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Towing characteristics of T.S. Seiyo-Maru Ⅱ, Tokyo University of Fisheries*

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Abstract: The training ship Seiyo-Maru Ⅱ of the Tokyo University of Fisheries was built in 1987. Among its propulsion characteristics, only estimated power curves obtained through water tank tests during construction are available, whereas its towing characteristics, required for the studies of fishing gears and oceanographic equipments and instruments have been left unknown. Therefore, the ship speed *Vs*, rotational speed of the main engine *N* in rpm and shaft horsepower *SHP* were measured to obtain the wake coefficient w, propulsive efficiency η and effective horsepower curves (*EHP* curves) using propeller efficiency η_0, all of which are parameters required to determine the ship's towing force. The relationships were determined among the ship's speed *Vs*, towing force in tonf, main engine power output *PS* and main engine speed *N*.

The results of these calculations showed that the towing force of the ship is 5.6~6.8 tonf at a propeller blade angle θ of 20 degrees, a shaft horsepower of 800 *PS* and a ship speed of 3.0~6.0 knots.

1. Introduction

The T/S Seiyo-Maru was built in 1987, having the gross tonnage of 167 t, the design full load displacement of 385.6 t, the main engine horsepower of 1,050 *PS* at 850/330 rpm, the diameter of the four-bladed propeller of 1.85 meters, and the skew angle of 25 degrees. Table 1 shows the ship's principal dimensions.

The estimated power curves were plotted by West Japan Fluid Engineering Laboratory Co., Ltd., the propeller manufacturer, using effective horsepower *EHP*, wake coefficient w, thrust deduction coefficient t, propulsive efficiency η, and thrust coefficient Kt and torque coefficient Kq of the manufacturer's propeller. Howeve, the towing characteristics required for the oceanographic survey and the netted fishing gear were not known. In 1992, as corrective measures for the ship, rubber mountings were installed to the main engine bed to improve vibration-damping performance and noise levels in the hull.

To investigate the ageing after construction and the towing characteristics, the speed *Vs* and main engine horsepower *SHP* of the ship were measured at sea by, taking propeller blade angle θ and rotational speed *N* in rpm of the main engine as parameters. The results of these and propeller efficiency η_0 were used to determine the wake coefficient w, propulsive efficiency η and effective horsepower *EHP* curves, all of which are required for estimating the towing characteristics.

The results of the estimation are reported in this paper together with some technical expertise gained.

2. Methods of measurements and results

Measurments were carried out in the sea area near Ukishima in Tokyo Bay on 6 July 1993. Sea conditions were calm and suitable during measurements. The draft fwd was 2.40 m, and aft 3.47 m. The hydrostatic curves and Bonjean curves show that the ship was approximately in conditions corresponding to its design full-load displacement.

Measurements were carried out at propeller blade angles θ of 14.0, 16.0, 18.0, 20.0 and 22.0 degrees at the indicator on the engine room console. For each of these blade angles, the ship was tested at main engine speed *N* of 850, 800, 750, 700 and 650 rpm.

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Table 1. Principal dimensions of hull, engine and propeller.

Hull		
Gross tonnage	GT	160.00 t
Length over all	Loa	35.50 m
Length between perpendiculars	Lpp	31.00 m
Breadth (moulded)	B	7.00 m
Depth (moulded)	D	3.40 m
Block coefficient	Cb	0.653
Main engine		
Type		NIGATA 5PA5L
Max. continuous output		1,050 PS×850/330 RPM
Propeller		
Diameter		1.850 m
Expanded area ratio		0.65
Skew angle		25°
Number of blades		4

Table 2. Relationships among blade angle θ, ship speed *Vs* at the main engine rotational speed *N* in rpm and measured values of *SHP* (*ps*).

Blade angle (θ)	*N* (rpm)	*Vs* (knots)	*SHP* (ps)
14.0°	650	7.30	171
	700	7.90	217
	750	8.40	267
	800	9.05	325
	850	9.40	400
16.0°	650	7.95	212
	700	8.65	256
	750	9.15	333
	800	9.60	414
	850	9.95	510
18.0°	650	8.75	261
	700	9.25	334
	750	9.70	423
	800	10.05	529
	850	10.40	651
20.0°	650	9.30	328
	700	9.75	422
	750	10.15	536
	800	10.50	671
	850	10.90	824
22.0°	650	9.75	412
	700	10.15	533
	750	10.55	676
	800	10.95	843
	850	11.40	1,032

The JRC's Doppler log, model "JNA-761" and the Furuno LORAN C "CI20H" were used to measure the speed *Vs* of the ship. To measure shaft horsepower *SHP*, the Soyo Engineering Co.'s horsepower meter was used.

To mitigate the adverse effects of tidal currents during measurements, the ship was maneouvred along the reciprocal courses between two points in the same sea area. Table 2 shows the mean values of the measurements.

3. Methods of analysis

Fig. 1 is a representative flowchart for determining relationships between ship speed, effective horsepower and main engine horsepower on the basis of propeller data (Koike *et al.*, 1992, 1993). When the ship proceeds at a propeller blade angle θ (degrees) and a speed *Vp* (m/s),

$$V \downarrow \quad Vp = V(1-W) \downarrow \quad J = Vp/nD$$

$$Kt \downarrow \quad T = Kt \cdot \rho n^2 D^4 \downarrow \quad R = T(1-t) \downarrow \quad EHP = RV/75$$

$$Kq \downarrow \quad Q = Kq \cdot \rho n^2 D^5 \downarrow \quad P = 2\pi n Q \downarrow \quad SHP = P/(75 \cdot \eta)$$

Fig. 1 Flowchart for determination of the relationships among parameters.

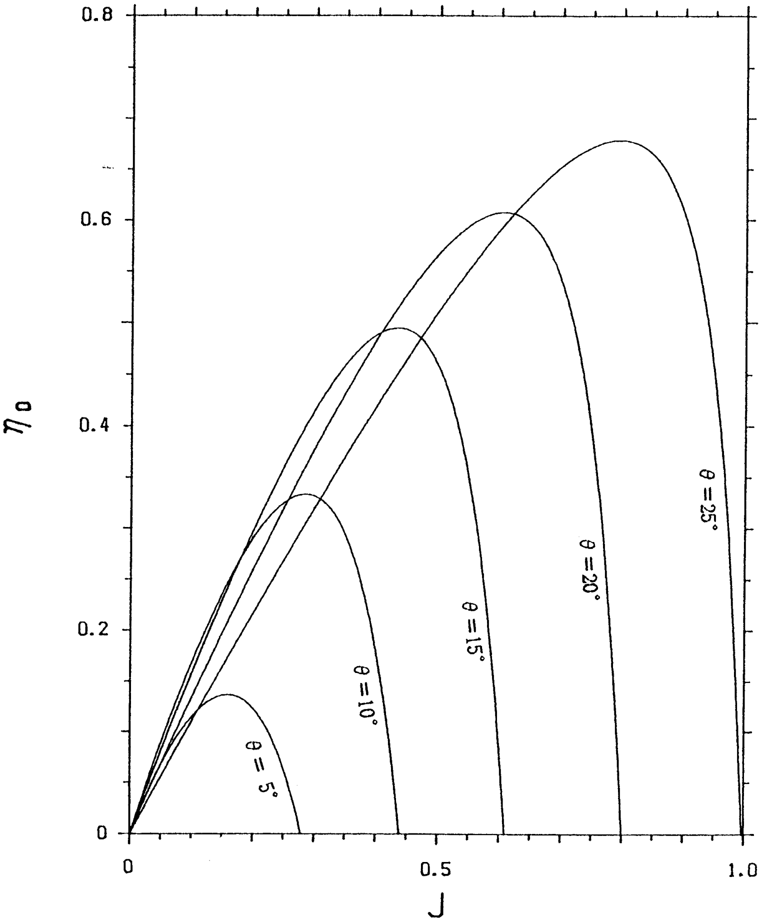

Fig. 2. Typical curves of popeller efficiency η_0.

the advance speed Vp (m/s) of propeller can be determined from wake coefficient w, and the propeller advance coefficient J, from propeller diameter D (m) and rotational speed n (rps). In these calculations, the propeller thrust coefficients Kt on the left and Kq on the right were determined by polynominal of J and propeller pitch ratio P/D $(0.7\pi\ \tan\theta_0)$:

$$Kt=\sum_{j=0}^{r}\sum_{j=0}^{s} A_{ij}\,(J)^i\,(P/D)^j, \qquad (1)$$

$$Kt=\sum_{j=0}^{r}\sum_{j=0}^{s} A_{ij}\,(J)^i\,(P/D)^j, \qquad (2)$$

where A_{ij} and B_{ij} are the coefficients of the polynominals for Kt and Kq, r and s are the orders of the polynominal for J and P/D (value=3 for r and s). Fig. 2 shows the coefficients Kt and Kq used for the ship as propeller efficiency η_0.

When the Kt value on the left is known, the overall hull resistance R (kgf) can be determined from liquid density ρ (kg/m^3), and n and D by way of thrust T (kgf) and thrust deduction coefficient t. The values of overall resistance R and ship speed V are used to calculate effective horsepower PS.

When the value of Kq on the right is known, torque Q (kgf-m) and power P (kgf-m/s) can be determined from Kq, ρ and n, and furthermore, shaft horsepower SHP (PS) can be determined from propulsive efficincy η.

Unkonown factors in the flowchart are wake coefficient w, coefficients of the propeller Kq and Kt, thrust deduction coefficient t and propeller efficiency η.

Propulsive efficiency η is as below:

$$\eta=\eta_R \cdot \eta_T \qquad (3)$$

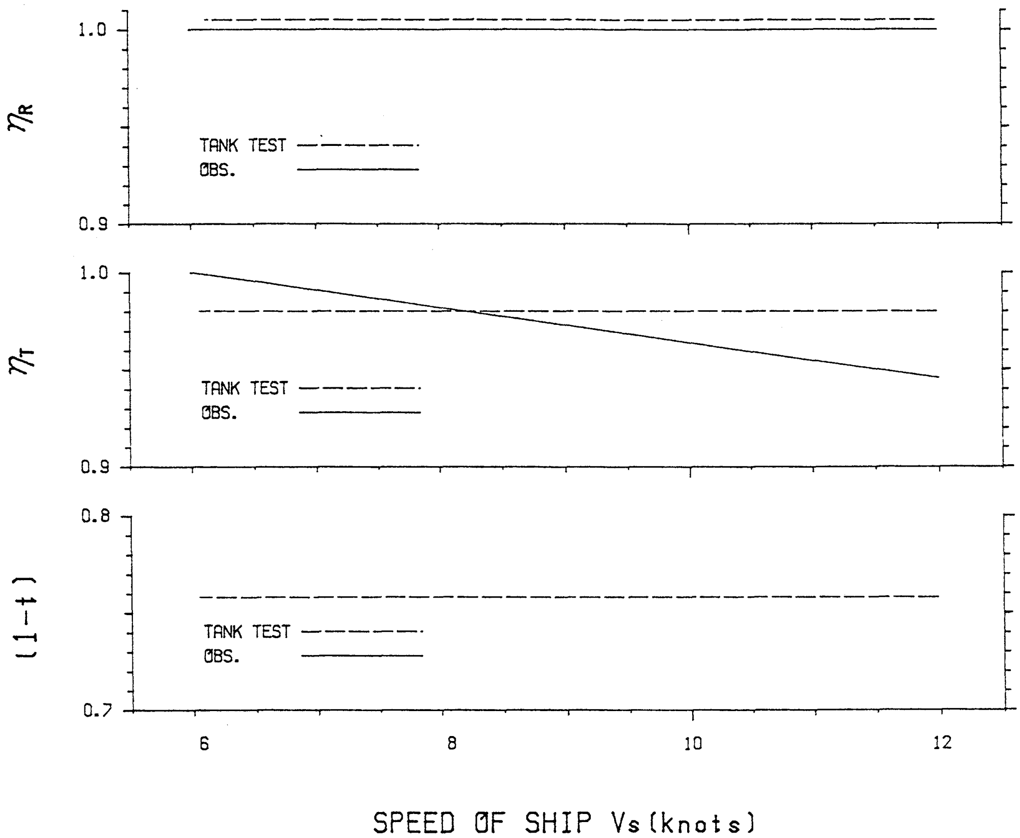

Fig. 3. Relationships between the ship speed Vs and the propeller efficiency ratio η_R, transmission efficiency η_T and thrust deduction factor $(1-t)$.

where η_R: propeller efficiency ratio and η_T: transmission efficiency.

On the assumptions that the propeller blade are intact, that propeller coefficients Kt and Kq and thrust deduction coefficient t during the self-propulsion test remain unchanged, and that the value of $t=0.242$ obtained from the full-load water tank test can be used, the unknown factors remaining in the flowchart are wake coefficient w and propulsive efficiency η.

Thus, when ship speed V at a given propeller angle θ, rotational speed of the propeller n and shaft horsepower SHP are known from measurements, wake coefficient w and propulsive efficiency η can be calculated using the relational expessions shown in the flowchart. Effective horsepower curves can also be plotted.

For navigation at a given propeller blade angle θ and a given rotational speed n_1, overall hull resistance R_1 can be determined from effective horsepower EHP curves. For towing an object at a given constant ship speed V and propeller rotational speed n_2, overall resistance R_2 for thrust T contains a resistance component of the towed object Xr_r and is expressed as a sum of hull resistance R_1 and towed object resistance Xr

Hence, when towed object resistance Xr is regarded as towing force TF,

$$TF=R_2-R_1 \qquad (4)$$

Because the maximum continuous rating of the main engine of the ship is 1.050 PS, shaft horsepowers of 1,000, 800 and 600 PS were selected. For these horsepower levels, relationships betweeen ship speed V_s, towing force TF and rotational speed of the main engine N were determined.

For practical reasons, various conversions were performed: form V (m/s) to Vs (knots)

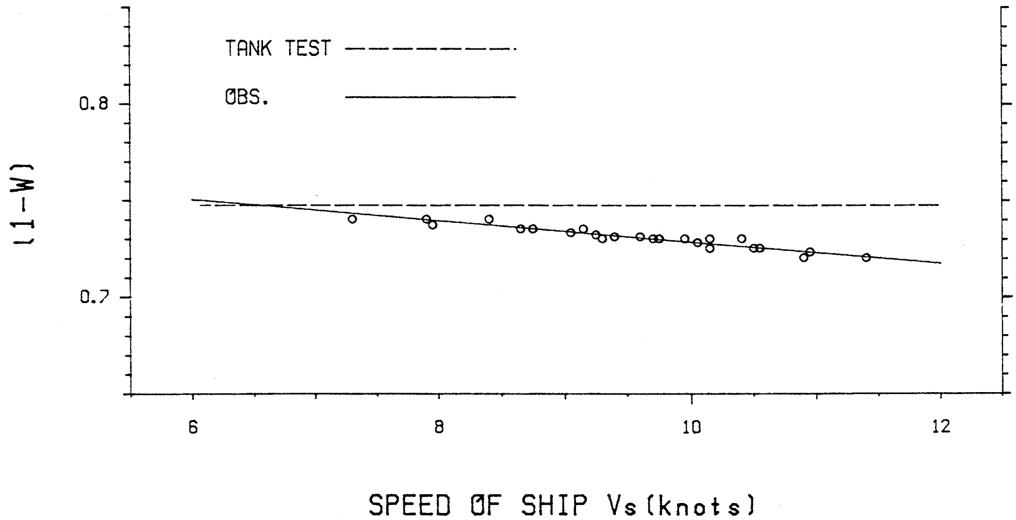

Fig. 4. Relationship between the ship speed *Vs* and the wake ratio $(1-w)$.

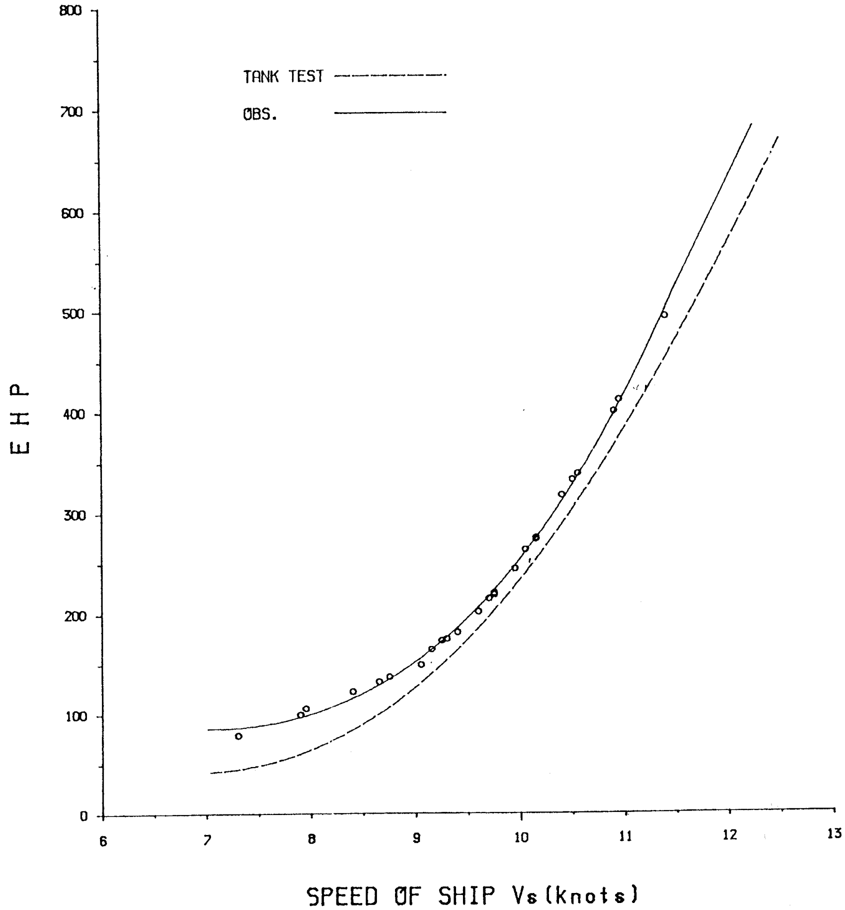

Fig. 5. Comparison of *EHP* curves.

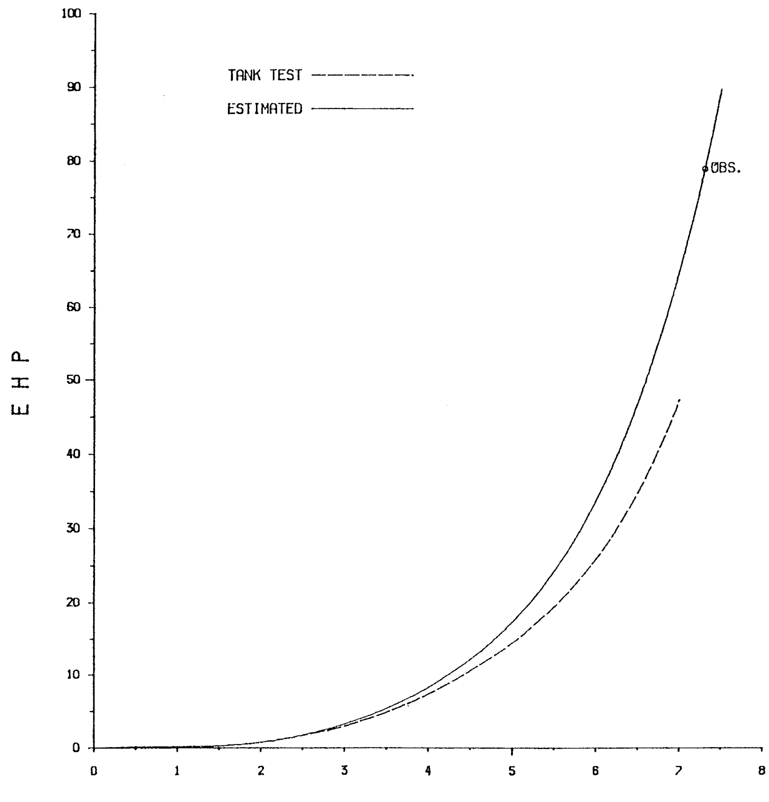

Fig. 6. Comparisonof *EHP* curves at low speed.

for ship speed, from kgf to tonf for towing force, and from n (rps) to N (rpm) via the reduction ratio for rotational speed of the main engine.

4. Results of analysis

Propeller efficiencies: The results of the full-load water tank test showed a propeller efficiency ratio η_R of 1.005 and a transmission efficiency η_T of 0.98. Generally, the η_R values are approximately 1.0 and the η_T values are within the range from 0.98 to 0.95. Therefore 1.0 was adopted as the η_R value to simplify the calculations.

Fig. 3 shows comparison of the results of the calculation based on observations (solid lines) with those obtained by the water tank test (dotted lines). η_T based on observations was 0.95 at *Vs* of 11.75 knots and 0.98 at 8.00 knots.

Wake ratio $(1-w)$: Fig. 4 shows a comparison of the wake ratio $(1-w)$ for given ship speeds *Vs*. The wake ratio $(1-w)$ based on observations (marked with circles) was 0.72 at *Vs* of 11.75 knots and 0.75 at 6.0 knots. As the ship speed is reduced, the wake ratio increases. Inquiries of several shipbuilders about this tendency were thus confirmed.

Effective horsepower *EHP*: Fig. 5 shows a comparison of the effective horsepower based on observations (marked with circles) in relation to given ship speeds with that obtained by the tank test (dotted curve). The solid line represents the effective horsepower determined by the method of least squares.

The values based on observations were higher than those obtained by the tank test by 30～50 *PS*, especially in the low speed range.

Fig. 6 shows the effective horsepower curves for the low speed range. In the estimation, the effective horsepower of 78.8 *PS* at 7.30 knots

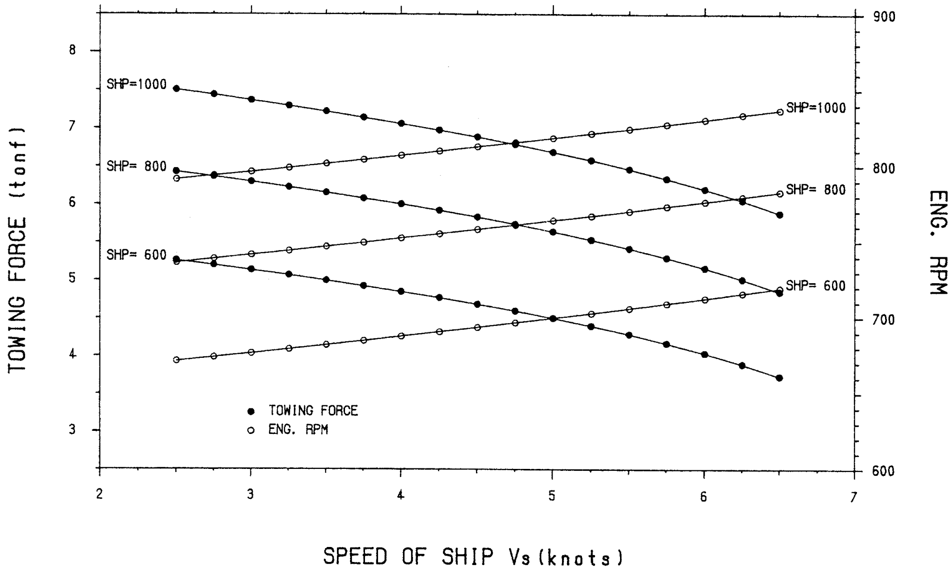

Fig. 7. Towing force curves.

was employed. The solid curve is estimated one based on observations and the dotted curve is that obtained by water tank test.

Estimated horsepower: The T/S Seiyo-Maru II usually navigates with a propeller blade angle θ of 20 degrees and at a main engine rotational speed N of 820 rpm. The water tank test shows a shaft horsepower of 690 *PS* and a ship speed of 10.76 knots, while the present measurements gave a shaft horsepower of 720 *PS* and a ship speed of 10.68 knots.

Towing force *TF*: The *EHP* curves shown in Fig. 6 were used to estimate the towing force *TF* (tonf) and the main engine rotating speed N (rpm) at ship speeds of 2.5~6.5 knots with the normal propeller blade angle of 20 degrees.

For the estimation, shaft horsepowers of 600, 800, 1,000, *PS*, the wake ratio of 0.75 and the propulsive efficiency of 0.95 (worse condition) were employed as navigation conditions. The results of estimation are shown in Fig. 7.

The training ship is usually operated at 3/4 of *MCR* of the main engine. At a shaft horsepower of 800 *PS*, the ship speed *Vs* is 3.0 knots, the towing force *TF* is 6.84 tonf, and the main engine rotaional speed N is 770 rpm. The ship speed of 6.0 knots corresponds to the towing force of 5.68 tonf and the main engine rotational speed of 805 rpm. The ordinary towing force of the T/S Seiyo-Maru II is estimated to be 5.6 to 6.8 tonf.

5. Conclusion

The ship speed *Vs*, main engine rotating speed N and shaft horsepower *SHP* of the T/S Seiyo-Maru II were measured at sea and the results were analyzed to make clear the characteristics required for towing various fishing gear and oceanographic survey instruments.

In the analysis, propulsive efficiency η and wake coefficient w were determined from thrust deduction coefficient t obtained in the water tank test. The estimated effective *SHP* was higher than that obtained by the water tank test by 30 to 50 *PS*. This demonstrates an increase in overall hull resistance R as compared with that at the completion of the ship.

The measurements and analysis showed that for the propeller blade angle of 20 degrees and main engine rotational speed of 820 rpm under normal navigation conditions, the ship speed *Vs* decreased by 0.1 knots and the shaft horsepower increased by 30 *PS* for 6 years after the condition.

For a propeller blade angle of 20 degrees and shaft horsepower of 800 *PS*, the towing force *TF* was within the range of 6.84~5.69 tonf at the ship speed of 3.0~6.0 knots and at the main engine rotational speeds of 770～805 rpm. Thus, the normal towing force of the T/S Seiyo-Maru II was estimated to be 5.6 to 6.8 tonf.

References

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東京水産大学研究練習船青鷹丸II世の曳航性能について

小 池 孝 知

要旨 1987年に建造された東京水産大学研究練習船青鷹丸II世の船速 *Vs*，主機関回転数 *N* および軸馬力 *SHP* の計測を行い，推定馬力および曳航力を試算した。その結果，通常航海におけるプロペラ翼角 θ が 20°，主機関回転数 *N* が 820 rpm では，建造直後と比較して船速 *Vs* は 0.1 knots ほど低下し，軸馬力 *SHP* は 30 *PS* ほど増大していることが分かった。また，プロペラ翼角 θ が 20°，軸馬力 *SHP* が 800 *PS* の場合には，船速 *Vs* 3.0～6.0 knots での曳航力は 6.8～5.6 tonf と推定された。