HYDRAULIC CHARACTERISTICS OF TURBINE IMPELLER* **

Ivan FOŘT and Jana MALÁ

Department of Chemical Engineering,
Prague Institute of Chemical Technology, 166 28 Prague 6

Received April 29th, 1980

The study concentrates on the determination of hydraulic characteristics (volumetric flow rate, power output, hydraulic efficiency and total head) of a standard, six-blade turbine impeller in a flat-bottomed cylindrical vessel provided with radial baffles at a turbulent regime of agitated liquid. The investigated characteristics are determined by means of a macroscopic balance of mechanical energy of the impeller region expressed in a dimensionless form. The results show that all the hydraulic characteristics investigated, with the exception of the dimensionless impeller total head, are independent of the impeller-to-vessel diameter ratio. The hydraulic efficiency of a standard turbine impeller is about 40%.

The purpose of this study is to determine hydraulic characteristics of a standard turbine impeller rotating in a cylindrical flat-bottomed vessel provided with radial baffles at the wall under turbulent flow regime, using mass and energy balances of the impeller region. The hydraulic characteristics obtained – volumetric flow rate, power consumption, total head and hydraulic efficiency – may serve as initial data for the assessment of the dispersing efficiency the investigated system.

Overall mass and energy balances of an open system have been described by Bird[1]. Nagata and coworkers[2,3], Cutter[4], Fořt and coworkers[5] and Möckel[6] carried out measurements of velocity distribution in the vicinity of a turbine or a flat-blade impeller with vertical or inclined blades. The experiments were undertaken with newtonian fluids in turbulent regime, with or without baffles. Using a mechanical energy balance of the cylindrical volume circumscribed by the impeller the energy dissipated in the impeller region as well as in the remaining liquid volume per unit time was determined. For a six-blade impeller with flat blades of inclination angle $\alpha = 45°$ in a vessel with radial baffles it was found[5] that in the rotor region about one third of impeller power input was dissipated, and for a standard turbine impeller with vertical blades the dissipation amounted almost to two thirds of impeller power input. Cutter[4] states that in the system with radial bafles only about one fifth of impeller input is dissipated in the rotor region of a standard turbine impeller. His conclusions, however, suffer from an error of neglecting in the mechanical energy balance the difference of the static pressure distribution in the rotor region from that of hydrostatic pressure. But under the influence of impeller rotation the pressure field in the rotor region changes.[7] Thus the true value of static pressure, or its profile along the boundaries of rotor region, has to be considered.

* Part LVIII in the series Studies on Mixing; Part LVII: This Journal *47*, 226 1982.

** Delivered at the Conference "Mixing Processes" (Mischvorgänge), Cologne 1980.

THEORETICAL

Let us consider an agitated system made up of a flat-bottomed cylindrical vessel provided with four radial baffles of width $b = 0{\cdot}1D$ (Fig. 1). The vessel is filled with a low-viscosity newtonian liquid up to the height H equal to the inner diameter D. A standard six-blade turbine impeller with a separating disc of diameter d rotates at the height $H_2 = D/3$ above the bottom. In the vessel a cylindrical coordinate system is introduced, its origin coinciding with the intersection of the symmetry axis of the vessel and the flat bottom. Thus the coordinate z coincides with the vessel symmetry axis.

In the agitated system the following regions are defined: rotor region (Fig. 2), cylindrical region of volume V_m delineated by horizontal cross sections I, I' and cylindrical surface II between them (lateral area of a cylinder between the two cross sections). The dimensions of this region (diameter and height) are almost equal to the impeller diameter and the height of its blades. For this region the following simplifying assumptions are introduced: *1*) The stirred liquid is incompressible, *2*) The rotor region is axisymmetrical. *3*) The flow in the rotor region is quasistationary *4*) The flow regime in the rotor region is fully turbulent. Thus, for the rotor region, the continuity equation may be written

$$2\pi \int_0^{d/2} [\bar{w}_{ax}(r)]_I \, r \, dr + 2\pi \int_0^{d/2} [\bar{w}_{ax}(r)]_{I'} \, r \, dr = \pi d \int_{-h/2}^{h/2} [\bar{w}_{rod}(z)]_{II} \, dz = \dot{V}_p \,, \qquad (1)$$

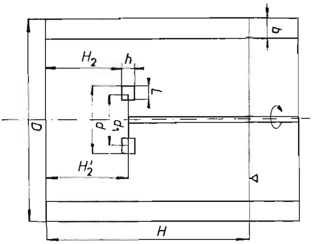

FIG. 1

Stirred system with standard turbine impeller. $h/d = 0{\cdot}20$; $L/d = 0{\cdot}25$; $d_1/d = 0{\cdot}75$

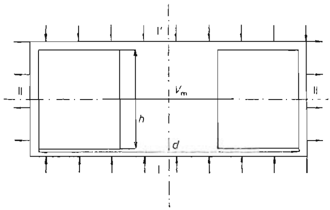

FIG. 2

Rotor region of turbine impeller

or in the dimensionless form, using the dimensionless Stokes function

$$2\pi\Psi_{\mathrm{I}} + 2\pi\Psi_{\mathrm{I'}} = 2\pi\Psi_{\mathrm{II}} = K_{\mathrm{p}} = \dot{V}_{\mathrm{p}}/nd^3\,, \tag{2}$$

the quantity K_p expressing the volumetric flow rate through the impeller $\dot{V}_p$ in a dimensionless form.

Considering assumptions *1–4*, the macroscopic balance of mechanical energy for the rotor region V_m may be expressed:

$$\dot{E}_{\mathrm{I}} + \dot{E}_{\mathrm{I'}} + e_{\mathrm{sp}}\dot{V}_{\mathrm{p}} = \dot{E}_{\mathrm{II}}\,, \tag{3}$$

where the energy inflows into the region V_m are

$$\dot{E}_{\mathrm{I}} = 2\pi \int_0^{d/2} [E(r)]_{\mathrm{I}}\,[\bar{w}_{\mathrm{ax}}(r)]_{\mathrm{I}}\, r\,\mathrm{d}r \tag{4}$$

and

$$\dot{E}_{\mathrm{I'}} = 2\pi \int_0^{d/2} [E(r)]_{\mathrm{I'}}\,[\bar{w}_{\mathrm{ax}}(r)]_{\mathrm{I'}}\, r\,\mathrm{d}r \tag{5}$$

and the energy outflow from this region is

$$\dot{E}_{\mathrm{II}} = \pi d \int_{-h/2}^{h/2} [E(z)]_{\mathrm{II}}\,[\bar{w}_{\mathrm{rad}}(z)]_{\mathrm{II}}\,\mathrm{d}z\,. \tag{6}$$

The radial profiles of the axial mean velocity component in the cross sections I and I′, $[\bar{w}_{\mathrm{ax}}(r)]_{\mathrm{I}}$ and $[\bar{w}_{\mathrm{ax}}(r)]_{\mathrm{I'}}$, as well as the axial profile of the radial velocity component in the cross section II, $[\bar{w}_{\mathrm{rad}}(z)]_{\mathrm{II}}$ are determined experimentally. Likewise, the energy per unit volume of liquid flow rate through the k-th cross section

$$[E(r)]_{\mathrm{k}} = [\bar{p}_{\mathrm{st}}(r)]_{\mathrm{k}} + \varrho/2\{[\bar{w}^2(r)]_{\mathrm{k}} + [\overline{w'^2}(r)]_{\mathrm{k}}\} + z_{\mathrm{k}}\varrho g\,,\quad [k = \mathrm{I}, \mathrm{I'}]\,, \tag{7}$$

or

$$[E(z)]_{\mathrm{II}} = [\bar{p}_{\mathrm{st}}(z)]_{\mathrm{II}} + \varrho/2\{[\bar{w}^2(z)]_{\mathrm{II}} + [\overline{w'^2}(z)]_{\mathrm{II}}\} + H'_2\varrho g\,. \tag{8}$$

are determined from the experimental results.

The quantity e_{sp} expresses the energy delivered by the impeller to the unit volume of liquid of the flow rate $\dot{V}_p$, reduced by the energy dissipated in the region V_m.

Then, the impeller energy output, *i.e.* the quantity of energy dissipated outside the rotor region per unit time, is

$$N_t = r_{sp}\dot{V}_p \tag{9}$$

and, finally, the impeller total head

$$h = e_{sp}/\varrho g \,. \tag{10}$$

Both the hydraulic characteristics of the turbine impeller defined above may be expressed in a dimensionless form as an output criterion

$$Po_t = N_t/\varrho n^3 d^5 \tag{11}$$

and as a dimensionless impeller total head

$$\varkappa = hg/n^2 d^2 \,. \tag{12}$$

If the dimensionless input criterion is introduced (by means of impeller input N)

$$Po = N/\varrho n^3 d^5 \,, \tag{13}$$

the hydraulic efficiency of the impeller may be calculated from the relation

$$\eta_h = Po_t/Po \,. \tag{14}$$

All the introduced hydraulic characteristics may generally be functions of the system geometry. At turbulent flow regime, however, they may not depend on the Reynolds mixing number.

EXPERIMENTAL

Hydraulic characteristics of the turbine impeller have been calculated from the known values of velocity and pressure on the boundaries of the rotor region. The investigation of the velocity field was carried out in cylindrical perspex vessels of inner diameters $D = 0{\cdot}290$ m and $D = 1{\cdot}0$ m. The vessels were provided with four baffles of width $b = 0{\cdot}1D$ reaching to the bottom, and filled with distilled water up to the height $H = D$. Standard six-blade turbine disc impellers of diameters $d = D/4$ and $d = D/3$ placed in the vessel axis were employed (Fig. 1). The elevation of the horizontal plane of separating disc above the vessel bottom was $H_2' = D/3$. The impeller was driven by a d—c electric motor, the frequency of revolution of which was controlled by an electromagnetic regulator connected with a photoelectric sensor, with an accuracy of $\pm 1{\cdot}0\ \text{min}^{-1}$.

As dependent variables the profiles of the quantity $[E(r)]_k$, $[k = \text{I}, \text{I}']$ in cross sections I and I′ and the profile $[E(z)]_{\text{II}}$ (Eqs 7 and 8) in cross section II were measured by the directional three- and five-hole Pitot tubes[8–9], respectively. The corresponding mean velocity components (axial components in the cross sections I, I′ and radial component in the cross section II) were determined by means of hot film anemometer probes,[8,10] using the information about the flow direction from the measurements by the Pitot tubes. As the measurements were undertaken at more than one impeller frequency (usually at two frequencies) in two geometrically similar systems of different dimensions, both the dependent and independent variables were expressed in a dimensionless form, with coordinates

$$R = 2r/d\,, \quad z = 2(z - H_2')/h \tag{15a,b}$$

and mean velocity components

$$W_{ax} = \overline{w}_{ax}/\pi dn\,, \quad W_{rad} = \overline{w}_{rad}/\pi dn\,. \tag{16a,b}$$

For the energy of unit volume of liquid flow rate through the k-th cross section the following expressions were used

$$[\varepsilon(r)]_k = [E(r)]_k/\varrho(\pi dn)^2\,, \quad [k = \mathrm{I}, \mathrm{I}']\,, \tag{17a}$$

$$[\varepsilon(z)]_{\mathrm{II}} = [E(z)]_{\mathrm{II}}/\varrho(\pi dn)^2\,. \tag{17b}$$

Thus, for the given relative vessel size d/D two velocity profiles, for cross sections I and II, were evaluated. Hydraulic characteristics of the turbine impeller were calculated from the relations introduced in the Theoretical. The mass and energy flow rates in the boundary cross sections of the region V_m were determined by the numerical integration over these cross sections, replacing their analytical expression (Eqs *1, 2*) and Eqs (*3–5*). The value of the input criterion *Po* for the investigated configurations of the stirred system at a turbulent regime was taken ever from the literature[11]

$$Po = 5{\cdot}6\,, \quad [Re_M > 1{\cdot}0\,.\,10^4]\,. \tag{18}$$

RESULTS AND DISCUSSION

Figs 3 and 4 show the examples of the profiles of axial mean velocity component in cross section I (Fig. 3) and radial mean velocity component in cross section II (Fig. 4), expressed in a dimensionless form (Eqs *15* and *16*). The radial profile of W_{ax} in cross sections I and I′ and the corresponding turbulentce intensity is low. On the other hand, the axial profile of W_{rad} in cross section II reaches its maximum in the vicinity of the axial coordinate of the impeller disc $[Z = 0]$ and the zero value close to the outer boundaries of the rotor region $[Z = \pm 1]$. The turbulenece intensity of the flow leaving the rotor region reaches considerable values[12].

The consistency of the used experimental data was checked by the comparison of volumetric flow rates through cross sections I and I′ with that through cross section II (Eqs *1* and *2*), *i.e.* by testing the validity of continuity equation of incompressible fluid. The results given in Table I show a good coincidence between the compared volumetric flow rates for the given relative size of the impeller d/D. In accordance with further quoted studies[9,12,13] the value of the flow rate criterion K_p increases with the increasing value of the ratio d/D.

Table II sums up the calculated values of the hydraulic characteristics of the turbine impeller. For the relative impeller size $d/D = 2/5$ the values are partly taken over, partly calculated from Möckel's data[6], The values of hydraulic characteristics in Table II imply that dimensionless input and hydraulic efficiency are practically independent of the impeller relative size, whereas the dimensionless impeller total head decreases with the increasing value of the ratio d/D.

With respect to the accuracy of the determination of the original quantities ($\overline{w}_{ax}$, $\overline{w}_{rad}$, E) the values of calculated hydraulic characteristics were determined with an accuracy of 15%–30%.

In accordance with the adopted assumption *4* the flow in the rotor region is supposed to be fully turbulent. This has been secured by choosing such frequency of revolutions and impeller sizes that the Reynolds mixing number values were always greater than forty thousand.

The incompressibility of the stirred liquid was secured by the used liquid (water) at a constant temperature without significant changes of the atmospheric pressure above the free liquid surface.

The quasistationary character of flow in the rotor region was attained by a careful impeller speed control as well as by choosing sufficiently long time interval, relative to the reciprocal value of impeller frequencies of revolution, to obtain the corresponding mean time values.

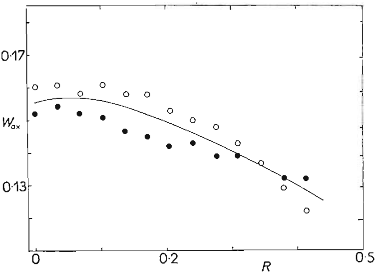

FIG. 3

Dependence of the dimentionless axial mean velocity component W_{ax} on the dimensionless radial coordinate R — region I. $d/D = 1/3$; $D = 0{\cdot}290$ m; ○ $n = 534\ \text{min}^{-1}$; ● $n = 750\ \text{min}^{-1}$

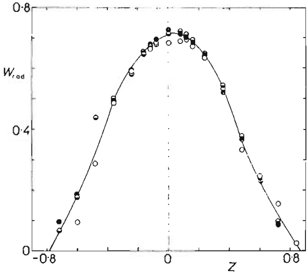

FIG. 4

Dependence of the dimensionless radial mean velocity component W_{rad} on the dimensionless axial coordinate Z — region II. $d/D = 1/4$; $D = 1{\cdot}0$ m; ○ $n = 100\ \text{min}^{-1}$, ◒ $n = 150\ \text{min}^{-1}$ ● $n = 200\ \text{min}^{-1}$

The assumption of axial symmetry of the region V_m was fulfilled by placing the impeller coaxially with the symmetry axis of the vessel. The impeller speed reached such values that, with regard to the number of turbine impeller blades (six), the region V_m could be considered compact.

The average value of the hydraulic efficiency of the standard turbine impeller is about 40%. This result corresponds well to the published data of Nagata and coworkers[2,3]. From the comparison of this value of η_h with that for an impeller with flat inclined blades (6 blades, $\alpha = 45°$), which is about 66% (ref.[5]), we may conclude that the turbine will be more suitable for producing intensive turbulence (about 60% of its power are dissipated in the rotor region V_m), whereas the impeller with flat inclined blades will be more efficient for attaining higher pumping effect at the given value of the impeller input. This difference may also be illustrated by the fact that the dimensionless total head $\varkappa$ (characterizing the energy dissipated outside the rotor region) in a vessel with the impeller with flat inclined blades is about twice as great as $\varkappa$ value in a system with a turbine impeller.

By comparison of the energy dissipation rate per unit volume in the volume V_m and in the remaining volume of liquid the extent of inhomogeneity of spatial distribu-

TABLE I

Testing of validity of continuity equation in the region V_m by means of the quantity Ψ

d/D	Input I + I′	Output II	K_p
1/3	0·148	0·151	0·937
1/4	0·108	0·103	0·660

TABLE II

Hydraulic characteristics of turbine impeller

d/D	Po_t	$\varkappa$	η_h
1/4	2·432	2·740	0·434
1/3	2·115	2·230	0·378
2/5[a]	2·223	2·127	0·398
Average value	2·259	—	0·403

[a] Taken over from study[6].

tion of mechanical energy dissipation in the system may be evaluated. This inhomogeneity is more than twice as high in a system with turbine impeller than in a similar one with an impeller with flat inclined blades. This proves that the turbine impeller is suitable for such processes, where turbulence in the rotor region (*e.g.* dispersion of gas into liquid, emulgation of immiscible liquids) is of major significance, whereas the impeller with flat inclined blades should be employed in processes requiring intense convective flow and turbulence in the continuous phase of the stirred liquid, as *e.g.* in homogenation of miscible liquids or in suspending of solid particles.

LIST OF SYMBOLS

b	baffle width, m
D	vessel inner diameter, m
d	impeller diameter, m
d_1	diameter of turbine impeller separating disc, m
E_k	energy per unit volume of liquid flow entering or leaving region V_m [k = I, I′], [k = II], resp., Nm
E_k	energy flow rate into/from impeller rotor region [k = I, I′], [k = II], resp. Nm
e_{sp}	energy per unit volume given by impeller to volumetric flow rate $\dot{V}_p$ reduced by energy per unit volume dissipated in impeller rotor region V_m, Nm
H	liquid still height, m
H_2	height of lower edge of impeller blade above vessel bottom, m
H_2'	height of turbine impeller separating disc above vessel bottom, m
h	impeller total head, m
h	impeller blade height, m
j	index
K_p	flow rate criterion
k	index
L	impeller blade length, m
N	impeller power input, W
N_t	impeller power output, W
n	impeller frequency of revolution, s^{-1}
Po	impeller input criterion
Po_t	impeller output criterion
p_{st}	mean static pressure, Pa
R	dimensionless radial coordinate
r	radial coordinate, m
Re_M	Reynolds mixing number
V_m	impeller rotor region volume, m^3
$\dot{V}_p$	volumetric flow rate through impeller rotor region or impeller pumping capacity, $m^3 s^{-1}$
W_j	dimensionless value of axial [j = ax] or radial [j = rad] mean velocity component
$\bar{w}_j$	axial [j = ax] or radial [j = rad] mean velocity component, m s^{-1}
w'	fluctuation velocity component, m s^{-1}
Z	dimensionless axial coordinate
z	axial coordinate, m
α	impeller blade inclination angle, °

ε_k dimensionless value of energy per unit volume in liquid flow entering or leaving region V_m [k = I, I′], [k = II], resp.
η_h hydraulic efficiency of impeller
$\varkappa$ dimensionless impeller total head
ϱ density, kg m^{-3}
Ψ' dimensionless Stokes stream function

REFERENCES

1. Bird R. B.: Chem. Eng. Sci. *6*, 123 (1957).
2. Nagata S., Yamamoto K., Ujihara M.: Mem. Fac. Eng., Kyoto Univ. *30*, 336 (1958).
3. Nagata S., Yamamoto K., Hashimoto K., Naruse Y.: Mem. Fac. Eng. Kyoto Univ. *31*, 260 (1959).
4. Cutter L. A.: AIChE J. *12*, 35 (1966).
5. Fořt I., Neugebauer E., Pastyříková H.: This Journal *36*, 1769 (1971).
6. Möckel H. O.: Chem. Tech. (Leipzig) in press.
7. Riet Van't K., Bruijn W., Smith J. M.: Chem. Eng. Sci. *41*, 40 (1976).
8. Fořt I., Hrach M., Obeid A.: Sb. Vys. Šk. Chemicko-Technol. Praze *K 15*, 37 (1980).
9. Fořt I., Drbohlav J., Krátký J., Grospičová M., Kroužilová Z.: This Journal *37*, 222 (1972).
10. Drbohlav J., Fořt I., Krátký J.: This Journal *43*, 696 (1978).
11. Czechoslovak Standard 69 10 21, Brno 1969.
12. Drbohlav J., Fořt I., Máca K., Ptáček J.: This Journal *43*, 3148 (1978).
13. Souza de A., Pike R. W.: Can. J. Chem. Eng. *50*, 15 (1972).

Translated by M. Procházka.