Side-Intake marine impulse thruster/PD pump of ultra-high capacity and its analysis from fundamentals

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Synopsis

The paper introduces a novel pump that works in a similar hydrodynamic principle as an axial piston pump, but different in that the new pump employs a unique concept of Side-Intake of water (fluid) which allows it to achieve a capacity coefficient much greater than existing pumps. Based on its working principle and applications for either marine propulsors or pumps, this new system is referred to as Side-Intake Marine Impulsive Thruster/Positive Displacement Pump of Ultra-High Capacity (SI-MIT/PDPUHC). The paper presents in detail the concept of the Side-Intake of water in SI-MIT/PDPUHC and a compact design of SI-MIT/PDPUHC for its practical utilization. An evaluation of SI-MIT/PDPUHC from efficiency, linearity, and effectiveness perspectives is provided. The approach for the evaluation is based on the analysis from the fundamental principles in fluid mechanics, and supported by marine animals, laboratory findings and existing industry systems.

Keywords: Pump; PD pump; Axial piston pump; Marine propulsor

1. Introduction

For marine vehicle propulsion, propellers or impeller-driven water jets are mostly being used. Simply put, to generate thrust to propel a marine vehicle requires to kick or pump water in the direction opposite to the vehicle's moving direction. Marine propellers or marine impeller-driven water jets all rely on spin of blades in water to pump large amount of water in the direction opposite to the vehicle's move for thrust. However, the spin of blades causes swirl of water, which doesn't contribute to thrust and is an energy waste. This principle-embedded energy waste due to the spin of blades leads to the fact that such propulsors can hardly reach the ideal efficiency of propulsion regardless of how much design optimization efforts are spent. The highly swirl velocity not only brings down the efficiency of a propulsor but also is a source of blade surface cavitation and the helical vortex system in the propulsor wake that generates water noise. Quoted from Terwisga (2013), Figure 1.1 provides an overview of various kinds of energy loss in a propeller. A rough estimate indicates that a complete elimination of rotational energy loss due to swirl in propeller can increase a propeller's efficiency up to an average of 20%.

Tremendous investment and research efforts were continuously spent on various kinds of Energy Saving Devices (ESD) since the use of propellers or impeller-driven water jets for marine propulsion. Examples are contra rotating propellers, static stators upstream or downstream, Mewis duct, Grim's vane wheel, just to name a few. All of them were designed with a main focus on recovering the swirl loss in propeller, but none of them is able to completely eliminate the swirl loss, and their performances are very non-linear, i.e., largely affected by the flow conditions in which the propeller and its powered vehicle operate.

Pumps can be conveniently classified into two categories, namely positive displacement (PD) pumps and dynamic (DN) pumps. Examples of PD pumps are axial piston pumps, reciprocating pumps and gear pumps. The hydrodynamic principle of a PD pump is straight forward. It uses a moving boundary to directly displace a fixed amount of fluid in each cycle. Such a principle is endowed with linear performance characteristics, specifically the fluid energy (including kinetic and pressure) displaced is proportional to the energy spent by the moving boundary, resulting in a hydrodynamic efficiency of a PD pump being more or less a constant for a wide range of load conditions. DN pumps rely on spin of blades in water to raise water kinetic energy that converts to pressure head. Examples are propellers or axial-flow pumps and impeller-driven water jets used in marine propulsion, and also include centrifugal and mixed-flow pumps. The hydrodynamics of pumping fluid through spin of blades attributes to the principle of a lifting foil. Such pumps require an optimal pitch angle distribution along a blade's radius in order to work at optimal angle of attack and maintain high hydrodynamic efficiency. For a given design of DN pump, it can hardly operate in optimal pitch angle distribution at all load conditions,

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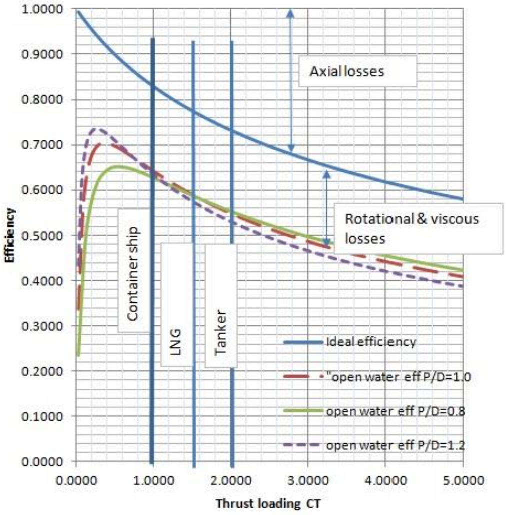

Figure 1.1.: An overview of various kinds of energy loss for B-series propellers from Terwisga (2013).

e.g., changes of low to high flow rates and pressure heads. In other words, the hydrodynamics of DN pump is extremely non-linear. The relation from input power to fluid output power can only be established through the solution to a highly nonlinear and delicate fluid dynamics problem. That explains a DN pump including propellers and impeller-driven water jets can only reach its highest efficiency at the design condition (point). As a marine propulsor, its propulsion efficiency degrades greatly when the powered vehicle operates at off-design conditions. Reflected in real life, a marine vehicle equipped with propulsors of DN pumps reacts prominently weak and sluggish during acceleration and maneuvering. DN pumps are not only very non-linear performers, but their energy efficiency is also lower than that of PD pumps in general.

While PD pumps are linear performers and their efficiency is higher than that of DN pumps, marine propulsors choose to use DN pumps (propellers and impeller-driven water jets), and DN pumps are also predominantly used in most industries for fluid pumping purposes. A major limitation to the applications of existing PD pumps is either a small capacity coefficient, $C_Q = Q / nD^3$ or a low pump speed limited by the small inlet valve opening that introduces throttling effect. The C_Q parameter, in which Q is the rate of discharged fluid volume, D is the system diameter, and n is the pump speed, measures the capability of a pump on how large an amount of fluid relative to its system size can be discharged in each revolution or cycle. A small capacity coefficient and/or a low limiting running speed drastically limit PD pumps' applications to small flow rate areas, e.g., hydraulic pressure systems. Comparatively, DN pumps possess higher limiting running speeds and/or relatively large capacity coefficients, and therefore are dominant in the pump market, because most pump applications demand great fluid capacity in addition to a pressure head rise. These qualities also make DN pumps be predominantly applied for marine propulsors.

In the digital era, there is a race in industries to make their products smart, but the pump industry falls far behind in the race. There are noticeable efforts in developing smart or intelligent pumps that tried to incorporate computerized digital control to optimize the operation of pumps. The main idea is to use smart motors as variable speed drives (VFDs) to adjust the pump speed and automatically ensure the pump operate in a safe and efficient state at all time. DN pumps dominates the pump market due to their quality of larger capacity than that of PD pumps but are very non-linear performers in hydrodynamics. When smart technology works with a DN pump, the pump may smartly meet the system load requirement through VFDs, but at the same time its operating point shifts away from its BEP (Best Efficient Point) and results in a reduction of energy efficiency. In other words, the highly non-linear characteristics in hydrodynamics of DN pumps makes it ineffective and difficult to work with modern smart motors and actuators for smart pumps and marine propulsors.

This paper introduces a novel PD pump that works in a similar hydrodynamic principle to an axial piston pump, but different in that the new pump employs a unique concept of Side-Intake of water (fluid) which allows it to achieve a capacity coefficient much greater than existing pumps and expects its inlet suction behaviour be comparable to DN pumps. Based on its working principle and applications for either marine propulsors or pumps, this new system is referred to as Side-Intake Marine Impulsive Thruster/Positive Displacement Pump of Ultra-High Capacity (SI-MIT/PDPUHC). A SI-MIT/PDPUHC is able to combine the superior characteristics of current PD and DN pumps and overcome their drawbacks. Namely, it not only preserves the superior characteristics of an

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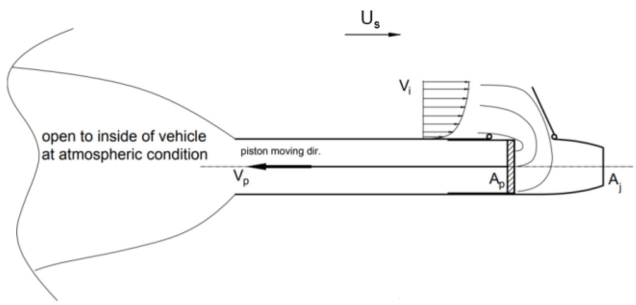
axial piston pump's linear performance and high efficiency, but also possesses a new characteristic of a much higher capacity coefficient than existing pumps and marine propellers. The linear performance in hydrodynamics and a reliance on reciprocating movement of piston-cylinder setups make the new pump a natural match to integrate modern smart linear motors for smart pumps and marine propulsors. When used as marine propulsors, it will completely eliminate the swirl loss that exists in propellers and impeller-based water jets.

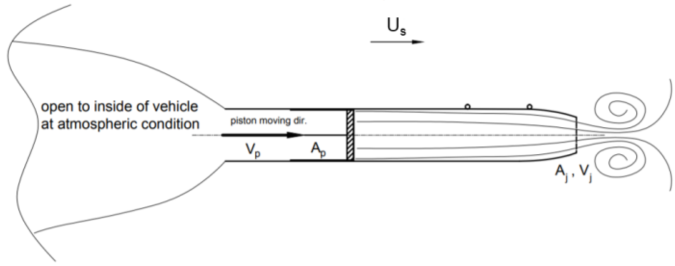
The concept of the Side-Intake applied in a piston pump for marine propulsor was first introduced in Huan (2015). A main contribution of this paper is to introduce a completely novel design of a 6-cylinder piston pump incorporated with the Side-Intake concept. The achievement is largely attributed to the design of a novel inner-ring rotary valve and thus makes a SI-MIT/PDPUHC become more appealing for practical utilization. An evaluation of SI-MIT/PDPUHC from efficiency, linearity and effectiveness perspectives is provided. The approach for the evaluation is based on the analysis from the fundamental principles in fluid mechanics, and supported by marine animals, laboratory findings and existing industry systems. Although most information in Section 1, 2 and 4 can also be found in Huan (2015), for the integrity of this paper they are preserved as necessary and modified appropriately in line with the discussion of the new system.

2. Principle of Side-Intake in SI-MIT/PDPUHC

A SI-MIT/PDPUHC relies on pistons' reciprocating motion in piston-cylinder sets to intake and discharge water (fluid). A unique feature of SI-MIT/PDPUHC is that it adopts the Side-Intake concept to achieve fluid intake from the side (lateral) in reference to fluid discharge in the axial direction of the cylinder or in line with the pistons' movement. The concept of Side-Intake of water using piston-cylinder setups is shown in a schematic diagram in Figure 2.1. Water intake openings are made on the circumferential wall of a cylinder near its discharge end. An open-close valve is required to open and close these Side-Intake openings during the piston's strokes for intake and discharge respectively. The discharge end is completed with a jet nozzle for marine propulsor. With a piston installed inside the cylinder, the system becomes a one-cylinder SI-MIT/PDPUHC. Figure 2.1(a) shows a water intake process. In this process, the valve is in fully open position and the piston takes the recovering stroke, namely the piston moves toward left as shown in the picture. Figure 2.1(b) shows the process of water discharge through the jet nozzle for thrust generation. In this process, the valve is in fully closed position and the piston takes the discharge stroke, namely the piston moves toward the right as shown in the picture. With the piston's reciprocating motion together with the valve open and close of the Side-Intake openings, the system achieves water pumping or thrust generation through fluid flow from a side-inlet to an axial-outlet that is in line with the cylinder body or the piston's movement.



(a) Intake



(b) Discharge

Figure 2.1.: A schematic diagram for the process of Side-Intake.

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Following the description from above, a SI-MIT/PDPUHC with one cylinder will have no water intake during discharge and also no water discharge during water intake. To keep a continuous water intake and discharge, a SI-MIT/PDPUHC shall take at least two pair of cylinders for a practical design, which is shown with a schematic diagram in Figure 2.2.

The purpose of adopting the Side-Intake concept is trifold. Firstly, the Side-Intake makes a SI-MIT/PDPUHC possible to best use the system's volume for fluid so that the pump's capacity coefficient can be maximized. The Side-Intake at the same time offers the feasibility to make the opening of the inlet valve for a SI-MIT/PDPUHC large enough so that the flow rate will not be restricted by the valve throttling area in a common PD pump. With the flexibility for a large inlet opening to lift the limiting running speed of a PD pump, it is expected that the inlet suction behaviour of a SI-MIT/PDPUHC can be made comparable to a conventional DN pump or marine propulsor. Secondly, it allows each piston to separate its fluid chamber into a dry and a wet compartment anytime as the piston moves. It is expected that when the piston heads during the recovering cycles confront air at atmospheric condition or ambient pressure instead of fluid the energy cost will be small. This feature makes a SI-MIT/PDPUHC similar to an axial piston pump. Specifically, the similarity is that both systems accomplish fluid intake during the recovering cycles of the pistons' movement, in which the pistons confront air instead of fluid because one side of the piston heads is in contact with air. Therefore, a SI-MIT/PDPUHC is basically an axial piston pump but with a capacity coefficient that can be made much greater than existing pumps. Lastly, the Side-Intake concept makes an axial piston pump capable for marine propulsors and eliminates completely the swirl loss in propellers or impeller-driven water jets, because the piston motion in a SI-MIT/PDPUHC only generates flows in line with the thrust.

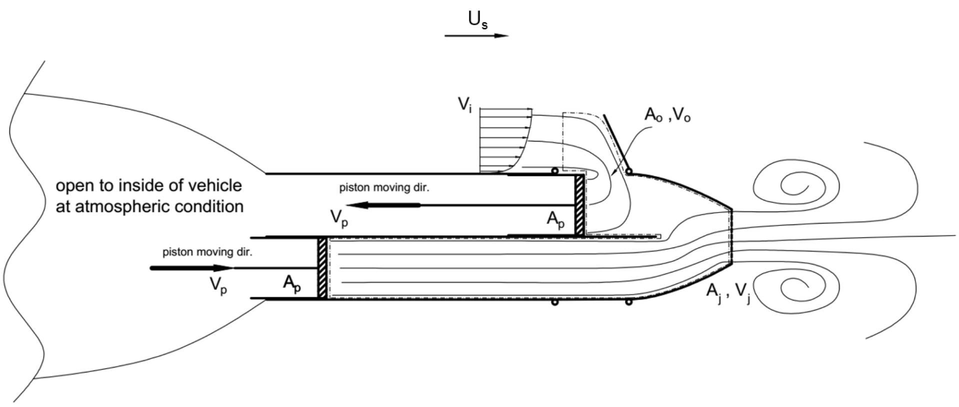

Figure 2.2.: A schematic diagram of a SI-MIT/PDPUHC for a continuous flow.

3. A compact SI-MIT/PDPUHC configuration

A design of SI-MIT/PDPUHC for practical utilization that works in the Side-Intake principle is presented in an exploded view in Figure 3.1 with its major components. Figure 3.2(a) and (b) show the assembled configuration when applied as a pump or a marine propulsor respectively.

The system has two piston sets that is shown in Figure3.1, namely, piston set-1 and piston set-2. Each piston set has three pistons with the cup heads of fan-shape. A radial-baffle is used to divide the inside volume of the cylinder body into six water chambers in fan-shape columns that match the piston heads. The two piston sets are alternatively settled in these water chambers as shown in Figure4.1. Each piston has its rod connected to the sliding mover of a linear motor. For the prime mover, one example is to use six independent linear motors to power the two sets of pistons and achieve the reciprocating motions of the pistons. More sophisticated design for one compact linear motor that possesses six movers specifically for a SI-MIT/DPDHUC system is desirable. The technology for design and manufacture of linear motors with great power density as prime movers was well established, for instance, Sato et al. (2013). The two sets of pistons together with their power system (the linear motors and their controls) constitute two sets of pumping mechanisms. During operation, the reciprocating movement of these two sets of pumping mechanisms work in 180^{o} or a half cycle phase difference, resulting in one set of the pumping mechanism discharges water while the other intakes water. Consequently, the system accomplishes a continuous water pumping as depicted in Figure 2.2. As shown in Figure 3.1, the radial-baffle is extended out of the cylinder body at one end of the cylinder. The extended part leaves the water chambers open both in the side (lateral) and the axial directions. When the inner-ring rotary valve is ridded seamlessly on the extended radial baffles, the water chambers open the side intake openings at the same time closing the axial discharge openings or vice versa depending on a rotary actuation for the inner-ring rotary valve.

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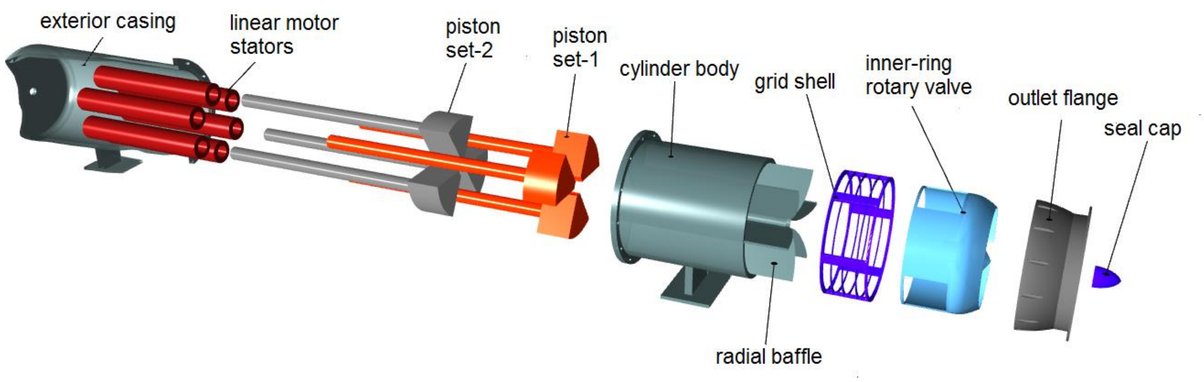
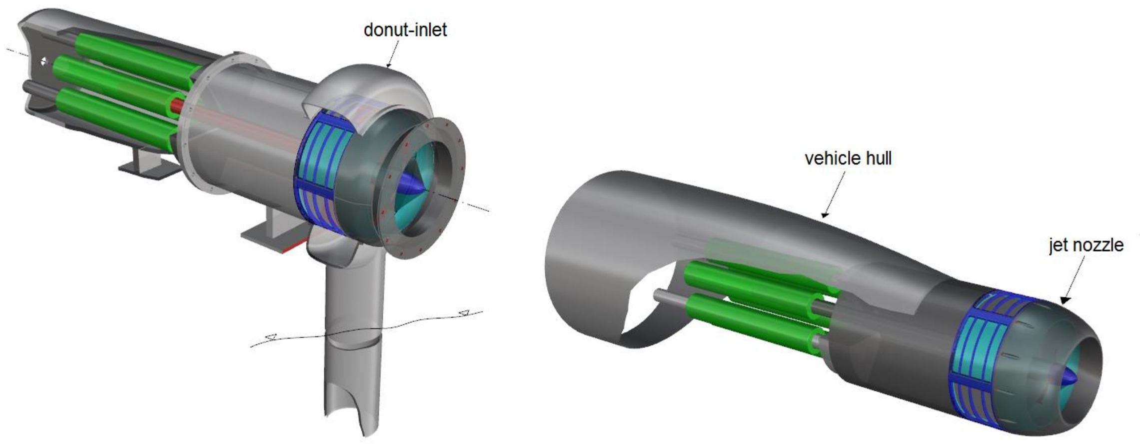


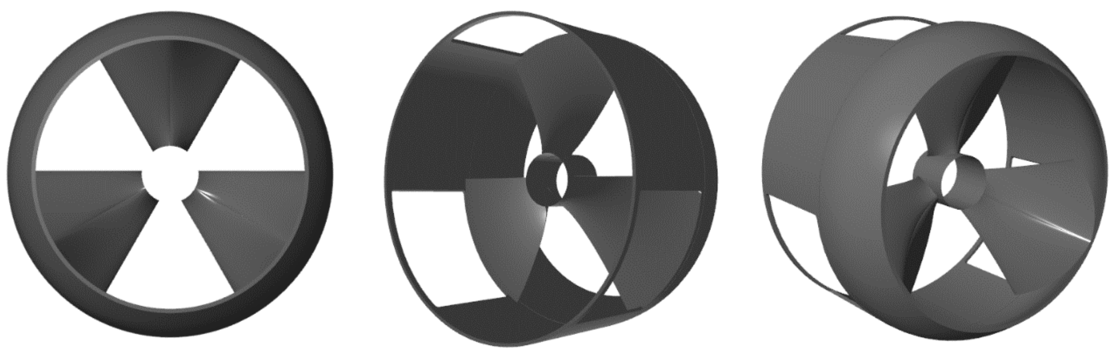
Figure 3.1.: An exploded view of a SI-MIT/PDPUHC.



(a) Applied as a pump

(b) Applied as a marine propulsor

Figure 3.2.: An assembled system of a SI-MIT/PDPUHC.



(a) Front

(b) Rear perspective

(c) Front perspective

Figure 3.3.: Views of the inner-ring rotary valve.

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The inner-ring rotary valve is a central piece of the system which makes the system achieve the Side-Intake concept. The detail structure of the valve is shown in Figure 3.3. The inner-ring rotary valve has three side openings equally distributed along the circumference of the cylindrical wall. Their sizes match the side openings of the water chambers left by the extended part of the radial baffle. The so-called Side-Intake of water uses these side openings to intake water into the water chambers. Associated with each side opening on the wall, internally in the axial direction of the valve, three fan-shape baffles are made to block the axial openings of the water chambers. On the other hand, associated to the rest of the impermeable part of the cylindrical wall, their respective internal axial direction is made free for the discharging water from the water chambers to pass through. In operation, the inner-ring rotary valve driven by a servo motor or a mechanical system rotates 60^o a time rendering three side openings for a set of three water chambers fully open and the rest three side openings for another set of three water chambers fully closed. After the valve actuates, one set of pistons associated to the side openings fully open does the recovering cycle for water intake, meanwhile the other set of pistons associated to the side openings fully closed does the stroke to discharge water out of the outlet flange or the jet nozzle. After this half cycle finishes, the valve actuates again and the two sets of pistons switch their stroke mode, i.e., one from intake to discharge and the other from discharge to intake, and continue their strokes until finish. This completes one cycle and then the process repeats. In general, the two sets of pumping mechanisms work in a half cycle phase difference and the inner-ring rotational valve rotates 60^o in each actuation.

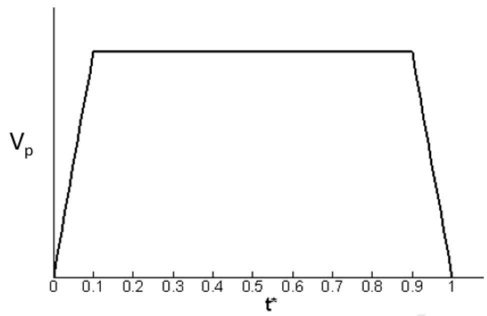

Figure 3.4.: An example of piston velocity program.

To explain the timing and coordination of the piston motion with the valve actuation, it is necessary to use an example of a piston velocity program. Piston movement is naturally impulsive, from zero velocity accelerating to the maximum then decelerating to a stop. For an illustration, let’s use a velocity program shown in Fig. 3.4, in which a non-dimensional time, $t^*=1$ indicates one complete stroke. In this example, the piston has 10% acceleration time and 10% deceleration time. With this example, the inner-ring rotary valve may start to actuate at the time the piston starts to decelerate in the last 10% stroke, and a 60^o rotation of the valve completes at the same time the piston just finishes the last 10% stroke. The reason for such a timing is that when the piston starts to decelerate, it already loses the strength to further pump water or generate thrust.

The position of the inner-ring rotary valve together with the cup heads of the pistons makes every piston always separate its water chamber into a wet and a dry compartment during its movement. The design achieved one of the purposes of using the Side-Intake concept presented in Section 2. The dry compartment is connected and open to the inside of a marine vessel or the power mechanism of a pump. Because of the presence of the dry compartment, the pistons’ energy consumption in the recovering cycles is expected to be small.

The rotational motion to open and close the inner-ring rotary valve cuts across the flow, which introduces resistive energy loss. Because the inner-ring is made thin and the cross area that cuts into the flow is small. Comparing to a gross secondary flow generation due to the entire blades spin in propeller or impeller-driven water jet, the cutting motion of the inner ring rotary valve only excites locally a minimum secondary flow. In addition, with the use of ball bearings for easy rotation, the energy cost to open and close the valve is expected to be very small.

It is not difficult to see that a SI-MIT/PDPUHC is a variation of an axial piston pump by changing an axial intake design in a conventional axial piston pump to the Side-Intake concept, and therefore expected to preserve its characteristics of linear performance and high efficiency. In addition, the Side-Intake concept makes a SI-MIT/PDPUHC be able to possess a capacity coefficient much greater than existing pumps and marine propellers. An analysis is given in Section 4.2.

4. Analysis of SI-MIT/PDPUHC from fundamentals

4.1. Efficiency and linearity of SI-MIT/PDPUHC

To facilitate an analysis with first principles, the schematic diagram in Figure 2.2 that represents a continuous inflow and outflow of a SI-MIT/PDPUHC is used. The first principle used in the analysis are the mass, momentum and energy conservation laws in a control volume. The control volume encloses the water region from water coming into the system to water leaving out of the system at the jet exit or outlet, which is shown in the dash-dotted line in Figure 2.2. Obviously, this volume is a constant at any moment of the pistons' motion. Neglecting the elevation difference between the intake and discharge as well as water viscous effect, applying the first principles to this control volume leads to,

$$Q = \rho \cdot A_p \cdot V_p = \rho \cdot A_o \cdot V_o = \rho \cdot A_j \cdot V_j \tag{1}$$

$$T = -Q \cdot (V_j - V_i) \tag{2}$$

$$W_p = Q \cdot (\frac{1}{2}V_j^2 - \frac{1}{2}V_i^2) \tag{3}$$

Note that V_j and V_i are the flow velocity in line with the thrust axis. Eqs. (1)-(3) govern the mass flow rate, the thrust generation and the required amount of mechanical work added to water from piston's locomotion. In Eq. (1), A_o is the valve opening. It is necessary to stress again that the Side-Intake concept offers the flexibility to make A_o large enough so that the flow rate in a SI-MIT/PDPUHC is not governed by the inlet opening velocity, V_o, but the piston velocity, V_p This is considered as a guideline to the valve design of a SI-MIT/PDPUHC for eliminating the inlet throttling effect. It can be seen that the piston's mechanical work on such a system is the work done on the boundary of the control volume and required mainly during the piston's forward stroke to discharge water for pumping or thrust generation, which is shown by the down-piston in Figure 2.2. During the piston's recovering (the backward) stroke to intake water shown by the up-piston in Figure 2.2, which moves to the left, the motion of that piston requires a negligible amount of work because the left compartment at ambient pressure makes the pressure on two sides of the piston more or less equal if not considering the 'added mass effect' due to the unsteady motion of the piston. The effect of unsteady motion of the piston on thrust and efficiency is discussed in Section 4.3.

In application, thrust and efficiency of a propulsor shall be considered as it is integrated with the vehicle it propels. Because of the thrust deduction factor, a balance of vehicle's resistance R and thrust comes to be,

$$\frac{R}{1-t} = -T \tag{4}$$

The useful work is the product of vehicle's speed and resistance, and the efficiency of the propulsor then leads to,

$$\eta = \frac{W_{useful}}{W_p} = (1-t) \cdot \frac{2U_s}{V_j + V_i} \tag{5}$$

In Eq. (5), U_s is the ambient water velocity, which is the same as the vehicle's speed but in the opposite direction when considering the vehicle is fixed. When a propulsor is integrated with a vehicle, V_i comes from the wake of the hull. Its relation with the vehicle speed is established through the wake fraction by V_i=(1-w)U_s. Because the area of the Side-Intake openings of the valve, A_o, could be made larger than the piston's bore area, from the law of mass conservation and a full account of the boundary layer ingestion, V_i is expected to be much smaller than U_s, i.e., (1-w) in SI-MIT/PDPUHC can be much smaller than that for a conventional propeller. Considering the wake fraction, Eq. (5) reduces to,

$$\eta = \frac{W_{useful}}{W_p} = \frac{1-t}{1-w} \cdot \frac{2V_i}{V_j + V_i} \tag{6}$$

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The first factor on the right side of Eq. (6) is the hull efficiency and the second factor is the well-known ideal (or jet) efficiency of propulsion for propeller or water jet. Eq. (6) can be simply written as,

$$\eta = \frac{W_{useful}}{W_p} = \eta_{hull} \cdot \eta_{ideal} \tag{7}$$

Eq. (7) says that if not considering loss between the input power (or the Shaft Horsepower) and the required amount of mechanical work for thrust, the efficiency of a SI-MIT achieves ideal efficiency when considered alone and becomes the product of the hull efficiency and the ideal efficiency when integrated with a vehicle.

The propulsive coefficient is used to measure the efficiency of a marine propulsor when it propels a vehicle and all losses are considered. Through an energy balance, it may be written as,

$$PC = \frac{\text{EffectiveHorsepower}}{\text{Shaft Horsepower}} = \eta_{hull} \cdot \eta_{pump} \cdot \eta_{swirl} \cdot \eta_{ideal} \cdot \eta_{mech} \tag{8}$$

where,

$$\eta_{pump} = \frac{\text{totalkineticenergy raised btw outletand inlet}}{\text{mechanicalwork done on fluid}} \tag{9}$$

$$\eta_{swirl} = \frac{\text{flow kineticenergy required for thrust}}{\text{totalkineticenergy rasied btw outletand inlet}} \tag{10}$$

and, η_{mech} accounts for mechanical loss before power delivered to the end of the shaft connecting to the propulsor. Assuming there is no more radial velocity at the propulsor outlet, η_{swirl} then accounts for the swirl loss due to the remaining tangential velocity at the jet outlet. By definition, it is the ratio of the required flow kinetic energy for thrust, which is contributed only from the flow velocities in line with the thrust axis at the outlet and inlet, to the total fluid kinetic energy raised between the outlet and inlet. As a pump, that total kinetic energy raised could turn to a pressure head rise as needed. However, the mechanical work done on fluid will not all convert to the total kinetic energy rise between the outlet and inlet because of other losses including volumetric loss due to fluid leakage, viscous shear loss, and these losses are taken into account in the pump efficiency in Eq.(9).

One of the major differences of SI-MIT/PDPUHC and a conventional marine propulsor is that for SI-MIT/PDPUHC, η_{swirl}=1 because SI-MIT/PDPUHC only generates flow velocity in the thrust axis, while for propeller or impeller-driven water jet, η_{swirl} is much less than one and varies with the rotor speed. The more off the design speed, the less η_{swirl}.

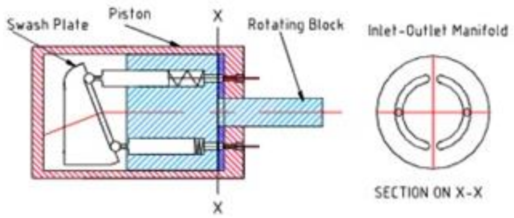

Figure 4.1.: Working principle of an axial piston pump.

A SI-MIT/PDPUHC is basically a variation of an axial piston pump but for large flow rate generation, because it is much more effective in the use of the space for fluid displacement than a conventional axial piston pump. However, as a pump, SI-MIT/PDPUHC works in the same principle as an axial piston pump used in industry. Figure 4.1 shows the working principle of a conventional axial piston pump. Its inlet and outlet openings are made on the same surface with a half circle shape. The piston-cylinder sets rotate with the shaft. The swash plate in the back of the piston shafts makes the pistons do reciprocating motion as the shaft rotates, and results in one cylinder intakes fluid while the other discharges fluid through the two half circle openings. SI-MIT/PDPUHC and axial piston pump are the same in that both systems keep ambient air pressure in the back of the pistons so that the recovering stroke is effortless. It is well recognized that an axial piston pump can easily achieve an efficiency of

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90%, i.e., η_{pump}=90%. Further, being a positive displacement pump, its efficiency is basically a constant as it works in different load conditions. In other words, an axial piston pump is a linear system or a linear performer. In terms of efficiency and linearity, SI-MIT/PDPUHC behaves the same as a conventional axial piston pump. However, SI-MIT/PDPUHC is different from a conventional axial piston pump in two accounts: (1) a conventional axial piston pump has inlet and outlet openings made on the same plane, which makes it applicable to a pump, but not applicable to net momentum generation as required for a marine propulsor; (2) because of the design of a conventional axial piston pump, the piston-cylinder sets are not able to take the maximum space of the system, which greatly reduces its effectiveness to generate flow rate. Because of these two reasons, a conventional axial piston pump is often used as a pump in applications where high pressure and low flow rate are required. For a marine propulsion system, a propulsor is required to generate flow rate in a very effective way. Taking a comparison of the design to a conventional axial piston pump, it is not difficult to realize that SI-MIT/PDPUHC is essentially an axial piston pump, but for applications to large flow rate pumping and linear momentum generation. Looking back to Eq. (8) and neglecting the mechanical efficiency for a moment, the propulsive coefficient for SI-MIT/PDPUHC can be expressed as,

$$PC_{SI-MIT} = \eta_{hull} \cdot \eta_{pump} \cdot \eta_{ideal} \tag{11}$$

where η_{pump} is a constant and its value is expected to be around 90%. η_{ideal} is determined by the ratio of the areas of inlet and outlet openings and also a constant. η_{hull} accounts for the coupling effect of the ship hull and the SI-MIT/PDPUHC, and it may vary a little depending on the vehicle's speeds. However, this nonlinearity doesn't come from the working principle of SI-MIT/PDPUHC.

Through the first principle analysis for SI-MIT/PDPUHC and its working principle comparison to axial piston pump used in industry, one can realize that SI-MIT/PDPUHC belongs to axial piston pump but for large flow rate pumping and linear momentum generation, and therefore SI-MIT/PDPUHC is ideal for marine propulsors. As a marine propulsor, the efficiency of SI-MIT/PDPUHC in open water is determined by,

$$\eta_{SI-MIT} = \eta_{pump} \cdot \eta_{ideal} \tag{12}$$

Again, η_{pump} is basically a constant and its value shall be close to the efficiency of a conventional axial piston pump. The high efficiency and linear characteristics of SI-MIT/PDPUHC can be seen from Eq.(12). For comparison, it is worthwhile to point out that the nonlinear characteristics of a propeller or impeller-driven water jet reside in η_{swirl} and η_{pump}. Both terms are nonlinear functions because of the existence of the tangential flow velocity due to the spin of rotor and its working principle is governed by the theory of non-linear lifting surface.

It is acknowledged that the analysis above neglects the energy cost in the open and close of the open-close valve for water intake and discharge and it shall be accounted for in the pump efficiency. To achieve high pump efficiency, a nontrivial question is to design an open-close valve that costs least energy. An inner-ring rotary valve is thus proposed and discussed in Section 3. This valve is expected to cost relatively a small amount of energy during the open and close processes. Remember that a conventional axial piston pump has a similar energy cost during the switching from intake to discharge of the cylinder through the rotation of the cylinders. However, its efficiency is still recognized to be the highest among any other types of positive and dynamic pumps.

The first principle analysis above is valid only for steady state or quasi-steady state flow within the control volume from the inlet to the outlet of SI-MIT/PDPUHC. In reality, the piston motion is unsteady. Recent studies have proven that the water jet generated from an unsteady piston motion is able to form vortex rings in the jet exit flow, which through entraining the ambient flow mass and being accelerated results in an additional increase of the axial flow momentum. Because of this reason, the vortex rings generated from the water jet of unsteady piston motion will contribute to an additional thrust and therefore give a further increase of the propulsion efficiency, which is not taken into account in Eqs. (1) to (3). A brief discussion on vortex ring and its effect on SI-MIT/PDPUHC is given in Section 4.3

4.2. *Effectiveness of SI-MIT/PDPUHC*

Effectiveness of a power machine is about the power density question. For a propulsor, it is ideally to have the most compact system to generate a given thrust power without scarifying its efficiency. Eq.(2) is a general equation to calculate thrust within a propulsor system between its inlet and outlet. There are two ways to increase thrust. One way is to increase the difference of the inlet and outlet velocities, which is equivalent to increasing the load on the propulsor. This is not desirable in that it will increase the linear slip loss, and therefore reduce the efficiency through a reduction of the ideal (or jet) efficiency term in Eq.(7). An ideal way to increase thrust is by increasing the flow rate, because flow rate doesn't directly appear in the efficiency equation and assuming that

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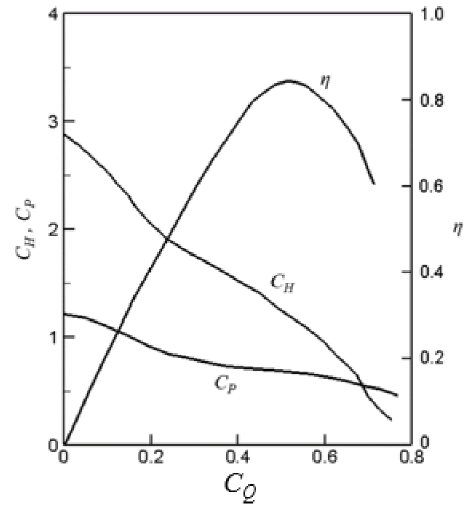

Figure 4.2.: A typical axial-flow pump curve from White (1986).

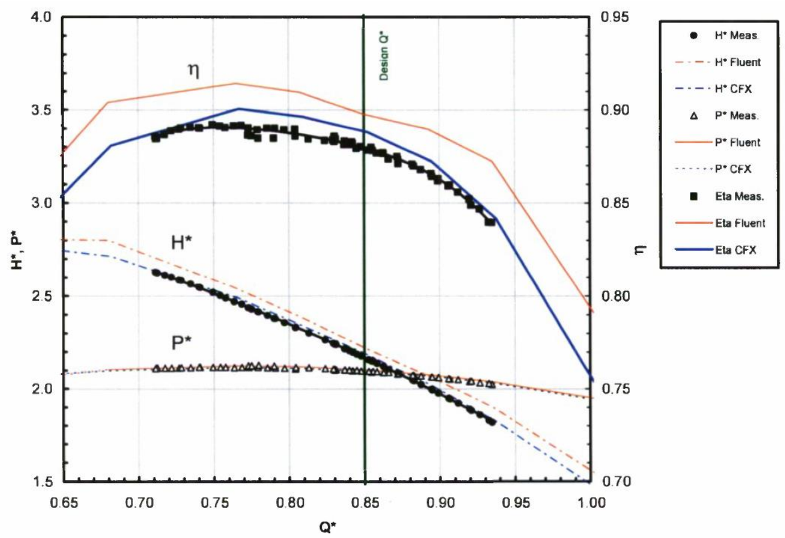

Figure 4.3.: ONR AxWJ-2 measured and CFD calculated pump curves from Chesnakas, et al. (2009).

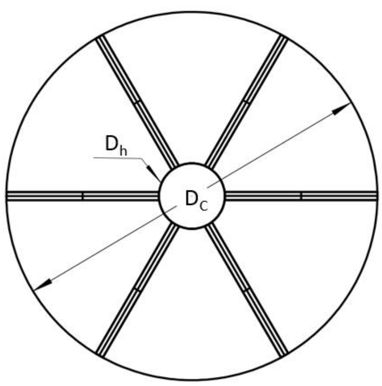

Figure 4.4.: Geometric parameters in a cross section of SI-MIT/PDPUHC.

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η_{swirl} and η_{pump} are not a function of the flow rate. As shown in Figure 4.2, for propeller and impeller-driven water jet, the relation of efficiency and flow rate is a bell-shape curve, but for SI-MIT/PDPUHC, efficiency is basically independent from flow rate because SI-MIT/PDPUHC is a linear performer.

If one doesn't like to see a reduction of efficiency in a propeller or impeller-driven water jet, then to increase flow rate means to increase the size of the propulsor and/or its running speed. The effectiveness question is then to answer: among the same size of propulsors, which propulsor can produce the most flow rate without scarifying efficiency. The capacity coefficient, which is defined as C_Q=Q/nD^3 and the limiting speed without causing inlet throttling may be the measures to determine the effectiveness of two propulsors or pumps. When fully appreciate one of the benefits of the Side-Intake concept, i.e., the feasibility to make the inlet valve opening comparable to the piston bore size, which was embedded in the design of the inner ring rotary valve, the inlet suction behaviour and cavitation requirement for a SI-MIT/PDPUHC can be made compatible to a conventional DN pump or marine propulsor. With this assumption, the capacity coefficient then becomes a unique parameter to compare the effectiveness of the two propulsors or pumps.

It is well known that axial-flow pumps are the most effective system for generating flow rate, and therefore often chosen for water jet propulsors. A propeller actually can also be viewed as an axial-flow pump. Because of the nonlinear behaviour, the relation between the capacity coefficient and the efficiency in propeller or impeller-driven water jet has no straight way to determine, but comes from the result of the design skill level and the model tests. Figure 4.2 quoted from White (1986) provided pump curves for a typical axial-flow pump. A rough estimate for C_Q at the best efficiency point is around 0.55 for a typical axial-flow pump. There were tremendous investment and research work on designing the best axial-flow pump jet for marine propulsor. Through a literature search and to the best knowledge of the author, it was found that ONR AxWJ-2 pump jet was able to achieve $C_{Q,ONR}$ =0.77 at its highest pump efficiency of 0.89.

The data is read from Figure 4.3 which was presented in Chesnakas, et al. (2009). The way SI-MIT/PDPUHC generates thrust is straightforward. It is just a linear displacement of water through a piston linear stroke motion. Therefore, its capacity coefficient is constant and easily determined. Figure 4.4 shows a section of the SI-MIT/PDPUHC presented in Section 3. Because there are three fan-shape pistons in a set in the cylinder body to discharge water, the pistons' total bore area, A_p to displace water is basically half the cylinder cross-section area minus the hub and radial baffle area, i.e.,

$$A_p = F_l \frac{\pi}{8}(D_C^2 - D_h^2) \tag{13}$$

where D_C and D_h are diameters of the cylinder and the hub, F_l is a reduction factor to consider the blockages of the radial baffle. An equivalent diameter of this area can also be calculated, $D_E^2 = 0.5F_i(D_C^2 - D_h^2)$. The capacity coefficient of the SI-MIT/PDPUHC can then be calculated by,

$$C_Q = \frac{A_p\bar{V}_p}{n_C D_C^3} = \frac{A_p 2n_C L_p}{n_C D_C^3} = \frac{2A_p L_p}{D_C^3} \tag{14}$$

where $\bar{V}_p$, n_C and L_p are the average velocity, CPM (Cycles Per Minute) and the stroke length of a piston respectively. Combining Eq.(13), Eq. (14) and D_E, the capacity coefficient reduces to,

$$C_Q = F_l \frac{\pi}{4}(1 - \frac{D_h^2}{D_C^2})\sqrt{\frac{F_l}{2}(1 - \frac{D_h^2}{D_C^2})}\frac{L_p}{D_E} \tag{15}$$

In the design, the L_p/D_E ratio is chosen to be 4 with the goal to generate maximum vortex rings for efficiency enhancement. When reasonably choosing D_h/D_C to be 1/6 and the area reduction factor, F_l =0.82 to take consideration of 18% area loss by the radial baffles, one obtains,

$$C_Q = \frac{Q}{n_C D_C^3} = 1.58 \tag{16}$$

for the SI-MIT/PDPUHC. The result showed that under the condition that the SI-MIT/PDPUHC runs at CPM the same as the RPM of ONR AxWJ-2, the capacity coefficient of the former can be made twice greater than that of the later. Further, because a SI-MIT/PDPUHC capacity coefficient is more or less a constant, the designers can always choose higher working CPM to make the system more compact if inlet suction loss and cavitation are under check. This character of SI-MIT/PDPUHC as the result of the Side-Intake concept qualifies SI-MIT/PDPUHC to be a pump or marine propulsor with ultra-high capacity. In the next section, it showed that a λ around 4 is just the formation number to generate maximum vortex ring in unsteady jet flow, which will further enhance propulsive efficiency for SI-MIT/PDPUHC.

Figure 4.5.: Vortex ring formation in the wake of a starting jet from Olcay and Krueger (2008).

4.3. Water jet from unsteady piston motion

Piston motion naturally employs a non-constant velocity profile or a velocity program, which is from velocity zero at the start of discharge to its peak and then back to zero at the end of discharge. The velocity program of the piston motion in turn creates an impulsive water jet or a starting jet at the jet exit. It is a well-known phenomenon that vortex rings will be generated when forcing fluid impulsively out of a nozzle into the ambient fluid. Thus, piston-cylinder arrangement is commonly used in experiments by researchers to generate starting jets and study vortex ring dynamics. Figure 4.5 taken from the work by Olcay and Krueger (2008) gives a typical view of a vortex ring formation and evolution as a column of water in the cylinder is ejected out by the piston to the ambient water.

Recently, there are increasing interests in the study of vortex ring formation in impulsive jet flow in relation to impulse, thrust and propulsive efficiency and its application to marine vehicle propulsion. One early research in this direction was reported in Gharib, et al. (1998). Using a piston-cylinder device to study vortex ring formation in starting jets generated from various short to long of the piston stroke to diameter ratio (L/d), Gharib, et al. (1998) observed that for L/d sufficiently less than 4, there is only a single vortex ring in the jet flow, while for L/d larger than 4, the jet flow will generate a pinched-off leading vortex ring followed by a trailing jet. After the leading vortex pinches off, the on-going trailing jet just behaves like a steady jet flow. The L/d ratio can also be viewed as a non-dimensional time scale for completing one piston stroke. Their observation concluded that when this non-dimensional time scale is around 4, referred to as the ‘formation number’ in the paper, a pinch-off of the leading vortex ring from a trailing jet will occur, indicating that the vortex ring can no longer absorb any more vorticities emanating from the jet flow. In other words, the leading vortex ring contains the maximum circulation a vortex ring is able to acquire. Again using the piston-cylinder device, Krueger and Gharib (2003) further studied the relative contributions of the leading vortex ring and the trailing jet to the total impulse provided by the impulsive jet flow at various L/d ratios. Note that thrust of an impulsive or unsteady jet is the time average of the total impulse. The experiment results from Krueger and Gharib (2003) showed that the thrust contributed from the vortex ring is much higher than the thrust contributed from the trailing jet, which represents a steady jet mode, and the maximum thrust of an impulsive jet can be achieved at a L/d ratio just after the leading vortex pinches off or equal to the formation number. According to Krueger and Gharib (2003), the fact that vortex rings contributes to an increase of thrust or total impulse over a steady jet for a given average piston velocity is attributed to ambient flow mass entrainment into the vortex ring and an acceleration of the vortex ring due to the over-pressure at the jet exit. The existence of an over-pressure at the jet exit is the result of the pulsed-motion of the piston.

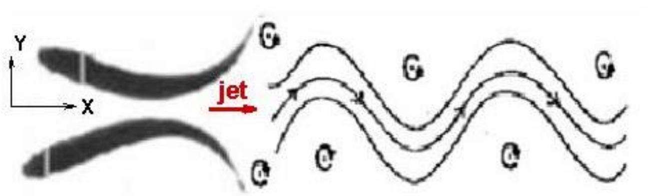

Figure 4.6.: Impulsive jet generated by fish caudal fin.

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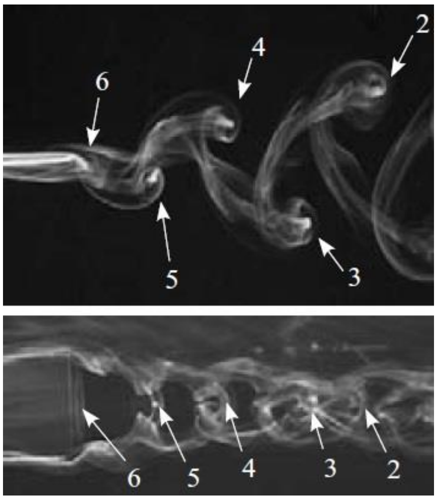

Figure 4.7.: 3-D reverse von Kármán vortex street from Buchholz and Smits (2006).

Nature provided the most compelling evidence that over millions of years' natural selection most of the fast moving biological animals choose impulsive mode to fly or swim. On the aquatic animal side, examples are various kinds of fish and squid. Figure 4.6 describes a 2-D view of the impulsive jet generated from an alternating impulsive flapping of the caudal fin of a fish. The jet is featured with reverse von Kármán vortex street that generates impulsive thrust for fish. The alternating impulsive flapping of caudal fin also controls the direction of the jet for fish to have an easy and fast maneuver. Many researches were done on fish swimming, e.g., Drucker and Lauder (1999). The DPIV observation in fish jet wake from Drucker and Lauder (1999) inferred that the alternating impulsive flapping of the caudal fin of a fish actually generates a system of vortex rings chained together with each ring oriented in the local mean jet flow. Buchholz and Smits (2006) studied the vortex structures in the wake of a pitching panel of low-aspect-ratio. Figure 4.7 taken from Buchholz and Smiths gave a visual look of a system of chain vortex rings, a 3-D reverse von Kármán vortex street, in the wake of the pitching panel. A squid ingests surrounding fluid into a large mantle cavity. It then uses its body muscle to eject the fluid out through a nozzle called the siphon in an impulsive manner. By comparison, it is not difficult to realize that the way SI-MIT/PDPUHC generates impulsive thrust is more like that of a squid. Examples of human beings naturally employing impulsive thrust were found in competitive sports, such as swimming and boat racing.

From the natural evidence of aquatic animal swim and the fundamental research finding that optimum vortex ring at the formation number generated from an impulsive jet can greatly increase propulsive efficiency, researches on using impulsive jet as a general marine propulsion system received ever greater attention around the world. Ruiz et al. (2011) designed and built an impulsive jet system to be installed in a laboratory-scale submarine vehicle to accomplish a self-propelled vehicle with impulsive propulsion. The goal of their research was to study the propulsion efficiency of impulsive jets under vehicle's self-propelled conditions. Openings were made as water jet inlet on the submarine hull along its circumference at about the mid body, and the end of the hull left open to water as a jet nozzle. A propeller was installed inside the aft-mid body of the submarine hull to generate water flow for thrust. The impulsive jet flow at the nozzle exit was achieved by a periodic close and open of the inlet openings through a rotating shell mounted on the hull. When the openings were constantly open, the system worked in a steady jet mode, which was the same as an impeller-driven water jet. When the openings were periodically open and closed during the spin of the propeller to drive the flow, the system worked in an impulsive jet mode. The experiment results from Ruiz et al. (2011) showed that the impulsive jet was able to increase propulsive efficiency by over 50% greater than that of the steady jet through an optimum use of vortex ring. Their analysis model also pointed out that a substantial propulsion efficiency enhancement from the impulsive jet by more than 50% over a steady jet were the results of two primary mechanisms: (1) ambient fluid entrainment into the forming vortex ring; (2) the added mass generated by the downstream acceleration of the vortex ring due to the over-pressure created by the pulsing jet flow at the nozzle exit. These findings were in agreement with the analysis from Krueger and Gharib (2003). However, it should be noted that their propulsion system whether operated in the impulsive jet mode or in the steady jet mode there always was swirl loss because of the use of propeller, and the baseline efficiency of their propulsive system in steady jet mode was not provided in the paper.

All the recent studies proved that impulsive jet or unsteady jet created from a typical piston-cylinder setup can have a substantial increase of thrust over a steady jet with the same power input especially when the impulsive jet operates around the formation number to generate pinched-off vortex rings. This additional thrust or efficiency increase is related to the vortex ring dynamics in the jet wake generated from the impulsive motion of the device, which is not quantified in the steady-state flow analysis for a system from inlet to exit.

Any practical fluid power system has to work on a cycle. A SI-MIT-PDPUHC's power cycle contains a discharge stroke and a recovering stroke of pistons. The benefit of an impulsive jet to propulsion efficiency is at best owed to the discharging stroke. The unsteady stroke of the piston also introduces impulsive flow or 'added mass effect' during the recovering stroke or the fluid intake process, which is by any means disadvantageous in view of the efficiency as well as the impact to the system. However, this effect can be the best alleviated by a design of 'impulse absorber', for instance a spring system, being built into the piston head.

5. Conclusions and future work

A novel PD pump named SI-MIT/PDPUHC is presented. This new pump is based on the unique concept of Side-Intake of water (fluid) and the details of the principle of the Side-Intake are discussed. A qualitative analysis for SI-MIT/PDPUHC on its efficiency being a pump or a marine propulsor is provided, though the effect of the unsteadiness of the flow due to the piston's impulsive motion is not taken into account at this time. It showed that SI-MIT/PDPUHC is a variation of an axial piston pump by changing the axial intake in a conventional axial piston pump to the Side-Intake idea, and therefore preserves its characteristics of linear performance and a high pump efficiency. The Side-Intake features three characteristics. One is that it achieves fluid intake from the side (lateral) in reference to fluid discharge in the axial direction of the cylinder; another is it allows a piston to separate its fluid chamber into a dry and a wet compartment anytime as the piston moves so that the energy consumption in the recovering cycles of pistons can be minimized; and lastly it allows to create as large as possible the volume of fluid chambers in the system so that the capacity coefficient can be maximized. A quantitative analysis for the last feature on a compact design of SI-MIT/PDPUHC for practical utilization is provided and the analysis shows that a SI-MIT/PDPUHC potentially qualifies as a pump or marine propulsor of ultra-high capacity. The linear performance in hydrodynamics and a reliance on reciprocating movement of piston-cylinder setups make the new pump a natural match to integrate modern smart linear motors for smart pumps and marine propulsors.

As a marine propulsor, the performance advantages of SI-MIT/PDPUHC are expected to include: (1) a complete elimination of the swirl loss that exists in propellers and impeller-based water jets; (2) being a linear thruster, referring to that the thrust power is able to maintain more or less a linear relation to the input power even when the vehicle's operational condition varies; (3) being potentially a very compact thruster because of its ultra-high capacity coefficient; and (4) a reliance on piston-cylinder setup for thrust makes it a natural and simplest device to generate and manipulate vortex rings for the maximum thrust enhancement from the vortex ring impulse.

Some concerns for SI-MIT/PDPUHC lie in the pistons' unsteady strokes that could introduce system vibration, possible cavitation and a performance reduction especially from an accelerating intake process of fluid. It is proposed to design an 'impulse absorber' in the piston head to minimize these negative effects, but as for how to achieve an optimal 'impulse absorber' and evaluate its effectiveness remains an interesting research topic.

Proposed future work includes: (1) develop a simple model that represents the hydrodynamic cycle featuring the Side-Intake concept in SI-MIT/PDPUHC for numerical studies. CFD tools will be applied to the simulations of the hydrodynamic cycles in SI-MIT/PDPUHC through this model for pump and propulsion efficiencies. It is expected that the CFD simulations will give an accurate account of the unsteady flow effect generated from the impulsive motion of the pistons on the hydrodynamic efficiency in SI-MIT/PDPUHC. However, the accuracy of the CFD tools is expected to be challenged by the highly transient and vortical nature of the impulsive jet flows in SI-MIT/PDPUHC; (2) design and build a prototype of SI-MIT/PDPUHC for pump as well as open water propulsion tests and compare its performance characteristics with those from the pumps and propellers of similar size.

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