

Description

The present invention relates to impellers and impeller systems for mixing and blending applications and particularly for mixing and blending liquids and liquid suspensions having a wide range of viscosities such that the flow developed by the impellers and impeller systems can be in the turbulent, the transitional, and the laminar flow regimes.

The invention is especially suitable for use in providing an impeller which is relatively small (e.g., 40.6cm (sixteen inch) diameter and less) and is adapted for use in portable mixers. Such mixers may encounter liquids and liquid suspensions of various viscosity, and because of their small size, are likely to encounter transitional and laminar flow conditions. The inventive impellers and impeller systems are adapted for efficient mixing and blending so as to minimize power requirements and avoid the need for large motors which would limit portability. The features of the invention in providing efficient flow over a wide range of flow conditions, extending from turbulent, through transitional to the laminar flow regime, and the efficiency of the impellers and impeller systems makes them applicable for other applications where large sizes may be desirable. The invention, however, affords effective mixing over a large range of flow conditions (viscosity of the medium being mixed) so as to provide complete mixing and blending rapidly with blend times being that of large conventional impellers used in similar flow regimes and at less power input as compared to such impellers.

The flow regime, in which flow is developed for mixing impellers, depends upon the ratio of inertial forces to viscous forces in the medium (the liquid or liquid suspension confined in a tank or vessel). This ratio is expressed as the mixing Reynolds number and is calculated as follows:

$$N_{Re} = \frac{10.76 \text{ NRPM } D^2 \text{ in (SG)}}{\pi \text{ cps}}$$

where SG is the specific gravity of the medium and $\pi \text{ cps}$ is the viscosity in mPa s. Reynolds numbers are of the order of 2×10^5 in the turbulent range. Reynolds numbers are from 10 to 1000 in the transitional regime. For Reynolds numbers below 10, the impeller is developing flow in the laminar regime. Normally, flow of water and similar materials of low viscosity is in the turbulent regime. Liquids of the viscosity of light oils are normally caused to flow in the transitional flow regime, when impeller driven. More viscous liquids, such as liquids having the viscosity of glycerine are normally driven to produce mixing Reynolds numbers corresponding to flow in the low end of the transitional flow regime and into the laminar flow regime.

The specific gravity, diameter and speed of the impeller affects the inertial forces, while the viscosity of the medium affects the viscous forces. However, in a small vessel and with a small impeller, the mixing Reynolds number, even for a low viscosity liquids such as light oils, can extend into the transitional flow regime. Thus, with small diameter impellers, for example, 7.6cm (three inch) diameter, the mixing Reynolds numbers, even with slightly viscous media, are in the transitional and may extend into the laminar flow regimes. Thus, the problems of mixing and blending in the transitional and laminar flow regimes are exacerbated, when small diameter impellers are used.

It is desirable for efficient mixing and rapid blending that the impeller produce axial flow. As the Reynolds number decreases, the media's resistance to flow increases and an axial flow impeller's discharge becomes more radial. In the laminar flow regime, the discharge flow from the impeller is radial. Then, on one side (below the impeller in a down pumping impeller) reverses, leaving regions which are not efficiently mixed or blended. In addition, vortices are produced, which move sluggishly if at all, thereby increasing the blending time. See, publication by R.J. Weetman, et al., entitled "The Use of Fluidfoil Impellers in Viscous Mixing Applications", presented at AIChE annual meeting, San Francisco, California, U.S.A., November 5-10, 1989. Axial flow impellers which provide effective and efficient mixing and blending in turbulent flow conditions, such as the impellers described in U.S. -A- 4,468,130 issued August 28, 1984 (the type A310 sold by the Mixing Equipment Company of Rochester, New York, USA, ("MIXCO") - a unit of General Signal Corporation); U.S. -A- 4,722,68 (the A6000 sold by MIXCO); U.S. -A-4,802,771 (the C104 also sold by MIXCO) and U.S. -A-4,896,971 (the A315 and A320 also sold by MIXCO) have approximately constant pitch or are under pitched in order to maintain a substantially constant angle of attack (the angle which the velocity vector of the media makes with the blade chord). With such axial flow impellers and even with pitch blades turbines (e.g., the MIXCO A200), the flow direction of the impellers discharged becomes radial in the transitional flow regime and is radial in the laminar flow regime. Such impellers operate

efficiently (the ratio of flow to input power is high). However, due to the generation of reverse flow and vortices, the blending time increases and blending becomes sluggish or not occur at all, even when the impeller is operated continuously for prolonged periods of time (e.g. of the order of an hour).

It is reasonable to compare an impeller or impeller system with a standard impeller for purposes of measuring its effective blending time. One such axial flow impeller which has long been in use is a constant pitch fluidfoil (with camber and twist) having three blades, 120° apart. The shape of this impeller is similar to that of a conventional marine propulsion propeller. Such a standard impeller is the type A100 also available from MIXCO.

A fluidfoil having twist is characterized by its pitch ratio (PR). The pitch is the distance a point on an impeller blade would move along the axis of rotation of the impeller in the manner of a screw thread while that point moves around the axis for one rotation of the impeller. PR is defined as the ratio of the pitch to the diameter D of the impeller, which is the diameter swept by the tip of the blades.

Mathematically,

$$PR = \text{Pitch}/D,$$

and from the geometry

$$PR = \frac{r}{R} \pi \tan \frac{\beta}{r}$$

where r is the radius from the axis of rotation of the impeller to a section of the blade where PR is determined, R is the radius from the axis to the tip (maximum impeller diameter) and β is the pitch or blade angle in degrees.

The pitch or blade angle is defined as the angle which the chord of the blade makes to a plane perpendicular to the axis of rotation of the impeller. The pitch angle is the arc tangent ($\tan^{-1}$) of the pitch ratio divided by the product of π (pi) and the ratio of the radius at the point where the pitch ratio is measured. In a constant pitch impeller, the pitch angle is such that the pitch and pitch ratio remains constant from the tip to a point near its base (or the hub) where the blade is connected to the drive shaft. The axis of the shaft is the rotational axis about which the impeller is driven. The diameter is the swept diameter from the axis to the point where the pitch ratio is measured. At the tip, the diameter is the swept diameter of the entire impeller. For typical axial flow impellers, a constant pitch or under pitch, where the pitch ratio near the hub (say at 0.4R, R being the radius of the impeller or half the swept diameter), is less than the pitch ratio at the tip.

In accordance with the invention, an impeller configuration has been discovered that provides efficient axial flow in the turbulent flow regime, and inhibits the tendency for the flow discharge to become radial in the transitional regime (while maintaining efficient operation under transitional flow conditions). The blades of the inventive impeller have twist to the extent that provides an over pitch condition. This condition is manifested by the pitch ratio of the blades near the hub (at 0.4R) being greater than the pitch ratio at the tip. Further, in accordance with the invention, the radial flow discharge inhibition property of the impeller in the transitional regime and to the laminar flow regime is enhanced by the use of a blade angle at the tip which is larger than in the conventional axial flow impeller and preferably is in the range from 18° to 30°. A further feature of the invention is to use a pair of impellers which are dependent (coupled so that the total power needed to drive the pair is less than the sum of the power to drive each impeller individually under the same conditions). Such dependent relationship preferably is obtained by spacing of the impellers at less than a diameter of the large of the pair. This arrangement enhances the axial flow and further inhibits the radial discharge in the transitional and laminar regimes. A further feature is to provide the blades of the impeller with different blade angles at the tip. Thus, the impeller may be a four-bladed impeller where two of the diametrically opposite blades are at a first pitch angle, while the other two diametrically opposed blades (90° apart from the first two) are at a different blade angle at their tip.

Without limiting the invention to any theory of operation, it is believed that the over pitch conditions and the use of different blade angles at the tip of the impellers inhibits the tendency for the discharge flow from the impeller to be radial, and enhances axial flow, by causing asymmetries which give rise to oscillations in the flow velocity, much like inherently exist in the turbulent flow regime. Such oscillations are shown in data presented in a paper by Ronald J. Weetman, et al., entitled "Power Flow and Shear Characteristics of Mixing Impellers" presented at the Sixth European Conference on mixing, Pavia, Italy, May 24-26, 1988.

Accordingly, it is a general object of the present invention to provide improved impellers and impeller systems which afford better mixing and more rapid and effective blending than conventional axial flow impeller systems not only in the turbulent flow regime, but also in the transitional and into the laminar flow

regime.

This and other objects of the invention are attained, in accordance with one aspect of the invention by an impeller which produces flow ranging from turbulent to laminar flow in bodies of liquid and liquid suspensions and which is effective in mixing and blending such bodies, said impeller having a plurality of blades rotatable about an axis, said blades having tips which define the radius of said impeller, said blades extending from said tips radially toward said axis, characterized in that said blades have twist defined by inclinations thereof with respect to a plane perpendicular to said axis at angles which increase smoothly along said blades toward said axis to provide an over pitched condition, where the pitch ratio of the blades at a section thereof spaced $0.4R$ from said axis is greater than PR at said tips, where PR is defined by the following equation:

$$PR = \frac{r}{R} \pi \tan \frac{\beta}{R}$$

where r is the radius from the axis of rotation to the section where PR is computed in accordance with said equation, and β is the angle of inclination at said section in degrees.

The foregoing and objects, features and advantages of the invention, and the design of impellers and impeller systems in accordance with presently preferred embodiments thereof, will become more apparent from a reading of the following description in connection with the accompanying drawings in which:

Fig. 1 is a perspective view of a mixing impeller which embodies the invention;
 Fig. 2 is a plan view of the impeller shown in Fig. 1;
 Fig. 3 is an elevational view of the impeller shown in Figs. 1 and 2, which is partially in section along one blade of the impeller, the section being taken along the line 3-3 in Fig. 2;
 Fig. 4 is an end view of the impeller shown in Figs. 1-3;
 Fig. 5 is a view showing sections through various foils of the impeller at different radial distances from the impeller axis, namely at the tip at $0.7R$ (where R is the radial distance from the tip to the axis - i.e., the impeller diameter D , divided by two) and $0.4R$, the view showing the twist of the impeller;
 Fig. 6 is a graph also showing the twist of the impeller in terms of the blade or pitch angle (the angle of the chord through a section to the radius);
 Fig. 7 is a plan view of a mixing impeller in accordance with another embodiment of the invention;
 Figs. 8 and 9 are end views illustrating the different blade angles at the tips of each pair of impeller blades of the impeller shown in Fig. 7;
 Fig. 10 is a plan view showing an impeller similar to the impeller shown in Fig. 1 to 4 with flow guides on the low pressure (top) surface of the blades;
 Fig. 11 is a view similar to Fig. 10 showing the use of turbulence generator wedges on the low pressure (top) surface of the blades;
 Fig. 12 is a bottom view of the impeller shown in Fig. 10 showing flow guides on the high pressure or bottom surfaces of the blades;
 Fig. 13 shows an end view of the impeller shown in Figs. 10 and 12 and an end view of the impeller shown in Fig. 11;
 Fig. 14 is a perspective view of an impeller system in accordance with the invention and particularly a pair of impellers, one of larger diameter and one of smaller diameter, similar in shape to the impellers shown in Figs. 1 through 5 and which are in dependent relationship; and
 Fig. 15 is a curve showing the blend time ratio (as compared to a standard A100 impeller obtained with a 17.5 cm (seven inch) impeller as shown in Figs. 1 through 5 in a medium which presents a mixing Reynolds number of approximately 200, the test set-up being similar to that used and described in the November, 1989 paper by Ronald J. Weetman, et al. entitled "The Use of Fluidfoil Impellers in Viscous Mixing Applications" which is referenced above.

Referring to Figs. 1 through 5, there is shown a mixing impeller 10 having three blades 12, 14 and 16 assembled 120° apart on a hub 18 which is attached to a shaft 20. The blades may be attached to the hub by bolts and brackets when the blades are made of metal. Preferably, the blades and hubs are intricately molded of composite material such as fiber reinforced plastic, for example by injection or compression molding of compounds containing fibers and plastic resin. The mold, therefore, defines the shape of the blades and the hub. The hub may be attached to the shaft 20 which extends to a motor directly or via drive gears. The motor and shaft may be part of the assembly of a portable mixer which may be clamped to walls of a tank containing the liquid or liquid suspension to be mixed. Alternatively, the motor and its drive assembly with the shaft depending therefrom, may be supported by a stand or other support mechanism.

Each of the blades 12, 14 and 16 is identical. The blades have tips 24 which may lie along a circumference of a circle defining the swept diameter of the impeller. The blades also have leading edges

26 and trailing edges 28. The blades are fluidfoils and have camber and twist except along about 10% of the radial length of the blades adjacent to the tips (see Fig. 6). This enables the tip end of the blade to be cut back so as to shorten the diameter of the impeller if desired thereby reducing the number of sizes of impellers which need to be molded. The blades may be further extended with constant pitch angle beyond (e.g., a further 10%) their nominal radial length R. The fluidfoil blades have opposite top and bottom surfaces 30 and 32 which are the low pressure and high pressure surfaces of the blades. The high pressure or pressure surface of the blade is the surface facing the direction in which the fluid is driven. Accordingly, when the impeller is rotated in a clockwise direction about the axis 34, an axial flow is produced in the downward direction (downward pumping) in the medium in the tank. Lines 36 extending radially from the center of the shaft. These lines 36, define the blade axes where maximum camber point on the chord (40) is approximately located. The maximum camber is located at approximately the 40% point (from the leading edge) of the chord (Fig. 5). The chord is drawn between points where the mean line through the blade cross-section intersects the leading and trailing edges (See mean line 53 in the foil 48 in Fig. 5). The blade or pitch angle is the angle between the chord and a plane perpendicular to the blade axis 34. A line 42 in Fig. 5 is in this plane. The three foil shapes shown in Fig. 5 represent cross sections through the leading 26 and trailing edge 28 points at swept radius of R, .7R and .4R as shown in Fig. 2. Foils at the tips is shown at 44. The foils at 0.7R are shown at 46 and the foils at 0.4R are shown at 48. The chords at the tip are shown at 40 and the chords 50 & 52 in the foils 46 at 0.7R and 48 at 0.4R are shown. Patterns for forming the molds for fabricating the impeller may be constructed by laminating or sandwiching together a series of foils 44, 46 and 48 or using stereo lithography techniques.

The width of the blades increases from the tip to 0.4R and then decreases to form transition between the blades and the hub 18. The principal working surface of the blades is from the tip to 0.4R. It is in this working surface where the twist providing the over pitch condition occurs.

The twist as defined as the difference between the blade angle at the tip and at 0.4R and is defined by the following equation:

$$\Delta\beta = \tan^{-1} \frac{PR'}{0.4\pi} - \tan^{-1} \frac{PR}{\pi}$$

where PR' is the pitch ratio at 0.4R and PR is the pitch ratio at the tip. The pitch ratio at 0.4R is greater than the pitch ratio at the tip to afford the over pitch condition.

Preferably, the twist, delta beta, is between 30° to 45°. The blade angle at the tip, also known as the tip chord angle (TCA) is preferably 20° and may be in the range from 18° to 30°.

The width of the blade increases smoothly from the tip to 0.4R. This smooth increase in width has been found to improve the efficiency of the impeller under turbulent flow conditions. In terms of the ratio of the length of the chord to the diameter of the impeller at the tips, the width variation is preferably 0.25 at the tip to 0.33 at 0.4R. This is a variation in width of approximately 30%. The width variation may be in the range from about 0.15 at the tip to 0.40 at 0.4R.

The thickness of the blades may also vary increasing from the tip to the base of the blades at the hub 18. As a function of the ratio of the maximum distance between the low and high pressure surfaces 30 and 32 to the chord length is preferably 0.1 at the tip and 0.14 at 0.4R. The blades may, however, be approximately equal thickness throughout their length. The thickness of the blade may increase to 0.22 in terms of the ratio of the maximum distance between the pressure surfaces 30 and 32 to the chord at 0.4R, and for the remaining length of the blade to the hub when it is desired to strengthen the connection of the blades to the hub. Such strengthening may be desirable under conditions where a motor directly drives the shaft which drives the impeller. While three blades are shown, there is no limitation on the number of blades.

The camber of the blades as expressed as the ratio of the maximum distance from the mean line (shown by way of example as the line 53 through the foil 48 in Fig. 5) to the chord divided to the chord length, may be preferably constant, and in a preferred embodiment is approximately 6%.

The blade angle, beta, varies in accordance with the twist. The variation may be smooth as shown in Fig. 6 for the preferred embodiment illustrated in Figs. 1 through 5.

The surprising results obtained through the use of sufficient pitch to provide an over pitched condition is illustrated in Fig. 15. There, the blend time is compared to a marine propeller type impeller, the A100. It is seen that as the twist increases, the blend time ratio dramatically decreases. This result may be observed

using an experiment similar to the one described in the Weetman, et al. article on viscous mixing applications. The colored tracer shows how the flow is maintained more axial than a conventional impeller in spite of transitional and laminar flow conditions in the tank, such that blending is not sluggish.

Referring to Figs. 7, 8 and 9, there is shown a mixing impeller 60 having a first pair of blades 62 and 64 disposed diametrically opposite to each other and a second pair of blades 66 and 68 also disposed diametrically opposite to each other. The shape of these blades may be similar to the shape of the blades 12, 14 and 16. The first pair of blades 60 and 62 are spaced 90° from the blades 66 and 68. However, their tip chord angles are different. The pair of blades 66 and 68, as shown in Fig. 8 for the blade 66, have a smaller tip chord angle than the blades 62 and 64. This angle is, for example, 20° in the case of the blades 66 and 68 and 25° in the case of the blades 62 and 64. The diametrically opposed blades have the same tip chord angle in order to reduce fluid forces acting to bend the shaft 70 of the impeller. The different angular orientations cause asymmetry in the flow pattern produced by the impeller 60 and further prevents the formation of vortices in the transitional and laminar flow regimes.

Introducing asymmetries provide improved mixing and blending in the transitional flow in the mixing tank. This is because when the flow around the immediate vicinity of the impeller is in transitional flow, the other parts of the tank can be in laminar flow. When there is laminar flow in parts of the tank, stationary vortices or circulation patterns will be set up that inhibit mixing, and particularly blending, which does not occur or takes place very sluggishly in these areas. This effect is shown in the above-referenced 1989 publication authored by R. J. Weetman et al. It is, therefore, important to introduce asymmetries in the flow field to distribute stationary circulation patterns or vortices. The improved impellers are over pitched and, in the embodiment of Figs. 7-9, also have different tip chord angles to obtain this important function. Asymmetries in the flow field may also be introduced by providing the opposite pairs of blades with different twist, camber and/or length. The difference may be of the same order of magnitude as illustrated in Figs. 7-9, wherein the tip chord angle in one pair of blades 62, 64 of the other pair 66, 68 increases the driving power for the pair 62, 64 about 25%-50% over the power for driving the other pair 66, 68. Although the above discusses similar blade characteristics for opposite pairs of blades, each blade of each pair may differ (e.g., in tip chord angle or any other characteristic) as long as the fluid forces did not limit the impeller's operation.

Figs. 10, 12 and 13A show impellers of similar design to the impellers shown in Figs. 1-5 and like parts are designated with like referenced numerals. A plurality of ribs 72, 74 which lie along circles centered at the axis of rotation of the shaft 20, define flow guides. The ribs 74 on the bottom or high pressure surface 32 extend on both sides of the blade axes 36. They inhibit the strong radial flow that occurs near the hub in the transitional flow regime. The ribs 72 on the upper surface 30 of the blades are substantially entirely between the blade axes 36 and the trailing edges of the blades. These guides further inhibit radial flow and separation in the turbulent flow regime.

Figs. 11 and 13B shows an impeller similar to the impeller shown in Figs. 1 to 5, and like parts are identified by like referenced numerals. A plurality of turbulence generating wedges 76 are disposed on the top low pressure side of the blades. These projections reduce the chance of separation of the flow and generate turbulence which enhances mixing in the turbulent flow regime. Instead of wedges, riblets may be used. The wedges may be combined with the flow guides which are shown in Fig. 12. The flow guides are provided by the plurality of ribs 74 on the high pressure surfaces 32 (see Fig. 13A). The turbulence generating wedges 76 are shown on the low pressure surfaces 30 in Fig. 11 and also in Fig. 13B.

Referring to Fig. 14, there are shown a larger diameter impeller 80 and a smaller diameter impeller 82 coaxially mounted on a common drive shaft 84. These impellers each have three blades and the blades are generally in alignment with each other. The shape of the blades is preferably as described in connection with Figs. 1 to 5. The blades are spaced sufficiently close together in axial separation along the shaft 84 as to be dependent. Dependency of operation is shown when the addition of the second impeller does not require a significant increase in driving power in order to rotate both impellers. For example, if the impellers were spaced apart more than one diameter of the larger impeller, the amount of power required to drive both impellers would be approximately the sum of the power required to drive each individually. Dependent impellers require, for the same flow velocity, similar or less power than for a single impeller alone. The impeller in the outlet flow (the lower, smaller impeller 82 in Fig. 14) changes the flow pattern for the upper impeller. In transitional and laminar flow regimes, where the discharge flow produced at the high pressure surfaces 86 of the upper impeller 80 tends to change to radial flow, because of the change in flow pattern produced by the lower smaller impeller, this radial flow is inhibited. The total flow pattern remains axial for a longer axial distance than would be the case for either impeller 80 and 82 alone. Accordingly, the use of the smaller impeller further enhances the axial flow characteristics of the impeller system.

embodiment, the upper impeller diameter is 17.5 cm (seven inches) and the lower 12.5 cm (five inches); both impellers providing an axial outlet flow in the downward direction as shown in Fig. 14. The spacing between the impellers is approximately one-half the diameter of the lower impeller, 8 cm (3.5 inches). The dual impeller system provides an increase in flow number N_Q of about 50% and a reduction in power number N_P . For a definition of N_Q and N_P , see the above description of the single impeller system. Weetman, et al. on viscous mixing applications. While the impellers are shown with straight blades, the blades may be staggered if desired. It may be advantageous to locate the larger impeller at the top. These advantages may also be obtained in up-pumping as well as down-pumping examples.

In the foregoing description, it will be apparent that there has been provided improved mixing systems which provide efficient operation in the turbulent flow regime, the transitional flow regime, and the laminar flow regime.

which produces flow ranging from turbulent to laminar flow in bodies of liquid and liquid bodies, said impeller (10) having a hub (12, 14, 16) rotatable about an axis (20), said blades having tips (24) which define the outer periphery of said impeller, said blades extending from said tips radially toward said axis, characterized in that said blades (12, 14, 16) have twist defined by inclinations thereof with respect to a plane perpendicular to said axis at angles which increase smoothly along said blades toward said axis to a pitched condition, where the pitch ratio (PR) of the blades at a section thereof spaced from said axis is greater than PR at said tips, where PR is defined by the following equation:

$$PR = \frac{r}{R} \pi \tan \beta$$

where r is the radius from the axis of rotation to the section where PR is computed in accordance with the above equation and β is the angle of inclination at said section in degrees.

According to claim 1, characterized in that said blades have a twist, the twist ($\Delta\beta$) being

$$\Delta\beta = \tan^{-1} \frac{PR'}{0.4\pi} - \tan^{-1} \frac{PR}{\pi}$$

where PR' is the pitch ratio at a radial distance of about 0.4R and PR is the pitch ratio at the tip and R is the radius of the impeller.

According to claim 1 claim 2, characterized in that $\Delta\beta$ is in the range of 30° to 45°.

According to any of claims 1 to 3, characterized in that the inclination of the blades (12, 14, 16) at said tips (24) is in the range of 18° to 30°.

According to any of claims 1 to 4, characterized in that the inclination of said blades (12, 14, 16) is constant for approximately 10% of the radial distance from their said tips to said axis.

According to any of claims 1 to 5, characterized in that said blades (12, 14, 16) are fluidfoils as well as said twist, and in that said pitch blade angles are measured between chords of said blades at a point at said tips and at a point at 0.4R.

According to claim 6, characterized in that said chords (40) at said tip (24) extend between intersections of radial lines of length R from said axis and leading and trailing edges of said blades and said chords at 0.4R, and between intersections of radial lines of length 0.4R from said axis and leading and trailing edges.



In a preferred embodiment, the upper impeller diameter is 17.5 cm (seven inches) and the lower impeller diameter is 12.5 cm (five inches); both impellers providing an axial outlet flow in the downward direction as viewed in Fig. 14. The spacing between the impellers is approximately one-half the diameter of the upper impeller 8.8 cm (3.5 inches). The dual impeller system provides an increase in flow number N_Q of approximately 60% and a reduction in power number N_p . For a definition of N_Q and N_p , see the above referenced article by Weetman, et al. on viscous mixing applications. While the impellers are shown with their blades in alignment, the blades may be staggered if desired. It may be advantageous to locate the smaller impeller above the larger impeller. These advantages may also be obtained in up-pumping as well as in the illustrated down-pumping examples.

From the foregoing description, it will be apparent that there has been provided improved mixing impellers and impeller systems which provide efficient operation in the turbulent flow regime, the transitional flow regime and into the laminar flow regime.

Claims

1. An impeller which produces flow ranging from turbulent to laminar flow in bodies of liquid and liquid suspensions and which is effective in mixing and blending such bodies, said impeller (10) having a plurality of blades (12, 14, 16) rotatable about an axis (20), said blades having tips (24) which define the radius (R) of said impeller, said blades extending from said tips radially toward said axis, characterized in that said blades (12, 14, 16) have twist defined by inclinations thereof with respect to a plane perpendicular to said axis at angles which increase smoothly along said blades toward said axis to provide an over pitched condition, where the pitch ratio (PR) of the blades at a section thereof spaced 0.4R from said axis is greater than PR at said tips, where PR is defined by the following equation:

$$PR = \frac{r}{R} \pi \tan \frac{\beta}{R}$$

where r is the radius from the axis of rotation to the section where PR is computed in accordance with said equation, and β is the angle of inclination at said section in degrees.

2. An impeller according to claim 1, characterized in that said blades have a twist, the twist ($\Delta\beta$) being defined by

$$\Delta\beta = \tan^{-1} \frac{PR'}{0.4\pi} - \tan^{-1} \frac{PR}{\pi}$$

where PR' is the pitch ratio at a radial distance of about 0.4R and PR is the pitch ratio at the tip and PR' is greater than PR.

3. An impeller according to claim 1 claim 2, characterized in that $\Delta\beta$ is in the range of 30° to 45°.
4. An impeller according to any of claims 1 to 3, characterized in that the inclination of the blades (12, 14, 16) at their tips (24) is in the range of 18° to 30°.
5. An impeller according to any of claims 1 to 4, characterized in that the inclination of said blades (12, 14, 16) is invariant for approximately 10% of the radial distance from their said tips to said axis.
6. An impeller according to any of claims 1 to 5, characterized in that said blades (12, 14, 16) are fluidfoils having camber as well as said twist, and in that said pitch blade angles are measured between chords (40) of said blades at a point at said tips and at a point at 0.4R.
7. An impeller according to claim 6, characterized in that said chords (40) at said tip (24) extend between points at intersections of radial lines of length R from said axis and leading and trailing edges of said blades, and said chords at 0.4R, and between intersections of radial lines of length 0.4R from said axis and said leading and trailing edges.



8. An impeller according to claim 6 or claim 7, characterized in that the width of said blades (12, 14, 16) in terms of the ratio of the length of said chord (W) to the diameter (D) of said impeller (W/D) at their said tips is about 0.25 at the tips, and at 0.4R is about 0.33.
9. An impeller according to claim 8 characterized in that the ratio of said thickness to the length of said chord (40) at said tip is about 0.1, and the ratio of said thickness, to the length of said chord at 0.4R is from about 0.14 to 0.22.
10. An impeller according to any of claims 1 to 9, characterized in that said blades (12,14,16) have opposite surfaces which respectively produce relatively high and relatively low pressure in said bodies, and projections defining flow guides on said surfaces.
11. An impeller according to any of claims 1-10, characterized in that said blades (12, 14, 16) have opposite surfaces which develop relatively high pressures in said bodies, and means for generating turbulence disposed on said low pressure surfaces.
12. An impeller according to any of claims 1-11, characterized by a second impeller (82), said second impeller having a plurality of blades rotatably disposed coaxially with said first named plurality of blades and spaced in dependent relationship therewith.
13. An impeller according to claim 12, characterized in that said first named plurality of blades and the blades of said second impeller have opposite surfaces, which develop relatively low and relatively high pressures in said bodies, said impellers being disposed with the relatively low pressure surfaces of said blades of said second impeller (82) adjacent said relatively high pressure surface, of said first named blades.

Patentansprüche

1. Rührflügel, der in Flüssigkeitskörpern und in Flüssigkeitssuspensionskörpern eine Strömung erzeugt, die von einer Laminarströmung bis zu einer Wirbelströmung reicht, wobei der Rührflügel für das Mischen und vermischen dieser Körper geeignet ist, wobei der Rührflügel (10) eine Mehrzahl von Schaufeln (12, 14, 16) umfaßt, die um eine Achse (20) drehbar sind, wobei die Schaufeln Spitzen (24) aufweisen, die den Radius (R) des Rührflügels definieren, wobei sich die Schaufeln von den Spitzen radial zu der Achse erstrecken, dadurch gekennzeichnet, daß die Schaufeln (12, 14, 16) eine Drehung aufweisen, die durch die Neigungen der Schaufeln im Verhältnis zu einer zu der genannten Achse senkrechten Ebene definiert sind, wobei die Winkel entlang der Schaufeln in Richtung der Achse allmählich größer werden, so daß ein überneigter Zustand vorgesehen wird, wobei der Neigungsgrad (PR) der Schaufeln an einem von der Achse um 0,4R entfernten Abschnitt größer ist als PR an den Spitzen, wobei PR durch die folgende Gleichung gegeben ist:

$$PR = \frac{r}{R} \pi \frac{\tan \beta r}{R}$$

wobei r die Radiusform der Rotationsachse zu dem Abschnitt ist, an dem PR gemäß der Gleichung berechnet wird, und wobei β den Neigungswinkel an dem Abschnitt in Grad darstellt.

2. Rührflügel nach Anspruch 1, dadurch gekennzeichnet, daß die Schaufeln eine Drehung aufweisen, wobei die Drehung ($\Delta\beta$) wie folgt definiert ist:

$$\Delta\beta = \tan^{-1} \frac{PR'}{0,4\pi} - \tan^{-1} \frac{PR}{\pi}$$



wobei PR' den Neigungsgrad in einem radialen Abstand von etwa 0,4R darstellt, und wobei es sich bei PR um den Neigungsgrad an der Spitze handelt, und wobei PR' größer ist als PR.

3. Rührflügel nach Anspruch 1 oder 2, dadurch gekennzeichnet, daß $\Delta\beta$ im Bereich von 30° bis 45° liegt.
4. Rührflügel nach einem der Ansprüche 1 bis 3, dadurch gekennzeichnet, daß der Neigungswinkel der Schaufeln (12, 14, 16) an deren Spitzen (24) im Bereich von 18° bis 30° liegt.
5. Rührflügel nach einem der Ansprüche 1 bis 4, dadurch gekennzeichnet, daß der Neigungswinkel der Schaufeln (12, 14, 16) über etwa 10% des radialen Abstands von den Spitzen zu der genannten Achse unveränderlich ist.
6. Rührflügel nach einem der Ansprüche 1 bis 5, dadurch gekennzeichnet, daß die Schaufeln (12, 14, 16) Flüssigkeitsfolien darstellen, die neben der genannten Drehung eine Wölbung aufweisen, und wobei die Schaufelneigungswinkel zwischen den Sehnen (40) der Schaufeln an einem Punkt an den Spitzen und an einem Punkt bei 0,4R gemessen werden.
7. Rührflügel nach Anspruch 6, dadurch gekennzeichnet, daß sich die Sehnen (40) an der Spitze (24) zwischen Punkten an Schnittpunkten der radialen Linien der Länge R von der Achse und den Hinterkanten der Schaufeln und den Sehnen bei 0,4R erstrecken und zwischen Schnittpunkten der radialen Linien der Länge 0,4R von der Achse und den genannten Vorder- und Hinterkanten.
8. Rührflügel nach Anspruch 6 oder 7, dadurch gekennzeichnet, daß die Breite der Schaufeln (12, 14, 16) als Verhältnis der Länge der Sehne (W) zu dem Durchmesser (D) des Rührflügels (W/D) an den Spitzen etwa 0,25 und bei 0,4R etwa 0,33 beträgt.
9. Rührflügel nach Anspruch 8, dadurch gekennzeichnet, daß das Verhältnis der Dicke zu der Länge der Sehne (40) an der genannten Spitze etwa 0,1 beträgt, und wobei das Verhältnis der Dicke zu der Länge der Sehne bei 0,4R etwa 0,14 bis etwa 0,22 beträgt.
10. Rührflügel nach einem der Ansprüche 1 bis 9, dadurch gekennzeichnet, daß die Schaufeln (12, 14, 16) entgegengesetzte Oberflächen aufweisen, die entsprechend in den Körpern verhältnismäßig hohe und verhältnismäßig niedrige Druckzustände erzeugen, und wobei sie Vorsprünge aufweisen, die auf den genannten Oberflächen Strömungsführungen vorsehen.
11. Rührflügel nach einem der Ansprüche 1 bis 10, dadurch gekennzeichnet, daß die Schaufeln (12, 14, 16) entgegengesetzte Oberflächen aufweisen, die in den Körpern verhältnismäßig hohe Druckzustände erzeugen, und wobei die Schaufeln eine Einrichtung zur Erzeugung einer Wirbelströmung aufweisen, die sich an den Niederdruckoberflächen befindet.
12. Rührflügel nach einem der Ansprüche 1 bis 11, gekennzeichnet durch einen zweiten Rührflügel (82), wobei der zweite Rührflügel eine Mehrzahl von Schaufeln aufweist, die drehbar coaxial zu der ersten genannten Mehrzahl von Schaufeln angeordnet sind und die in abhängigem Verhältnis zu diesen beabstandet positioniert sind.
13. Rührflügel nach Anspruch 12, dadurch gekennzeichnet, daß die erste genannte Mehrzahl von Schaufeln und die Schaufeln des zweiten Rührflügels entgegengesetzte Oberflächen aufweisen, die in den genannten Körpern verhältnismäßig niedrige und verhältnismäßig hohe Druckzustände erzeugen, wobei die Rührflügel so angeordnet sind, daß sich die Oberflächen der Schaufeln des zweiten Rührflügels (82) mit verhältnismäßig niedrigen Druckzuständen neben der Oberfläche der erstgenannten Schaufeln mit verhältnismäßig hohem Druckzustand befinden.

Revendications

1. Roue à aubes qui provoque une circulation allant d'un écoulement turbulent à un écoulement laminaire de masses de liquide et de suspensions liquides et qui assure efficacement le mélange et le brassage de telles masses, la roue (10) ayant plusieurs aubes (12, 14, 16) qui peuvent tourner autour d'un axe



(20), les aubes ayant des extrémités (24) qui délimitent le rayon (R) de la roue, les aubes étant dirigées radialement vers l'axe depuis les extrémités, caractérisée en ce que les aubes (12, 14, 16) ont une torsion déterminée par leur inclinaison par rapport à un plan perpendiculaire à l'axe avec des angles qui augmentent progressivement le long des aubes vers l'axe pour donner une condition d'inclinaison excessive, et le rapport d'inclinaison (PR) des aubes dans une section qui se trouve à une distance égale à 0,4R de l'axe est supérieur au rapport PR au niveau des extrémités, le rapport PR étant défini par l'équation suivante :

$$PR = (r/R)\pi\text{tg}(\beta r/R)$$

r étant le rayon compris entre l'axe de rotation et la section dans laquelle le rapport PR est calculé d'après l'équation précédente, et β étant l'angle d'inclinaison dans ladite section, exprimé en degrés.

2. Roue à aubes selon la revendication 1, caractérisée en ce que les aubes ont une torsion, cette torsion ($\Delta\beta$) étant définie par la relation

$$\Delta\beta = \text{tg}^{-1}(PR'/0,4\pi) - \text{tg}^{-1}(PR/\pi)$$

PR' étant le rapport d'inclinaison à une distance radiale approximativement égale à 0,4R et PR étant le rapport d'inclinaison à l'extrémité des aubes, PR' étant supérieur à PR.

3. Roue à aubes selon la revendication 1 ou 2, caractérisée en ce que $\Delta\beta$ est compris entre 30 et 45°.
4. Roue à aubes selon l'une quelconque des revendications 1 à 3, caractérisée en ce que l'inclinaison des aubes (12, 14, 16) à leur extrémité (24) est comprise entre 18 et 30°.
5. Roue à aubes selon l'une des revendications 1 à 4, caractérisée en ce que l'inclinaison des aubes (12, 14, 16) est invariable sur 10 % environ de la distance radiale depuis les extrémités et vers l'axe.
6. Roue à aubes selon l'une des revendications 1 à 5, caractérisée en ce que les aubes (12, 14, 16) sont des organes hydrodynamiquement profilés ayant une cambrure en plus d'une torsion, et en ce que les angles des aubes inclinées sont mesurés entre les cordes (40) des aubes en un point qui se trouve aux extrémités et en un point qui se trouve à une distance à 0,4R.
7. Roue à aubes selon la revendication 6, caractérisée en ce que les cordes (40) aux extrémités (24) sont comprises entre des points d'intersection de segments de droite radiaux de longueur L compris entre l'axe et les bords d'attaque et de fuite des aubes, et les cordes à une distance 0,4R entre des points d'intersection de segments de droite radiaux de longueur 0,4R compris entre l'axe et les bords d'attaque et de fuite.
8. Roue à aubes selon la revendication 6 ou 7, caractérisée en ce que la largeur des aubes (12, 14, 16) exprimée par le rapport de la longueur de la corde (W) au diamètre (D) de la roue à aubes (W/D) aux extrémités est d'environ 0,25 aux extrémités et d'environ 0,33 à une distance de 0,4R.
9. Roue à aubes selon la revendication 8, caractérisée en ce que le rapport de l'épaisseur à la longueur de la corde (40) à l'extrémité est d'environ 0,1 et le rapport de l'épaisseur à la longueur de la corde à une distance égale à 0,4R est compris entre environ 0,14 et 0,22.
10. Roue à aubes selon l'une quelconque des revendications 1 à 9, caractérisée en ce que les aubes (12, 14, 16) ont des surfaces opposées qui créent respectivement des pressions relativement élevée et relativement faible dans les masses, et des saillies délimitant des guides d'écoulement sur les surfaces.
11. Roue à aubes selon l'une des revendications 1 à 10, caractérisée en ce que les aubes (12, 14, 16) ont des surfaces opposées qui créent des pressions relativement élevées dans les masses, et un dispositif générateur d'une turbulence placé sur les surfaces à basse pression.



12. Roue à aubes selon l'une des revendications 1 à 11, caractérisée par une seconde roue à aubes (82), la seconde roue à aubes ayant des aubes disposées afin qu'elles tournent coaxialement au premier ensemble d'aubes et espacées avec une relation de dépendance par rapport à celles-ci.

- 5 13. Roue à aubes selon la revendication 12, caractérisée en ce que le premier ensemble d'aubes et les aubes de la seconde roue à aubes ont des surfaces opposées qui créent des pressions relativement faible et relativement élevée sur les masses, les roues à aubes étant disposées de manière que les surfaces de pression relativement faible des aubes de la seconde roue à aubes (82) soient adjacentes à la surface de pression relativement élevée des premières aubes.

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