. It was found that the rotor-stator did not influence the power consumption of the Paravisc turbine in relation the poor pumping capacity of the rotor-stator. The overall mixing process and the influence of rotor-stator on the mixing process were investigated. It was found that the rotor-stator not only initiates the mixing early but also improves rate of mixing. To ensure the best mixing efficiency, the present study highlights that the Paravisc needs to be operated in such a way that the contribution of the Paravisc power draw on the overall power consumption is predominant.

Keywords: dual shaft; stirred vessel; power consumption; mixing; laminar flow.

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INTRODUCTION

Stirred vessels are commonly used in the chemical, petrochemical and cosmetic industries for the production of liquid dispersions and emulsions. The quality of the liquid dispersion and emulsion is controlled by the combination of the bulk flow and shear generated by the impellers in the vessel. This does not create major difficulties with low viscosity fluids as it is possible to operate the mixer in the turbulent flow regime. However, in many situations of practical interest involving for instance the mixing of high viscosity fluids and/or significant viscosity increase during processing, turbulence cannot be achieved. In such cases, the simultaneous generation of bulk flow and shear becomes more challenging using a single impeller system.

To address this issue, stirred vessels with independent multiple (two or more) impellers have been proposed. They comprise typically a close-clearance impeller (anchor or helical ribbon) rotating at low speed to generate the bulk flow and a high-speed impeller (rotor-stator or dispersing turbine) to produce the intense shear needed for dispersion and emulsion tasks. Two mixer configurations are used in practice. The most common approach is to mount the two impellers on a coaxial concentric shaft (see e.g., Espinosa-Solares *et al.*, 2001, 2002; Foucault *et al.*, 2006; Kohler *et al.*, 2006). Although this configuration provides the high flexibility required in operation, certain limitations are apparent. A pseudo-cavern can be formed in the high shear region of the high speed impeller that impedes a thorough homogenization in the vessel. Moreover, the gearbox and the sealing systems in coaxial mixers are costly to manufacture and maintain. In an alternate configuration, the close-clearance impeller is mounted on the central shaft, whereas the high speed impeller is installed on an eccentric shaft. The presence of eccentric impeller not only breaks the symmetry of the system but it also plays the role of a baffle. These hardware modifications influence the fluid dynamics prevailing in the vessel and lead to significant improvement in the mixing process (Kohler *et al.*, 2006). The dual shaft arrangement also simplifies the mechanical design of the mixer.

Although the dual shaft arrangement is an attractive design as compared to coaxial mixer, to our knowledge no information is available in literature on their hydrodynamic characteristics. The present work was therefore carried out to understand the hydrodynamics of such a system where a twisted anchor impeller has been combined with a rotor-stator turbine.

EXPERIMENTAL SET-UP AND METHODS

A dual shaft mixer was designed and built [Figure 1(a)]. It comprises a 50 litre cylindrical vessel with a dished bottom and two actuated shafts. A home-made Paravisc impeller (Ekato, Germany) is mounted on the low

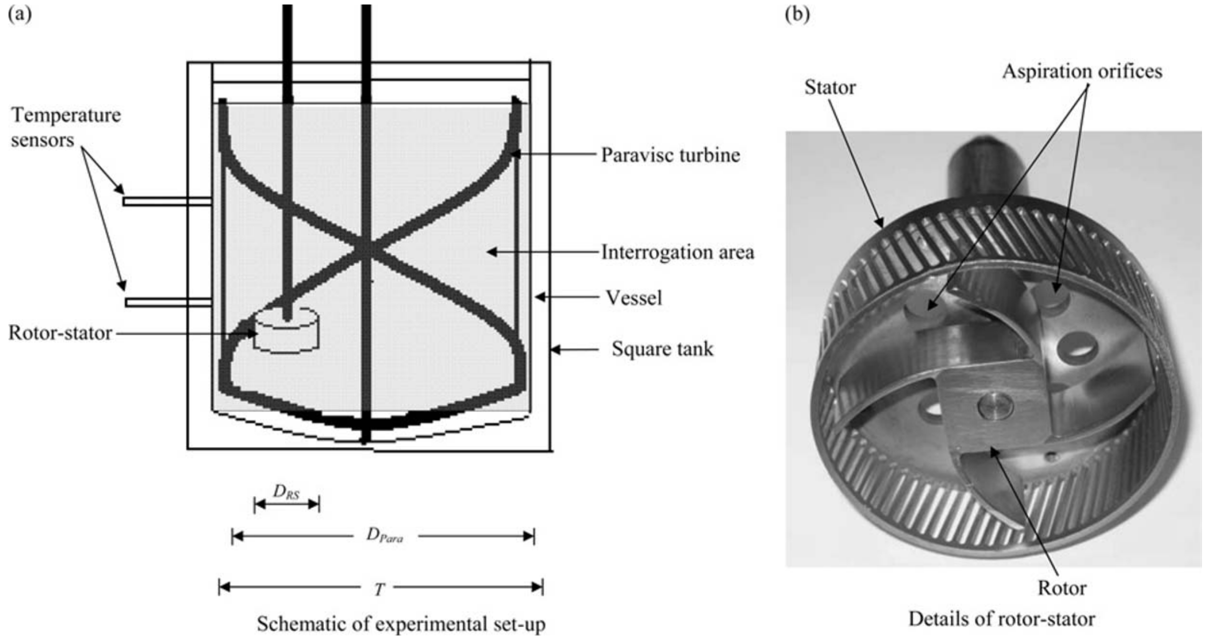

Figure 1. Experimental set-up.

speed shaft and a rotor-stator turbine (VMI-Rayneri, France) on the high-speed shaft. The impeller shafts are driven by two variable-speed motors with an accurate speed control. The rotor-stator turbine is shown in Figure 1(b). The rotor-stator turbine has four curved blades rotating in a fixed stator. The stator includes 72 inclined slotted orifices (an angle of 60° with horizontal) on the peripheral wall, eight aspiration orifices on the upper side and a free opening at the bottom. The gap width (δ) between the tip of rotor and stator is equal to 2 mm. The primary shaft supporting the Paravisc impeller is concentric with the reactor axis and extended till the bottom of the reactor. The rotor-stator shaft is eccentrically mounted and extended 0.15 m from the bottom of the reactor. The bottom clearance C equal to $T/3$ is kept constant in all the experiments. The dimensions of the vessel, impellers and shafts are summarized in Table 1.

In order to determine the power consumption of the Paravisc and the rotor-stator, a torque meter (Lorenz Messtechnik, Germany) was mounted on each shaft. The measurement range was 0–30 N-m for the Paravisc torque meter and 0–10 N-m for the rotor-stator torque meter, both with 0.1% full scale accuracy. The instantaneous torque values were measured with a sampling frequency of 2 Hz for 200 s. Two temperature sensors were also fitted to the vessel wall to monitor the fluid temperature. A LabVIEW

Table 1. Details of experimental set-up.

Vessel diameter, T	0.4 m
Liquid height, H	0.47 m
Radius of bottom curvature, Rc	0.4 m
Diameter of primary shaft, d_{s1}	0.024 m
Diameter of secondary shaft, d_{s2}	0.032 m
Diameter of Paravisc, D_{Para}	0.374 m
Height of Paravisc, H_{Para}	0.45 m
Width of Paravisc blade, W_{Para}	0.05 m
Diameter of rotor-stator, D_{RS}	0.09 m
Gap width between rotor and stator, δ	2 mm

(National Instruments, USA) data acquisition and control system has been used to adjust the rotating speeds and record the torque and temperature values. The measurements were carried out for different values of rotational speed varied between 0–70 rpm for the Paravisc and 0–1000 rpm for the rotor-stator. All the measurements were carried out at a vessel temperature equal to 22°C (±1°C).

The average power consumption was obtained from the measurement of instantaneous values of torque. The impeller power consumption was calculated as

$$P = \frac{2\pi N}{m}\sum_{i=1}^{m} T_i' \qquad \text{and} \qquad N_P = \frac{P}{\rho N^3 D_i^5} \tag{1}$$

where T_i' is the instantaneous values of torque (obtained after subtracting the residual torque value), N is the impeller rotational speed, D_i is the diameter of the impeller and m is the number of torque values collected over 200 s. The measurements were performed three times for each trial to check the reproducibility of the results. The average values calculated using the three sets are reported in the present study.

Both Newtonian and non-Newtonian fluids were considered. The rheological properties of these fluids were determined using a Bohlin AR-2000 viscometer with the Couette configuration. An aqueous solution of glucose (partially hydrolysed corn syrup) at various concentrations was used as Newtonian fluid. The viscosity (μ) of glucose solution was varied from 2.3 to 10.4 Pa s by varying the concentration of water and the density ranged between 1372–1384 kg m^{-3}. An aqueous solution of carboxy methyl cellulose (CMC) at two different concentrations (1.0 and 1.6% wt/wt in water) was used as non-Newtonian fluid. It was found that the CMC solution ($\leq$1.6% wt/wt in water) did not exhibit elasticity and its viscosity could be well described by the Carreau model (Ulbretch and Carreau, 1985). The shear thinning index (n) of the 1.0% and 1.6% CMC solution