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

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Abstract: - Very basic background of theoretical design methods of axial flow pump, such as that of worlds most famous design method of axial flow pump and that introduced in part I of this paper under minimizing the hydraulic energy losses along the whole flow fassages, are examined from various viewpoints based upon the considerations due to practical experiences. And the results concluded are that they are not only unreasonable but also unsufficient and useless, both theoretical and experimental view points.

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

1. Introduction

One of the worlds most famous design methods of the axial flow pump is the method established based upon the application of aerofoil theory to the fluid flow in the rotating flow passage of axial flow pump. Which is therefore introduced in the textbooks and used most popularly among engineers as the most trustful design method of the axial flow pump [1,2,3,4]. In this design method, the geometrical shape of flow passage between impeller blasdes, which is basically formed in three-dimensional, is luckily and conveniently or successfully developped on two-dimensional flat plate and aerofoil theory is applied.

This method is quite the same to theoretical result of this investigation, introduced in part I: Theoretical Analysis under the consideration minimizing the hydraulic energy losses along the whole flow passages of the axial flow pump [5]. Background of these concepts is therefore re-checked and re-examined theoretically and experimentally in detail in this paper, part II: Inspection. Those results are reported.

2. Comparison of Velocity Distribution at Design Flow Rate

Let us consider two axial flow pumps, A and B, whose geometrical sizes are equivalent except the location of the maximum solidity point of the impeller blades. The maximum solidity point locates at the distance 1/3 of the blade length from the leading edge of impeller inlet for pump A and 1/2 of the blade length for pump B. Let us assume here that because of these differences of the maximum solidity points of the impeller blades, magnitudes of the maximum overall pump efficiency differ between them in the practical operation of axial flow pump and the maximum overall pump efficiency for pump A is better than that for pump B.

Overall pump efficiency η is determined by

$$\eta = \frac{\rho g Q \cdot H}{E_M} \times 100 \qquad (1)$$

Where η is the overall pump efficiency in percent, ρ is the density of fluids [kg/m^3], g is the gravitational acceleration [m/sec^2], Q is the flow rate [m^3/sec], H is the pump head [m], and E_M is the mechanical energy supplied to the pumping system [kw].

Equation (1) indicates that if the pump head is compaired due to the overall pump efficiency at a certain flow rate, for example, at the design flow rate $Q_{\eta MAX}$, it indicates that the pump head H_A of good efficiency pump A is higher than that (H_B) of poor efficiency pump B.

In the previous discussion in part I: Theoretical Analysis, we assumed uniform velocity distribution at the design flow rate to minimize the hydraulic energy losses. We considered that to produce the overall pump efficiency at a high value we need to minimize the hydraulic energy losses along the flow passage as much as possible. And we had tried to form the velocity distribution uniform along the axial flow passage as much as possible. In the discussion, the distribution of axial component of velocity was assumed constant between the casing wall and the hub along the flow passage in axial direction regardless whether the flow passage is rotating or not and it was believed that such an uniform distribution of axial component of velocity makes the hydraulic energy losses the minimum and causes the overall pump efficiency high. Is this consideration and assumption reasonable? Let us re-check about these.

The overall pump efficiency η_A of good efficiency pump A is, as described above, higher than that (η_B) of poor efficiency pump B at the maximum efficiency design flow rate $Q_{\eta MAX}$, see Fig. 1.

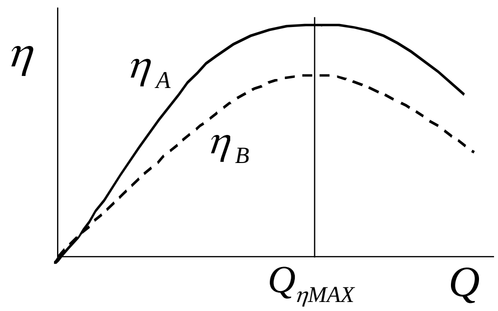

Fig.1 Efficiency curves at design flow rate.

Let's assume here that velocity distribution of good efficiency pump A is formed uniform, like line A in Fig. 2. Then, how do you image the geometrical formation of velocity distribution for poor efficiency pump B? Try to image velocity distribution for poor efficiency pump B and illustrate it in Fig. 2.

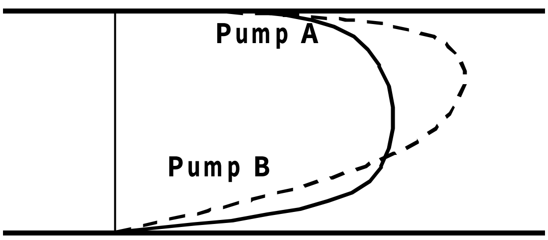

Fig. 2 Illustration of velocity distributions of good and poor efficiency pumps A and B.

① Do you think the formation of velocity distribution for poor efficiency pump B is exactly the same to that of good efficiency pump A at the design flow rate and it becomes uniform, like the line shown in Fig. 2?

If your answer is yes, you might considered as follow: Geometrical formation of velocity distribution does not change or need not change by the pump efficiency. Geometrical formation of velocity distribution is the same, or has to be exactly the same, to that of good efficiency pump A at the design flow rate. Only the magnitude of pump head H_B produced by poor efficiency pump B is lower than that (H_A) of good efficiency pump A. Then, what is the ogiginal physical source of pump head?

In this comparison of two velocity distributions of good and poor efficiency pumps, their flow rates at the maximum efficiency point, that is, the design flow rate is assumed equivalent between them. This indicates that their summation of flow rates in the flow passage between the casing wall and the hub has to be the same regardless the grade of overall pump efficiency. However, the formation of velocity distribution between the casing wall and the hub is not necessary to be the same between those pumps A and B by the grade of overall pump efficiency. Then, why is it possible to say that geometrical future of velocity distribution, has to be the same, or does not need to change by the efficiency at the design flow rate?

In the previous discussion, it was discussed as that if the pump efficiency is good, its velocity distribution becomes uniform between the casing wall and the hub along the axial flow passage. However, if the pump efficiency is poor, its hydraulic energy losses are caused due to the poor designing and the poor production. Therefore, the collision of fluid particles with the blade surfaces and the separation of fluid particles from the blade surfaces are caused in the rotating flow passage of impeller blades, and the centrifugal force of rotating impeller blades acts on those separated fluid particles radial outward in the flow passage. Therefore, fluid particles, separated from the blade surfaces, tends to flow radial outward in the rotating flow passage. And, the velocity distribution in the rotating flow passage tends to shift radial outward. This was the expression against the fluid particles flow conditions change for the decrease in flow rate.

Then, same discussion could be said even at the design flow rate. For example, if the impeller blade is not designed very well at the design flow rate or if the production of impeller blade is not made very well, the fluid particles collision with the blade surfaces and the separation of fluid particles with the blade surfaces might be caused at the rotating flow passage more than those for the good efficiency pump even at the design flow rate. Then, the effect of centrifugal force for those separated fluid particles might be caused stronger for poor efficiency pump B than that for good efficiency pump A. Then, the velocity distribution for poor efficiency pump B might be formed to tend to shift radial outward more than that for the good efficiency pump A. If the velocity distribution of good efficiency pump A is assumed uniform like the line shown in Fig. 2, then it might be formed as it becomes like a dotted line, shifted radial outward, as shown in Fig. 2 for the poor efficiency pump B.

②If you want to say that geometrical formation of velocity distribution of poor efficiency pump B differs from the uniform of good efficiency pump A, then, how do you want to image its formation of velocity distribution? Do you agree to above discussion this time? Are you sure? Because it may has problem.

In general concept, if two fluid particles are rotating at different radiuses in the same rotating flow passage of impeller blade, the fluid particle rotates at outer radius could be understood as that it is affected centrifugal force more than that at the inner radius. Therefore, if the velocity distribution shifts radial outward in the rotating flow passage, it indicates that fluid particles are effected centrifugal force stronger than those that distribute radial inward. In other words, as the flow rate is equivalent between those two axial flow pumps A and B, it seems that if the velocity distribution of poor efficiency pump B shifts radial outward than that of good efficiency pump A, it might be affected centrifugal force stronger than that of good efficiency pump A. Therefore, it seems that the pump head H_B of poor efficiency pump B becoms higher than that (H_A) of good efficiency pump A at that design flow rate. Is this correct? Could not be. If this result is applied to equation (1), it becomes that the overall pump efficiency η_B of poor efficiency pump B is higher than that (η_A) of good efficiency pump A. What is this?

Or you may agree to that if the velocity distribution shifts radial outward, the effect of centrifugal force on the fluid particles becomes stronger. However, you may want to say that it does not mean that the pump head becomes higher. Is this mean that even if the fluid particle flows at the outside radius in the rotating flow passage and the centrifugal force act stronger on the fluid particles, its pressure head is possible to be lower than that which flows inside radius in the same rotating flow passage of impeller blade?

3 Comparison of Velocity Distribution at Off Design Flow Rate

Let us compare the formations of velocity distribution of good and poor efficiency pumps at off design flow rate smaller than the design flow rate. Let us consider two axial flow pumps A and B, again. Their maximum overall pump efficiencies $\eta_{A\eta MAX}$ and $\eta_{B\eta MAX}$ are assumed equivalent this time at the design flow rate. However, the efficiency curve of pump A is assumed flatter than that of pump B at off design flow rate, that is, overall pump efficiency η_{AX} of pump A is higher than that (η_{BX}) of pump B at off design flow rate Q_X. See Fig. 3.

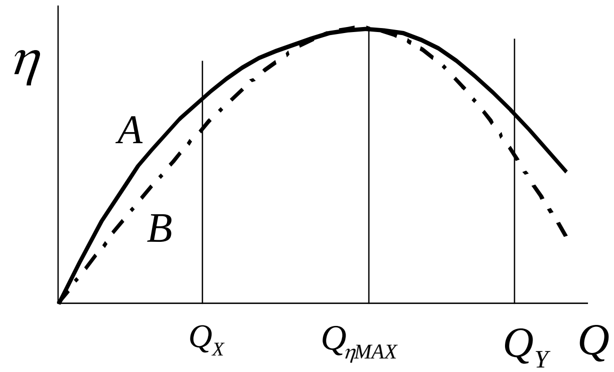

Fig. 3 Overall pump efficiencies of pumps A and B.

Fig. 4 shows illustrations of test facilities for pumps A and B at the design flow rate $Q_{\eta MAX}$. As their maximum efficiencies are equivalent at the design flow rate, their valve openings are equivalent. Now, let us assume here that their geometrical formations of axial component of velocity are equivalent and uniform at the design flow rate. By assuming this uniform velocity distribution at the design flow rate, let us re-check the previous definition that the better the pump efficiencies becomes, the formation of velocity distribution becomes uniform. Or that the uniform distribution of axial velocity in the flow passage makes hydraulic energy losses the minimum and the overall pump efficiency the maximum.

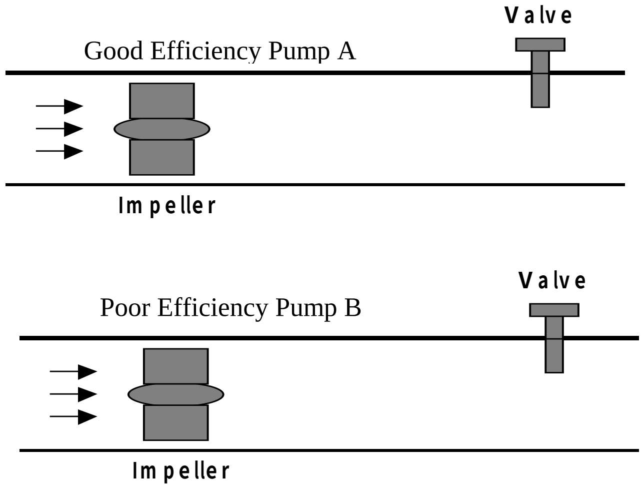

Fig. 4 Illustration of test facilities of pumps A and B
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Under above setting of test facilities, let us estimate their formation of velocity distribution at off design flow rate Q_X smaller than the design flow rate. If we close their discharge valve, their valve openings decrease. Then, at the flow rate Q_X, which discharge valve has been closed more? Or, which valve opening has been set smaller? In other words, to produce the same flow rate Q_X, which discharge valve should we need to close more, pump A or B?

Overall pump efficiency η_{AX} of pump A is higher than that (η_{BX}) of pump B at off design flow rate Q_X. This indicates that pump head H_{AX} of pump A is higher than that (H_{BX}) of pump B at off design flow rate Q_X. This might be clear from the equation (1). This indicates that discharge valve of pump A has to be closed more than that of pump B to produce the same flow rate Q_X because valve openings at the design flow rate were the same between them. That is, the valve opening of pump A has to be smaller than that of pump B. This indicates that pressure head P_{AX} of pump A is higher than that (P_{BX}) of pump B. Then, discharge valve opening has to be closed more than that of pump B to produce the equivalent flow rate Q_X. That is, average axial compponent of velocity V_{AX} of pump A is faster (larger) than that (V_{BX}) of pump B at the discharge valve. In other words, discharge valve for pump A had changed its valve opening more than that for pump B to produce the same flow rate Q_X. Therefore, cross area A_{AX} of discharge valve for pump A is smaller than that (A_{BX}) for pump B at off design flow rate Q_X.

These indicates that if overall pump efficiency at off design flow rate Q_X is very high for pump A and very low for pump B, the change of valve opening from that at design flow rate to that at off design flow rate differ between them. The change of valve opening for pump A is very large and made more than that for pump B.

Then, the change of geometrical formation of velocity distribution from that at the design flow rate $Q_{\eta MAX}$ to that at off desig flow rate Q_X might differ between them. Which velocity distribution had changed its geometrical formation more, A or B? The pump A might changed its geometrical formation of velocity distribution more than that of pump B. Then, which pump had kept its uniform velocity distribution longer for the decrease in flow rate from that at the design flow rate $Q_{\eta MAX}$ to that at off design flow rate Q_X? The pump B had kept its uniform velocity distribution longer for the decrease in flow rate. This result of discussion indicates that the pump B that has poor overall pump efficiency at off design flow rate has kept uniform velocity distribution longer for the decrease in flow rate from the design flow rate $Q_{\eta MAX}$ to the off design flow rate Q_X.

Then, do you think the better the overall pump efficiency becomes, the flatter the velocity distribution becomes? The better the pump efficiency becomes, the velocity distribution becomes uniform? If the velocity distribution becomes uniform, overall pump efficiency becomes high?

There indicates that if you are a good engineer and if you understand that if the velocity distribution becomes uniform, the pump efficiency becomes high, and if you expect to try to design a high quality pump, which produces high efficiency not only at design flow rate but also at off design flow rate, that is, if you want to design an excelent pump, which produces a flatter efficiency curve at whole flow rates at design and off design flow rates, and if you try to make the velocity distribution uniform not only at the design flow rate, but also at off design flow rate, and if you try to do your very best to form the uniform velocity distribution, not only at design flow rate but also at off design flow rate, it indicates that you are doing your very best to produce a poor reduced efficiency pump.

4. The Pure Reason of Occurrence of Downstream Backflow Phenomenon at Off Design Flow Rate

Let us consider pure reason of the occurrence of downstream backflow phenomenon in the rotating flow passage of axial flow pump at off design condition next.

We can see the pure reason of occurrence of downstream backflow in the previous discussion. In the previous discussion it became clear that the better the overall pump efficiency becomes, the valve is closed more than that for poor efficiency pump to produce the same flow rate. This indicates that the better the overall pump efficiency becomes, the velocity distribution shifts radial outward more than that for poor efficiency pump. This indicates that the better the overall pump efficiency becomes, axial component of velocity becomes more larger at outer radius near the casing wall and more smaller at inner radius near the hub than those of poor efficiency pump. This indicates that the better the overall pump efficiency becomes, the occurrence of downstream backflow is caused at a higher flow rate than that of poor efficiency pump at off design flow rate.

In the previous discussion in part I, the occurrence of downstream backflow is discussed as if it is caused due to the poor designing of impeller blades and the poor production of impeller blades. And the pure source of occurrences of downstream backflow is descrived as if it is caused by the fluid particles effected by centrifugal force due to the fluid particles collision with the blade surfaces and the separation of fluid particles from the blade surfaces in bad meanings. However, these are all wrong. It is true that the main source of downstream backflow is due to the fluid particles effected by centrifugal force. However, it is not in bad meaning, but in good meaning. Pump efficiency becomes better indicates that fluid particle tends to flow radial outward due to the effect of centrifugal force and the velocity distribution shifts radial outward. That is, the axial component of velocity becomes larger at outer radius near the casing wall and smaller at inner radius near the hub wall in good meaning. Therefore if this tendency becomes strong with the decrease in flow rate, the downstream backflow starts at inner radius near the hub. If the flow rate decreases this tendency becomes stronger.

From above discussion, it could be understood that the pump head is produced by the centrifugal pump. By moving the velocity distribution radial outward in the rotating flow passage, that is, by shifting the velocity distribution radial outward, fluid particles are effected centrifugal force stronger from the impeller blades, and increase their pressure head and the pump head. Therefore, if the effect of centrifugal force becomes very good at off design floe rate, the better the overall pump efficiency becomes, and the velocity distribution shifts radial outward, and the axial component of velocity becomes larger at the outer radius near the casing wall and the smaller at the inner radius near the hub. That is, the better the overall pump efficiency becomes, the downstream backflow occurs at a larger flow rate and its rotational area becomes larger and stronger in the rotating flow passage of impeller blades of the axial flow pump.

5. The Pure Reason of Occurrence of Upstream Backflow Phenomenon at Off Design Flow Rate

Let us consider the pure source of occurrence of upstream backflow, which is observed at the leading edge of impeller inlet next.

In the previous discussion in part I, upstream backflow is explained as if it is caused by the effect of increased downstream backflow with the decrease in flow rate at the downstream of impeller outlet. If the downstream backflow increases its rotational flow region in the direction radial outward and axial upward directions in the rotating flow passage of impeller blades, the fluid flow at upstream flow passage, especially the fluid flow at the leading edge of impeller inlet is effected its condition by the flow condition at the downstream of impeller outlet. Therefore, it is explained as that the upstream flow condition, especially the flow condition near the leading edge of impeller inlet, is effected by the flow condition of the downstream backflow and the fluid flow at upstream of impeller inlet, not only decrease the axial component of velocity at the outer radius near the casing wall of leading edge of impeller inlet, but also increases its radial outward component of velocity at the inner radius near the hub with the decrease in flow rate. And if this tendency becomes stronger with the decrease in flow rate, the upstream backflow is induced in the outer radius near the leading edge of impeller inlet.

However, this expression is not correct.

True physical source of occurrence of upstream backflow is not due to the existence of downstream backflow in the domain of flow passage at the downstream of impeller outlet. It is due to the separation of fluid particles from the blade surfaces at the leading edge of impeller inlet for the decrease in flow rate. In other words, upstream backflow is caused at the leading edge of impeller inlet due to the poor designing of the impeller blades.

If the impeller blade is designed very poor, the separation of fluid particles from the balde surfaces are caused at the leading edge of impeller inlet. This separation of fluid particles at the leading edge of impeller inlet is especially large at the outer radius near the casing wall. The fluid particles flow condition is strongly effected not only by those of fluid particles collision with the blade surfaces and the separation of fluid particles from the blade surfaces due to the mismatch (disagreement) of flow direction with the direction of blade angles, but also by the effect of boundary layer developed on the surface along the casing wall and the effect of clearance flow caused between the impeller blades tip surface and the casing wall surfaces. Therefore, it is very easy to loose its axial component of velocity at outer radius near the casing wall more than that at inner radius near the hub for the decrease in flow rate. In addition to thses, if the flow rate decreases, the remaining time of fluid particles in the rotating flow passage, that is the rotational motion of fluid particles together with impeller blades increase, which results to cause radial movement on fluid particles due to the rotational motion of the impeller blades. Therefore, the radial outward component of velocity is induced in the rotating flow passage with the decrease in flow rate. All these are the terms to increase the radial outward component of velocity in the rotating flow passage if the blade design is made poor. Therefore, it is possible to say that these are the pure sause of occurrence of upstream backflow.

6. Interrelation between Upstream and Downstream Backflows at Off Design Flow Rate

The relationship between the upstream and the downstream backflows is discussed many times in literatures. Most of the cases, it is discussed as that the upstream backflow is induced by the effect of downstream backflow and the downstream is induced by the upstream backflow. Their expression methods are as follows: If the reason of occurrence of upstream backflow is described first in detail, the reason of occurrence of downstream backflow is explained as that it is induced by the effect of upstream backflow. If the reason of occurrence of downstream backflow is described first in detail, the reason of occurrence of upstream backflow is explained as that it is induced by the effect of upstream backflow. These are wrong.

Some of the impeller blades cause upstream backflow phenomenon first at a large flow rate after a slight decrease in flow rate from that at the design flow rate. The domain of rotational motion of upstream backflow expanded with a decrease in flow rate. However, it did not induce the downstream backflow in the downstream flow passage of impeller discharge. Its occurrence was very late at off design flow rate. The flow rate at which downstream backflow was induced in the rotating flow passage was very small. It was caused after a far decrease in flow rate from that at the design flow rate [6].

Conversely, Some of the impeller blades cause downstream backflow phenomenon first at a large flow rate after a slight decrease in flow rate from that at the design flow rate. The domain of rotational motion of downstream backflow expanded with a decrease in flow rate. However, it did not induce the upstream backflow in the downstream flow passage of impeller discharge. Its occurrence of upstream backflow was very late at off design flow rate. The flow rate at which upstream backflow was induced in the rotating flow passage was very small. It was caused after a far decrease in flow rate from that at the design flow rate [6].

In addition to these, some of the impeller blades induced upstream and downstream backflows at a same flow rate at off design flow rate. These indicate that even if the flow condition at off design flow rate differs by the change in flow rate, it is not possible to say that one backflow phenomenon induces the other backflow phenomenon in the rotating flow passage. For example, above discussions are made for the decrease in flow rate. Then, try to explain these processes of occurrence of upstream and downstream backflow phenomena conversely for the increase in flow rate. Then we can recognize above results.

7. Conclusions

The background of theoretical design methods of axial flow pump, such as the worlds most famous design method of axial flow pump and that of part I of this study developped under minimizing the hydraulic energy losses along the whole flow fassages, are examined from various viewpoints based upon the considerations due to practical experiences. And the results concluded are that they are not only unreasonable but also unsufficient and useless, both theoretical and experimental view points.

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