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Depth Enhancement of an Underwater Towed System using Hydrodynamic Depressor

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ABSTRACT

The underwater towed system described here consists of tow cables, a towed body, an acoustic module and tail rope towed behind a surface ship. The required depth at a particular speed of the towing ship is obtained by paying out specified length of cable from the winch. However the excessive drag forces on the various components of the towed system results in impractically large values of cable length, especially at higher speeds. A hydrodynamic depressor is designed to improve the depth performance. The design is evolved based on numerical analysis and towing tank tests. Estimation of depth attained is carried out based on steady state theory of tow cables. Validation of the numerical analysis results is carried out through field evaluation of depressor performance during sea trial of the towed system.

Keywords: Towed system, hydrodynamic depressor, numerical analysis, towing tank tests, steady state theory of towing cables, sea trial

1. INTRODUCTION

Underwater systems are towed behind surface ships for various civilian, military and paramilitary applications. They are employed for geophysical explorations, coastal surveillance, patrolling of offshore installations and detection of underwater targets. The cables help to tow the system against the hydrodynamic forces and also facilitates electrical and signal transmissions between the component modules and the towing ship. Schematic of the towed system is shown in Fig. 1.

Oceanographic parameters such as temperature and salinity have significant influence on the propagation characteristics of acoustic waves through the sea water medium. These parameters are known to vary with respect to depth of water. During deployment of towed system in to sea, the optimum depth at which maximum acoustic performance can be obtained from the acoustic modules is calculated based on the prevailing oceanographic conditions. Hence it is important that the various acoustic modules of the towed system are placed at the required depth below the sea surface during towing operation. This is achieved by varying the speed of the towing ship or length of tow cable paid out from the winch. However, limitations on the length of cable that can be accommodated on the winch and the large drag forces impose restrictions on the maximum depth that can be achieved by the system.

Hydrodynamic depressor is a device which generates downward lift force and pulls the towed system to higher depth. Becker[1] reported the development of a high-speed hydrodynamic depressor for a surface ship towed sonar array. Rispin and Diggs[2] brought out the sea trial evaluation of hydrodynamic performance and stability of a depressor towed array system. Dynamic stability of the system is established up to speed of 30 knots. Wu and Chwang[3] developed an experimental set-up for monitoring the performance of an underwater towed system in a towing tank. The experimental set up was used for evaluating the hydrodynamic response of a two part towed system. Jithin and prakash[4] reported the numerical analysis of a hydrodynamic depressor which form part of a two part towed system. Numerical evaluation of depressor lift is reported to be in close agreement with experimental results. Wu and Chwang[5] proposed a hydrodynamic model of a two part underwater towed system in which a depressor is equipped with active control surfaces.

In the present case, the existing flat plate hydrodynamic stabiliser fins of the towed body are reconfigured to act as hydrodynamic depressors. The lift generated by the depressors at various speeds is estimated through numerical analysis and validated through experiments in towing tank. The effect of hydrodynamic depressors on depth is evaluated based on steady state theory of tow cables. The system is subjected to sea trials during which the analysis results are validated.

2. DESIGN OF HYDRODYNAMIC DEPRESSOR

Design of depressor is carried out by reconfiguring the stabiliser fins existing on the towed body. Since the towed body is required to pass through the handling system existing onboard the ship, outer dimensions of the depressors is required to be maintained the same as that of the existing stabiliser fins. NACA 0012 profile is adopted for the depressor cross section[6]. NACA0012 sections are thick enough to meet the structural strength requirements of the present project and have relatively constant centre of pressure. They also offer reasonable drag

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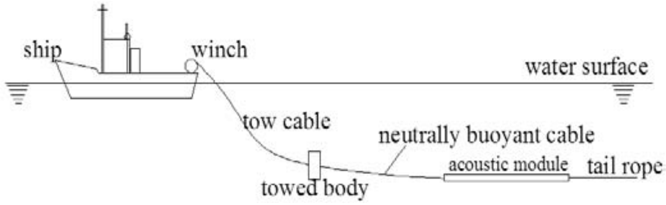

Figure 1. Schematic of the towed system.

characteristics[7]. Dimensions of the depressor are shown in Fig. 2. It has a span of 900 mm and an angle of attack of 9.0° is provided. Since the space constraints in the system does not permit installation of a mechanism to change the angle of attack and considering the stall angle of the NACA 0012 section which is around 12.0° a permanent angle of attack of 9.0° is chosen. The depressor is permanently welded to the towed body symmetrically at the centerline through suitable structural members.

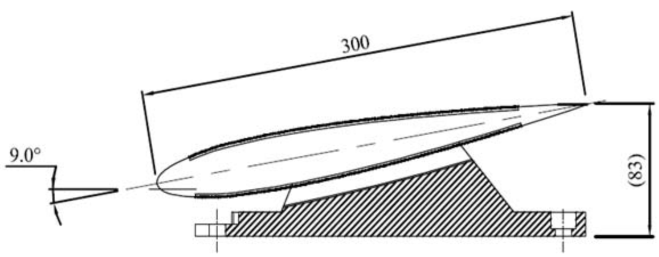

Figure 2. Dimensions of the depressor (mm).

3. NUMERICAL ANALYSIS OF DEPRESSOR

Lift generated by the depressor is estimated through computational fluid dynamics analysis using ANSYS FLUENT software. Steady state analysis is carried out using implicit formulation and standard k-ε turbulence model. Standard k-ε turbulence model is chosen since the fluid flow over the depressor is not expected to have much separation and no sharp variations in the mean strain rate. This is model is known to give satisfactory results under such flow conditions. In the present case the experimental results justify the application of this turbulence model. SIMPLE scheme is used for the pressure-velocity coupling[8]. Structured hexahedral mesh is used with a minimum size of about 1mm near the wall, the size gradually increasing away from the wall.

The inlet of the computational domain is located at two times the chord length of the depressor. The outlet is located at five times chord length. Top and bottom of the domain are located at two times chord lengths. Grid independence of the results is ensured by systematically increasing the mesh density. At the inlet, velocity vector is defined and the outlet is specified with outflow boundary condition. The analysis is carried out for speeds varying from 2 knots to 12 knots. In the present analysis, only the flow over the depressor is considered. The presence of towed body is not included in the computational model. Figure 3 shows the mesh around the depressor and Fig. 4 shows the velocity profile at 10 knots which correspond to a Reynold's number of 1.93 x 10^6 and a Froude's number of 2.998 wrt chord length. The analysis is carried out for an angle of attack of 9.0°. In Fig. 4, the flow direction is from right to left and the filled contour plot indicates the flow velocities in

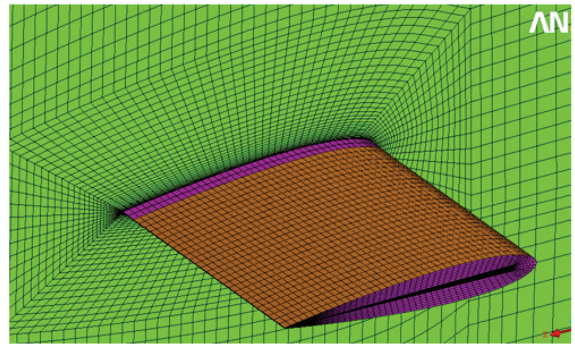

Figure 3. Mesh around the depressor.

Figure 4. Velocity distribution (m/s) at 10 knots.

m/s at different points around the depressor on a vertical plane passing through the centre line of the depressor. Results of numerical simulation are shown in Fig. 6.

The lift to drag ratio of NACA0012 depressor obtained from CFD analysis is about 75 at an angle of attack of 9.0°. This indicates a drag value of 87 N compared to a lift value of 6500 N at 12 knots. Hence the influence of drag force is neglected in the analysis.

4. EXPERIMENTAL MEASUREMENT OF LIFT FORCE

The numerical analysis results are validated through experimental measurement of lift force in towing tank. The tests are carried out in the high speed towing tank (HSTT) at Naval Science and Technological Laboratory, Visakhapatnam, India. Major features of HSTT are given below.

Length	: 500 m
Breadth	: 8 m
Depth	: 8 m
Maximum speed	: 20 m/s

A single linear variable differential transformer (LVDT) based submerged force gauge is used to measure the lift force. This gauge has a capacity to measure lift force up to 2000 N. Parameters such as drag and moments are not measured in the present experiment. Acquisition of experimental data and post processing are done onboard the towing carriage by a PC based 16 Channel data acquisition system and LabVIEW 6.1v software with the necessary analysis software developed in-house. The speed of towing carriage is measured by a wheel with an optical encoder.

During experiment, the full scale towed body fitted with depressor is rigidly attached to the towing carriage using a streamlined vertical strut. The streamlining is provided to the strut to minimize the interference effects. The modular force gauge is mounted on a horizontal plate inside the towed body at its mid depth and at mid length. The bottom face of the force gauge is attached to the horizontal plate and the top face of the gauge is attached to the vertical strut. Figure 5 shows the towed body fitted with the depressor undergoing lift force measurement in the towing tank. The top of the vertical strut is fixed to the towing carriage. Using this test set up, the downward lift force acting on the towed body due to the depressor is measured during the forward motion of the carriage at various speeds. Accuracy of the force gauge is established through calibration using known weights prior to the towing experiment. The resultant of weight and buoyancy of the towed body are subtracted from the measured lift force. Figure 6 shows the plots of lift force due to depressor obtained from numerical simulation and towing tank measurements at various speeds.

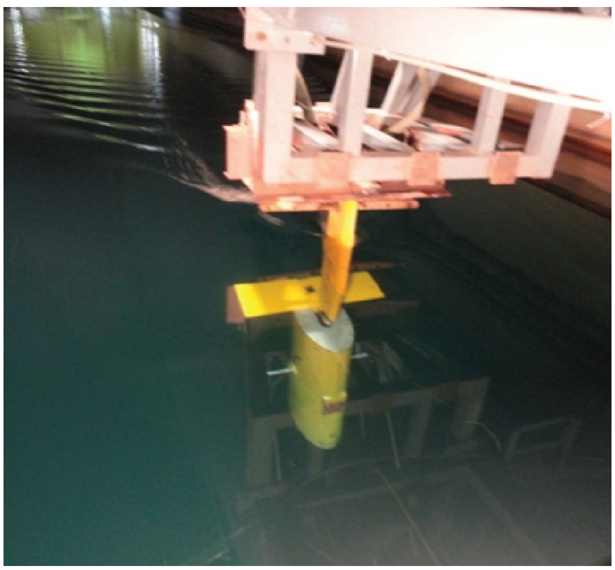

Figure 5. Towed body with depressor during test.

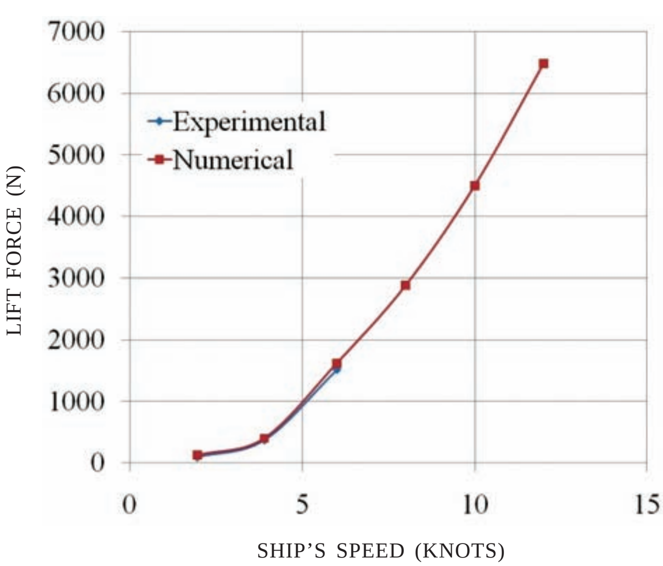

Figure 6. Lift force *vs* ship's speed.

5. ESTIMATION OF DEPTH ACHIEVED BY TOWED SYSTEM

Pode[9] carried out pioneering research work in the area of steady state analysis of underwater cables. He brought out detailed tables which facilitate the determination of configuration and tensions of a flexible cable moving in a fluid. In the present analysis, the depth attained by towed system is estimated based on steady state theory of tow cables using OrcaFlex software[10]. OrcaFlex is a non-linear finite element program which is capable of dealing with large deflections of flexible cables. Lumped mass element is used for analysis. The long cable is divided in to a number of small mass less segments. A node is positioned at each end of the segment. Properties such as mass, weight, buoyancy and hydrodynamic drag of the segments are lumped to the end nodes. Bending stiffness is represented by rotational spring dampers between the nodes and segments. 3D buoys available in the OrcaFlex software are used to model the towed body and tail rope. Figure 7 shows the arrangement segments, nodes and bending spring dampers in the finite element model[10].

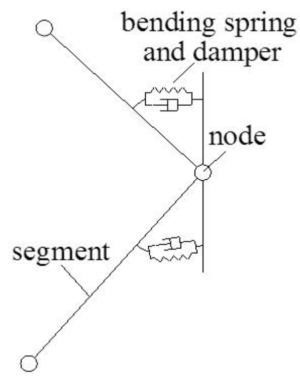

Figure 7. Arrangement of bending spring dampers in the finite element model.

The depressive force estimated from CFD analysis at various speeds is added to the downward weight force of the towed body in the Orcaflex model. Figure 8 shows the converged spatial geometry of the towed system simulated by the OrcaFlex model at a steady speed. The model shows the ship moving from left to right, towing behind it, the underwater system consisting of the tow cable, the towed body, the neutrally buoyant cable, the acoustic module and the tail rope in sequential order starting from the ship. The tail rope is modelled as a lumped mass and hence is not visible in the model.

Parameters used for the analysis are shown in Tables 1 and 2. In the steady state analysis of the towed system, the catenary shape assumed by the tow cable at a specific speed is decided by the hydrostatic and hydrodynamic forces experienced by the various components of the towed system attached to the end of the tow cable. Hence the tension estimated at the ship end of the tow cable includes contributions from weight, buoyancy and drag contributions from tow cable and all other components of the towed system. Figures 11 to 14 shows the comparison of depth and tension predicted based on numerical analysis compared with sea trial data for 130 m and 250 m pay out lengths of the tow cable and at ship's speeds of 8 knots, 10 knots, and 12 knots.

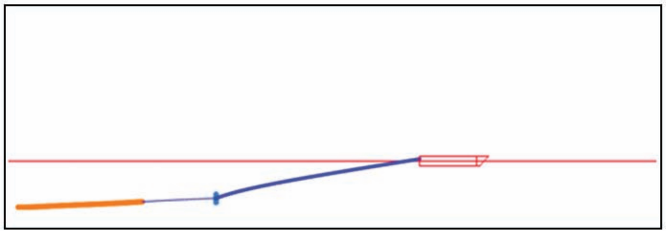

Figure 8. Orcaflex model of the towed system.

Table 1. Line parameters

Parameter	Tow cable	Neutrally buoyant cable	Acoustic module	Tail rope
Length (m)	50-500	190	165	30
Diameter (mm)	31	32	82	31
Mass in water (kg/m)	2.60	0	0	0
Normal drag coefficient	1.0	1.0	1.0	1.0
Axial drag coefficient	0.002	0.002	0.002	.003
Bending Stiffness (kNm^2)	7.3	-	-	-

Table 2. Parameters of towed body

Parameter	Value
Frontal area (m^2)	0.54
Wetted surface area (m^2)	2.43
Block coefficient	0.76
Drag coefficient	0.50
Weight in water (kg)	400
Buoyancy (free flooded)	Negligible

6. EVALUATION OF HYDRODYNAMIC PERFORMANCE OF DEPRESSOR THROUGH SEA TRIAL

Hydrodynamic performance of the depressor is evaluated through sea trial of the towed system on board a ship. The ship is equipped with specialised handling systems for deployment and retrieval of the complete towed system including the towed body fitted with depressor. The total length of the towed system is wound on a winch prior to the commencement of the sea trial. The system is then deployed systematically and then towed behind the ship at various specified cable payout lengths and speeds. Depth sensors are attached in the towed body and depth data are measured under steady state conditions. Load cell attached to the winch system measures the total tension at the winch. The measured value of total tension at the winch includes the effect due to all hydrodynamic and hydrostatic loads acting on the various components of the towed system. Sea state was approximately 2 during the trial. Figure 9 shows the handling system deploying the towed body fitted with flat plate stabiliser fins during the sea trial. Figure 10 shows the deployment of towed body where in the flat plate stabiliser fins replaced by NACA0012 depressor.

The sensor used to measure depth is a compact microprocessor-controlled temperature, depth, pitch and roll recorder with electronics housed in a water-proof housing. Calibration of the sensors is carried out prior to the sea trials. Depth accuracy of the sensors is +/- 0.4 per cent of the selected range which is 200 m during the present sea trials.

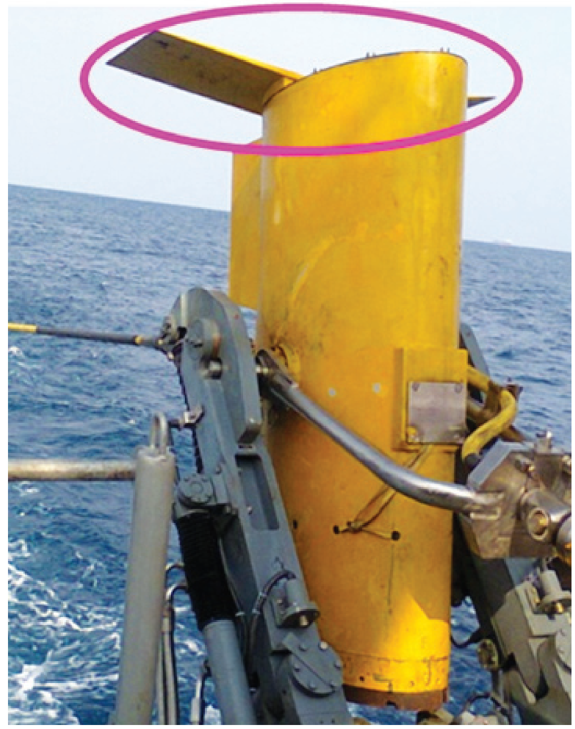

Figure 9. Towed body with flat plate fins.

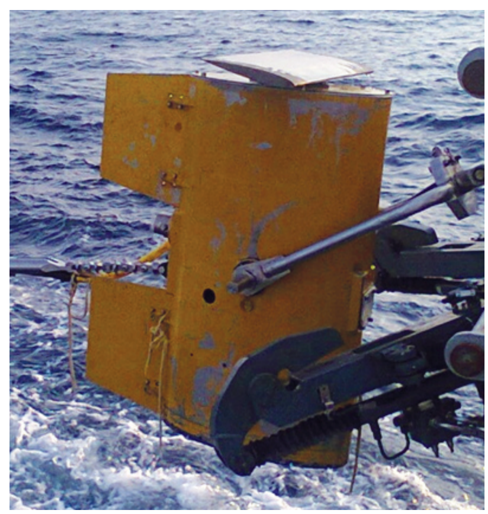

Figure 10. Towed body with depressor.

7. RESULTS AND DISCUSSION

In Fig 6, the lift force estimated based on numerical simulation is compared with the measured values obtained during towing tank experiments. Due to limitations in the towing tank facility, the towing tests could be conducted in the speed range of 2 knots to 6 knots only. However the low speed experimental values are used to validate the numerical analysis results. It is observed that the numerical and experimental results are in agreement with maximum deviation of about 6 per cent at 6 knots. Numerical estimates are found to be slightly on the higher side at all speeds. Probable reason for this could be due to the presence of towed body in the experimental set up. The presence of towed body in the vicinity of the depressor may have some influence on the flow around the depressor. In addition, the vertical strut used in the experimental set up, which is used to rigidly connect the towed body to the towing carriage, will also modify the flow around the depressor to some extent. In the numerical model of the depressor, effects due to both the towed body and the vertical strut are not considered.

Figures 11 and 12 show the depth attained by the towed system with and without depressor when the cable pay out length is 130 m and 250 m, respectively. Figures 13 and 14 show the tension on the winch. It is observed that the numerical results and sea trial data are in agreement. It is also observed that significant improvement in depth is achieved through the use of depressor.

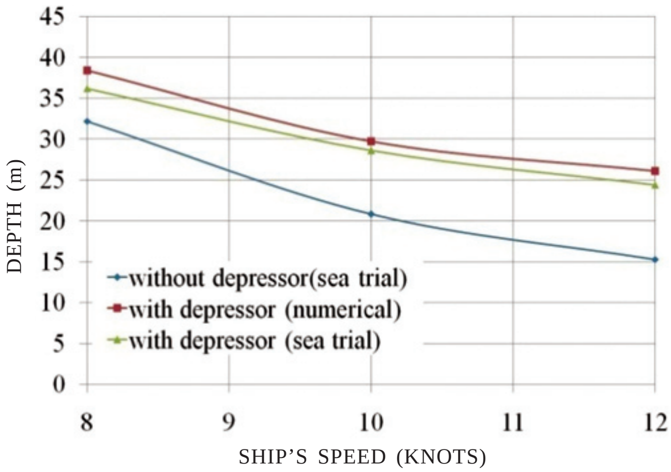

Figure 11. Depth *vs* speed (130 m cable pay out).

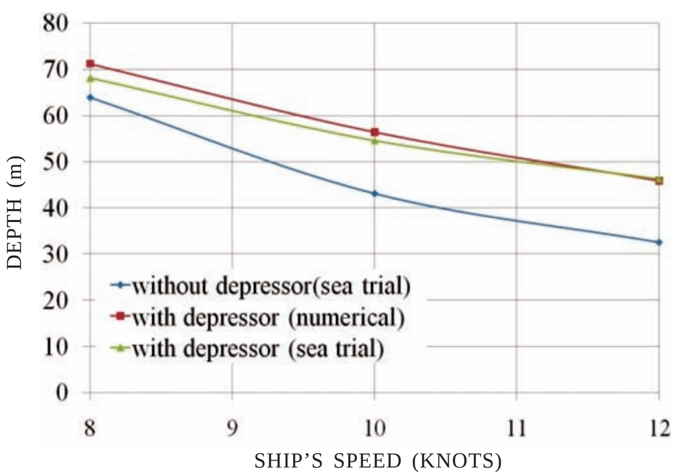

Figure 12. Depth *vs* speed (250 m cable pay out).

8. CONCLUSIONS

Hydrodynamic depressor is employed successfully to improve the depth attained by an underwater towed system. Existing flat plate stabiliser fins are replaced with hydrodynamic depressor having NACA0012 cross section maintaining the overall dimensions same. Lift force generated by the depressor is estimated through numerical analysis for various speeds. The lift force estimated through numerical analysis is validated through towing tank measurements. The numerical and experimental results are found to be in agreement.

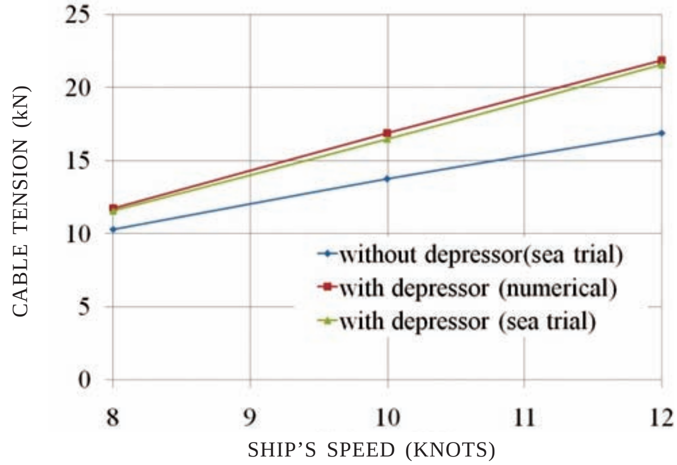

Figure 13. Tension *vs* speed (130 m cable pay out).

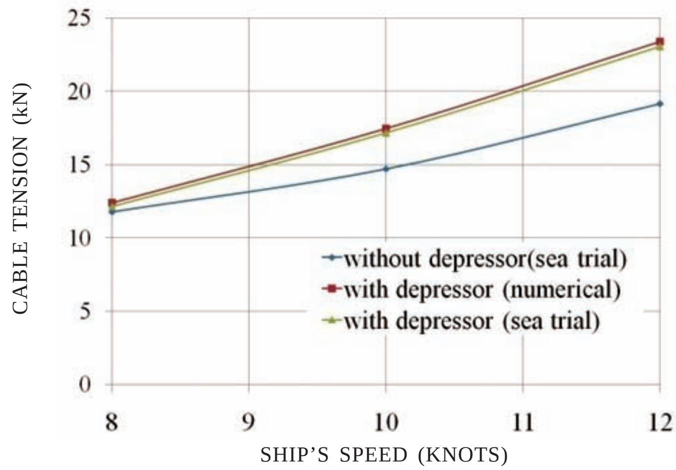

Figure 14. Tension *vs* speed (250 m cable pay out).

The depth attained by the towed system is numerically estimated based on steady state theory of tow cables. The towed system is subjected to sea trial with and without depressors and the effect of hydrodynamic depressor on the depth attained by the system is quantified through depth measurement. The numerically estimated depth and cable tension values are found to be in agreement with the data obtained during sea trial.

Depth data collected during the sea trial indicate that the effect of depressor increases the depth attained by the towed system by 12 per cent at 8 knots speed and by 60 per cent at 12 knots while the cable pay out length is 130 m. Similarly, the depressor increases the depth by 6.5 per cent at 8 knots speed and by 42 per cent at 12 knots while the cable pay out length is 250 m. The significant percentage increase in depth at high speeds is due to the increased lift generated by the depressor which varies as the square of the speed.

Presence of hydrodynamic depressor is found to increase the cable tension acting on the winch. This is because, the net force acting on the towed body increases as the depressor generates additional downward force.

REFERENCES

1. Becker, R.F. High speed sonar array depressor program final report. Prepared for Office of Naval Research, 1950.
2. Rispin, P. & Diggs, J.S. At sea evaluation of hydrodynamic performance of a simulated depressor-towed array. Office of Naval Research, 1981, Report No. DTNSRDC/SPD-**0953**-01.
3. Wu, J. & Chwang, A.T. Experimental investigation on a two-part underwater towed system. *Ocean Engineering.*, 2001, **28**, 735–750.
4. Jithin, P.N. & Senthil Prakash, M.N. Numerical investigation of the hydrodynamic behavior of the depressor. *Int. J. Mech. Eng. Robotics*., 2014, **3**(1). 256-264.
5. Wu, J. & Chwang, A.T. Investigation on a two-part underwater maneuverable towed system. *Ocean Engineering*, 2001, **28**, 1079–1096
6. Abbot, I.H. & Von Doenhoff, A.E. Theory of wing sections. Dover Publications., 1960.
7. Lewis, E.V. Principles of naval architecture. *SNAME*, 1989, **3**.
8. Fluent Version 6.2. User Manual., 2005.
9. Pode, L. Tables for computing the equilibrium configuration of a flexible cable in a uniform stream. David Taylor Model Basin, 1951, Report **687**.
10. Orcaflex Manual. Orcina Limited, 2013.

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CONTRIBUTORS

Mr P. Vinod obtained his BTech (Mechanical Engineering) from College of Engineering, Trivandrum and MTech (Mechanical Engineering) from IIT, Madras. Presently he is heading the Ocean Engineering Division at Naval Physical and Oceanographic laboratory, Kochi. His areas of interest include: Design and development of sonar handling systems, underwater towed bodies and underwater housings. He has taken the lead role in this work and put forward the idea of augmenting the depth attained by the towed system through various possible hydrodynamic means. He has drawn up the design envelop within which the depressor has to operate.

In the current study, he planned and coordinated with various agencies to facilitate the experimental and sea trial parts of the work.

Mr Roni Francis, obtained his graduation in Naval Architecture from Cochin University of Science and Technology and MSc from Indian Institute of Technology, Madras. Presently he is working as scientist at Naval Physical and Oceanographic Laboratory, Kochi where he is involved in the design and development of underwater systems. His area of research includes: Hydrodynamics of towed systems and computational fluid dynamics. He has published 12 papers in national and international journals and conferences.

In the current study, he generated the design and drawings of the hydrodynamic depressor and evaluated its performance through CFD and orcaflex analysis. He coordinated the fabrication of the depressor. He also conducted the experimental evaluation of the depressor.

Mr P. Prabhasuthan, obtained his BTech (Production Engineering) from Sree ChithraThirunal College of Engineering, Thiruvananthapuram. He is currently working as Scientist at Naval Physical and Oceanographic Laboratory (NPOL), Kochi. His significant contributions and interest are in the field of designing and developing SONAR towed bodies and handling systems.

In the current study, he designed, developed and coordinated the fabrication of the towed body on to which the hydrodynamic depressor is fitted. He also performed the assembly, integration and testing of the total system. He planned, coordinated and conducted the various sea trials of the system.

Dr O.R. Nandagopan, received BE (Mechanical Engineering) from Thiagarajar College of Engineering, Madurai, in 1983, ME (Production Engineering) from Anna University, Chennai, in 1985 and PhD (Structural Response of water backed Perforated plate for Underwater Non-contact Explosion) from Cochin University of Science & Technology, Kochi, in 2011. Presently he is the Director at Naval Science and Technological Laboratory (NSTL), DRDO, Kochi. He is Fellow of Institution of Engineers (India) and Member of Ocean Society of India.

In the current study, he took lead role in the material selection, structural design and analysis of the towed body and the hydrodynamic depressor. He worked out and implemented the development plan of the total system for various stages of design, development, fabrication, experiments and sea trials.