The Great Mystery of Theoretical Application to Fluid Flow in Rotating Flow Passage of Axial Flow Pump, Part I: Theoretical Analysis

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Abstract: - Theoretical design method was introduced under the condition of minimizing the hydraulic energy losses along the flow fassages. Theoretical results reached are as follows: Fluid flow in three-dimensional flow field between impeller blades of axial flow pump is possible to be considered on a flat plane as the fluid flow caused in two-dimensional flow field. Theoretical design methods due to the application of aerofoil theory or blade-to-blade theory are recommended. And, these theoretical results are compaired with the experimental results and they are confirmed because they consist perfectly with those experimental test results.

Key-Words: - Axial Flow Pump, Establishment of Design Method, Minimization of Hydraulic Energy Losses

1 Introduction

Some of today's theoretical design methods of axial flow pump are established based upon the application of aerofoil theory or the application of blade-to-blade theory. Those design methods are mostly accepted and used as the reasonable and trustful methods. And those support today's practical theoretical design methods in the world [1,2,3,4].

However, their applications are theoretically and experimentally not correct. More than that, they are not only incorrect in physical meanings but also bring the pump efficiency to a very poor one in the practical designing. Therefore, it is very important to re-examine the background of design method whether those applications are really reasonable or not.

Key word to this problem for the examination is "The Great Mystery" in the title. This particular word implicitly suggests that theoretical discussion is made seemingly correct. However, readers may not aware the facts that discussions are made seemingly correct, although they are theoretically and experimentally unreasonable. In addition, it reaches to a seemingly correct result at the end of discussion, which is of course not correct again. Nevertheless many readers might agree to it, because they are supported by experimental examples and they are satisfied very well although those examples are not enough to support. In addition, the method of application is not reasonable. More than that, to tell a truth, unfortunately the acceptance of that theoretical result is extremely convenient and comfortable among users, engineers and scientists because it allows us all to consider the three-dimensional internal flow condition as the problem caused in two-dimensional flow field. This migh be the another reason why that wrong concept is so easily accepted among us. Hence, many of the readers might not even know that the result is physically incorrect and insufficient unless the appointments are made concretely in the separated paper, part II: Inspection [5]. Therefore, it would be reasonable to say that this lecture includes triple mysteries. These are the main reason that the word "The Great Mystery" was brought in the title.

Because of this, at the very beginning of this lecture, the author would like to dare ask all the readers to pay careful attention on theoretical discussion and to make a careful check on each physical expressions whether they are truly and physically correct or not to findout those mysterious wrong parts in the lecture. From these viewpoints and situations, the author would like to say all the readers "Enjoy the careful readings and Good luck for the exciting adventure".

2 Very Basic Concept

As described above, this study deals with the background of today's modern pump design methods, which are the worlds most famous design methods of the axial flow pump and all the theoretical descriptions and discussions are made on very basic and very fundamental subjects. Therefore, in this lecture, part I, to make sure readers consideratrion, it would be suitable to start from the very beginnings.

Now, let us consider a horizontal pipeline filled with water. An axial flow pump is set in it and a valve is also set at a far downstream of impeller discharge to control the flow rate. See Fig. 1.

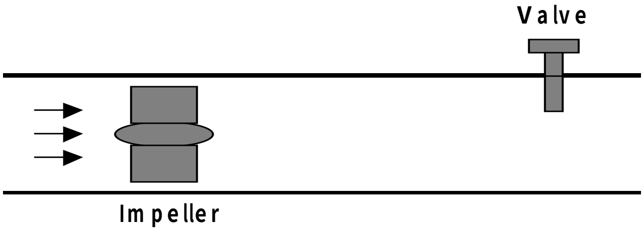

Fig. 1. An axial flow pump set in a horizontal pipeline.

Flow rate is of course due to the valve openings and the flow rate is comparable with the magnitude of absolute velocity at the leading edge of impeller inlet. Then, magnitude of absolute velocity V_1 at the leading edge of impeller inlet is the largest at the maximum flow rate, and zero at the smallest (zero) flow rate.

On the other hand, impeller blade rotates at a constant rotational speed regardless the valve openings. Therefore, peripheral velocity U does not change by the flow rate. These indicate that fluid particles flow direction at the leading edge of impeller inlet against the impeller blades peripheral velocity U, that is, relative flow angle β_1'changes with the change in flow rate. Relative flow angle β_1' becomes large with the increase in flow rate. It is the largest at the maximum flow rate and the smallest (zero) at the smallest (zero) flow rate. See Fig. 2.

However, impeller blade is generally fixed to the hub at a certain blade angle β_1. Therefore, most of the cases, the magnitude of blade angle is maintained at a constant value for the practical change in flow rate.

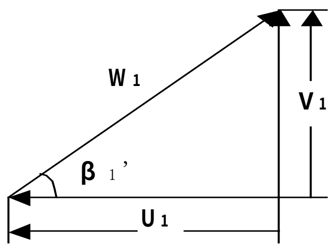

Fig.2 Velocity triangle at impeller inlet.

This indicates that relative flow angel β_1' against the impeller blades peripheral velocity U varies by the valve openings. That is, relative flow angel changes by the flow rate in the range between that at the maximum flow rate and that at the smallest (zero) flow rate. And its magnitude consists with that of blade angle β_1 at a certain flow rate between those of the maximum and the minimum (zero) flow rates. This indicates that flow condition at the leading edge of impeller inlet becomes the optimum at that flow rate and the fluid particle becomes possible to get into the impeller blades rotating flow passage without collision and separation. Therefore, it could be said that hydraulic energy losses Δh becomes the minimum at that flow rate. In other words, the overall pump efficiency η becomes the maximum at that flow rate in the practical operation of axial flow pump. See Fig.3.

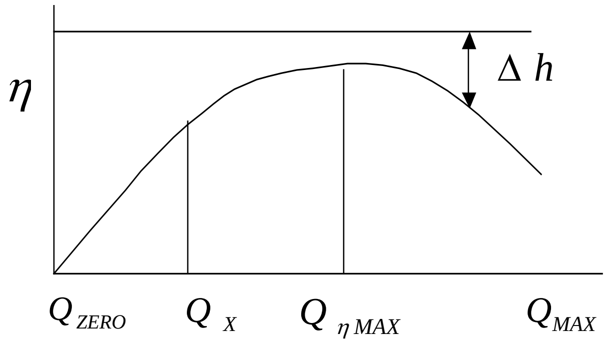

Fig.3 Illustration of overall pump efficiency curve.

Therefore, if axial flow pump is operated at this optimum flow rate, hydraulic energy losses Δh become the minimum, that is, overall pump efficiency becomes the maximum. Then, it could be said that this operating condition must be the most economical design point. Therefore, to cause the hydraulic energy losses the minimum and the overall pump efficiency the highest at the design flow rate, most of the axial flow pump designers may try to design their pumps at that optimum flow rate. In this meaning and viewpoints, it could be said that in the practical design process of the pump it is very important to minimize all the hydraulic energy losses at the design flow rate.

3 Design of Axial Flow Pump

Now let us assume here that you reader is a pump designer and going to design an axial flow pump. Then, you may want to try to minimize all the hydraulic energy losses at each flow passages as much as possible and design it at a high efficiency. Fig. 4 shows illustration of axial flow pump. At the upstream of impeller inlet, at location 0, what kind fluid particles movement and velocity distribution might be expected to minimize the hydraulic energy losses? The answer could be selected from the followings.

A: Fluid particle tends to flow radial outward and velocity distribution shifts radial outward.

B: Fluid particle tends to flow radial inward and velocity distribution shifts radial inward.

C: Fluid particle flows axial direction and velocity distribution is uniform between hub and casing wall.

Most of reader’s selections might be made on C.

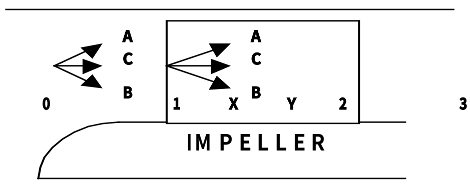

Fig. 4 Illustration of flow patterns in axial flow pump.

Then, how about the fluid particle’s flow condition at location 1 at the leading edge of impeller inlet? Which direction does the fluid particle flow and how the velocity distribution is formed between hub and casing wall in the flow passage? The selection might be made from the followings:

A: Flow direction tends radial outward and velocity distribution shifts radial outward.

B: Flow direction tends radial inward and velocity distribution shifts radial inward.

C: Flow direction does not change at all and remains velocity distribution uniform as it was at upstream.

Among these, in case of A it might indicate the optimun flow condition as follow: The fluid particle, which flowed axial direction and formed uniform velocity distribution at upstream flow passage, may change its flow direction and velocity distribution radial outward at location 1 at the leading edge of impeller inlet due to the effect of rotational motion of the impeller blades. And this tendency may not limit to the location 1 at the leading edge of impeller inlet, but also continue in the whole rotating flow passages through out the locations X, Y, and 2. Then, final velocity distribution at location 2 at the trailing edge of impeller outlet might become as a shifted one radial outward toward the casing wall. At the downstream of impeller outlet, at location 3, rotational part no longer exists in the flow passge. Therefore, its flow direction and velocity distribution may change again from radial outward towards the axial direction so that fluid flow becomes uniform between hub and casing wall along the axial flow passage at the downstream of impeller outlet, and these changes of velocity distribution might make the hydraulic energy losses the minimum in the practical operation of axial flow pump.

In case of B it might indicate the optimum flow condition as follow: The fluid particle, which flowed axial direction and formed velocity distribution uniform along the flow passage at upstream of impeller inlet, may change its flow direction and velocity distribution radial inward at location 1. And this tendency may not limit to the locasion 1 and may continue in the whole rotating flow passages at locations X, Y, and 2. Then, final velocity distribution at location 2 at the trailing edge of impeller outlet may shift radial inward. At the downstream of impeller outlet, at location 3, rotational parts no longer exists. Therefore, it may change its flow direction again from radial inward toward the axial direction and the velocity distribution formed might become uniform at the downstream of impeller outlet, and these changes of velocity distribution might make the hydraulic energy losses the minimum.

In case of C it might indicate the optimum flow condition as follow: As the fluid particle flowed axial direction and the velocity distribution was formed uniform at the upstream of impeller inlet, it may maintain its axial flow direction and uniform velocity distribution, not only at upstream of impeller inlet, but also in the rotating flow passage passing through the locations 1, X, Y, and 2, and this condition might be maintained at the downstream of impeller outlet. Then, it could be said that parallel flow condition might be kept along the whole flow passages; regardless whether the flow passage is rotating or not. And, these may make the hydraulic energy losses the minimum.

In addition, following additional idea might assist the selection of an answer. For example, consider a fluid flow in a horizontal pipeline. See Fig. 5.

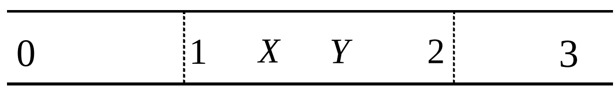

Fig.5 Illustration of fluid flow in a horizontal pipeline.

If no axial flow pump is set in the horizontal pipeline and if uniform velocity distribution is formed at the upstream of impeller inlet at location 0, uniform distribution might be kept at all locations 1, X, Y, 2 and 3 throughout the whole flow passages.

Now, let us image an axial flow pump whose hub diameter is infinitely small and blade thickness is zero. And let us set it in the horizontal pipeline. Leading edge of impeller inlet might be set on location 1 and the trailing edge of impeller outlet on location 2 of the pipeline. If uniform velocity distribution is assumed at upstream of impeller inlet, it may also form uniform velocity distribution throughout whole flow passages, at the leading edge of impeller inlet at location 1, at the middle locations X and Y in the rotating flow passage, at the trailing edge of impeller outlet at location 2, and at the downstream of impeller outlet at location 3. Then, this corresponds exactly the case C.

In this case, if we consider the hydraulic energy change, hydraulic energy of fluid particles at the upstream of leading edge of impeller inlet at location 0 is maintained constant at a value E_0 till the leading edge of impeller inlet. At the leading edge of impeller inlet at location 1, hydraulic energy jump is caused from E_0 to E_1. And it increases continuously at middle flow passage in the rotating flow passage passing through the locations X and Y in order E_X and E_Y, and reaches to the maximum value E_2 at location 2 at the trailing edge of impeller outlet. And the maximum value of hydraulic energy E_2 at the trailing edge of impeller outlet at location 2 is maintained and does not change at all even at location 3 at the downstream of impeller outlet, because at the downstream of impeller outlet there is no rotational part.

That is, even if the change of velocity distribution is not caused at all along the rotating flow passage, hydraulic energy change is possible to be caused along the flow passage as much equivalent to increased hydraulic energy, which corresponds to the energy transferred from mechanical to hydraulic energy by the action of impeller blades to fluid particles in the rotating flow passage of axial flow pump. This hydraulic energy change without the velocity distributions change could be considered as it might cause the hydraulic energy losses the minimum.

From these expressions and considerations, most of readers might agree to select the case C to minimize the hydraulic energy losses.

This indicates that if the hydraulic energy losses are the minimum, the formation of velocity distribution does not need to change at all along the flow passage, and uniform velocity distribution at the upstream of impeller inlet could be remained as it is along the whole flow passages in the rotating flow passage regardless whether the flow passage is rotating or not. Only the axial component of velocity, whose distribution is uniform, changes its magnitude by the flow rate. If the flow rate becomes large, the magnitude of axial component of velocity becomes large. The magnitude of axial component of velocity is the largest at the largest flow rate and the smallest (zero) at the smallest (zero) flow rate. This indicates that radial outward and inward components of velocity, formed in the rotating flow passages, are zero. In other words, to cause the hydraulic energy losses the minimum, radial outward and inward components of velocity have to be assumed zero.

Therefore, to see the flow condition between impeller blades and to consider the interrelation between impeller blades blade angle and fluid particles flow angle due to the change of flow rate; flow passage has to be cut along the casing wall on a circular cylindrical surface and developped on a flat plate. Then, a blade row can be obtained, see Fig. 6. This geometrical treatment is of course reasonable, because radial outward and inward velocity components are assumed zero at the design flow rate in the discussion. This indicates that fluid particles flow condition in three-dimensional flow field is possible to be considered on a flat plate as the fluid flow caused in two-dimensional flow field.

Impeller blade is operated at a constant rotational speed in the practical operation regardless the valve openings, therefore, rotational speed is regardless to flow rate. Its peripheral velocity U is therefore constant and the direction of fluid particles absolute velocity V_1 due to flow rate Q is perpendicular to impeller blades peripheral velocity U. Then, velocity triangle at the leading edge of impeller inlet gives magnitude of fluid particles relative velocity W_1 and its flow direction β_1' against the impeller blades peripheral velocity U. Then, if we consider the relative velocity W_1, instead absolute velocity V_1, we need not worry about the impeller blades rotational speed.

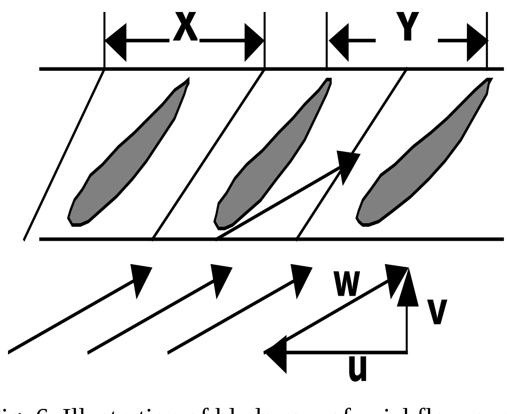

Fig. 6. Illustration of blade row of axial flow pump.

On the other hand, equally spaced impeller blades row appears at equivalent distances L on the flat plate. This indicates that flow condition at interval X or that at interval Y is repeated on the flat plate surface. Therefore, if relative velocity W_1 distributes uniform at upstream of impeller inlet, and if we consider the interval X in Fig. 6, impeller blade can be considered as a single vane located in the uniformly distributed relative velocity W_1, which leads to the application of two-dimensional aerofoil theory. If we consider the interval Y in Fig. 6, the fluid flow can be considered as a fluid flow caused between two impeller blades, which leads us to allow the application of two-dimensional blade-to-blade theory. Anyhow, by considering the methods minimizing the hydraulic energy losses along the whole flow passages of the axial flow pump, it became clear that fluid flow in three-dimensional flow field between impeller blades became possible to be considered on a flat plane as the fluid flow caused in two-dimensional flow field.

This result of consideration might be a wonderful and exciting theoretical result and discovery for all the axial flow pump users, designers, engineers, and investigators. Then, we need to check and examine this result whether it really corresponds with the experimental results or not, and make sure whether this is truly applicable to practical fluid flow caused in the axial flow pump.

4 Statement of Fluid Particles Flow Condition at Design Flow Rate

In the practical operation of axial flow pump, it is very well known that upstream and downstream backflows, which have radial outward and inward components of velocity, appear in the rotating flow passage of axial flow pump at off design condition. Then, why these upstream and downstream backflow phenomena, which have radial outward and inward components of velocity, have appeared in the rotating flow passage of axial flow pump? They are the fluid flow caused in three-dimensional flow field. It doesn't seem very strange? Impeller blade is originally designed so that it does not cause radial outward and inward movements at the design flow rate.

This might be a very good question to examine the theoretical result. The reason for this can be explained as follow: Oh don't worry about it. As it is point out, impeller blade is originally designed so that fluid particle does not cause radial outward and inward movements at the optimum design flow rate. Radial outward and inward velocity components are zero. Therefore, fluid particles flow angle consists with impeller blade's blade angle at the leading edge of impeller inlet at the design flow rate. And the fluid particle flows axial direction along the blade surface. And the fluid particles collision with the blade surfaces and the separation from the blade surfaces are not caused at all at the design flow rate. Therefore, hydraulic energy losses are the minimum and all the fluid particles flow axial direction along the blade surface. Therefore, fluid particles radial outward and inward movements, that is, radial outward and inward velocity components are not caused at all at the design flow rate in the rotating flow passage. This statement and realization on the fluid particles flow condition at the design flow rate is very important before the estimation of theoretical results.

5 Examination of Theoretical Results with Experimental Results at Off Design Condition

Let us estimate the theoretical result at off design condition. If the flow rate decreases from that at the design flow rate, the magnitude of absolute velocity, that is, the magnitude of axial component of velocity V_1 becomes smaller. Then, fluid particles relative flow angle against the impeller blades peripheral velocity U becomes smaller than that of impeller blades blade angle β_1. This indicates that mismatch (disagreement) is caused between the fluid particles relative flow angle β_1' and the impeller blades blade angle β_1 at the leading edge of impeller inlet. Because of this, fluid particles collision with the blades surface and separation from the blade surfaces are caused at the leading edge of impeller inlet.

Impeller blade is originally designed so that fluid particle flows along the blade surface at the design flow rate. Therefore all the fluid particles flow axial direction along the blade surfaces along the designed flow passages. If fluid particles collision with the blade surfaces and separation from the blade surfaces are caused, it effects to internal flow condition in the flow passage. Therefore, if the hydraulic energy losses are caused, overall pump efficiency decreases as much. In other words, if the fluid particle flows perfectly along the blade surface, no centrifugal force acts on it. Impeller blade is designed so that all the fluid particles flow axial direction. Therefore, all the fluid particles flow axial direction. However, if fluid particles collision with the blade surfaces and separation from the blade surfaces are caused in the rotating flow passage, impeller blades rotational motion affects them the centrifugal force. Separated fluid particles therefore have radial outward velocity component and they are forces to move radial outward in the rotating flow passage. With a decrease in flow rate, the fluid particles collision with the blade surfaces and the separation of fluid particles from the blade surfaces become large and this tendency becomes strong. For example, at flow rate Q_X in Fig. 3, hydraulic energy losses are caused larger than that at the maximum efficiency point and the drop of overall pump efficiency becomes larger in scale as much. This indicates that fluid particles collision with the blade surfaces and separation of fluid particles from the blade surfaces have increased and the radial outward component of velocity has increased as much in the rotating flow passage of axial flow pump.

If this tendency becomes strong with a decrease in flow rate, as the velocity distribution shifts radial outward in the rotating flow passage and it reaches to the maximum value at the outer radius near the casing wall and reaches to the minimum value at the inner radius near the hub at the trailing edge of impeller outlet, finally at a certain flow rate backflow from downstream to upstream, so called "downstream backflow" is induced near the hub at the impeller outlet in the rotating flow passage of axial flow pump.

If this tendency becomes stronger with a decrease in flow rate, downstream backflow region becomes large, not only radial outward direction, but also axial direction and it affects to upstream flow region. Then, fluid flow at upstream of impeller inlet starts to change its flow direction and velocity distribution with a decrease in flow rate. It starts to reduce axial component of velocity at outer radius and tends to have radial outward velocity component at inner radius due to the effect of rotational motion of the downstream backflow and tends to flow radial outward from hub side towards the casing wall side at the impeller inlet section, and shifts its velocity distribution radial inward, and finally upstream backflow is caused near the casing wall at the leading edge of impeller inlet.

Therefore, if the flow rate decreases more, these upstream and downstream backflows might increase their rotational regions. Downstream backflow therefore increases its rotational region radial outward toward the casing wall and the upstream backflow increases its rotational region radial inward toward the hub. They also increase their rotational region axial direction in their rotating flow passages. Therefore, upstream and downstream backflow phenomena are absolutely possible to be caused in the rotating flow passage of axial flow pump.

However, they are a kind phenomena caused at off design flow rate because of the fluid particles collision and separation at the rotating flow passage of impeller blades. Therefore, they are a kind result of poor designing and poor production of the impeller blades. Hence, if the impeller blade is designed very well and produced very well, fluid particles may flow axial direction along the blade surfaces and the separation and collision may not be caused at all and the radial outward and inward components of velocity may not appear in the rotating flow passage. Therefore, it is not necessary to worry about the appearance of radial outward and inward velocity components and the possibility of application of aerofoil theory to the design of impeller blades of axial flow pump.

From these viewpoints, it could be concluded that fluid flow, caused in the rotating flow passage of axial flow pump, is possible to be considered equivalent to that caused in two-dimensional flow passage.

6 Conclusions

1. Fluid flow between impeller blades of axial flow pump in three-dimensional flow field is possible to be considered on a flat plane as the fluid flow caused in two-dimensional flow field.
2. Theoretical discussion of internal flow condition between rotating impeller blades due to application of aerofoil theory or blade-to-blade theory is reasonable.
3. Theoretical results are very much consists with experimental test results.

★ However, it is regretful to tell the truth that these results are theoretically and practically not correct. The reasons for this would be shown in part II [5].

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