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DESIGN OF A SUBMARINE MAIN BALLAST TANK FLOOD AND VENT OPENINGS MECHANISMS

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ABSTRACT

Besides the piping systems used for blowing the main ballast tanks of a submarine by feeding compressed air, a great importance must be given to the flood and vent openings and the mechanisms that operate the covers. From an engineering standpoint, the flood openings must be as small as possible to minimize the noise generated by the water flow.Their number and cumulated area must be determied as to avoid over-pressure when feeding compressed air into the ballast tanks that may lead to structural damage. The requirements for designing of such systems are covered by the Class and Administration rules and regulations.

Keywords: submarine, submerged vehicle, ballast system, main ballast tank, flood port cover, vent opening cover, rotary vane actuator.

1. INTRODUCTION

The process of diving and surfacing constitutes a highly important maneuver in the movement of a submarine. During diving operation, the center of gravity and the center of buoyancy, which are, let’s say, “special” for this kind of boat, are in constant change while seawater floods the main ballast tanks. This maneuver must be completed in the smallest amount of time possible with the smallest amount of noise. Therefore, the design of the flood and vent openings have a great importance that reflects on the speed at which the submarine dives or surfaces. This paper will approach a design of these mechanisms used in a single hull, diesel-electric submarine, able to dive at a maximum depth of 270 meters.

2. DESIGN OF THE FLOOD PORTS AND VENT VALVES

The main aspect when designing the flood ports for a submarine main ballast tanks is that they must allow for a rapid water flow and to avoid overpressure inside the tanks. The desired water flow is around 2÷3 m/s, depending on the volume of the tank and the time required for the maneuver, otherwise it can be calculated using formula 1.

$$Q_{MBT} = \frac{V_{MBT}}{t}\left[\frac{m^3}{min}\right] \quad (1)$$

After the values regarding the tank volume and the desired time for the maneuver are plugged in, the flow rates for fore MBT and aft MBT are determined. The resulted flow of water for each tank will be further used to determine the necessary total area for the flood ports openings. This area can be calculated using Toricelli's Law for oriffices (2), from which we can extract the area.

$$Q = c_d \cdot A \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}} \equiv A = \frac{Q_{MBT}}{c_d \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}}} \; [m^3] \qquad (2)$$

where **Q** – water flow [m^3/min], $\mathbf{c_d}$ – discharge coefficient, **A** – area of the flood ports [m^2], **Δp** – pressure difference [Pa], ρ - fluid density [kg/m^3].

The results for the area of the flood ports for each main ballast tank, will have to be divided into several smaller openings, due to the constraints imposed by the internal framing structures, but also because of noise reasons, thus if larger the flood ports are used, the more noisy will be the flow of water. Another reason for dividing the total area into smaller, similar areas, is that the size and the complexity of the opening and closing mechanisms for the flood ports covers, will render them infeasible. Therefore, the flood ports will be of a rectangular shape and will be posted in between the structural frames of the ballast tanks, at their bottom most part, near the center line with one flood port on each side of the center line and in between each two framings, thus avoiding the possibility that water may accumulate and prevent an incomplete blow of the tanks, which may be leading to the possibility of an incomplete surfacing or ship instability. For each ballast tank, we will assume a distance between each framing to be at 600 mm, and the frame shape is that of a "T" profile, thus leading to the need of having a certain clearance between the flood port cover and the structural framing profile. Therefore, the adopted length of the flood ports openings (as considered from aft to fore of the MBT), for each MBT, will be around 400 mm. To calculate the width of the individual flood port in relation to the adopted length, first we must divide the total area calculated using the formula 2 to an even number of ports (one for each side of the center line). In short, the individual area of each port will be divided by the adopted length for inbetween the tank framing. In figures 1 and 2, an example of such flood ports arrangement can be observed.

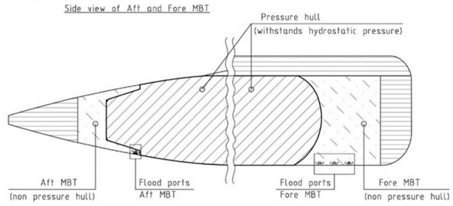

Fig. 1. Arrangement of the flood ports inside Aft and Fore MBTs (side view)

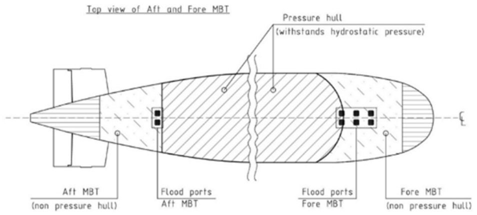

Fig. 2. Arrangement of the flood ports inside Aft and Fore MBTs (top view)

One problem that is to be solved regarding the flood ports, for the water to be able to enter or to exit the tanks, represents the choice of a system for opening the covers, comprised of actuators that use compressed air or hydraulic fluid. In this paper the system used for opening the flood ports covers are hydraulic actuators comprised of rotary vane actuators and the afferent piping to carry the hydraulic fluid. These rotary vane actuators are of a double vane type and can rotate at a 90° angle. The rotary vane will be most of the time in a wet environment, thus, the materials constituting the actuator must be corrosion resistant. From within a sizing perspective, the main forces that act on the flood port cover and for which the actuator must withstand, are generated by the water pressure at periscopic depth, which are by far, greater than the force required for lifting the cover. These forces can be defined by calculating the pressure difference between the hydrostatic pressure at periscopic depth and the pressure within the tank at a given water level. In this case, we will assume the hidrostatic pressure calculated with the formula 3 is 2 bar and the pressure inside the tanks is 1 bar, resulting in a pressure difference of 1 bar.

$$\Delta p = \rho \cdot g \cdot h \text{ [bar]} \quad (3)$$

where, **ρ** - water density [kg/m^3], **g** – acceleration due to gravity, **h** – depth of the submarine measured from the base line.

When pluging in the water density, it must be regarded that the water density around the globe is not the same, thus, in the calculation the water with the highest density can be used, implying that the submarine will navigate in all seas and oceans around the globe. In this case, the pressure acting on the ballast tanks is of 1 bar, meaning 100.000 N/m2. Having the pressure acting on the ballast tanks, we can determine the force acting on the cover, by multiplying it with the surface of the cover and taking into account a safety factor, using the formula 4.

$$F = (\Delta p - p_{tank}) \cdot A_{cover} \cdot f_s \; [N] \quad (4)$$

where, **F** – force [N], **Δp** – hidrostatic pressure [N/m^2], $\mathbf{p_{tank}}$ – pressure at the surface of the water inside the tank [N/m^2], $\mathbf{A_{cover}}$ – area of the flood port cover [m^2], $\mathbf{f_s}$ – safety factor.

For sizing the actuator, the first step is to determine the rotary vane actuator's torque which can be determined using the formula 5, pluging in the force determined with formula 4 and the arm of the force, determined from the center of the actuator's shaft to the center of gravity of the flood port cover as in figure 3.

$$T_{actuator} = F \cdot r_{cover} \; [Nm] \quad (5)$$

where, $\mathbf{T_{actuator}}$ – actuator torque [Nm], **F** – force acting on the cover [N], $\mathbf{r_{cover}}$ – arm of the force [m].

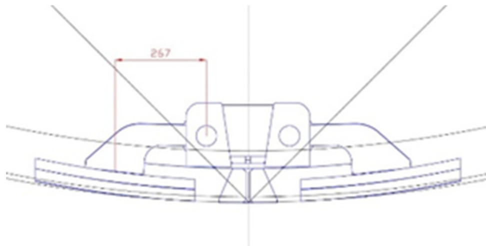

Fig. 3. Measuring the arm of the force

One important aspect in sizing the actuator, from which other elements, such as the vanes, hydraulic chamber (inside diameter) and the outer diameter of the body, is the dependence of the diameter of the turning shaft. To have the smallest possible diameter for the shaft that can support the required torque is in direct relation with the materials that the shaft is made of. Thus, the shaft will use metals that have a high yield strength. In this example, the 22Cr-0.5Ni Stainless Steel was used as it has a yield strength of 505 MPa. It must be taken into account that the turning shaft of the actuator will operate on one axis of movement, resulting in a force that creates uniaxial stresses. To determine the maximum applied torque in which the shaft won't be able to overcome the forces acting on it and in consequence the deformations will become plastic, the von Mises criterion that predict the moment of failure must be applied and the maximum admissible torque for the chosen material can be calculated.

$$T_{adm} = \sigma_v \cdot \sigma_\gamma \quad (6)$$

where, $\mathbf{T_{adm}}$ – maximum admissible torque [Nm], $\boldsymbol{\sigma_v}$ – von Mises criterion factor, $\boldsymbol{\sigma_\gamma}$ - yield strength of the material.

To determine the diameter of the shaft, we can use the maximum admissible torque calculated with formula 6 and equal it to the ratio between the torque of the actuator and the radius of the shaft to the polar moment of inertia of the shaft as stated in the formula 7.

$$T_{adm} = \frac{T_{actuator} \cdot r}{J} \quad ; J = \frac{\pi}{2} \cdot r^4 \qquad (7)$$

where, $\mathbf{T_{actuator}}$ is calculated in formula 4, **r** - the radius of the shaft and **J** is polar moment of inertia of the shaft.

By substituting all the values and determining the radius, the diameter of the shaft can be determined. The next step is to calculate the length of the vanes of the actuator (view figure 6), depending on the adopted number, the hydraulic pressure applied over the vanes and also the diameters of the rotor (shaft) and the stator, as follows:

$$T_{actuator} = \frac{z \cdot p_{hydraulic} \cdot L}{2} \cdot (D_s^2 - d_r^2) \qquad (8)$$

where **z** - the number of vanes, $\mathbf{p_{hydraulic}}$ - the pressure applied to the vanes [N/m^2], $\mathbf{D_s}$ - the internal diameter of the stator [m], $\mathbf{d_r}$ - the diameter of the rotor (shaft) [m].

It must be noted that all the elements of the calculation are playing an important role in determining the overall size of the actuator and any modification to a factor will, in consequence, affect the others significantly. For example, a large the number of vanes will impact not only the rotation capability of the shaft, making it rotating only for a small angle, but it will also determine that the diameter of the stator to be rather large, thus, resulting in a large body of the actuator. The hydraulic pressure has a similar role. Having a low hydraulic pressure will determine the need for a larger area of the vanes and on the contrary, having a higher hydraulic pressure will determine a smaller surface area needed to rotate the shaft. When using a higher pressure it must be noted that these actuators have several seals which must resist the forces acting on them, thus limiting the hydraulic pressure inside the system at the maximum pressure supported by the seals.

After determining the length and the width of one vane, all what is left, is to determine the thickness of the vane, for which it can withstand the hydraulic pressure inside the system by using the formula 9 for determining the force applied on the vane.

$$F_v = p_{hydraulic} \cdot A_v = p_{hydraulic} \cdot (L \cdot l) \; [N] \qquad (9)$$

where, $\mathbf{F_v}$ – force acting on the vane [N], $\mathbf{p_{hydraulic}}$ – hidraulic pressure [N/m^2], $\mathbf{A_v}$ – area of the vane [m^2], **L** – length of the vane [m], **l** – width of the vane [m].

The bending moment can be calculated using formula 10.

$$M_v = F_v \cdot r \; [Nm] \qquad (10)$$

Similar to the calculation of the shaft, knowing the force that is applied on the vane and the maximum allowable stress of the material, for which the vane will bend and the deformation will become plastic, the thickness of the vane can be determined using formula 11 by rewriting formula 10.

$$M_v = M_{adm} \cdot J_z \quad ; J_z = \frac{L \cdot t^2}{6} \qquad (11)$$

where, $\mathbf{M_v}$ – bending moment calculated [Nm], $\mathbf{M_{adm}}$ – maximum allowable stress according to von Mises [N], $\mathbf{J_z}$ – moment of inertia on z axis, **L** – length of the vane [m], **t** – thickness of the vane [m].

Rearanging the factors in formula 11, the thickness t of the vane can be calculated. A design proposal for the flood port covers can be observed in figures 4 and 5.

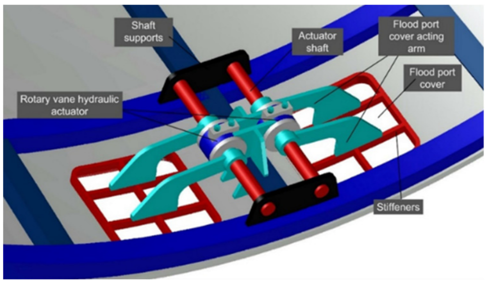

Fig. 4. Flood port cover assembly

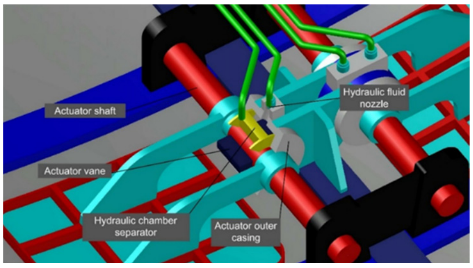

Fig. 5. 3D view of the rotary vane actuator components

The vent openings area for releasing the air to allow the diving procedure, will be calculated similarly to the flood ports openings, while the devices used for operating the vent covers may be linear hydraulic actuators, because the cover size is much smaller than that of the flood ports, thus resulting in a much simpler actuating system. The vent valves are located at the uppermost part of the tanks and are not subjected to high forces when submerged because of the fact that when the ballast tanks are filled with water, the pressure inside and outside the tank will be the same, nor the fact that when surfaced, the inside and outside pressures purshing against the vent cover is that of the atmospheric pressure. So, in this case, the linear hydraulic actuator will be sized in such a manner to support the forces for moving the vent cover and to be able to withstand the hidraulic pressure from wihin the system (the hidraulic pressure will be higher than the hydrostatic pressure at the maximum diving depth). When designing the vent openings and covers, it must be accounted that when the emergency condition blowing occurs, because of the high air pressure, their shape must be chosen in such a way to prevent the blowing of the cover outside of the tank and therefore rendering the submarine unable to surface. One more thing to take into consideration is that the air particles are very small, and the possibility of leakage between the vent opening and the vent cover can occur, regardless of how perfect they overlap. To avoid this from happening, a rubber seal or any other suitable sealing device must be installed on the contact surface between the cover and the opening. An example of such a system for vent valves are presented in figures 6 to 10.

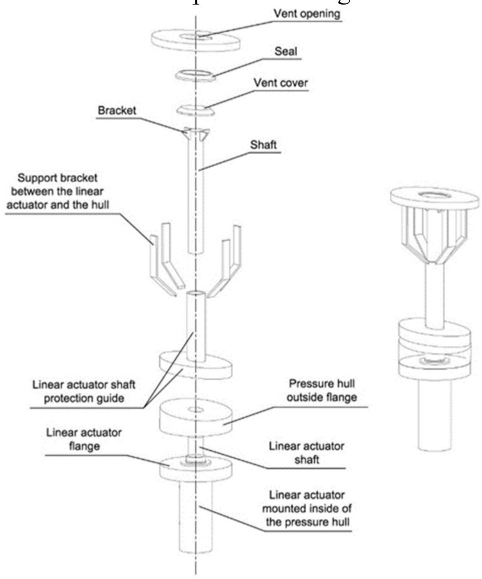

Fig. 6. Aft ballast tank vent port cover assembly inside of the pressure hull (exploded view)

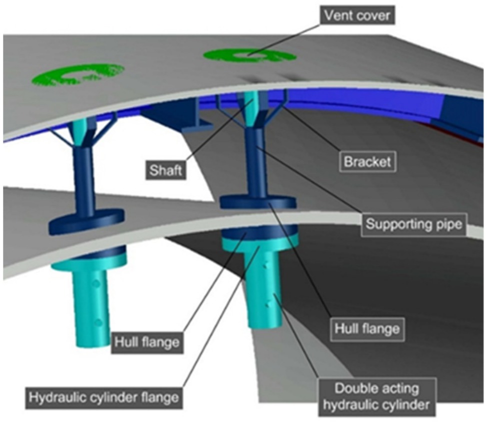

Fig. 7. Vent port cover assembly inside of the pressure hull (model view)

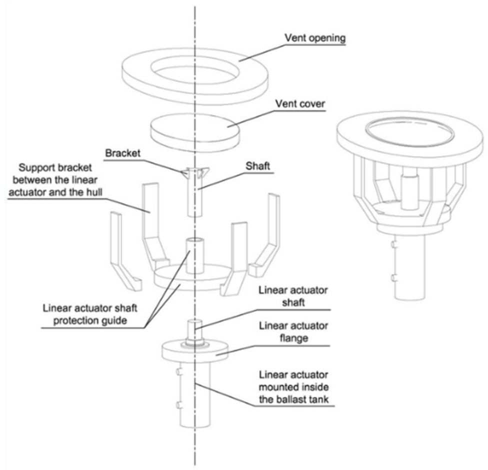

Fig. 8. Fore ballast tank vent port cover assembly inside of the ballast tank (exploded view)

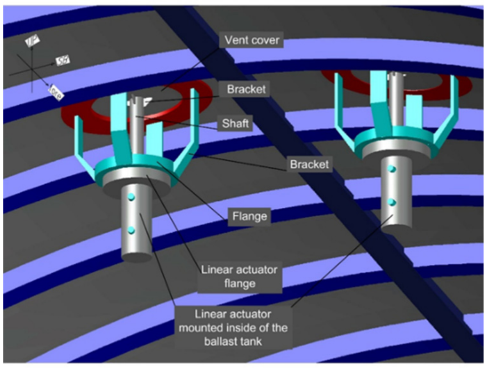

Fig. 9. Vent port cover assembly inside of the ballast tank (model view)

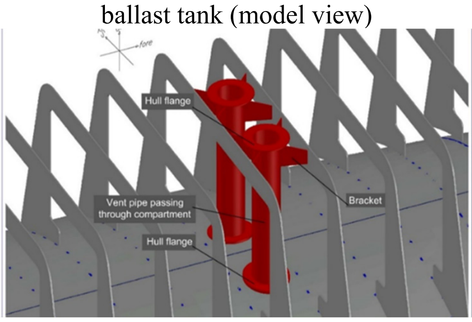

Fig. 10. Pipe connection between vent opening and outside passing through compartment (model view)

3. CONCLUDING REMARKS

When designing flood ports, the main scope is to minimize the noise generated by the flow of the water passing through the port. This can be done by minimizing the opening area and placing as many flood openings as needed to allow for a large flow of water passing in the smallest amount of time possible and to avoid any overpressure when compressed air is fed into the tanks. From a fail-safe perspective, in the case of the flood port cover malfunctioning, failsafe mechanisms must be installed to prevent the opening of the cover when the submarine navigates at the surface. Another aspect regarding the well functioning of the vent and flood openings cover mechanisms is to not allow for any debree to enter the ballast tanks, by installing steel grates or other such devices and the actuators used must be corrosion resistant.

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