ROTODYNAMIC MACHINE COMPRISING HELICO RADIO AXIAL IMPELLERS WITH INTERFACIAL SLIP CONTROL

Technical area

The invention relates to a rotodynamic machine comprising one or more helico radio axial impellers with interfacial slip control which can generally be used for the simultaneous compression of a liquid phase and a gaseous phase. These applications are found, in particular, in oil production, petrochemicals or in chemistry.

Prior art

By two-phase mixture is meant, in this document, a mixture composed of two phases, a relatively incompressible fluid (liquid) and a compressible fluid (gas) of lower density, each phase being insoluble in the other. Gas Liquid Ratio (GLR) is the ratio between the volume flow rate of the gas phase and that of the liquid phase. The density (actually, volumetric mass) ratio (RoLG) is the ratio of the density of the liquid phase to that of the gaseous phase. Interfacial slip refers to the velocity difference between the gas phase and the liquid phase. By extension, multiphase mixtures refer to mixtures composed of several immiscible liquid phases and a single gaseous phase.

Compression machines operate essentially according to two principles: rotodynamic and volumetric. In the first case (mainly, radial and axial machines), the energy communicated to the compression shaft is transmitted to the fluid, at first, in a kinetic form (through a moving element in rotation) then, in a second step, converted to potential energy (pressure in a fixed element). In the second case (mainly, screw, piston, lobe and diaphragm machines), the energy imparted to the shaft is converted directly into potential energy.

The moving element of a rotodynamic machine is called an “impeller” or a “wheel” while the fixed element is called a “diffuser” or a “rectifier”. The term "hydraulic cell" refers to the assembly of a fixed element and a moving element.

The side walls of an axial or a helico radio axial hydraulic channel are constituted, firstly, by a casing (or cover - outer diameter) and a hub (inner diameter) for, respectively, the furthest and closest parts from the axis of rotation and, secondly, by an intrados and an extrados for, respectively, the inner and outer parts of two adjacent blades. The ortho radial area of a channel is defined in a radial plane by the area between two adjacent blades. The orthogonal area of a channel is defined in a plane normal to two adjacent blades by the area between these two blades.

The **two-phase loss coefficient**, Phi, measures the ratio between pressure losses relative to a two-phase flow (separated phases) and those relating to a homogeneous flow (closely mixed phases appearing as a monophasic flow) under the same flow condition. Due to the existence of interfacial losses, this coefficient is always greater than 1.

The **two-phase efficiency** of a machine (pump or compressor) is the ratio between the energy imparted to a mixture of fluids and the energy transmitted to the shaft moving the impellers of the machine. The **two phase efficacy** for manometric head (respectively, energy) is the ratio between the head (respectively, energy) communicated to a two-phase mixture and that communicated to a homogeneous single-phase under the same operating condition (speed, volume flow rate, density, viscosity). Due to the existence of interfacial losses, the two phase efficacy is always less than 1.

Rotodynamic machines designed for the compression or the pumping of a monophasic fluid (radial and axial compressors and pumps) are totally unsuitable for the compression of a two phase gas - liquid mixture with a large density contrast. Beyond a volume gas fraction of the order of 5% (variable according to the ratio of the densities) the gas creates a bottleneck (restricted area) at the inlet of an impeller. This results in a significant degradation of performance, consequently, the two-phase manometric head and efficiency tending towards zero.

In the 1980s (Patents FR2474614, US4310335, WO8702117), a helico axial machine was developed to overcome this problem allowing the compression of a two-phase mixture independently of the gas fraction (from a completely gaseous phase to a phase totally liquid). Although this invention has provided a great improvement in the compression of a two-phase mixture, this machine presents two main defects: a degraded monophasic efficiency of the order of 30% compared to a conventional monophasic machine and a progressive degradation of the two-phase efficacy as the two phase condition moves away from monophasic conditions (minimal efficacy when the GLR is between 2 and 6). This degradation increases when the density ratio (RoLG) increases. The monophasic and diphasic efficiencies of a helico axial machine (rotodynamic type) are also lower than those of a two-phase volumetric machine.

A helico axial machine however has advantages over a volumetric machine: better resistance to abrasive or erosive elements, an ability to operate over a very long time regardless of the volume gas fraction, the absence of vibrations and a lower requirement in terms of maintenance.

The two-phase performance of radial, axial and helico axial machines can be analyzed as follows:

- In a radial machine, the denser liquid phase is expelled radially by the centrifugal force while the gaseous phase remains blocked at the inlet of an impeller. Conversely, in a fixed, horizontal, rectilinear and constant section pipe, calculation and observation show that the gas phase moves faster than the liquid phase, mainly because of a lower density and a lower viscosity generating less viscous dissipation. The displacement of the phases in a helico axial channel (cylindrical outer surface) in rotation is similar to that occurring in the pipe mentioned above where the liquid phase has a velocity lower than that of the gas phase. This difference in velocities increases when the ratios of densities and viscosities of the phases increase.
- By design, the blade of a helico axial impeller is not very curved generating a lower head coefficient compared to a radial impeller.
- By design, the blade of a helico axial impeller is very long generating a very high viscous dissipation leading to a decrease in monophasic efficiency and, consequently, in two-phase flow.
- Interfacial sliding inside the impellers and the diffusers generates different angles of incidence for each phase upstream of the blade leading edge and, therefore, input losses that do not exist for volumetric machines.

Attempts have been made using helico radio axial (HRA) machines to provide better performance compared to that of a helico axial machine (HA). An example of an HRA machine is presented in the patent FR2899944. Several factors may explain the resulting poor performance:

- The invention describes an angle Delta (formed in a meridian plane between the impeller outside _ casing internal and the axis of rotation) varying between -20 and +20 degrees at the inlet of the impeller. It should be noted that a negative angle prevents most of the liquid flow to move forward, therefore, to a slowing of the liquid phase and consequently to an increase in the interfacial slip (the opposite of the desired effect). An angle greater than a few degrees leads to a fast expulsion of the liquid phase to the impeller outlet as it occurs with radial impellers (blocking the gas at impeller inlet).

- The impeller outside presents a strictly concave shape (increasing angle Delta from the inlet to the outlet) not allowing a control of phase sliding at any point in the axial direction. An equality in velocity displacement for the two phases can be reached only at a point but not over a large distance with a strictly concave shape.
- Angle Delta is described by intervals of a few tens of degrees (between - 20 and + 20 at inlet and between 0.1 and 70 at outlet) while the sensitivity to the radial acceleration should lead to angles 10 to 100 times lower than the extreme values mentioned above (20 and 70 degrees).

The purpose of the present invention is precisely to improve the performance relative to the design of helical axial and helical radio axial machines of the prior art.

Summary of the invention

The invention relates to a rotodynamic machine for compressing a multiphasic mixture comprising at least one gaseous phase and one liquid phase, said machine comprising at least one impeller rotating around an axis and mounted inside a casing and at least one diffuser integral with the casing, said impeller comprising at least two blades to form at least two channels delimited by the hub, the casing and the two so-called blades, characterized in that:

- The outside of said impeller (casing inside) comprises at least two sections each with a curved shape in the meridian plane, a first concave part located on the inlet side of said impeller for an acceleration of the liquid relative to the gas and a second convex part located on the outlet side of said impeller to maintain the velocity of the liquid at a value roughly equal to that of the gas.
- The mean angle Alfa of the impeller outside (casing inside) is between 1 and 4 degrees, said angle Alfa being included in a meridian plane and formed by the tangent to the impeller outside and the axis of rotation (z axis) and said average angle Alfa being defined by the two sides of a right-angled triangle whose opposite side is equal to the difference between the outlet and inlet radius of the impeller and the adjacent side is equal to the axial length of the impeller, the inner radius of the casing (outer radius of the blades) being continuously increasing from the inlet to the outlet.
- According to a mode of realization, the angle Alfa defining the shape of the impeller outside (casing inside) in a meridian plane is locally equal to the following values: 0.2, 4.0, 3.5, 1.5 and 0.3 degrees, respectively, at the following axial positions: 0.0, 0.4, 0.6, 0.8 and 1.0, the latter values representing the ratio between the axial distance to the inlet and the axial length of the impeller while angle values may vary within 50 percent and linearly with the above angle values.
- According to a mode of realization, at the inlet of said impeller, the angle Beta of the blade, formed in a plane tangential to a cylinder of revolution, by the projected tangent to the blade with a line located in a radial plane and tangent to the cylinder of revolution, is between 4 and 30 degrees.
- According to a mode of realization, at the outlet of said impeller, the angle Beta of the blade is between 10 and 60 degrees, this angle increasing continuously from the inlet to the outlet in an interval of plus or minus 10 percent.
- According to a mode of realization, said rectifier comprises at least two blades, the height of said blades at the inlet of said rectifier being equal to the height of the blades at the impeller outlet in a range of ten percent and that at the outlet of said rectifier being equal to that at the impeller inlet in a range of ten percent.

Brief description of the figures

Other features and advantages of the invention will be better understood and become clearly apparent on reading the description, given by way of illustration and in no way constituting a limitation, made hereinafter with reference to the drawings, among which:

- Figure 1 represents, in meridian section, two impellers according to the invention of a multiphase compressor (pump) with a diffuser between these two impellers.
- Figures 2 and 3 show the variation of a two-phase loss coefficient, Phi, as a function of the GLR and the RoLG for a rectilinear channel in rotation with, respectively, a zero angle (Figure 2) and a positive angle (Figure 3) inclination relative to the axis of rotation.
- Figure 4 defines the x, y and z axes, a radial plane, a plane tangent to a cylinder of revolution and the Beta angle of a channel in this tangent plane. The Beta angle of a blade is formed in a tangential plane, firstly, by the projected tangent to the mean direction of the channel flow and, secondly, by the axis x. The axis x located in a radial plane (normal to the axis of rotation) is tangent to the cylinder of revolution. The y axis located in a meridian plane (passing through the axis of rotation) is perpendicular to the axis of rotation. The z axis parallel to the axis of rotation is normal to the other two axes. The x, y and z axes form an orthogonal system.
- Figure 5 defines a velocity triangle in a plane tangent to a cylinder of revolution.
- Figure 6 defines the angle Alfa of inclination of the casing inside (impeller outside) with respect to the axis of rotation.
- Figure 7 represents the variation of the angles Alpha and Beta as a function of the axial position in the context of an application.
- Figure 8 represents the variation of the outer and inner diameters as a function of the axial position as well as the five sections defining the shape of the outside diameter.
- Figures 9 and 10 show the variation of the two-phase efficacy as a function of the GLR and the RoLG, respectively, for a conventional helico axial impeller and for an impeller according to the present invention.

Detailed description

The present invention relates to a rotodynamic machine represented by figure 1 and, more particularly, to an internal casing shape (or impeller outside, 6) of the two-phase compression impeller 2 in a meridian plane of said impeller associated with an angle Alfa of said casing according to the axial direction so that, from the inlet to the outlet of the impeller, the velocity of the liquid is as close as possible to that of the gas thereby limiting the interfacial losses (dissipation in the form of heat) at the interface of the gas and liquid phases and consequently limiting also the energy losses at the inlet of the diffuser by a better adaptation of the inlet angle of the blades of the diffuser 3 to the angle of incidence of the liquid and the gas. This process is called interfacial slip control or phase velocity control in the impeller, the object of this control being to reduce the slip to a value as close as possible to zero. This control is permitted by the component in the flow direction of the centrifugal force exerted on the fluids (vectorial sum of three components) perpendicular to the axis of rotation.

This mode of control of the interfacial slip allows the use of more curved impeller blades than with the prior techniques allowing an increase of the monophasic efficiency and the pressure coefficient. This results, in two-phase, to an increase in efficiency, pressure coefficient and efficacy.

Before explaining the characteristics of the impeller according to the invention a few reminders are given on the behavior of a two-phase fluid in more simple cases.

The invention relating to a two-phase compression impeller with interfacial slip control will be better understood by the **description of a two-phase flow in a rectilinear channel subjected to rotation**, the channel being located in a meridian plane passing through the axis of rotation and at a constant distance from the axis during rotation. The channel is of constant area (perpendicular to the flow) maintaining a constant velocity for each phase between the inlet and the outlet of the channel. The characteristics of the flow are presented below in two cases: firstly, for a zero inclination and, secondly, for a tilted channel (non zero inclination relative to the axis of rotation).

As for a channel at rest, when the inclination of the rotating channel is zero (the distances from the inlet and the outlet of the channel to the axis of rotation are equal), the velocity of the liquid phase is lower than that of the gas, the difference in velocities increasing with the density ratio RoLG. The interfacial losses are measured by the Phi factor for the same operating condition (volume flow and rotational speed) for several RoGL and GLR values. The factor Phi 26 is shown on figure 2. It is maximum when the GLR 25 is between 2 and 2.5 (regardless of the value of RoLG). The Phi factor is maximum when the RoLG ratio is the highest, ie forty (21 - Curve with triangles) and minimum when this ratio is the lowest, ie five (24 - Curve with squares). It continuously increases from the minimum value to the maximum value of RoLG. The intermediate values of RoLG represented are ten, 23 and twenty, 22.

The inclination is said to be "negative" when the distance to the axis of rotation is greater at the inlet than at the outlet of the channel. In this configuration, the component along the direction of the channel of the centrifugal force created by the rotation of the channel is directed from the outlet to the inlet of the channel slowing the drive of the liquid to the outlet of the channel. This results in an increase in the Phi factor corresponding to an increase in the slip factor between the two phases.

When the inclination is “positive” (distance to the axis of rotation smaller at the inlet than at the outlet of the channel), the liquid is accelerated relative to the gas (component of the centrifugal force, along the direction of the channel, directed from the entrance to the exit of the channel). The Phi factor 34 is represented on figure 3 as a function of the GLR 33 in the case of a 0.4 degree inclination. This factor is maximum when the GLR is between 1.5 and 2. It is considerably reduced, for all the GLR and RoLG values compared to figure 2, indicating a liquid phase velocity relatively close to that of the gas phase. Note that the first velocity is still slightly less than the second one. The extreme values of RoLG are represented for values of forty, 31 and five, 32 (figure 3).

To approach an equality between the velocities of the two phases, it is necessary to use an inclination of 0.5 degree (not shown here). In this latter case, the two phases (separated phases) flow in a manner similar to a uniform monophasic flow. As a result, the interfacial losses are minimum and the loss coefficient Phi is close to 1 for all RoLG and GLR values.

When a rotating rectilinear channel is located in a meridian plane passing through the axis of rotation (previous section) only a single centrifugal force (due to the rotation of the channel) is present, the vectors representing the flow velocities for each phase being located in a plane passing through the axis of rotation. **In this case, there is no Coriolis effect.**

Figure 4 defines a radial plane 42 normal to the axis “z” of rotation 45, a tangential plane 43 to a cylinder of revolution 41 of radius R, 40, the axis “x” 44 located both in a radial plane and in a tangential plane, the axis “y” 46 normal to the “x” and “z” axes and the angle Beta 50 formed by the tangent 49 to the projected 48 in the tangent plane of the direction of the channel and by the axis “x” 44. The “generator” 47 represents the line of tangency between the tangent plane 43 and the cylinder of revolution 41.

Figure 5 defines a triangle of velocity of a flow projected on a plane tangent 43 to a revolution cylinder at any point of a helical channel. In this triangle, the velocities W 53 and V 52 refer, respectively, to the relative (relative to the channel) and absolute (relative to a fixed frame) velocities of the flow. U 51 is the drive speed of the channel. Wx 55 and Vx 54 refer to the velocities W and V projected along the axis x 44 and Wz 56, the velocity W projected along the z 45 axis.

Two centrifugal forces and a centripetal force (Coriolis) are generated by the rotation of the channel and by the displacement of the fluid inside the channel given the non-zero component in the radial plane of Wx velocity. These three forces, F1, F2 and F3, all perpendicular to the rotation axis, are due to:

- The rotation of the channel; $F1 = Omg^2 * R$, centrifugal,
- The displacement of the fluid in a direction transverse to a meridian plane (the component of the velocity W in the radial plane); $F2 = | Wx |^2 / R$, centrifugal,
- The Coriolis force ; $F3 = 2 * Omg * | Wx |$, centripetal,

Where Omg is the angular velocity, R the distance of the channel to the axis of rotation. The forces F1, F2 and F3 are oriented in the direction y 46. The sum of the forces F1 and F2 is greater (in absolute value) or equal to F3. Consequently, the resultant of the forces F1 + F2 + F3 is always directed in the direction of the centrifugal forces. F1+F2 is equal to F3 only when $|Wx| = Omg * R$.

It appears in this section that the centrifugal force acting on the fluid is not only dependent on Omg and R but also on W and Beta. It is this last observation which is implemented in the channel of a helico radio axial impeller.

The two-phase compression / pumping machine shown in figure 1 consists of at least one helico radio axial impeller 2 described below followed by a rectifier 3. The impeller comprises blades 4 and 12 mounted on a hub 5 which is mounted on a drive shaft 16. The interior of the vanes constitutes a channel in which moves a fluid.

The channels are closed (a cover mounted on the outside of the impeller) or opened, as shown on figure 1, the fixed part opposite to the impeller constituting the upper part of the channels (the casing, 6). In the latter case, in order to limit the leakage of the fluid between two channels, the radial distance 13 between the tip of the blades and the casing is of the order of a few tenths of a millimeter. Arrows 14 and 15 represent the direction of the flow from the inlet of the first impeller to the outlet of the last impeller (figure 1) or rectifier (diffuser).

The geometric shape of the blades 12 in an orthogonal plane (perpendicular to a blade or to the channel flow) is designed, in the manner known to those skilled in the art, so as to provide sufficient mechanical strength to the blades (forces applied by the fluid and resulting from the rotation). The leading and trailing edges of the blades are designed to minimize hydraulic blockage and reduce losses between fixed and moving parts (rectifier and impeller).

An impeller is followed by a rectifier constituting a hydraulic cell. Several hydraulic cells are connected in series from the inlet to the outlet of the two-phase compression machine so as to satisfy the manometric head requirements (pressure elevation). The impellers and the rectifiers are separated by intervals of the order of a millimeter to limit fluid leakage between fixed and rotating elements, same for the interval 11 separating a rectifier from the axis of rotation. The inlet and outlet piping and scrolls are designed as is known to those skilled in the art.

The outside diameter and the height of the blades at the impeller inlet and the rotational speed define the volume flow rate as is known to those skilled in the art.

The overall shape of the channels is designed differently in the context of the invention by comparison with a strictly helico axial channel (HA) or a helico radio axial channel (HRA) not designed for the control of interfacial slip:

- ***Conventional Helico Aaxial channel*** – In this case, phase mixing is permitted as long as GLR, RoLG, viscosities and surface tension allow. This is usually possible when the GLR is less than 2 (this value depends on the RoLG). This mixing is facilitated by designing the channel so that the accelerations in an orthogonal system are as low as possible in three directions (respectively, longitudinal to the flow, radial and normal to the blades). This results in extremely long channels leading to high viscous losses, a lower velocity of the liquid relatively to the gas inside the impeller and also at the impeller outlet leading to high losses in the diffuser inlet. In the case of an HA impeller, the outside diameter is constant (the component, in the meridian plane, of the flow, at the level of the casing, is made along a direction 63 parallel to the axis of rotation z 45, figure 6). With these impellers, the hub presents a concave shape delaying the expulsion of the liquid phase from the hub to the casing (it happens only for low GLR). Unfortunately, this design causes an inhomogeneity in the variation of the flow area from the inlet to the outlet generating an inhomogeneity in the flow rate, therefore, additional losses.
- ***Helico Radio Axial channel with a concave outside (cover) -*** In figure 6, the component, in the meridian plane, of the flow, at the level of the casing, is made along a direction 62 with an inclination continuously increasing with the distance from the impeller inlet. This does not permit to control the interfacial slip as the liquid phase is in continuous acceleration from the impeller inlet to the outlet.
- ***HRA impeller channel according to the invention with interfacial slip control*** - In this type of channel, the phase mixture is expected when the GLR is relatively low (less than 1 or 2) while the control of the interfacial slip is exerted when phase separation is unavoidable. The phase control is based on an acceleration of the liquid phase using the component in the flow direction of the resultant centrifugal and centripetal forces. In figure 6, the component in the meridian plane of the flow, at the casing 61, is made in a direction 62 making an angle Alfa 64 with the axis of rotation z 45. This acceleration is controlled by the shape of the casing (successively concave and convex) in order to avoid an excessive liquid velocity relative to that of the gas as it occurs with radial impellers or in HRA impellers with a steep and unique slope. According to this principle of operation, it is no longer necessary to minimize the acceleration in the flow direction (resulting from the change in flow area) nor the acceleration transverse to the blades allowing the use of more curved blades and shorter blades than the ones used in previous designs. Moreover, the shape of the hub is determined by a regular orthogonal area variation from the inlet to the outlet (limitation of diffusion losses) rather than by the creation of a centripetal force to delay the expulsion of the liquid towards the casing (impeller outside).

From the design of an impeller with control of the interfacial slip, it results several advantages:

a. Interfacial losses in the impeller greatly reduced

b. Reduced two-phase losses at the rectifier inlet and interfacial losses inside the rectifier

c. Viscous losses (monophasic and diphasic) inside the impeller channel reduced

d. An increased pressure coefficient (monophasic and diphasic).

Items a and b mentioned above lead to an increase of the two phase efficacy independently of an improvement of the monophasic performances.

Items c and d mentioned above lead, in monophasic as well in diphasic, to an increase of the efficiency and of the compression ratio. These improvements are in addition to the performances listed in the previous paragraph concerning, strictly, the improvement of the two-phase efficacy (Items a and b).

These performances are obtained by the invention defined as follows and according to figure 7 representing the variation of the angles Beta 71 and Alfa 72 in degrees 74 as a function of the axial position 73, in meters, and according to figure 8 representing the variation of the outside diameter 6 and the inside diameter 5, in meters, as a function of the axial position 82, in meters (It should be noted that in figures 7 and 8 the dimensions are relative to a particular application given as an example - Average external diameter of the impeller of 0.25 m):

- The impeller outside (or casing inside) 6 comprises, in the meridian plane, from the inlet to the outlet at least two sections each with a curved shape, a first part concave 84 (summit directed towards the axis of rotation - internally) located on the inlet side of the said impeller for an acceleration of the liquid relative to the gas and a second part convex 86 (summit directed opposite to the axis of rotation - externally) located on the outlet side of said impeller for reducing the acceleration of the liquid with respect to the gas.
- At the impeller outside 6 and at the impeller inlet, therefore, upstream of the concave section, the section 83 is substantially parallel to the axis of rotation. At the impeller outside and at the impeller outlet, therefore, downstream of the convex section, the section 87 is substantially parallel to the axis of rotation. The derivatives as a function of the axial position of the equations describing the concave and convex shapes are equal at each connection as well at at the upstream side (between 83 and 84) as the downstream side (between 86 and 87) as in the intermediate portion (85, between 84 and 86) in order to limit losses resulting from discontinuity of shape or shape variation.
- The mean angle Alfa 64 at the impeller outside, in the meridian plane, is included between 1 and 4 degrees (preferably 2 degrees), the said angle Alfa being defined by the two sides of a right angle triangle whose opposite side is equal to the difference between the outlet and inlet radius of the impeller and the adjacent side is equal to the axial length of the impeller.

At the impeller outside 6, the radius is continuously increasing from the inlet to the outlet of the impeller. A further element of the invention consists in an accentuation of the curved shape of the blades in order to increase, relatively to a conventional hydraulic HA, the angle difference between the exit and entry of the blades (Beta), therefore, reducing the length of the blades. This action promotes both the pressure coefficient and the efficiency. These performance improvements are obtained in single phase as well as in two phase flows.

The number of blades n is chosen so that the orthogonal distance between two adjacent blades approximates, in the middle of the axial distance, the blade height measured by the distance separating the casing to the hub in a direction normal to the axis of rotation. However, the number of blades may be greater or smaller by one unit than the number of blades defined above. The blades are preferably mounted radially to the hub.

The angle Beta of the blades preferably varies linearly between the inlet and the outlet of the impeller.

The ratio between the outside diameter at the inlet and the axial length of the impeller is taken between 4 and 6 and is preferably equal to 5. The orthogonal area between two blades increases continuously from the entrance to the exit with a ratio included between 1.5 and 2.1 (preferably 1.8) taking the ratio between the exit and inlet areas.

At the inlet, the external diameter of the impeller, the blade height and the angle Beta are defined as is known to those skilled in the art according to the data relating to the volume flow of the multiphasic mixture and the speed of rotation.

The laws of variation of the external diameter, the angles Alfa and Beta as well as the variation of orthogonal area define the internal diameters as well as the shape of the blades at any point of the impeller.

The performances resulting from the invention are given in the case of a particular application that does not limit the scope of the invention, figures 7 to 10:

The average angle Beta, 71 is 8 degrees at the inlet and 32 degrees at the outlet of the impeller. It evolves linearly along the axial direction.

At the impeller outside, the angle Alfa, 72 takes the values as shown below.

Position	Inlet		Middle		Outlet
Casing shape	Linear	Concave	Linear	Convex	Linear
Dist Fraction	0.0	0.4	0.6	0.8	1.0
Alfa Degrees	0.2	4.0	3.5	1.5	0.3

Where "Dist Fraction" represents the distance to the entry as a fraction of the axial length and where "Casing Shape" indicates the corresponding shape at a specific location (Linear, Concave or Convex).

The two-phase efficacies for both a conventional helico axial impeller and a helico radio axial Impeller with interfacial slip control are presented, respectively, in figures 9 and 10. The results are as follows:

- The two-phase efficacy is minimum for both types of impeller and for all density ratio RoLG when the GLR is close to 2.
- For a large density ratio RoLG (value of 40), the two-phase efficacy is increased significantly from 0.23 (conventional HA – 91, figure 9) to 0.60 (HRA according to the invention – 101, figure 10).
- For a low density ratio RoLG (value of 5), the two-phase efficacy is increased from 0.63 (conventional HA – 94, figure 9) to 0.80 (HRA according to the invention – 104, figure 10).

It appears through this numerical example that the interfacial slip control is considerably more effective when the density ratio RoLG is high, that is to say also when the phase separation is unavoidable.

Variation of the pressure coefficient according to the present application: The pressure coefficient is increased from 0.50 (conventional HA) to 0.60 (HRA according to the invention) allowing a reduction in the number of compression stages of the order of 20% for a given manometric head.

CLAIMS

1 - Rotodynamic machine (1) for compressing a multiphasic mixture comprising at least one gaseous phase and one liquid phase, said machine comprising at least one impeller (2) rotating around an axis (16) and mounted in a casing (6) and at least one diffuser (3) integral with the casing, said impeller having at least two blades (4 and 12) in order to form at least two channels delimited by the hub (5), the casing and the said two blades, characterized in that:

- The said casing (6) comprises, in the meridian plane, at least two sections characterized by a curved shape, a first concave part (84) located at the inlet side of the said impeller for accelerating the liquid relative to the gas and a second convex part (86) located at the outlet side of the said impeller for maintaining the liquid velocity at a value substantially equal to that of the gas,
- The mean angle Alfa (64) of the casing, in the meridian plane, is between 1 and 4 degrees, the said angle Alfa being defined by two sides of a right angle triangle where the opposite side is equal to the difference, taken on the impeller outside, between the outlet and inlet radius and the adjacent side is equal to the axial length of the impeller, the inner radius of the casing (outer radius of the blades) continuously increasing from the inlet to the outlet.

2 - Rotodynamic machine according to claim 1 characterized in that the angle Alfa defining the shape of the casing in a meridian plane is locally equal to the following values: 0.2, 4.0, 3.5, 1.5 and 0.3 degrees, respectively, at the following axial positions: 0.0, 0.4, 0.6, 0.8 and 1.0, the latter values representing the ratio between the axial distance to the inlet and the axial length of the impeller, the angle values may vary in a range of 50 percent and substantially linearly in the intermediate axial positions.

3 - Rotodynamic machine according to claim 2 wherein, at the entry of the impeller, the angle Beta of the blade, formed in a plane tangential to a cylinder of revolution by the projected tangent line to the blade with a line located in a radial plane and tangent to the cylinder of revolution, is between 4 and 30 degrees.

4 - Rotodynamic machine according to claim 2 wherein, at the outlet of the impeller, the angle Beta of the blade is between 10 and 60 degrees such that this angle increases continuously from the inlet to the outlet within a range of plus or minus 10 percent.

5 - Rotodynamic machine according to one of the preceding claims characterized in that the number of blades n of the impeller is equal to or greater than 2 and is chosen so that the orthogonal distance between two adjacent blades approximates the height of the blade measured by the distance separating the casing to the hub in a direction radial to the axis of rotation.

6 - Rotodynamic machine according to one of the preceding claims characterized in that the number of blades is greater or less than one unit to the number of blades n corresponding to the approximation between the orthogonal distance taken between two adjacent blades and the height of the blades.

7 - Rotodynamic machine according to one of the preceding claims characterized in that the orthogonal area between two blades increases continuously from the inlet to the outlet and in a ratio between 1.5 and 2.1 (preferably 1.8) between the exit and entrance areas.

8 - Rotodynamic machine according to one of the preceding claims characterized in that the rectifier comprises at least two blades, the height of said blades at the inlet of said rectifier being equal to the height at outlet of the impeller blades in a range of ten percent and that of said blades at the outlet of said rectifier being equal to the height at the inlet of the impeller blades in a range of ten percent.

9 - Rotodynamic machine according to one of the preceding claims characterized in that the impeller comprises a cover on its outside part the said cover being fixed to the impeller blades.

10 - Rotodynamic machine according to one of the preceding claims comprising several impellers separated by rectifiers.