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A Computing Model for Design of Flexible Buoyancy System for Autonomous Underwater Vehicles and Gliders

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ABSTRACT

Modern design approaches are conceived and utilised in an integrated loop covering system statics, dynamics, optimisation, and others. In this regard this paper presents a computing based integrated design approach for a flexible buoyancy system (FBS) aimed towards the applications in autonomous underwater vehicles and gliders. The primary design alternatives for the FBS are: piston and pump driven and both are investigated. The primary design of autonomous underwater vehicles and gliders is computed from first principle of mechanics and defined in the computer aided design model and it is implemented in the Matlab*TM. Lastly, to show the application of the present approach, a design example is presented for a water depth of 6000 m.

Keywords: Flexible buoyancy system; Underwater vehicle; Computing model; Piston driven flexible buoyancy system; Pump driven flexible buoyancy system

1. INTRODUCTION

Although, water bodies cover almost 71 per cent of earth's surface, so far only around 5 per cent of the world's water bodies have been explored, i.e. Oceans, seas, rivers, great lakes, etc. They are critically important to all the known forms of life and forms part of the carbon cycle, and influences climate and weather patterns. They are also habitat for variety of species, minerals, ores and this scenario demands that they are surveyed and mapped for efficient utilisation of underwater resources. Furthermore, the coastal security of nations also demands constant surveillance and defence capabilities that can be deployed along the coast and deep in to the ocean/sea/lake, e.g. 0 m to 100 m low operating water depth (*h*), 100 m to 400 m medium (*h*), and higher than 400 m is high (*h*). Additionally, in high operating water depths, there are: high ($400\,m < h \leq 3000\,m$), very high ($3000\,m < h \leq 6000\,m$) and ultra high ($h > 6000\,m$).

Primarily, the underwater vehicles (UVs) are aimed towards the ocean related applications. Their applications include both the civilian and naval operations and they can be classified into: Tethered and un-tethered vehicles and further based upon their sizes into: Vehicle - large size, and Glider - small size. A tethered vehicle can use the umbilical cable for drawing the power and/or real time data transmission. An un-tethered vehicle needs to carry power on-board in form of batteries and for them total installed on-board power is critical. Furthermore, the UVs are restricted in the size. And, because of this limited size they cannot accommodate a large space exclusively for the power storage. Also, the UVs are designed to maneuver, change depth, launch/recover, sit/crawl/move on ocean/sea/river bed and hover at some specific water depth. These operational capabilities imply that the design of UVs is flexible in the weight and or buoyancy.

Technically, for any floating or submerged body its buoyancy can be changed through operation of a thruster. However, it has disadvantages: high power consumption by the thrusters and depending upon the thrust direction adverse effect on drag and weight. In this regard, although the efficient control of buoyancy through flexible buoyancy system (FBS) is gaining critical importance in applications, the design process is not known and the existing research results only present the design summary, rather than the design approach[1-3]. Encouraged from the critically important need towards the design and development of new generation energy efficient AUVs/AUGs and to address the gaps in research, we present an integrated design approach. Herein, we aim to explore a computing based integrated design approach for a FBS targeted towards the applications in autonomous underwater vehicles/gliders.

2. INTEGRATED DESIGN MODEL

The integrated design model is integrated with the basic design requirements of UVs, e.g. total change of buoyancy required, rate of change of buoyancy required, and depth of operation, etc. These are represented in the CAD model. The implementation is done in the Matlab*TM and the overall design approach is as shown in Fig. 1 (a) and the control system architecture in Fig. 1 (b).

Our design model and methodologies are built on the principles of engineering and fluid mechanics[4-6] and the details are reported in *Appendices A.1* and *A.2*.

We focus on two design alternatives: Piston operated FBS,

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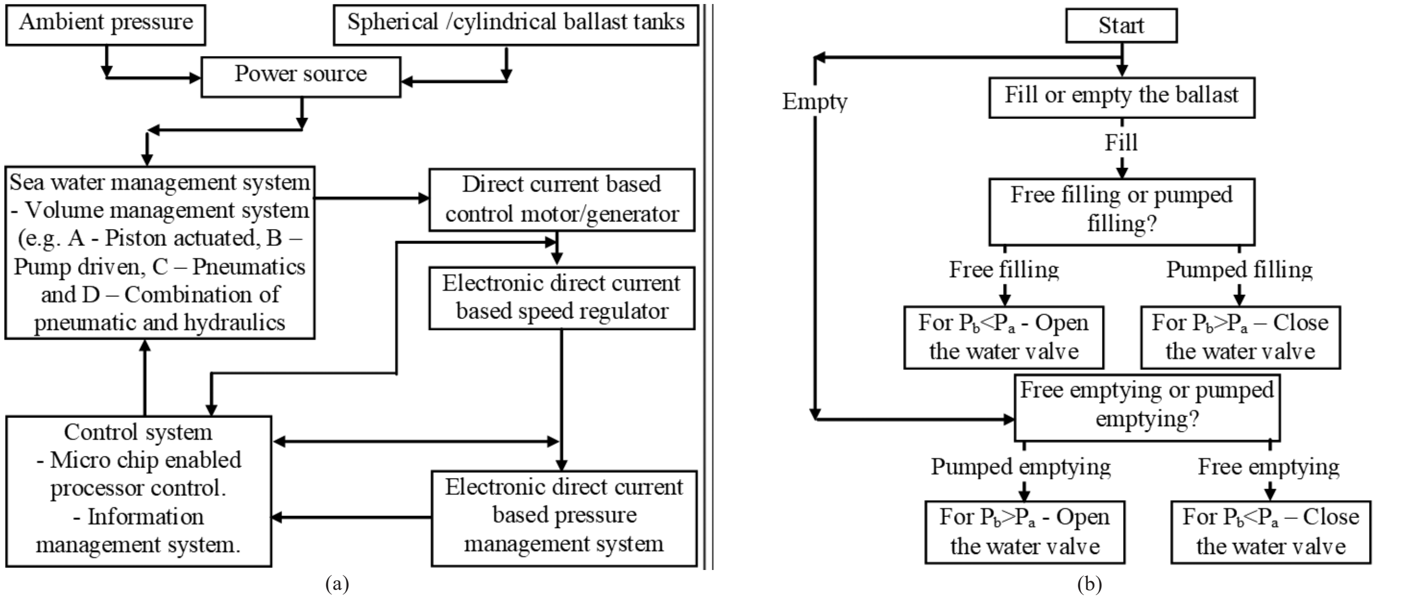


Figure 1. (a) Descriptions about our conceptual design approach and (b) Control system mechanism of ballast tank.

and pump driven FBS. The main components of piston operated FBS are: linear actuator, cylindrical ballast tank, flow control valve, piston actuating inside ballast tank, DC motor, and the check valve, etc. These details are as shown later in Fig. 4 (a). The size of cylindrical ballast tank is a function of capacity of the required buoyancy change. The primary function of linear actuator is to transfer the rotary motion of motor shaft to the motion of translation of piston actuating inside the cylindrical ballast tank, i.e. to convert electrical energy to mechanical work. In our work, the size of ballast tank is computed by optimising length (l) to diameter (d) ratio of the ballast tank. The computed l/d ratios for the required capacities of 10 kg and 18 kg are as shown in Figs. 2 (a) and 2 (b) and we can observe that a favourable range of the l/d ratio is 1.30 - 1.90.

Normally, a fluid will move from high pressure to low pressure and this concept is utilised to ballast the UV. In this process the fluid potential energy (i.e. hydrostatic pressure) will be converted into the kinetic energy of the fluid, which can be used for free filling the ballast tank in order to change the buoyancy. This can be written as follows:

Fluid Kinetic Energy = Fluid Potential Energy

$$\frac{1}{2}\rho_{fluid} \times v^2 = \Delta P = P_o - P_i = (\rho_{fluid} \times g \times h + P_a) - P_i \qquad (1)$$

assuming that the $P_i = P_a$, then we get the following:

$$v = \sqrt{2 \times g \times h} \qquad (2)$$

and the mass flow rate through the valve can be given as follows:

$$\dot{m} = \rho_{fluid} \times \frac{\pi}{4} \times d_{valve}^2 \times v \qquad (3)$$

and now from Eqn. (3) we get the following:

$$d_{valve} = \sqrt{\dot{m} \Big/ \left(\rho_{fluid} \times \frac{\pi}{4} \times v\right)}$$

where P_o is the pressure outside the ballast tanks and it is equal to the sum of hydrostatic and atmospheric pressure, P_i is the pressure inside of the ballast tanks, P_a is the atmospheric pressure, ρ_{fluid} is the fluid density, h is the depth of operation, g is the gravitational acceleration, A_{valve} is the valve area, v is the fluid flow velocity and d_{valve} is the valve diameter. The computed ratios of the depth versus diameter of flow control valve for various mass flow rates are as shown in Fig. 3 (a). Here, our aim is to show that when the ballast tanks are filled freely or emptied freely then how the diameter of the flow control valve is required to be changed to either decrease or increase the buoyancy. The values of the required thrust (T)

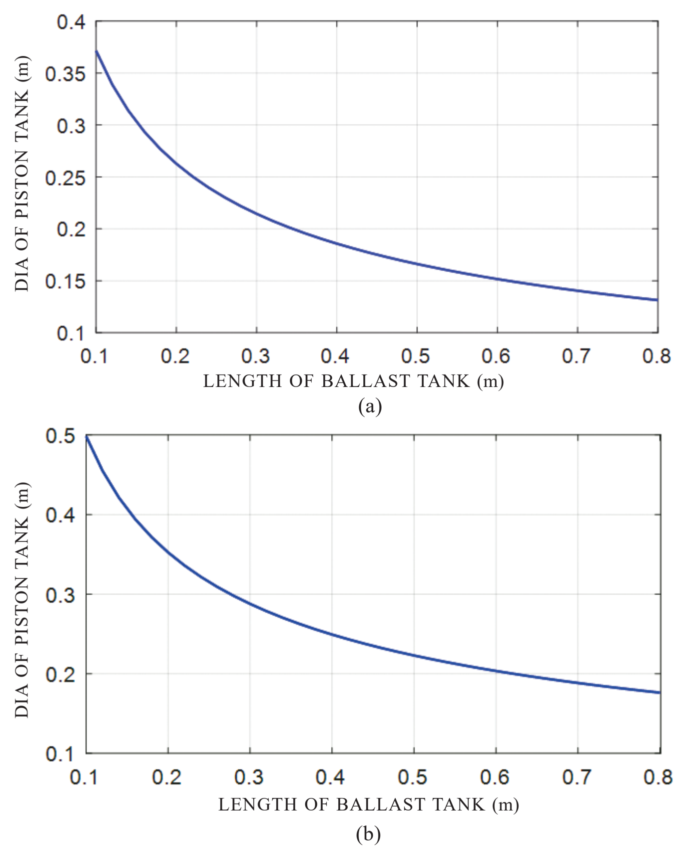
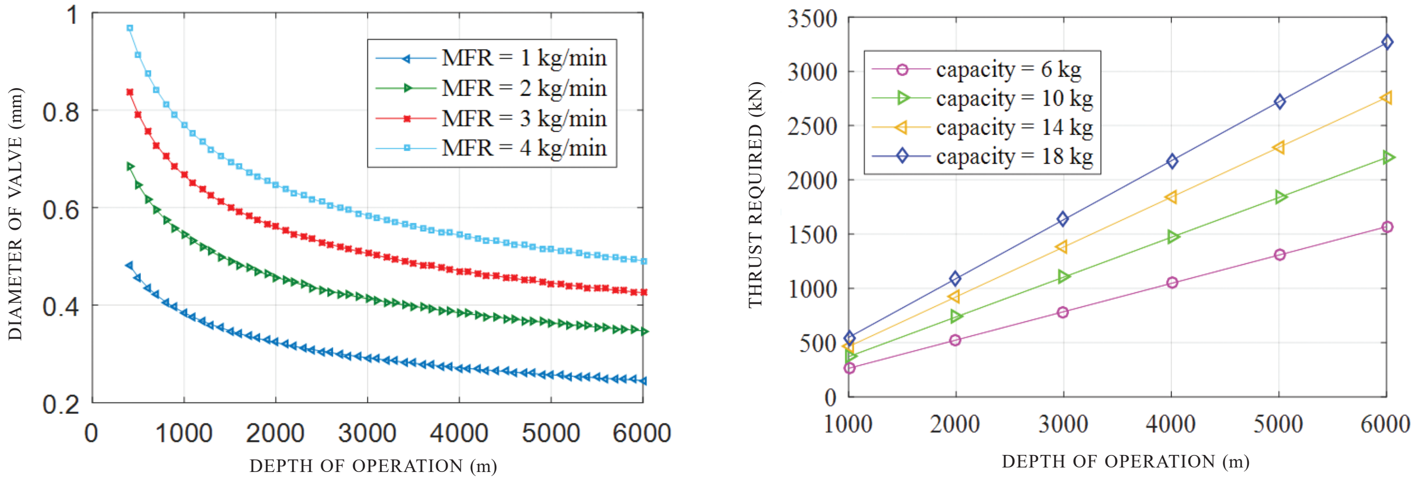


Figure 2. Computed l/d ratios for the (a) 10 kg and (b) 18 kg capacity ballast tanks.



(a) (b)

Figure 3. (a) Computed depth versus diameter of flow control valve for various MFRs and (b) Computed depth versus thrust required for various capacities of ballast tank.

are computed using a procedure from Wang[7], *et al.* and this is written as follows:

$$T = P_T \times A_{cp} = (P_a + \rho_{fluid} \times g \times h) \times$$
$$(0.25\pi \times d^2) = (P_a + \rho_{fluid} \times g \times h) \times (0.25\pi \times d^2) \quad (4)$$

and the net buoyancy of piston operated FBS (B_{PBFBS}) is defined as follows:

$$B_{PBFBS} = \forall \times \rho_{fluid} \times \eta_{vp}$$
$$= \frac{\pi}{4} \times d^2 \times l \times \rho_{fluid} \times \eta_{vp}$$
$$= \frac{\pi}{4} \times d^3 \times k \times \rho_{fluid} \times \eta_{vp} \quad (5)$$

and now by using Eqns. (5) in (4), we get the following:

$$T = \left(P_a + \rho_{fluid} \times g \times h\right) \times 0.25\pi \times$$
$$\left(B_{PBFBS} / \left(0.25\pi \times k \times \rho_{fluid} \times \eta_{vp}\right)\right)^{2/3} \quad (6)$$

where P_T is the total pressure, A_{cp} is the cross-section area of the piston moving inside the cylindrical ballast tank, k is the length to diameter ratio of the cylindrical ballast tank, η_{vp} is the volumetric efficiency of the piston operated FBS (i.e. the ratio of the volume of cylindrical ballast tank which can be filled/emptied to control the buoyancy to the overall volume of the piston operated cylindrical ballast tank) and other parameter are same as define before. From Eqn. (6) we observe that the required thrust at piston head is the function of ballast tank size and depth of operation and that increases with increase in depth of operation or with increase in capacity of the ballast tank. The thrust required at depth of 6000 m for a cylindrical ballast tank of capacity 18 kg at k = 1.46 is 3.3 MN, i.e. 3300 kN. The computed thrusts required versus depth for various capacities of ballast tank are as shown in Fig. 3 (b). Furthermore, the variation of depth versus thrust at various k is as shown in Fig. 4 (c). We can observe from Fig. 4 (c) that for a cylindrical ballast tank of the capacity of 18 kg, the thrust required at piston head is very high at the operating depth of 6000 m with k = 1.3 than with k = 1.9, e.g. with k = 1.9, at the operating depth of 6000 m the thrust is 2.737 MN and it is 788 kN less than the thrust with k = 1.3. However, it needs to be noted here that still the required thrust for piston driven system is very high and it increases exponentially with operating depth. Hence, we cautiously conclude that the piston driven system is energy efficient only at lower ranges of operating depths and for the higher ranges a pump driven system offers better energy efficiency.

Now, on a general note, for a piston based system we can see that the required thrust capacity is very high. Therefore the design of linear actuator which can produce this thrust will demand a large length and that will be difficult from strength and cost points of view. Furthermore, a large piston length cannot be accommodated inside a short vehicle. Also, a large linear system will have high chance of failure because large *l/d* ratio implies that the piston will tend to behave like a slender structure and under the requirements of high speed of movement and thrust it will likely fail in buckling. Hence, we cautiously conclude that the piston operated FBS can be used for fast buoyancy control, but it is restricted to small buoyancy capacity and low to medium range of depth of operation, i.e. less than 400 m. On the other hand, the pump driven FBS can be used for large buoyancy capacity and high depth of operations, i.e. more than 400 m.

In any FBS, the essential systems are: piston/pump with either hydraulic or mechanical actuation, direct current motor, valves for controlling incoming water and/or releasing water and mechanism for pressure control. In our designs the pump is with hydraulic and the piston is with mechanical actuations and water entry and exit are controlled through valves. The tanks are of e-fiber or glass fiber materials and of spherical shape for the pump driven FBS. The buoyancy is varied by changing the volume of water in the tanks. In the pump operated FBS, the main components are: Pump, a direct current motor, control and relief valves, and pressure intensifier. In our design, alternatively the water is drawn from the tank and pumped outside and via the processes of ‘pumping out’ and ‘pumping in’ with the reverse process, i.e. the buoyancy is varied by either ‘pumping in’ or ‘pumping out’ the water in to/from two spherical ballast tanks.

We assume that the half-filled tanks result into the neutral buoyancy of AUV and then we analyse the design for varying

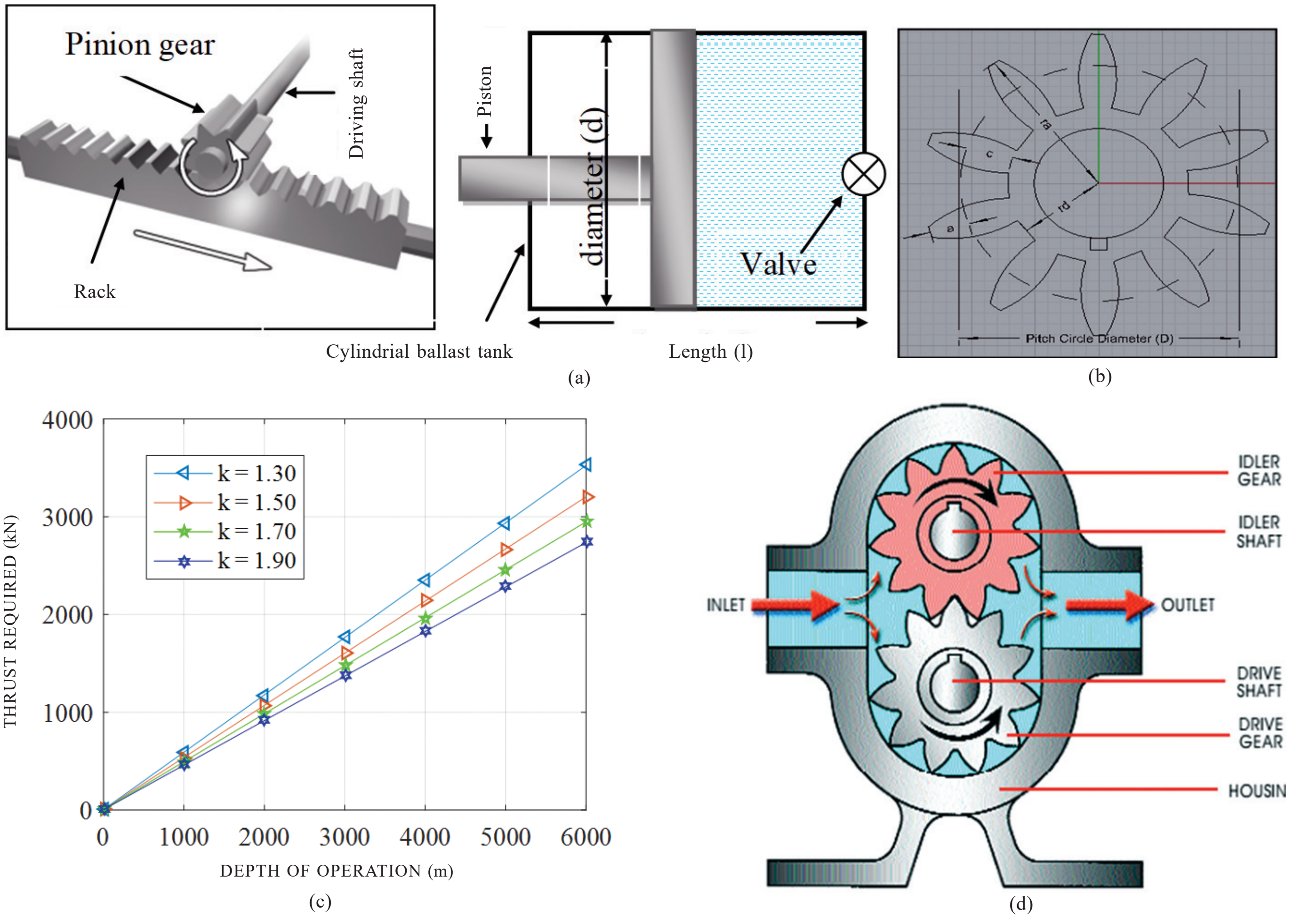


Figure 4. (a) Details of the major component of piston operated FBS, (b) Depiction of the r_a, r_d and *D* etc. for the pump, (c) Variation of depth versus thrust at various *l/d* ratios, and (d) A typical cross section of external gear pump adapted from Liquiflo[13] and Karassik[14], *et al.*

ambient conditions. The pump is highly powered and controlled either autonomously or in the master/slave mode through an embedded command system interface. The FBS is designed to 'pump in' and 'pump out' seawater from a ballast tank to the ambient and in this process the buoyancy is changed, i.e. either increased or decreased or kept neutral. The process of 'pumping in' is implemented through a valve of diversion. The fluid circulation around a circuit is used to actuate the pressure intensifier through hydraulic process and the motor drives pump, pump drives the pressure intensifier and the sea water is pumped out from the ballast tank. The conceptual design is as shown earlier in Fig. 1(a) and we control the system through a 'micro chip enabled controller' and 'electronic pressure transducer' that measures pressure. In the mode of stand-alone operations, a schedule of the mission is designed in prior through pre-programming. This pre-programming is implemented through command system architecture and stored on a memory card and read with an embedded software application. Also, this approach of pre-programming is used to select the initial parameters to start the operations and control the pump, motor and the operation of schedule. If the FBS is not required the program enters into the 'low power mode (LPM)' and enters into the mode of re-activation depending upon the operation of schedule. In the master/slave mode a command system protocol can be used for direct command control resulting into targeted command transfer from the AUV/AUG to the FBS and this result into direct control of the FBS. The primary specifications of the pump operated FBS are as listed in Table 1.

3. DESIGN CONCEPTS, APPROACHES AND SIMULATION RESULTS

In the pump operated FBS, the selection of pump for FBS mainly depends upon: Pressure against which it works and the MFR required. The other parameters that influence the selection are: Property and characteristics of fluid, i.e. density, viscosity, temperature, and corrosiveness, etc. The pumps are mainly classified: Positive displacement pump - PDP - (equal amount of discharge for each revolution) such as lobe pump, vane pump, and gear pump, etc.; and non-positive displacement pump - NPDP - in which volume of flow discharge is not constant such as centrifugal pump. The PDP is used for small volume discharge at high pressure and the NPDP is used for heavy amount of discharge and it works well even at low pressure as compared to the PDP. In our work we focus on PDP and the classification of PDP is adopted from Petersen and Jacoby [8] and accordingly it is listed in Table 2.

Table 1. Specifications of the pump operated FBS.

Depth rating	6000 m
Buoyancy capacity (B±)	18 kg
Capacity of the ballast*	0.0219 m^3
Actual capacity of pump (Q_{act})	1.2 ml/rev
Maximum speed	1200 RPM
Mass flow rate	1.47 kg/min
Sea water density	1025 kg/m^3
Power required	1.98 kW

*Total capacity of the ballast tank is higher than the buoyancy capacity of the tank because it can only be filled up to 80% only, i.e. considering the 20% margin for the ballast tank.

Table 2. Classification of the PDP

Positive displacement pump selection				
Type of PD pump	**Type of liquid**		**Abrasive**	**Differential pressure**
	Thin	**Viscous**		
Internal gear	G	E	G	G
External gear	G	G	P	E
Lobe	A	E	G	G
Vane	E	A	P	A

In Table 2, the A, E, G and P stand for the average, excellent, good and poor performance indices respectively. We can clearly observe that for high pressure difference, the external gear pump is excellent and for the pump operated VBS to be used at high water depth the external gear PDP is the most suitable option. Following Worall[9], *et al.* we define the following:

- Rate of change of buoyancy ($B\pm$): Mass flow rate of external gear positive displacement pump

$$= Q_{act} \times N \times \rho_{fluid} = Q_{design} \times N \times \rho_{fluid} \times \eta_{vol} \tag{7}$$

where η_{vol} is the volumetric efficiency of external gear positive displacement pump and from AIChE[10] it is defined as follows:

$$\eta_{vol} = \frac{Q_{act}}{Q_{design}} = \frac{Q_{design} - Q_{slip}}{Q_{design}} \tag{8}$$

where Q_{act} is the actual displacement volume by the pump per revolution (ml/rev), Q_{slip} is the amount of volume slip back from pressure to suction side due to low viscosity of the fluid or very high differential pressure, Q_{design} is the designed displacement volume of pump (ml/rev), N is the speed of motor (rpm) and ρ_{fluid} is the sea water density.

Now, following Egbe[11] and Chapple[12], for the design of external gear PD pump we define the following:

$$Q_{design} = \pi \times b \times (r_a^2 - r_d^2)/2 = \pi \times b \times D^2 \times (9n - 2.35)/8n^2 \tag{9}$$

$$r_a = 0.5D + a, r_d = 0.5D - c, a = D/n,\ c = 1.25D/n \tag{10}$$

where b is the face width, n is the number of teeth, r_a is the addendum radius, r_d is the dedendum radius, and D is the pitch circle diameter. The detailed description of the r_a, r_d and D, etc. is as shown in Fig. 4(b) for the pump. A typical cross section of the external gear pump adapted from Liquiflo[13] and Karassik[14], *et al.* is as shown later in Fig. 4 (d).

The variations in pump speed and on-board voltage with an increase in the ambient pressure are as shown in Figs. 5 (a) and 5 (b), respectively. The power supply from electrical control panel to the FBS is through a cable. The voltage monitoring is done for onboard panels and based upon the onboard voltage the pump speed is regulated. The initial setting of pump speed is done at the pre-selected ambient pressure and later when the ambient pressure increases then the total work required for pumping out the water increases, e.g. at the zero bar ambient pressure the pump speed is set at 1200 rpm with 21 V and this value is based upon the specifications that we are taken from the product catalogue of the motor GEMS[15] and ATO[16] .

The onboard voltage is expected to vary with changes in the ambient pressure, i.e. the voltage will decrease with an increase in the ambient pressure. It happens because the required power to change the buoyancy increases with an increase in the ambient pressure and the current drawn by the motor of pump also increases. This increase in the drawn current, results into the decrease in voltage. Overall efficiency η_o of the FBS is defined as the ratio of the output power (i.e. power delivered) to the input power (i.e. of the sum of the delivered power at a specific depth of operation and the power required to overcome the frictional/constant loss W_{fl}) and now we define the η_o as follows:

$$\begin{aligned}\eta_o &= \frac{W_{out}}{W_{in}} = \frac{\Delta p \times A \times v}{W_{out} + W_{fl}} \\ &= \frac{\rho_{fluid} \times g \times h \times (\dot{m}/\rho_{fluid})}{W_{out} + W_{fl}} \\ &= \frac{\dot{m} \times g \times h}{\dot{m} \times g \times h + W_{fl}}\end{aligned} \tag{11}$$

where Δp is the differential pressure, A is the cross-sectional area of the exit section of the pump, v is the flow velocity, W_{out} is the power delivered, ρ_{fluid} is the fluid density, g is the gravitational acceleration, h is the depth of operation and $\dot{m}$ is the mass flow rate. As shown in Eqn. (11), η_o of the FBS varies with the ambient pressure and at the low ambient pressure the system requires high power to counter the friction losses than the power delivered. If the ambient pressure increases, the power delivered increases but the friction losses remain constant. This will result in an increase in the system efficiency and this is as shown in Fig. 5 (c) for three different W_{fl} . These result in to 75 per cent, 80 per cent and 85 per cent efficiency and required power versus speed at various depths of operation is as shown in Fig. 5 (d).

The variation of speed versus MFR for different designs of pump is as shown in Fig. 5 (e) and we observe that for a designed PDP of four different capacities (e.g. 1.2 ml/rev, 2.4 ml/rev, 3.6 ml/rev, and 4.8 ml/rev) the MFR of pump increases with an increase in the speed of motor shaft that is connected to the drive gear of the PDP. The actual MFR of pump depends upon speed of the motor and volumetric efficiency of the pump. In this work we assume a volumetric efficiency of 80 per cent for the pump and with that we have report the results to control the mass flow rate and buoyancy of the vehicle.

Finally, the basic structure of micro chip enabled processor controller is as shown in Fig. 5(f). The basic architecture of the proposed computer aided design model of this work is modular and it is as shown in Fig. 5(g).

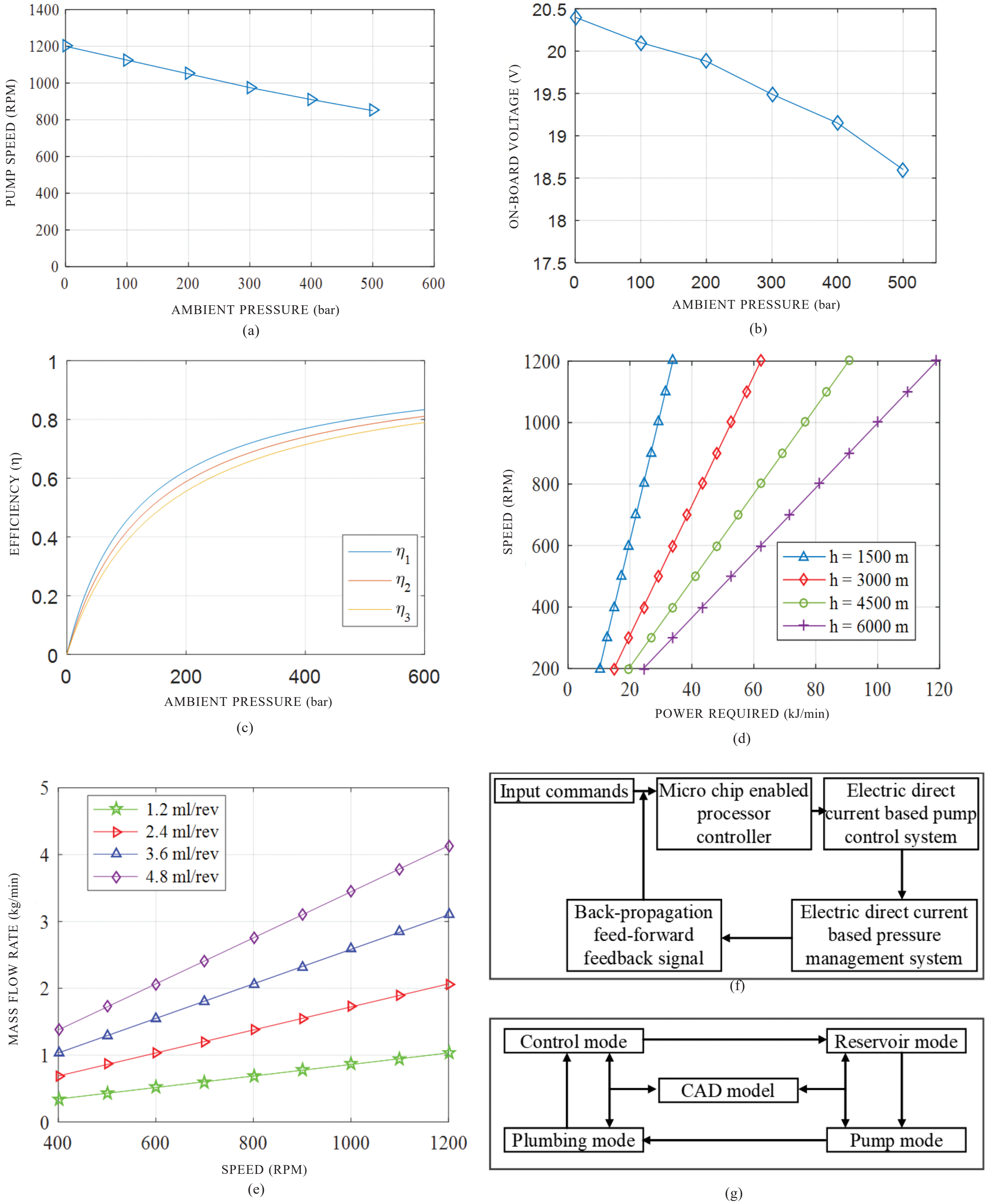


Figure 5. (a-b) Variations in pump speed and on-board voltage with an increase in the ambient pressure, (c) Computed efficiency versus ambient pressure, (d-e) Required power versus speed at various depths of operation and of speed versus MFR for different designs of pump, (f) Basic structures of micro chip enabled processor controller, and (g) The proposed computer aided design model.

4. CONCLUSIONS

This paper has presented a computing based simulation model for dynamic study of flexible buoyancy system (FBS) used for autonomous underwater vehicles/gliders (AUVs/AUGs). We considered two primary design alternatives for the FBS: piston and pump driven and both have been investigated in this paper. We computed the primary design of AUVs/AUGs from first principles of mechanics and have defined the basic design in the computer aided design model with implementation in Matlab*TM. Our model showed a seamless integration and through the presented design example for depth rating up to 6000 m we can conclude that the FBS is a feasible design concept which is applicable from low to medium to high ranges of operating depths. However, we have not reported a detailed energy efficiency analysis in this paper.

An important limitation of the present work is that it does not optimise any parameters that might be of critical interest in the design of UVs, e.g. neither minimisation of system weight or energy consumption. From the long range and endurance perspectives, it is important to design a FBS that offers the desired +/- B with lesser weight and lower energy consumption. Also, cost can be other parameter of interest. Furthermore, the product realisation is to be explored in the future.

REFERENCES

1. Sun, Q.; Zheng, R.; Ren, F.; Li, M. & Liang, H. The design and analysis of variable buoyancy system of AUV. *In* the Proceedings of 2nd Asia-Pacific Conference on Intelligent Robot Systems, 2017, Wuhan, China, pp. 259-263. doi: 10.1109/ACIRS.2017.7986104.
2. Medvedev, A.V.; Kostenko, V.V. & Tolstonogov, A.Y. Depth control methods of variable buoyancy AUV. *In* the Proceedings of IEEE Underwater Technology 2017 (IEEE UT2017) . Busan, South Korea, pp. 1-5. doi: 10.1109/UT.2017.7890333.
3. Sylvester, A.H.; Delmerico, J.A.; Trimble, A.Z. & Bingham, B.S. Variable buoyancy control for a bottom skimming autonomous underwater vehicle. *In* the Proceedings of IEEE MTS Oceans - St. John's 2014. doi: 10.1109/OCEANS.2014.7003058, 2014.
4. Sharma, R.;Kim, Bo-ram & Kim, Tae-wan Development of a computer-aided design (CAD) model for variable buoyancy engine (VBE) for autonomous underwater vehicles/gliders. *In* the Proceedings of the International Conference on Ship and Offshore Technology: Technical Innovations in Shipbuilding, RINA, UK, 2013, pp. 197-209.
5. Hibbeler, R.C. Engineering mechanics: Statics and dynamics. Ed. 14. Prentice Hall, USA, 2015.
6. Munson, B.R.; Rothmayer, A.P.; Okiishi, T.H. & Huebsch, W.W. Fundamentals of fluid mechanics. Ed. 7. Wiley, USA, 2012.
7. Wang, W.H.; Engelaar, R.C.; Chen X.Q. & Chase, J.G. The state-of-art of underwater vehicles - theories and applications. *In* Mobile Robots - State of the Art in Land, Sea, Air, and Collaborative Missions, edited by X. Q. Chen, ISBN: 978-953-307-001-8, InTech Publishers, UK, 2009.
8. Petersen, J. & Jacoby, R. Selecting positive displacement pumps. *Plant Engineering*, 1993, **7**, 138 - 140,
9. Worall, M.; Jamieson, A.J.; Holford, A.; Neilson, R.D.; Player, M. & Bagley, P.M. A variable buoyancy system for deep ocean vehicles. *In* the IEEE Oceans 2007 – Europe, pp. 1-6, 2007. doi: 10.1109/OCEANSE.2007.4302317
10. AIChE. Positive Displacement pumps: A guide to performance evaluation. Wiley-AIChE, USA, 2007.
11. Egbe, E.A.P. Design analysis and testing of a gear pump. *Res. Inventy: Int. J. Eng. Sci.*, 2013, **3**(2), 1-7.
12. Chapple, P. Principles of hydraulic systems design. Ed. 2. Momentum Press, USA, 2014. Http://vp.salesmrc.com/pdfs/Comparing_Four_Types_of_PD_Pumps.pdf (Accessed on 18 October 2018).
13. Liquiflo, External gear pump principle of operation and advantages. 2018. www.michael-smith-engineers.co.uk/mse/uploads/documents/Liquiflo-Catalog-2009-Engineering-Section.pdf (Accessed on 18 October 2018).
14. Karassik, I.J.; Messina, J.P.; Cooper, P. & Heald, C.C. Pump handbook. Ed. 3. McGraw-Hill Professional, USA, 2000.
15. Global Electric Motor Solutions (GEMS). Model no. GM86BLF. http://gemsmotor.com/12v-24v-48v-brushless-bldc-motor (Accessed on 18 October 2018).
16. Automation Technologies Online (ATO). Model no. ATO-130WDM10020.www.ato.com/2-kw-24v-brushless-dc-motor (Accessed on 18 October 2018).
17. Jensen, H.F. Variable buoyancy system metric. Massachusetts Institute of Technology and Woods Hole Oceanographic Institution, USA, 2009. pp. 1-112. (MS Thesis)
18. Dyer, I.; Taylor, R.E.; Newman, J.N. & Price, W.G. Concepts in submarine design. Press Syndicate, Cambridge University, UK, 1994.

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In the current study, he has implemented the formulation, performed all the analyses and prepared the final manuscript.

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In the current study, he has conceived the problem, provided overall guidance and supervision in the current study.

Appendix A

A.1 Following Jensen[17] and Dyer[18], *et al*. for a submerged/floating body the forces acting on it are defined as follows:

$$\Delta B = (F_B - F_G)/g = \nabla \times \rho_{fluid} - M \tag{12}$$

where ΔB is the net buoyancy force in kg, F_G is the gravitational force, F_B is the buoyant force because of the fluid on submerged/floating object, M is the mass of the floating object, ∇ is the volume of the fluid displaced by the object, and ρ_{fluid} is the density of the fluid. From Equation (12), we can observe that the ΔB can be controlled either by changing the M or by changing the value of ∇, because in general the fluid's density cannot be changed. Again, it is difficult to change ∇ because for a submerged body it is equal to the volume of body itself and that cannot be changed, therefore in our work we focus on change in the M .

A.2 Now, for a piston operated FBS (i.e. PBFBS), the net buoyancy can be controlled by either displacing the fluid from vehicle to outside (de-ballasting) or allowing fluid from outside to the vehicle (ballasting). The total buoyancy of PBFBS can be written as follows:

$$\begin{aligned} B_{PBFBS} &= \forall \times \rho_{fluid} \times \eta_{vp} \\ &= \frac{\pi}{4} \times d^2 \times l \times \rho_{fluid} \times \eta_{vp} \\ &= \frac{\pi}{4} \times d^3 \times k \times \rho_{fluid} \times \eta_{vp} \end{aligned} \tag{13}$$

where η_{vp} is the volumetric efficiency of piston operated FBS (i.e. the ratio of the volume of cylindrical ballast tank which can be filled/emptied to control the buoyancy to the overall volume of the piston operated cylindrical ballast tank), $\forall$ is the volume of piston driven cylindrical ballast tank, l and d are the length and diameter of the piston driven cylindrical ballast tank respectively, k is the length to diameter ratio of the piston and other parameters are as defined before.