An Investigation on Occurrence of Backflow Phenomena Caused in Axial Flow Pump, Part I: Downstream Backflow

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Abstract: - Physical source of occurrence of downstream backflow is investigated in this paper. Result indicates that it might be the impeller blades impelling action on the main flow, flowing along the blade surfaces in the rotating flow passage, but might not be the rotating impeller blades action on the separated fluid particles from the blade surfaces caused due to fluid particles collision with blade surfaces and disagreement of fluid particles flow angle with impeller blades blade angle which increases with the decrease in flow rate.

Key-Words: - Axial Flow Pump, Downstream Backflow Phenomenon, and Off Design Condition

1 Introduction

Hydraulic turbomachinery is the machine that transfers energy from mechanical to hydraulic energy and from hydraulic to mechanical energy. The typical of turbomachinery, which transfers energy from mechanical to hydraulic energy, is the pumps. The typical of turbomachinery, which transfers energy from hydraulic to mechanical energy, is the water turbines.

Type number classifies geometrical formation of pumps. If the type number is small, pump is designed as the centrifugal pump. If the type number is middle size, pump is designed as the mixed flow or semi mixed flow pumps. If the type number is large, pump is designed as the axial flow pumps.

These classifications are also possible depend upon the maximum overall pump efficiency of which we can accomplish today at the design flow rate. The maximum overall pump efficiency is high for centrifugal pumps. It is the highest. However, it is lower for mixed flow and semi mixed flow pumps. It is the lowest for axial flow pumps. See Fig.1 [1].

Because of these viewpoints, many researchers, engineers, and pump users have put their remarkable attention on theoretical design methods of the axial flow pumps to improve their overall pump efficiency for many years. The world's most trustful theoretical design method of today on the axial flow pump is the design method due to application of aerofoil theory to the fluid flow between impeller blades.

This theoretical design method is basically established upon the application of lift and drag interrelationship to the fluid flow in the rotating flow passage of axial flow pump. Free vortex motion, that is, uniform distribution of axial component of velocity is assumed between hub and casing wall across the flow passage for the fluid flow in the rotating flow passage of impeller blades. Hence, effect of centrifugal force due to impeller blades rotational motion is not considered at all and neglected from the discussion in the theory.

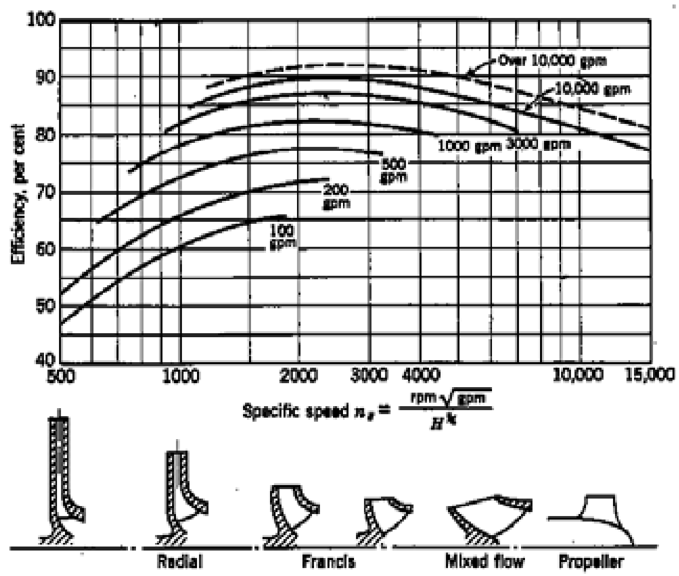

Fig. 1. Type number and overall pump efficiency.

While, in the theoretical design method of centrifugal pump, the effect of centrifugal force is fully introduced and discussed, and succeeded to accomplish practical high overall pump efficiency at the design flow rate. In addition to this, some of the investigators have reported that the background of design method of which aerofoil theory is applied is not reasonable both theoretical and experimental view points [2,3,4,5,6].

From these viewpoints, reasons of occurrence of upstream and downstream backflow phenomena are investigated in this investigation because they are the typical of internal flow condition of axial flow pumps. Therefore, it is expected that the result of those investigations might bring us some significant information on this problem. From these viewpoints, the reasons of occurrence of downstream backflow are merely discussed in this paper.

2. General Concept on the Occurrence of Downstream Backflow Phenomena

In most literatures, the physical source of occurrence of downstream backflow phenomenon at off design flow rate are explained as that: If the impeller blade is excelent and designed very well and produced very well, disagreement between fluid particles flow direction and impeller blades blade angle is not caused at all at the leading edge of impeller inlet. However, if the impeller blade is designed very poor and produced very poor, disagreement between fluid particles flow direction and impeller blades blade angle is caused at the leading edge of impeller inlet. This leads to cause fluid particles collision with blade surfaces and separation of fluid particles from blade surfaces toward the rotating flow passage at the leading edge of impeller inlet. These phenomena are therefore considered, not only as the source to cause poor overall pump efficiency, but also as the source to cause downstream backflow at off design flow rate.

The reason for these is explained as follow: If the collision of fluid particles with the impeller blade is caused and the separation of fluid particles from the blade surfaces is caused, energy transfer from mechanical to hydraulic energy, that is, energy transfer from impeller blade to fluid particles may not be caused very well. And it appears as the hydraulic energy losses on overall pump efficiency curve as the poor grade in performance characteristic curve. This disagreement between fluid particles flow angle and impeller blades blade angle becomes large with the decrease in flow rate because magnitude of fluid particles axial component of velocity, that is, magnitude of fluid particles absolute velocity becomes small with the decrease in flow rate while impeller blades peripheral velocity is remained constant. That is, fluid particles relative flow angle against the impeller blades peripheral velocity becomes small with the decrease in flow rate. Usually, impeller blades blade angle is designed against the flow rate at the design flow rate so that it meets at the flow rate between the maximum and the smallest (zero) flow rates and it is kept constant for the change in flow rate in the practical operation of axial flow pumps.

Therefore, amount of disagreement between fluid particles flow angle and impeller blades blade angle becomes large, not only with the decrease in flow rate from the design flow rate toward the small flow rate, but also with the increase in flow rate from the design flow rate toward the large flow rate. Therefore, if the operating condition moves from the design flow rate, for example, toward the small flow rate at off design flow rate, it is obvious that disagreement of fluid particles flow angle with the impeller blades blade angle increases with the decrease in flow rate. If fluid particles collision with the blade surfaces and separation of fluid particles from the blade surfaces are not caused at all and if all the fluid particles flow along the blade surfaces perfectly, energy transfer from mechanical to hydraulic energy, that is, energy transfer from impeller blade to fluid particle is accomplished perfectly, and fluid particle flows axial direction along the blade surfaces as it is designed, and radial outward movement may not be caused at all in the rotating flow passage of impeller blades.

However, if the fluid particles collision with the blade surfaces and the separation of fluid particles from the blade surfaces to the rotating flow passage between impeller blades are caused, the centrifugal force due to impeller blades rotational motion acts on it radial outward. Fluid particle, which is affected radial outward forces by the rotating impeller blades, tends to flow radial outward in the rotating flow passage. This tendency becomes strong with the decrease in flow rate because magnitude of disagreement of fluid particles flow direction with the impeller blades blade angle becomes large. This is especially true and obvious for the poor efficiency pump. And these are considered as the source to cause the downstream backflow in the rotating flow passage at off design flow rate..

In other words, fluid particles collision with the blade surfaces and the separation of fluid particles from the blade surfaces are caused in the rotating flow passage due to disagreement of fluid particles flow direction with the impeller blades blade angle. Then rotating impeller blade acts centrifugal force on the separated fluid particles radial outward. That is, separated fluid particles are forced to move radial outward in the rotating flow passage by the centrifugal force due to impeller blades rotational motion, and tend to flow radial outward and the distribution of axial component of velocity, which was originally formed uniform, tends to shift radial outward in the rotating flow passage with the decrease in flow rate.

Therefore, the magnitude of axial component of velocity at impeller discharge increases at outer radius near the casing wall and decreases at inner radius near the hub with the decrease in flow rate. In other words, the distribution of axial component of velocity between hub and casing wall across the outlet flow passage shifts radial outward with the decrease in flow rate. If this tendency becomes strong with the decrease in flow rate, backflow phenomenon, which is so called “downstream backflow” or “downstream return flow”, is induced near the hub in the rotating flow passage of axial flow pump. This might be general expression for the fluid particles flow conditions change for the decrease in flow rate from that at the design flow rate.

The thing we need to put our special attention in this theoretical expression is the source of downstream backflows occurrence. It is obviously put on the fluid particles, which are separated from the blade surfaces due to collision with the blade surfaces and the disagreement of fluid particles flow direction with the impeller blades blade angle. In this theory, it is explained as that if the separation of fluid particles from impeller blade is caused in the rotating flow passage, rotating impeller blade may act centrifugal force on them to move radial outward. Therefore, separated fluid particles, which are effected radial outward force by the impeller blades, tend to move radial outward and this tendency becomes strong with the decrease in flow rate. Therefore, magnitude of axial component of velocity becomes large at outer radius near the casing wall and that becomes small at inner radius near the hub with the decrease in flow rate.

The concept that locates behind this theory might be as follows: Impeller blade is designed so that it forms uniform distribution of axial component of velocity across the flow passage. Therefore, if the main flow flows perfectly along the blade surfaces, it may flow axial direction and form the shape of free vortex motion. It is therefore believed that if the fluid particle flows along the blade surfaces, radial outward component of velocity may not be caused at all in the rotating flow passage. In other words, radial outward force, that is, centrifugal force due to impelling action of rotating impeller blade may not be caused at all on the main flow. However, if fluid particle separates from the blade surfaces and if it does not flow along the blade surfaces, the effect of rotating impeller blades, that is, radial outward force due to centrifugal force may act on them radial outward, which may result to cause the downstream backflow.

Are these practically true? Let us check this next.

3. Experimental Verification for the Occurrence of Downstream Backflow

Let us consider two axial flow pumps A and B. Their practical flow rates at the maximum efficiency operating condition $Q_{A\eta MAX}$ and $Q_{B\eta MAX}$ are equivalent and their maximum pump efficiencies $\eta_{A\eta MAX}$ and $\eta_{B\eta MAX}$ are also equivalent at the design flow rate. However, geometrical shape of efficiency curve of pump A is flatter than that of pump B at off design flow rate. That is, 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. 2.

Now, let us compare geometrical formation of velocity distributions at off design flow rate Q_X in the range smaller than the design flow rate.

To simplify the discussion, let us consider that difference of overall pump efficiency at off design flow rate between axial pumps A and B is caused due to the difference of the maximum solidity point of the impeller blades although their geometrical sizes are equivalent. Let us assume here that the maximum solidity point for pump A locates at the distance 1/3 of the blade length from the leading edge of impeller inlet and 1/2 of the blade length for pump B. In more simple expression, pump A has good efficiency and pump B has poor efficiency at off design flow rate.

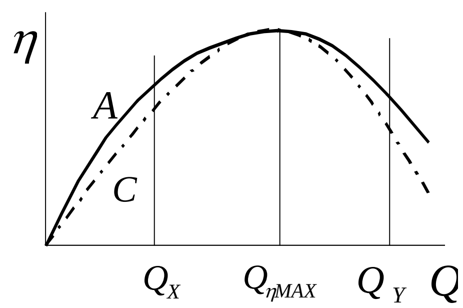

Fig. 2. Overall efficiency curves of pumps A and C.

Fig. 3 shows illustrations of practical test facilities for pumps A and B operated at the design flow rate $Q_{\eta MAX}$. As their maximum efficiencies are assumed equivalent at the design flow rate, their valve openings are set equivalent at that flow rate. Now, let us assume here that their geometrical formations of velocity distribution, that is, distributions of axial component of velocity are also equivalent and uniform between hub and casing wall across the flow passage at the design flow rate. Under these statements of uniform velocity distribution across the flow passage at the design flow rate for both pumps, let us re-check general concept that the better the pump efficiency becomes, the geometrical formation of velocity distribution, that is, the distribution of axial component of velocity becomes uniform between hub and casing wall across the flow passage, and that the uniform distribution of axial component of velocity across the flow passage makes hydraulic energy losses the minimum and the pump efficiency the maximum.

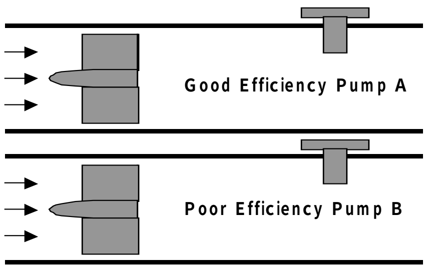

Fig. 3. Illustration of practical test facilities.

If we close those discharge valves valve openings, flow rate decreases. Then, which discharge valves valve opening shall we set smaller between pumps A or B at flow rate Q_X? In other words, to produce the same flow rate Q_X, which discharge valves valve opening shall we need to close more?

From the definition of overall pump efficiency, overall pump efficiency η_{AX} of pump A is higher than that (η_{CX}) 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_{CX}) of pump B at off design flow rate Q_X. This indicates that discharge valves valve openings were equivalent between them at the design flow rate. Therefore, discharge valves valve opening of pump A has to be closed more than that of pump B to produce the same flow rate Q_X. That is, discharge valvces valve opening of pump A has to be smaller than that of pump B.

In other words, head H_A of pump A is higher than that (H_B) of pump B. That is, pressure head P_{AX} of pump A is higher than that (P_{BX}) of pump B. Then, to produce the equivalent flow rate Q_X at the location of discharge valve, discharge valves valve opening has to be set smaller than that of pump B. Average magnitude of axial compponent of velocity V_{AX} of pump A has to be larger than that (V_{BX}) of pump B at the location of discharge valve. In other words, to produce the same flow rate Q_X at the location of discharge valve, the discharge valves valve opening for pump A has to be changed more than that for pump B. Therefore, the cross area A_{AX} of discharge valve for pump A is smaller than that (A_{BX}) of pump B at off design flow rate Q_X.

These indicates that if overall pump efficiency for pump A is higher than that of pump B at off desig flow rate Q_X, discharge valves valve openings from that at the design flow rate $Q_{\eta MAX}$ to that at off design flow rate Q_X have to differ between them and the change of discharge valves valve opening for pump A has to be made larger than that for pump B.

Then if the operating conditions change is made for those pumps from that at the design flow rate $Q_{\eta MAX}$ to that at off desig flow rate Q_X, their geometrical formations of velocity distribution, that is, distributions of axial component of velocity between hub and casing wall across the flow passage might differ between them. Then which geometrical formation of velocity distribution had changed more? Pump A might change its geometrical formation more than that of pump B. Then, which pump had kept its uniform velocity distribution longer for equivalent 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 might kept its uniform velocity distribution longer for the unit decrease in flow rate.

This result of discussion indicates that the pump B that has poor overall pump efficiency at off design flow rate had kept uniform velocity distribution longer for the decrease in flow rate. That is, the distribution of axial component of velocity between hub and casing wall was kept longer for pump B for the same decrease in flow rate. In other words, the velocity distribution of poor efficiency pump B was more uniform than that of good efficiency pump A at off design flow rate Q_X.

Then, is it still possible to say that the better the pump efficiency becomes, the flatter the velocity distribution becomes? Or that the better the pump efficiency becomes, the velocity distribution becomes uniform? Then, is it possible to say that if the velocity distribution becomes uniform, pump efficiency becomes high?

4. Practical Experimental Data on the Occurrence of Downstream Backflow

Experimental performance characteristic curves for five axial flow pumps tested and the flow rates at which upstream and downstream backflows occurred are shown in Fig. 3 [5].

It indicates that geometrical formation of overall efficiency curve for DP-1 pump is the flattest of the five and that of DP-2 pump comes next. DP-3 pump has a poor overall efficiency curve and DP-4 pump has a reduced one on the efficiency curve and that of DP-0 pump was a reduced poor efficiency curve at off design flow rate. The ratios of flow rate at which downstream backflow was observed to that at the design flow rate for DP-0, DP-1, DP-2, DP-3, and DP-4 pumps were 0.341, 0.653, 0.655, 0.667, and 0.698, and those at which upstream backflow was observed were 0.790, 0.573, 0.603, 0.672, and 0.762, respectively. The difference of flow ratios between the flow ratios at which downstream backflow occurred and that at which upstream backflow was observed are -0.449, 0.080, 0.052, -0.005, -0.064 for DP-0, DP-1, DP-2, DP-3, and DP-4 pumps, respectively.

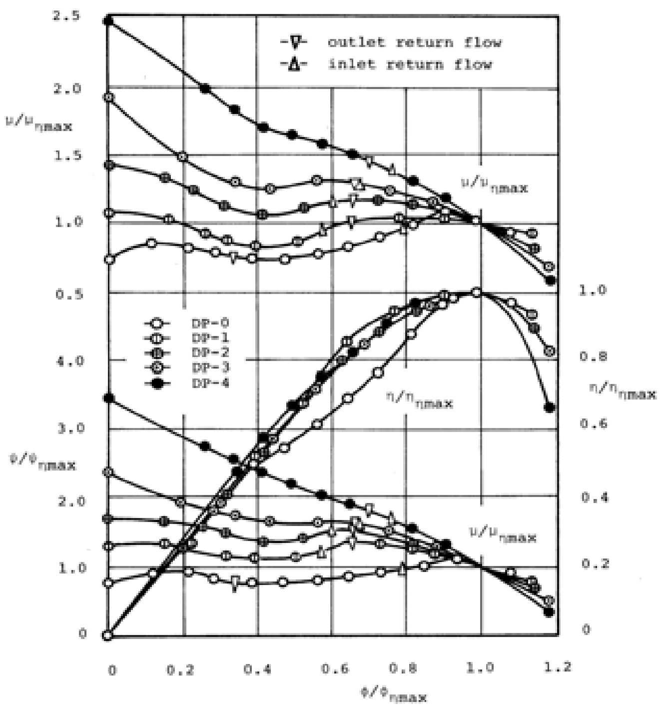

Fig. 4 . General performance characteristic curves of five semi-mixed flow type axial flow pumps tested and flow ratios at which upstream and downstream backflows started by T. Tanaka [5].

This experimental data indicate that downstream backflow occurred first at a large flow rate near the hub in the impeller discharge before the upstream backflow appeared in the impeller inlet for DP-1 and DP-2 pumps. DP-1 pump had occurred downstream backflow at a fairly large flow ratio before the occurrence of upstream backflow. It had the smallest flow rate at which the upstream backflow appeared of all the five. The occurrence of downstream backflow for DP-2 pump was the largest of the five. However, the occurrence of upstream back flow was larger than that of DP-1 pump. On the other hand, downstream backflow was occurred after the occurrence of upstream backflow at a small flow ratio for DP-0, DP-4, and DP-3 pumps. Downstream backflow occurred just after the upstream backflow aoccurred and at approximately the same flow rate for DP-3 pump. The downstream backflow for DP-4 pump appeared sometime after the upstream backflow occurred. On the DP-0 pump, upstream backflow occurred first at a fairly large flow rate and the downstream backflow occurred with a further decrease in flow rate. The flow rate at which upstream backflow occurred was the largest and that of downstream backflow occurred was the smallest of the five.

These indicates that occurrence of downstream backflow has strongly interrelated with the grade of pump efficiency. If the occurrence of downstream backflow is caused at a fairly large flow ratio compared with that of occurrence of upstream backflow, geometrical formation of overall pump efficiency curve becomes a flatter one. Overall efficiency curve of DP-1 pump is the example of this. Geometrical formation of overall pump efficiency curve becomes a reduced one with the occurrence of upstream backflow becomes first. Overall efficiency curve of DP-2 pump is the example of this. Occurrence of downstream backflow was the firstest of the five. It was occurred at flow ratio 0.655. However, upstream backflow was caused at a large flow ratio 0.603 soon after the downstream backflow had started. The difference of flow ratios between the flow ratios at which downstream backflow occurred and that at which upstream backflow was observed are 0.080 for DP-1 pump and 0.052 for DP-2 pump. On the other hand, if the upstream backflow starts first at a large flow rate and the occurrence of downstream backflow becomes very late at a small flow rate, geometrical formation of efficiency curve becomes a reduced one as it is seen in DP-3, DP-4, and DP-0 pumps.

These result of observations obviously indicate that the occurrence of downstream backflow is not due to the disagreement of fluid particles flow direction with the impeller blades blade angle. That is, the occurrence of downstream backflow is not caused due to the fluid particles collision with the blade surfaces and the separation of fluid particles from the blade surfaces into the rotating flow passage between blades. In other words, the occurrence of downstream backflow is not caused due to the action of centrifugal force on fluid particles separated from the blade surfaces due to poor designing and poor production of the impeller blades.

These indicates that downstream backflow is caused by the centrifugal force due to impeller blades regular impelling action on fluid particles in the rotating flow passage of impeller blades.

5. Pure Reasoning of Occurrence of Downstream Backflow at Off Design Flow Rate

Let us consider a very good efficiency pump, whose overall pump efficiency is very good at design and off design flow rate. This indicates that it has a very flat overall pump efficiency curve at the design and off design flow rates. In other words, it has high overall pump efficiency not only at the design flow rate, but also at off design flow rate.

From previous discussion, it is clear that discharge valves valve opening for good efficiency pump is smaller than that of poor efficiency pump at off design flow rate. Velocity distribution for good efficiency pump shifts radial outward more than that for poor efficiency pumps at outer radius near the casing wall. That is, if the pump efficiency is better, the axial component of velocity becomes larger at outer radius near the casing wall but the axial component of velocity becomes smaller at inner radius near the hub than those of poor efficiency pump at an equivalent off design flow rate. That is, if the pump efficiency is better, the amount of fluid particles, which flows outer radius near the casing wall is larger and that, which flows inner radius near the hub is smaller. By shifting the amount of fluid particles flow rate radial outward near the casing wall, flowing fluids are obtaining impeller blades stronger effect on centrifugal forces and high pressure head in the rotating flow passage. Therefore, in the rotating flow passage of better efficiency pump; radial outward forces act on fluid particle stronger than that for poor efficiency pump. This indicates that the occurrence of downstream backflow is caused in the flow passage near the hub at a larger flow rate more than that of poor efficiency pumps at off design flow condition. Therefore, if the pump efficiency is better, increasing rate of axial component of velocity is larger at outer radius near the casing wall and smaller at inner radius near the hub compared with those of poor efficiency pumps for the unit decrease in flow rate. Therefore, it could be said that not only expanding speed of rotational region of downstream backflow becomes faster than that of poor efficiency pump in the rotating flow passage, but also its strength becomes stronger than that of poor efficiency pump for the unit decrease in flow rate. That is, the better the overall pump efficiency becomes, not only the flow rate at which downstream backflow occurred becomes larger at high flow rate, but also the expanding speed of rotational region of downstream backflow becomes larger and its strength becomes stronger in the rotating flow passage. That is, the better the overall pump efficiency becomes, the downstream backflows rotational region becomes larger.

In this discussion geometrical shape of overall pump efficiency curve is assumed very flat. This indicates that impeller blade is designed so that it keeps high efficiency even at a very small off design flow rate. That is, the separation of fluid particles from the blade surfaces is not caused easily. In other words, fluid particle does not separate easily from the blade surfaces. Hence impeller blade can keep its strong impelling action and can push the fluid particle behind the impeller blade at high efficiency even at the very small flow rate compared with those of poor efficiency pump.

From these discussions, it would be clear that the grade of overall pump efficiency is strongly interrelated with the flow rate of occurrence of downstream backflow, its expanding speed for the decrease in flow rate, and the strength of rotational motion. If the pump efficiency is better than the others, it indicates that its downstream backflow might be caused at a high flow rate compared with those of poor efficiency pumps. And the expanding speed for the unit decreases in flow rate and the strength of rotational motion of downstream backflow are larger than those of poor efficiency pumps.

6. Conclusion

Physical source of occurrence of downstream backflow might be the impeller blades impelling action on the main flow, but might not be the action on the separated fluids from the blade surfaces due to fluid particles collision and disagreement of fluid particles flow angle with the impeller blades blade angle which increases with the decrease in flow rate.

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