TURBULENT CHARACTERISTICS OF THE VELOCITY FIELD IN A SYSTEM WITH TURBINE IMPELLER AND RADIAL BAFFLES*

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The velocity field is examined in the paper in the stream of liquid ejected from the blades of a turbine impeller located axially in a cylindrical vessel with radial baffles. Attention was paid to turbulent characteristics in the examined stream: the intensity of turbulence, the energy spectrum of turbulence and the Lagrange macro-scale of turbulence. The experiments were carried out by means of a constant temperature hot-film anemometer. The experimental results indicate a strong effect of the source of turbulence — the impeller — on its character. The intensity of turbulence in the examined stream reaches values an order of magnitude higher (*i.e.* 40–80%) over those found in a pipe, and, moreover, displays conspicuously non-uniform profile across the stream. From the energy spectra it follows that the character of turbulence, and particularly in the proximity of the impeller, is markedly non-isotropic and non-homogeneous. Yet, no significant profile of the normalized spectra across the stream was found. The spectra were normalized by the frequency averaged fluctuation energy in a given position. The analysis of the spectra further indicates the existence of both the inertia and the dissipation subrange of the equilibrium region. The former appears near the frequency of the impeller blades. The effect of the geometry (*i.e.* the size the shape of the impeller), expressed in terms of the Lagrange macro-scale of turbulence, reaches near the outher edge of the blades practically their width. From here on this characteristic decreases strongly with radial distance.

The paper deals with energy properties of the turbulent flow in a system with turbine impeller and radial baffles. The measurements were performed in the stream ejected from the blades of the rotating impeller.

The intensity of turbulence in a mixed system with radial high-speed impellers and radial baffles was investigated by Oldshue[1], Liepe, Möckel and Winkler[2], Sato and coworkers[3], Mujumdar and coworkers[4], Rao and Brodkey[5] and Cho, Armanath and Becker[6] (Table I). The last authors report the intensity of turbulence an order of magnitude lower than the others. Aside from the above authors using constant temperature anemometers one could further put forth the paper of Cutter[7] who used a photographic technique. The Lagrange scale of turbulence has been determined by a number of authors, *e.g.* ref.[2,3,5,6], from the time course of the autocorrelation function and the known average velocity in a given position within the stream. The integration of the Lagrange's autocorrelation function $r_L = r_L(\tau)$ though covered only the range where the random component had been correlated and not the range affected by the periodic

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component of the frequency of the impeller. A similar approach was used also by Mujumdar and coworkers[4] to calculate the intensity of turbulence in the stream of liquid ejected from the blades of a rotating turbine impeller; the measured intensity was corrected on the effect of the mentioned periodic component of the fluctuation. Thus they take into consideration only that component of the intensity induced by random fluctuations. The corrected intensity of turbulence is then considerably lower than that measured. The energy spectra of turbulence* have become the most widely used characteristic of turbulence in mixed systems and one has thus a large number of experimental results available in the literature. These results[2-6,8] may be summed up as follows: The largest fraction of energy supplied by the impeller is carried by large vortices originated by the blades of the impeller. The energy spectrum exhibits a maximum for frequency f_m of the impeller blades (the frequency of revolution of the impeller multiplied by the number of the blades). On the obtained spectra one can also see a local maximum at frequency equalling $2f_m$, see *e.g.* ref.[2-5], provided that the measurements were carried out in a horizontal plane of sym-

TABLE I

Radial Profiles of Intensity of Turbulence Published in Literature

Reference	Operating conditions and radial profiles of intensity of turbulence	Experimental method
1	$D = 0.442$ m, $d/D = 1/3$, $H/D = 10/9$, $H_2/D = 1/3$, $b/D = 1/10$, $R = 0.083$, $Z = 0$, $n = 131$ min^{-1}, $\delta_w = 0.45$, $n = 251$ min^{-1}, $\delta_w = 0.44$	anemometer in water[a]
2	$D = 0.4$ m, $d/D = 0.35$, $H/D = 1$, $H_2/D = 0.35$, $b/D = 1/10$, $R = 0.20$, $Z = 0$, $n = 400$ min^{-1}, $\delta_w = 0.39$	anemometer in water[a]
7	$D = 0.294$ m, $d/D = 1/3$, $H/D = 10/9$, $H_2/D = 0.5$, $b/D = 0.10$, $Z = 0$, $R = 0.065, 0.037, 0.24$, $n \in (200; 600)$ min^{-1}, $\delta_w = 0.41, 0.38, 0.49$	photographing in water[a]
3	$D = 0.21$ m, $d/D = 1/2$, $H/D = 1$, $H_2/D = 1/2$, $b/D = 1/13.3$, $R = 0.28, 0.52$, $Z = 0$, $n = 72$ min^{-1}, $\delta_w = 0.35, 0.35$	anemometer in kerosene
4	$D = 0.384$ m, $d/D = 1/3$, $H/D = 1/3$, $H_2/D = 7/15$, $b/D = 1/10$, $R = 0.5, 0.10$, $Z = 0$, $n \in (600; 700$ min$^{-1})$, $\delta_w = 0.39, 0.38$.	anemometer in air; liquid level simulated by upper lid
5	$D = 0.295$ m, $d/D = 1/3$, $H/D = 1.5$, $H_2/D = 1/2$, $b/d = 1/8$, $R = 0.05, 0.2, 0.4, 0.6$, $Z \in \langle -0.8, 0.8 \rangle$ $n = 300$ min^{-1}, $\delta_w \in 0.50, 0.60$	anemometer in water

[a] All measurements with a six-blade turbine except ref.[3,5] with eight-blade turbine and vertical blades.

* In view of the experimental technique we are talking about the one-dimensional energy spectra.

metry of the impeller and in the proximity of the rotating blades. In accord with the general laws, see *e.g.* ref.[9,10], the spectra of turbulence may be divided into following three regions:

1) A region of large vortices starting from zero frequency up to that close to the frequency of blades. These vortices carry a relatively small fraction of total energy. The normalized dissipated energy, $E^*(f)$, increases or remains approximately constant with increasing frequency.

2) A region of vortices carrying the major fraction of energy appears around the frequency f_m. $E^*(f)$ reaches here its maximum.

3) Equilibrium region consisting of two subranges: the inertia and the dissipation one. From the cited papers it follows that in the second of the subranges even highly nonisotropic systems displays isotropic behaviour and many authors[2–4] find the assumptions of isotropic behaviour well satisfied even in the first of the subranges. From the results of Kim and Manning[8] it further follows that the normalized energy spectra of turbulence (Eq. (*3*)) are very similar regardless of the size of the impeller, its r.p.m., and, within certain limits, also of the direction of the vector of local average velocity. The maximum of the energy supplied by the impeller at its frequency and its doubled frequency, however, sharply decreases with the radial distance from the outer edge of the blades. These conclusions have been also confirmed by Liepe and coworkers[2] who, moreover, found that with increasing radial distance from the rotating impeller the region of isotropic turbulence shifts toward lower frequencies eventually up to the inertia subrange. All the so far published results were obtained on laboratory-scale apparatuses, *i.e.* under conditions substantially different from those in full-scale systems. As a consequence, the analyses of turbulence that have been performed to date have been those at high r.p.m. of the impeller and little attention has been paid to low frequency region of the energy spectra. Equally so, the results of measurements in the proximity of the rotating blades have not been so far made available. Accordingly, the goal of this study is to supplement and extend the investigations in the proximity of the impeller in a system nearing in size the full-scale equipment.

EXPERIMENTAL

The equipment for calibrating the hot-film anemometer consisted of a duct with circulating distilled water. In the circulation loop there was an axial pump, a honeycombed flow-directing device and a confusor with the calibrated probe on its end. The selected shape of the confusor ensured potential flow of liquid around the probe[11]. The temperature was held constant within $\pm$ 0·05°C, the r.p.m. of the pump to $\pm 0{\cdot}5\%$ of the preset value.

The measurements were carried in a cylindrical perspex vessel with flat bottom 1 m in diameter equipped with four bottom reaching radial baffles. The width of the baffles, b, was $b = 0{\cdot}1\ D$. The vessel was filled with distilled water ($H = 1$ m). An axially located six-blade turbine impeller with flat vertical blades and a separating disc of $d = 0{\cdot}25$ m was in all experiments located with the bottom edge of the blades 0·308 m above the bottom of the vessel. The ratio of the characteristic dimensions was $d : h : L = 20 : 4 : 5$. The r.p.m. of the impeller were held constant and equal $n = 200\ \text{min}^{-1}$. The driving unit of the impeller, the technique of measurement of r.p.m. as well as other parts of the experimental set up, including the mounting of the probe enabling its traverse and rotation within the system, have been described in detail in one of the preceding papers[12] of this series. To measure the turbulent characteristics a Disa 55A01 anemometer was used jointly with a film wedge-shaped probe of the Disa 55A81 or 55A83 type. The measurements were carried out in distilled water at $20 \pm 0{\cdot}05$°C and constant temperature of the heated film of the probe. The degree of overheating was selected 0·05 and held constant.

The anemometer used was not designed for measurement in the velocity field of considerable intensity of turbulence as it is the case in the examined stream ejected from the impeller blades.

To determine the time-averaged voltage of the anemometer bridge a new auxiliary device was developed. The signal of the anemometer was brought *via* voltage divider onto the averaging amplifier (for wiring see Fig. 1) consisting of an amplifier with feedback formed by a capacitor and a resistance in parallel. The frequency to be smoothed by this device was given by the time constant depending on the capacitor in the feedback circuit only. The processed DC signal was brought into a digital voltmeter connected to a typewriter. This combination enabled the average voltage of the anemometer bridge output to be determined. For the purpose of determining the intensity of turbulence and its energy spectra the AC signal from the anemometer was recorded on a tape recorder *via* frequency modulator. The signal of the demodulator was then fed into a precision AC voltmeter Orion EMG TS 1201 enabling the square root of the mean square fluctuation voltage to be read off. The ratio of this quantity and the average voltage provided the sought intensity of turbulence in the location of the probe. To obtain the energy spectra the signal of the tape recorder was processed by the above mentioned modulator and fed into a frequency analyser Bruel and Kjaer 2109. The spectra were thus directly analyzed starting from a minimum possible frequency f_1 up to a frequency f_2 given by the level of the noise. The measurement was terminated when the intensity of the signal was comparable with the background noise. In order to obtain the energy spectra at very low frequencies the following procedure was used: The signal of the anemometer was recorded on the tape recorder at 4·75 cm/s and played back at 19·05 cm/s. This transformation enabled analysis of the richest part of the spectrum in the range of the frequency of the impeller because the center of the band of the lowest filter of the analyser used was 25 Hz which corresponded to the true frequency of 6·25 Hz. The attenuation of a third-octave filter of the frequency analyser was 3 db per 1/3 of octave and then 40 db per octave.

The measurement in the mixed system consisted of mounting the hot-film probe in a preselected point in the stream of liquid ejected from the blades of the impeller and recording the output signal of the anemometer bridge. The height of liquid in the vessel as well as the distance of the lower edge of the blades from the bottom were held constant. For independent variables we took

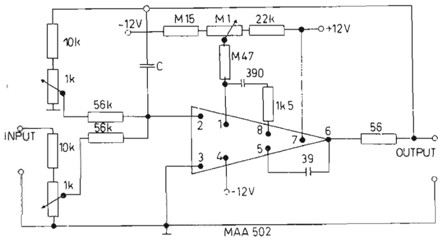

FIG. 1

Wiring of the Integrating Amplifier

C, μF	Time constant, s
2	5
4	10

the frequency of the impeller, n, the radial distance of the probe from the axis, r, the axial distance of the probe from the horizontal plane of symmetry of the impeller, z, the angle between the axis of the probe and a vertical plane on one hand and a horizontal plane on the other hand, and, finally, the temperature of the batch. The accuracy of setting is the same for all variables as in the already cited paper[12] of this series. The dependent variable — the fluctuation voltage of the anemometer bridge — was recorded on the tape recorder. In assessing the accuracy of determining as well as processing this quantity one has to consider the accuracy of single devices used. From the data supplied by manufacturers and the tests follow these accuracies: Anemometer Disa 55A01 $\pm 1\%$, frequency modulator Aritma TKM-2: 2% (up to 1·5 kHz), tape recorder Tesla B 43: $\pm 5\%$ (tape Agfa PA 41), integration circuit: $\pm 0{\cdot}5\%$ and digital voltmeters Tesla BM 480 and Tesla BM 445 E: $\pm 0{\cdot}5\%$. All the above accuracies are mutually practically independent.

The probe was calibrated by taking the following relation between the average voltage of the anemometer and the average velocity of water passing around the hot-film probe as[13,14]

$$U^2 = A + B\bar{w}^{1/2} . \tag{1}$$

The results of the calibration indicate that the empirical relation (*1*) suits well and the average velocity can be determined in the range of 0·3—0·6 m/s with relative deviation better than $\pm 3\%$; above 0·6 m/s with the accuracy better than 2%. A comparison of the constants A, B at three different temperatures of liquid suggests that these are directly proportional to the temperature difference between the hot film and the liquid. This finding is in agreement with the theoretical analysis[15] of temperature properties of the constants of the King's law at constant temperature of the hot film. The results indicate that a correction of these parameters on the temperature of

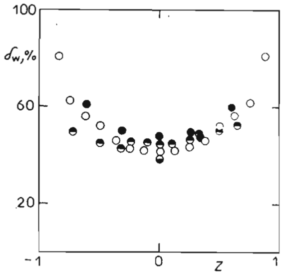

FIG. 2

Axial Profiles of the Intensity of Turbulence in the Stream of Liquid Ejected from the Blades of the Turbine Impeller

○ Cutter[7], ◒ Mujumdar and coworkers[4], ◓ Oldshue[1], ● this work.

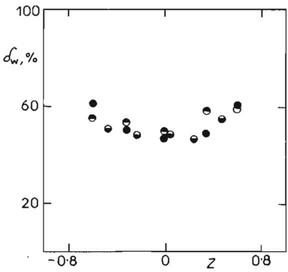

FIG. 3

Axial Profiles of the Intensity of Turbulence in the Stream of Liquid Ejected from the Blades of the Turbine Impeller

● $R = 0{\cdot}04$, ◓ $R = 0{\cdot}24$, ◒ $R = 0{\cdot}36$.

liquid is necessary in order to preserve the accuracy of velocity measurements. The accuracy of the calibration relation requires correction for temperature changes as little as $\pm 0{\cdot}2°C$. The same applies also to the calculation of the average and the fluctuation velocity component.

RESULTS

THE TURBULENT CHARACTERISTICS OF LIQUID EJECTED FROM THE BLADES OF A TURBINE IMPELLER

Intensity of Turbulence

The measured axial profiles of the intensity of turbulence, defined by

$$\delta_W = \left(\overline{w'^2}\right)^{1/2} / \overline{w} \tag{2}$$

agree well with the experimental data published in the literature[1,4,7] although the sizes of impellers in the cited papers (d/D) is different from the relative size of the impeller used in this work (Table I and Fig. 2). From the results it follows that the intensity of turbulence reaches a minimum in the center of the stream just departing from the blades of the turbine impeller and increases the absolute value of the coordinate Z. This fact confirms the assumption that large vortices may exist on the limits of an examined radius but the high average velocity makes their existence near the center impossible. The axial profiles of the intensity of turbulence do not change significantly with radial coordinate (Fig. 3) which suggests that the structure of the examined stream is preserved in the whole interval of investigated radius R.

The Energy Spectrum of Turbulence

The obtained spectra of turbulence expressed in terms of the normalized dissipated energy, defined by

$$E^*(f) = \left[E(f)/\Delta f\right] \Big/ \left[1/(f_2 - f_1) \int_{f_1}^{f_2} E(f)\, df\right] \tag{3}$$

are plotted as functions of the axial coordinate with the radial coordinate R as a parameter in Figs 4–6. The obtained spectra confirm the results of Kim and Manning[8] who discovered their similarity in the normalized form regardless of the radial or axial distance from the impeller as long as the measurements are carried out within the stream ejected from the blades of the impeller. At the same time the fractional energy belonging to the frequency of the impeller, $f_m = 20$ Hz and $2f_m$ decreases with increasing radial distance from the impeller as well as the absolute value of the coordinate Z. The figure shows that starting already from relatively low frequencies (c 200 Hz) the system becomes almost isotropic., *i.e.* satisfies the relations[9,10] valid for isotropic turbulence in the inertia region

$$E^*(f) = K_1 f^{-5/3} \tag{4}$$

and the dissipation region

$$E^*(f) = K_2 f^{-7} . \tag{5}$$

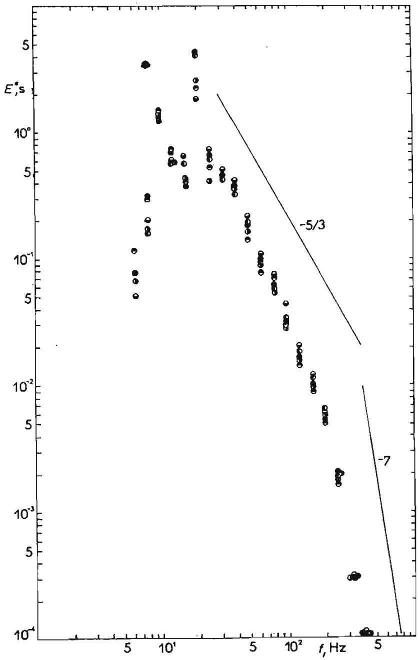

FIG. 4

Energy Spectra of Turbulence in a Mixed System with the Turbine Impeller and Radial Baffles ($d/D = 1/4$, $R = 0{\cdot}04$ m, $n = 200$ min^{-1})

◑ $Z = 0{\cdot}60$, ◓ $Z = 0{\cdot}36$, ● $Z = 0{\cdot}00$, ◒ $Z = -0{\cdot}36$, ◐ $Z = -0{\cdot}60$.

The integral of the quantity $E(f)$ over the whole range of frequencies equals, according to the definition of an average squared quantity, the average value of the observed fluctuation stress component on the anemometer voltage output. At a similar conclusion one can arrive in the study of the energy spectra published by Liepe and coworkers[2] and Kim and Manning[8]. A more detailed study, however, shows that aside from the given frequency f_m other peaks appear on the course of the function $E^*(f)$:

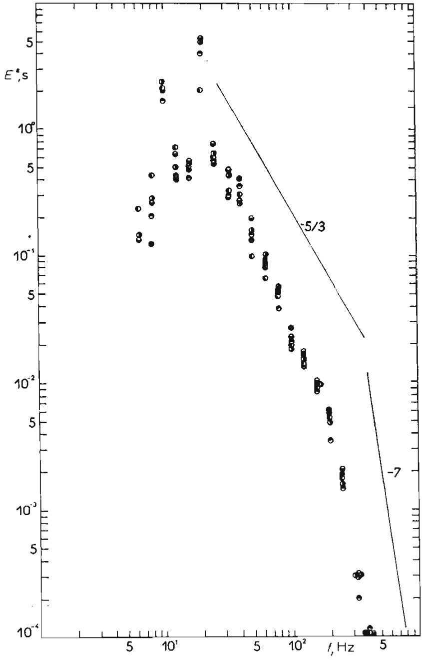

FIG. 5

Energy Spectra of Turbulence in a Mixed System with the Turbine Impeller and Radial Baffles ($d/D = 1/4$, $R = 0{\cdot}24$, $n = 200\ \text{min}^{-1}$)

◑ $Z = 0{\cdot}60$, ⊖ $Z = 0{\cdot}36$, ● $Z = 0{\cdot}00$, ⊖ $Z = -0{\cdot}36$, ◐ $Z = -0{\cdot}60$.

for $f_m/2$ and $2f_m$, *i.e.* at subharmonic and second harmonic frequency of the r.p.m. of the impeller. The existence of the first of the peaks has not been mentioned in the literature to date but the function $E^*(f)$ displayed, excepting the given frequency, a monotonous (decreasing) course. The shape of the found function, however, differs in the low frequency range ($f < f_m$) markedly from the above course where, excepting the neighbourhood of $f_m/2$ and f_m, $E^*(f)$ has a monotonously increasing

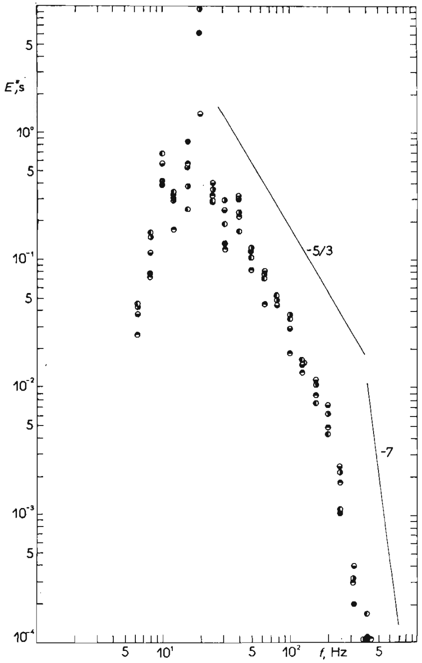

FIG. 6

Energy Spectra of Turbulence in a Mixed System with the Turbine Impeller and Radial Baffles ($d/D = 1/4$, $R = 0{\cdot}36$, $n = 200\ \text{min}^{-1}$)

◑ $Z = 0{\cdot}48$, ◓ $Z = 0{\cdot}24$, ● $Z = 0{\cdot}00$, ◒ $Z = -0{\cdot}24$, ◐ $Z = -0{\cdot}48$.

course. Moreover, the value of $E^*(f)$ depends also on vertical position, Z, in the stream. This fact is apparently associated with the behaviour of the large-scale system used in this work.

In the analysis of the results of the present work it was found that the frequency limit from which Eqs (4) and (5) for isotropic turbulence begin to hold depends, in contrast to other apparatuses, neither on the radial nor the axial coordinate of the position within the stream ejected from the blades of the impeller. With increasing R and Z decreases only the fractional energy, E^*_m, defined by

$$E^*_m \equiv \frac{1}{f_2 - f_1} \frac{E(f_m) - E_{it}(f_m)}{\int_{f_2}^{f_1} E(f)\, \mathrm{d}f} \tag{6}$$

belonging to the frequency of the impeller blades. The values of $E_{it}(f_m)$ (*e.g.* energy belonging to the isotropic turbulence energy spectra at the frequency of the impeller) were estimated from Figs 4–6. The quantity E^*_m reaches a maximum (about 15%) in close proximity of the impeller and then decreases with increasing R. A value smaller than 1% is reached for $R < 0{\cdot}60$ as was found by extrapolation. From here on the examined stream is no longer affected by the source of mechanical energy – the impeller. These conclusions comport well with the results of Liepe and coworkers[2] which indicate that for $d/D = 1/3$ the fractional energy E^*_m at $R = 0{\cdot}25$ belonging to the frequency of the blades amounts to 12% and below 1% decreases at $R < 0{\cdot}65$. It is proper to mention that with regard to the structure of the energy spectra in the stream ejected from the blades of the turbine impeller, the above approach of Mujudmar and coworkers[4], neglecting the energy contribution of that part of the spectrum below f_m is not adequate.

A Model of the Turbulent Characteristics in a System with Turbine Impeller and Radial Baffles

The authors of some of the cited papers attempted to generalize their results. As a basic and relatively successful approach to data processing appeared rendering the obtained quatities dimensionless, *e.g.* in terms of the intensity of turbulence δ_W, or a suitable normalization of the quantity as was the case of $E^*(f)$. These approaches were taken over also in this work. From the results and a comparison with the published data (Fig. 2) it follows that despite of certain differences in the geometry of the system the scale-up from a laboratory experiments is possible at least to the pilot-plant dimensions. Similarly one can process the results from the literature[2,3,6] of the Lagrange macro-scale of turbulence, Λ_L, normalized by the width of the blades. Fig. 7 shows the results of normalization and it is apparent that in the limit one can put

$$\lim_{R \to 0} \Lambda_L/h = 1\,. \tag{7}$$

This fact plays an important role in mixed system modelling mainly in evaluation of the effect of hydrodynamic conditions on homogenization of miscible liquids in systems where the size of the largest vortices departing from the rotating impeller is comparable with the size of elements of mixed liquid[2,16]. However, the course of the energy spectra in laboratory models is not identical to that in pilot-scale equipment in the whole frequency range even in the normalized form. This applies to that part of the energy spectra below the frequency of the impeller blades, or its second harmonic. Here, the course of $E^*(f)$ is different in case of systems of different size and hence the average value of this function cannot be used as a reference value for normalizing the spectrum[6]. Yet, the part above f_m (or $2f_m$) does permit certain conclusions to be made: The inertia subregion extends over the frequency range of $f_m(2f_m) - 10f_m$ and the dissipation subregion over the range $10f_m - 10^3$ Hz. The latter region thus reaches the frequencies where the experimental error is comparable with the measured energy. The absolute values of the normalized energies $E^*(f)$ themselves depend strongly on radial distance from the source of mechanical energy (the impeller) for frequencies below the second harmonic of the impeller blades and above it the difference is no longer important in view of the accuracy of the experimental technique. The proposed method of processing the experimental energy spectra in the stream ejected from the blades of a rotating turbine impeller is thus suitable for modelling their course in the frequency region above the frequency of the impeller blades or its first harmonic. On the contrary, for frequencies below f_m a comparison of the laboratory results with the pilot-scale results does not suffice to determine a suitable way of modelling the energy spectra in a mechanically mixed batch.

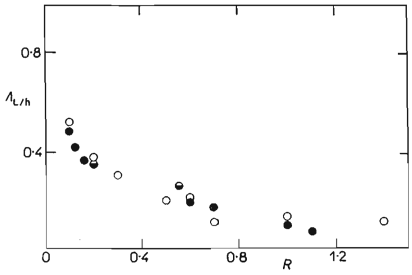

FIG. 7

Radial Profiles of the Dimensionless Lagrange Macro-Scale of Turbulence λ_L/h Determined from Published Results ($Z = 0$)

○ Liepe and coworkers[2], ⊖ Sato and coworkers[3], ● Cho and coworkers[6].

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LIST OF SYMBOLS

A	constant (V^2)
B	constant ($V^2\ s^{1/2}\ m^{-1/2}$)
b	width of radial baffle (m)
D	vessel diameter (m)
d	impeller diameter (m)
$E(f)$	spectral density of energy (kg $m^2\ s^{-2}$)
$E_{it}(f)$	energy belonging to isotropic course of the energy spectrum (kg $m^2\ s^{-2}$)
$E^*(f)$	normalized spectral density (s)
E^*_m	dimensionless quantity defined by Eq. (*6*)
f	frequency (s^{-1})
f_1	lower limit of frequency analysis (s^{-1})
f_2	upper limit of frequency analysis (s^{-1})
f_m	frequency of impeller blades (s^{-1})
Δf	bandwidth of frequency analysator (s^{-1})
H	clear liquid height (m)
H_2	height of lower edge of the blade above bottom (m)
K_1	constant ($s^{-2/3}$)
K_2	constant (s^{-6})
N	frequency of axial pump of the calibrating device (s^{-1})
n	r.p.m. of impeller (s^{-1})
$R = (2r-d)/d$	dimensionless radial coordinate
r	radial coordinate (m)
$r_L(\tau)$	autocorrelation function
t	temperature (deg)
U	time-averaged voltage (V)
$\bar{w}$	mean velocity of liquid (m s^{-1})
w'	fluctuation velocity of liquid (ms^{-1})
$Z = 2z/h$	dimensionless axial coordinate
z	axial coordinate (m)
δ_w	intensity of turbulence
τ	time (s)

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