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Pump for conveying a highly viscous fluid

The invention relates to a pump for conveying a highly viscous fluid in accordance with the preamble of the independent claim.

10 Pumps for pumping highly viscous fluids are used in many different industries, for example in the oil and gas processing industry for conveying hydrocarbon fluids. Here, these pumps are used for different applications such as extracting the crude oil from the oil field, transportation of the oil or other hydrocarbon fluids through pipelines or within refineries. But also in other industries for example the food industry or the chemical industry there is often
15 the need for conveying highly viscous fluids.

The viscosity of a fluid is a measure for the internal friction generated in a flowing fluid and a characteristic property of the fluid. Within the framework of this application the term “viscosity” or “viscous” is used to designate the kinematic viscosity of the fluid and the term “highly viscous fluid” shall be
20 understood such, that the fluid has a kinematic viscosity of at least $10^{-4} \text{ m}^2/\text{s}$, which is 100 centistokes (cSt).

For the pumping of highly viscous fluids it is known to utilize centrifugal pumps. Pumping highly viscous fluids with centrifugal pumps requires considerably more pump power than for example pumping water. The higher
25 the viscosity of the fluid becomes the more power the pump needs to deliver the required pumping volume. Especially in the oil and gas industry the main focus – at least in the past – has been on pumping volume, i.e. the flow generated by the pump, and on the reliability of the pump rather than the

efficiency of the pump. However, nowadays a more efficient use of the pump is strived for. It is desirable to have the highest possible ratio of the power, especially the hydraulic power, delivered by the pump to the power needed for driving the pump. This desire is mainly based upon an increased awareness
5 of environment protection and a responsible dealing with the available resources as well as on the increasing costs of energy.

To improve the efficiency of a pump for pumping highly viscous fluids it is known to use specific impeller designs, especially impellers with high head coefficients. The head coefficient of the impeller can be increased for example
10 by increasing the blade outlet angle or the number of blades or the impeller outlet width. Despite of these measures there is still a need to even more improve the efficiency of a pump for pumping highly viscous fluids.

Therefore, it is an object of the invention to propose a new pump for conveying highly viscous fluids that has a better efficiency, i.e. an increased
15 ratio of the power delivered by the pump when pumping the fluid to the power that is supplied to the pump for driving the pump

The subject matter of the invention satisfying this object is characterized by the features of the independent claim.

Thus, according to the invention a pump for conveying a highly viscous fluid is
20 proposed, comprising a casing with at least a first inlet and an outlet for the fluid, an impeller for conveying the fluid from the inlet to the outlet, wherein the impeller is arranged on a rotatable shaft for rotation around an axial direction, and comprises a front shroud facing the first inlet of the pump, wherein the casing is provided with a stationary impeller opening for receiving the front
25 shroud of the impeller and having a diameter, wherein the front shroud and the stationary impeller opening form an gap having a length in the axial direction, wherein the ratio of the length of the gap and the diameter of the impeller opening is at most 0.092.

The invention is in particular based upon the finding that the pump efficiency
30 may be increased when pumping highly viscous fluids by designing the gap

between the front shroud of the impeller and the stationary impeller opening considerably shorter than it has been done in the prior art.

5 The gap which is sometimes also designated as the labyrinth is needed for sealing the high pressure side of the impeller, more particular the side room, against the inlet of the pump. The impeller is arranged in the stationary impeller opening which is a part of the pump that is stationary with respect to the casing and adapted to receive the impeller. In the mounted state the impeller is located in said impeller opening such that there is the gap or the
10 labyrinth between the outer circumferential surface of the impeller's front shroud and the inner circumferential surface of the stationary impeller opening. This gap has a length in the axial direction which provides a sealing between the side room on the high pressure side of the impeller and the inlet of the pump, which is the low pressure side of the pump.

15 During operation of the pump a back flow is generated flowing from the high pressure side of the impeller, which is for a single stage pump the region near the outlet of the pump, through the side room, and through the gap between the front shroud and the stationary impeller opening back to the low pressure side of the impeller.

20 The gap or the labyrinth, respectively, is designed as a radial clearance seal or labyrinth, i.e. it provides a clearance with respect to the radial direction. Therefore the main flow through the gap is in axial direction, i.e. parallel to the shaft. This has to be differentiated from an axial clearance seal or labyrinth that extends perpendicularly or obliquely to the shaft, thus the main flow through an axial clearance seal is in radial direction or oblique with respect to
25 the radial direction. In an axial clearance seal the clearance in axial direction changes upon a relative movement of the stationary part and the rotating part in axial direction, wherein in a radial clearance seal the clearance in radial direction changes upon a relative movement of the stationary part and the rotating part in radial direction.

30 An essential finding is that by the short axial length of the gap (i.e. the labyrinth) proposed by the invention the power losses across the gap are decreasing inter alia due to the reduced drag in the side room. On the other

hand one may expect that the shortening of the gap would result in a reduced sealing action thus increasing the back flow in the pump. However an increase in the back flow rate reduces the pump efficiency and thus contravenes an improved efficiency. Therefore the unexpected finding is that
5 by shortening the gap with respect to the axial direction the overall pump efficiency increases despite the risk of an enhanced back flow rate.

According to the invention the length of the gap shall not exceed 0.092 times the diameter of the impeller opening.

10 The optimal length of the gap depends on several factors for example the viscosity of the fluid. Thus, depending on the specific application it may be preferred that the ratio of the length of the gap and the diameter of the impeller opening is at most 0.073 and preferably at most 0.055.

There are also applications for which it is advantageous when the ratio of the length of the gap and the diameter of the impeller opening is at most 0.037
15 and preferably at most 0.019.

For practical reasons there is also a preferred lower limit for the length of the gap. According to the preferred design, the ratio of the length of the gap and the diameter of the impeller opening is at least 0.0001.

20 In order to generate the desired sealing effect by the gap it is preferred to have a radial clearance between the front shroud and the impeller opening which is at most 0.0045 times the diameter of the impeller opening. The radial clearance is the extension of the gap with respect to the radial direction, i.e. perpendicular to the axial direction, and may be considered as the width of the gap. This radial clearance is the minimum distance between the outer
25 circumferential surface of the impeller's front shroud and the inner circumferential surface of the stationary impeller opening along the gap.

The two surfaces delimiting the gap may be designed as even surfaces.

According to another embodiment the gap comprises a plurality of lands consecutively arranged with respect to the axial direction, wherein two

adjacent lands are respectively separated by a groove. In such an embodiment the two surfaces delimiting the gap are not even. The part of the outer circumferential surface of the impeller's front delimiting the gap or the part of the inner circumferential surface of the stationary impeller opening
5 delimiting the gap may be provided with a plurality of lands and grooves there between. In such an embodiment the length of the gap in axial direction is defined as the sum of the lengths of all individual lands in the axial direction. The grooves do not contribute to the overall length of the gap in axial direction.

- 10 According to a preferred embodiment the stationary inlet opening comprises a wear ring delimiting the gap with respect to the radial direction, the wear ring being arranged stationary with respect to the casing.

- Supplementary or as an alternative measure it is also possible that the impeller comprises a wear ring delimiting the gap with respect to the radial
15 direction, the wear ring being arranged stationary with respect to the impeller.

- The invention is especially suited for many types of centrifugal pumps. The pump may be designed for example as a single suction pump or a double suction pump, as a single stage pump or as a multistage pump. When the pump is designed as a single suction pump it may have a rear shroud on the
20 impeller in addition to the front shroud. In such a design it is also possible that the rear shroud of the impeller forms a gap with a part being stationary with respect to the casing. This gap at the rear shroud may be designed in an analogously same manner as it is explained with respect to the gap at the front shroud of the impeller.

- 25 According to a preferred embodiment the pump is designed as a double suction pump, having a second inlet for the fluid being arranged oppositely to the first inlet of the pump, wherein the impeller is designed as a double suction impeller comprising vanes for conveying the fluid both from the first inlet and from the second inlet to the outlet.

- 30 For such a design as a double suction pump it is preferred, that the impeller comprises a second front shroud facing the second inlet of the pump, wherein

the casing is provided with a second stationary impeller opening for receiving the second front shroud of the impeller and having a diameter, wherein the second front shroud and the second stationary impeller opening form a second gap having a length in the axial direction, and wherein the ratio of the length of the second gap and the diameter of the second impeller opening is at most 0.092.

Depending on the specific application it may be preferred that also the ratio of the length of the second gap and the diameter of the second impeller opening is at most 0.073 and preferably at most 0.055.

There are also applications for which it is advantageous when the ratio of the length of the second gap and the diameter of the second impeller opening is at most 0.037 and preferably at most 0.019.

Also for the second gap it is advantageous, when there is a radial clearance between the second front shroud and the second impeller opening which is at most 0.0045 times the diameter of the second impeller opening.

It is an especially preferred measure when the gap and the second gap are designed essentially in an identical manner.

According to an essential application the pump is designed for the use in the oil and gas industry.

Further advantageous measures and embodiments of the invention will become apparent from the dependent claims.

The invention will be explained in more detail hereinafter with reference to the drawings. There are shown in a schematic representation:

Fig. 1: a cross-sectional view of an embodiment of a pump according to the invention,

Fig.2: an enlarged representation of detail I in Fig. 1,

Fig. 3: a sketch of the front shroud and a wear ring as part of the stationary impeller opening,

Fig. 4: as Fig. 3, but for a variant of the embodiment,

5 Fig. 5: a second variant for the design of the gap between the front shroud and the stationary impeller opening, and

Fig. 6: an illustration of a comparison of a pump according to the invention with prior art pumps.

Fig. 1 shows a cross-sectional view of an embodiment of a pump according to the invention which is designated in its entity with reference numeral 1. Fig. 2
10 shows an enlarged representation of detail I in Fig. 1. The pump 1 is designed for conveying a highly viscous fluid, whereas the term "highly viscous" has the meaning that the kinematic viscosity of the fluid is at least $10^{-4} \text{ m}^2/\text{s}$, which is 100 centistokes (cSt).

In this embodiment the pump 1 is designed as a double suction single stage
15 centrifugal pump. This design is one preferred embodiment which is in practice useful for many applications. Of course, the invention is not restricted to this design. A pump according to the invention may also be designed as a single suction centrifugal pump or as a multistage centrifugal pump or as any other type of centrifugal pump. Based upon the description of the embodiment
20 shown in Fig. 1 and Fig. 2 it is no problem for the skilled person to build a pump according to the invention, that is designed as another type of pump, especially centrifugal pump, for example a single suction pump.

The double suction pump 1 comprises a casing 2 with a first inlet 3, a second inlet 3' and an outlet 4 for the fluid to be pumped. The fluid may be for
25 example crude oil, oil or any other hydrocarbon fluid being highly viscous. The pump 1 has an impeller 5 with a plurality of vanes 51 for conveying the fluid from the first inlet 3 and the second inlet 3' to the outlet 4. The impeller 5 is arranged on a rotatable shaft 6 for rotation around an axial direction A. The axial direction A is defined by the axis of the shaft 6 around which the impeller
30 5 rotates during operation. The shaft 6 is rotated by a drive unit (not shown).

The direction perpendicular to the axial direction A is referred to as the radial direction.

5 The first inlet 3 and the second inlet 3' are arranged oppositely to each other with respect to the axial direction A. Thus, according to the representation in Fig. 1, the fluid is flowing both from the left side and from the right side in axial direction A to the impeller 5, whereas the fluid from the first inlet 3 is flowing in opposite direction to the impeller 5 as the fluid from the second inlet 3'. The impeller 5 conveys both the fluid coming from the first inlet 3 and the fluid coming from the second inlet 3' into the radial direction to the outlet 4 of the
10 pump.

The impeller 5 comprises a front shroud 7 covering the vanes 51 and facing the first inlet 3 of the pump 1. Since in this embodiment the impeller 5 is designed as a double suction impeller 5 it comprises a second front shroud 7' facing the second inlet 3' and covering the vanes 51 on the side of the
15 impeller 5 which faces the second inlet 3'.

The casing 2 is provided with a stationary impeller opening 8 for receiving the front shroud 7 of the impeller 5. The stationary impeller opening 8 is stationary with respect to the casing 2 of the pump 1 and has a circular cross-section with a diameter D, whereas the diameter D designates the smallest diameter
20 of that part of the stationary impeller opening 8 which receives the front shroud 7.

In an analogous manner the casing 2 comprises a second stationary impeller opening 8' for receiving the second front shroud 7' of the impeller 5.

25 In the mounted state the impeller 5 is arranged coaxially within the stationary impeller opening 8 such that the outer circumferential surface of the front shroud 7 faces the inner circumferential surface of the stationary impeller opening 8. Thus, the front shroud 7 and the stationary impeller opening 8 form a gap 9 (see also Fig. 3) between the front shroud 7 and the stationary impeller opening 8. The gap 9 is also called labyrinth. It has an essentially
30 annular shape and provides sealing action as will be explained hereinafter. The gap 9 has a length L which is the extension of the gap 9 in the axial

direction A. The gap 9 extends parallel to the shaft 6 or parallel to the axial direction A, respectively. Thus, the back flow is flowing through the gap 9 parallel to the shaft 6 and in the opposite direction as the fluid flowing through the respective inlet 3. Thus, viewed in the main flow direction of the fluid
5 entering through the respective inlet 3 the starting position of the gap 9, i.e. the opening through which the fluid enters the gap 9, is arranged behind the ending position of the gap 9, i.e. the opening through which the fluid leaves the gap 9.

10 In an analogous manner a second gap 9' is formed between the second front shroud 7' and the second stationary impeller opening 8'. The second gap 9' has a length L' in the axial direction A and the second stationary impeller opening 8' has a diameter D'. The gap 9' extends parallel to the shaft 6 or parallel to the axial direction A, respectively. Preferably, but not necessarily, the length L' equals the length L and the diameter D' equals the diameter D.
15 Since the design and the arrangement of the second gap 9' may be identical as the gap 9 the following description will only refer to the gap 9. It shall be understood that this description applies in an analogously same manner also for the second gap 9'.

20 The gap 9 or the labyrinth 9 seals a side room 10 located on the high pressure side of the impeller 5 against the low pressure side of the impeller 5 which is located at the inlet 3. The side room 10 is located at the high pressure side of the impeller 5 near the outlet 4 of the pump 1 and delimited by the front shroud 7 of the impeller 5 as well as by the casing 2 of the pump
25 the outlet 4 through the side room 10. The back flow passes the gap or the labyrinth 9 flowing essentially in the axial direction A, i.e. parallel to the shaft 6 and reaches the low pressure side of the impeller 5 next to the first inlet 3. It is obvious that the back flow reduces the efficiency of the pump 1.

30 Thus, it is one of the functions of the gap 9 to provide some sealing action to limit the back flow. That is the reason why the gap 9 is also called labyrinth.

It is the basic idea of the present invention to shorten the lengths L (see Fig. 2 and Fig. 3) of the gap 9 in the axial direction A as compared to solutions

known from the prior art. Although one could expect that a shortening of the length L would result in an increased back flow which in turn reduces the pump efficiency, it has been realized that by shortening the length L of the gap 9 the overall efficiency of the pump 1 may be increased.

5 Referring to Fig. 2 and Fig. 3 the design of the gap 9 will now be explained in more detail. In the embodiment according to Fig. 1 the stationary inlet opening 8 comprises a wear ring 11 delimiting the gap 9 with respect to the radial direction. The wear ring 11 faces the outer circumferential surface of the front shroud 7 that is inserted in the stationary inlet opening 8. The wear ring 11 is
10 fixedly mounted to the casing 2, thus, the wear ring 11 is stationary with respect to the casing 2.

Fig. 3 shows a sketch of the front shroud 7 and the wear ring 11 as part of the stationary impeller opening 8 to more clearly understand the dimensions of the gap 9.

15 According to the invention the length L of the gap 9 is designed such that the ratio of the length L and the diameter D of the impeller opening 8 is at most 0.092, i.e. $L/D \leq 0.092$. As already said, the diameter D designates the smallest diameter of the stationary impeller opening 8, i.e. the diameter at that location where the wear ring 11 comes closest to the outer circumferential
20 surface of the front shroud 7. The length L of the gap 9 is the extension in axial direction A of that region where the stationary impeller opening 8 and the front shroud 7 come closest to each other.

In the arrangement shown in Fig. 3 the wear ring 11 is designed with a protrusion 111 in radial direction. Accordingly the length L of the gap 9 is
25 equal to the extension of the protrusion 111 in the axial direction 9.

The second parameter defining the geometry of the gap 9 is the radial clearance R between the front shroud 7 and the stationary impeller opening 8 or the wear ring 11, respectively, along the axial extension of the gap 9. The radial clearance R designates the minimum radial clearance along the gap 9.

In practice it has been proven as advantageous, when the radial clearance R does not exceed 0.0045 times the diameter D of the stationary inlet opening 8, i.e. preferably the condition $R/D \leq 0.0045$ is fulfilled.

The optimal length L of the gap 9 depends on the respective application.

- 5 There are several factors influencing an appropriate choice of the length L of the gap 9, for example the kinematic viscosity of the specific fluid to be pumped, the pressure increase generated by the pump, the flow through the pump or other operational parameters of the pump 1.

- 10 For a given set of operational parameters of the pump 1 the lengths L of the gap 9 should preferably be reduced with increasing viscosity of the fluid to be pumped.

- 15 In practice and depending on the application it may be preferred that the ratio L/D does not exceed 0.073 or more preferred does not exceed 0.055, or even more preferred does not exceed 0.037 or specifically preferred does not exceed 0.019.

According to the preferred embodiments of the pump 1 the minimum ratio L/D is 0.0001, i.e. the length L of the gap 9 is preferably at least 0.0001 times the diameter of the stationary impeller opening 8 or the wear ring 11, respectively.

- 20 Fig. 4 shows in a similar representation as Fig. 3 a variant of the embodiment of the pump 1. According to this variant the impeller 5 and more particular the front shroud 7 of the impeller 5 comprises a wear ring 11' delimiting the gap 9 with respect to the radial direction. The wear ring 11' is fixedly connected to the impeller 5 and rotating with the impeller 5. In this variant the stationary impeller opening 8 may comprise a wear ring 11, too, but may also be
25 designed without a wear ring.

- Fig. 5 illustrates a second variant for the design of the gap 9 between the front shroud 7 and the stationary impeller opening 8. According to the second variant the stationary impeller opening 8 or the wear ring 11, respectively, or as an alternative (not shown) the front shroud 7 is designed such that the gap
30 9 comprises a plurality of lands 12 consecutively arranged with respect to the

axial direction A, wherein two adjacent lands 12 are respectively separated by a groove 13. In such a design the total length L of the gap 9 is the sum of the individual lengths L1, L2, L3, L4, L5 of all lands 12 in the axial direction. The extension of the grooves does not contribute to the total lengths L of the gap 9, i.e. $L=L1+L2+L3+L4+L5$. It shall be understood that the number of lands and grooves as well as their geometric design shown in Fig. 5 has only exemplary character.

The pump 1 according to the invention has a better pump efficiency as compared to pumps known from the state of the art. The pump efficiency designates the ratio of the power delivered by the pump and the power input for the pump, i.e. the power that is used to drive the pump. The power delivered by the pump is usually the hydraulic power generated by the pump 1.

Fig. 6 illustrates a comparison of a pump according to the invention with prior art pumps. The graph shows the pump efficiency P as a function of the viscosity V of the fluid conveyed by the pump. For the purpose of a better understanding the graph is standardized such that the pump efficiency P of the prior art pumps equals the horizontal viscosity axis V, i.e. the pump efficiency P for the pump according to the prior art lies always on the V-axis for each viscosity. Thus, the graph directly shows the increase of the pump efficiency of the pump 1 according to the invention as compared to a prior art pump. The pump efficiency of the pump according to the invention is represented by the curve K. As can be clearly seen, as soon as the viscosity of the fluid is greater than a specific value V1 the pump 1 according to the invention has an increased pump efficiency compared to the prior art pump. The efficiency gain is increasing with the viscosity of the fluid. The specific value V1 of the viscosity where the pump 1 according to the invention becomes more efficient than the prior art pump is usually smaller than the value of $10^{-4} \text{ m}^2/\text{s}$. Thus, for a highly viscous fluid the pump 1 according to the invention has a higher pump efficiency than the prior art pump.

Although specific reference has been made for the purpose of explanation to an embodiment, where the pump 1 is designed as a double suction single stage centrifugal pump the invention is in no way restricted to such

embodiments. The pump according to the invention may also be designed as any other type of centrifugal pump, for example as a single suction pump or as a multistage pump. In particular, the invention is applicable both to centrifugal pumps with a closed impeller, i.e. an impeller having a front shroud and a rear shroud, and to centrifugal pumps with a semi-open impeller, i.e. having a rear shroud but no front shroud. In such designs where the impeller has a rear shroud or a rear shroud only, the design of the gap 9 according to the invention may be used for the rear shroud in an analogously same manner as herein described with reference to the front shroud.

Patent claims

1. A pump for conveying a highly viscous fluid comprising a casing (2) with at least a first inlet (3) and an outlet (4) for the fluid, an impeller (5) for conveying the fluid from the inlet (3) to the outlet (4), wherein the impeller (5) is arranged on a rotatable shaft (6) for rotation around an axial direction (A), and comprises a front shroud (7) facing the first inlet (3) of the pump, wherein the casing (2) is provided with a stationary impeller opening (8) for receiving the front shroud (7) of the impeller (5) and having a diameter (D), wherein the front shroud (7) and the stationary impeller opening (8) form a gap (9) having a length (L) in the axial direction (A), characterized in that the ratio of the length (L) of the gap (9) and the diameter (D) of the impeller opening (8) is at most 0.092.
2. A pump in accordance with claim 1, wherein the ratio of the length (L) of the gap (9) and the diameter (D) of the impeller opening (8) is at most 0.073 and preferably at most 0.055.
3. A pump in accordance with anyone of the preceding claims, wherein the ratio of the length (L) of the gap (9) and the diameter (D) of the impeller opening (8) is at most 0.037 and preferably at most 0.019.
4. A pump in accordance with anyone of the preceding claims, wherein the ratio of the length (L) of the gap (9) and the diameter (D) of the impeller opening (8) is at least 0.0001.
5. A pump in accordance with anyone of the preceding claims having a radial clearance (R) between the front shroud (7) and the impeller opening (8) which is at most 0.0045 times the diameter (D) of the impeller opening (8).
6. A pump in accordance with anyone of the preceding claims, wherein the gap (9) comprises a plurality of lands (12) consecutively

arranged with respect to the axial direction (A) and wherein two adjacent lands (12) are respectively separated by a groove (13).

- 5 7. A pump in accordance with anyone of the preceding claims, wherein the stationary inlet opening (8) comprises a wear ring (11) delimiting the gap (9) with respect to the radial direction, the wear ring (11) being arranged stationary with respect to the casing (2).
- 10 8. A pump in accordance with anyone of the preceding claims, wherein the impeller (5) comprises a wear ring (11') delimiting the gap (9) with respect to the radial direction, the wear ring (11') being arranged stationary with respect to the impeller (5).
- 15 9. A pump in accordance with anyone of the preceding claims being designed as a double suction pump, having a second inlet (3') for the fluid being arranged oppositely to the first inlet (3) of the pump, wherein the impeller (5) is designed as a double suction impeller (5) comprising vanes (51) for conveying the fluid both from the first inlet (3) and from the second inlet (3') to the outlet (4).
- 20 10. A pump in accordance with claim 9, wherein the impeller (5) comprises a second front shroud (7') facing the second inlet (3') of the pump, wherein the casing (2) is provided with a second stationary impeller opening (8) for receiving the second front shroud (7') of the impeller and having a diameter (D'), wherein the second front shroud (7') and the second stationary impeller opening (8') form a second gap (9') having a length (L') in the axial direction (A), and wherein the ratio of the length (L') of the second gap (9') and
25 the diameter (D') of the second impeller opening (8') is at most 0.092.
11. A pump in accordance with claim 9 or 10, wherein the ratio of the length (L') of the second gap (9') and the diameter of the second impeller opening (8') is at most 0.073 and preferably at most 0.055.

12. A pump in accordance with anyone of claims 9-11, wherein the ratio of the length (L') of the second gap (9') and the diameter (D') of the second impeller opening (8') is at most 0.037 and preferably at most 0.019.
- 5 13. A pump in accordance with anyone of claims 9-12 having a radial clearance between the second front shroud (7') and the second impeller opening (8') which is at most 0.0045 times the diameter (D') of the second impeller opening (8).
- 10 14. A pump in accordance with anyone of claims 9-12, wherein the gap (9) and the second gap (9') are designed essentially in an identical manner.
15. A pump in accordance with anyone of the preceding claims being designed for the use in the oil and gas industry.