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Comparative Analysis of Engine Propeller Matching (EPM) Characteristics Using Empirical Methods and Computational Fluid Dynamics (CFD) for a 5 GT Fishing Vessel

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KEYWORDS
Fishing Vessel;
Engine Propeller Matching;
Operating Point;
Wageningen B Series;
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ABSTRACT – The compatibility between engine and propeller characteristics is a critical factor in determining the performance of a ship's propulsion system, particularly for small-scale fishing vessels. This study aims to analyze engine-propeller matching (EPM) on a 5 GT fishing vessel using an empirical method based on the Wageningen B-Series and to perform hydrodynamic calculations via Computational Fluid Dynamics (CFD) simulation. Ship resistance calculations were performed using the MARIN (DESPPC) method to determine the required thrust at various operating speeds. Propeller characteristics were determined using Wageningen B-Series data (B4-30, AE/AO = 0.30, P/D = 1.0), while engine characteristics were modeled using a constant power approach.The analysis results show that the optimal operating point occurs at 834 rpm with a torque of 389 Nm. Under these conditions, the propeller is capable of generating a thrust of approximately 3950 N, so the propulsion system is deemed capable of meeting the ship's operational requirements. CFD simulation results yielded a thrust of 3666.17 N. A comparison between the EPM and CFD methods revealed a 7.2% discrepancy based on thrust. This discrepancy is attributed to the EPM method's reliance on idealized flow conditions, whereas CFD accounts for viscosity effects, turbulence, and flow distribution more realistically under open-water conditions. Overall, the Engine–Propeller Matching method proved to be effective as an initial approach in the analysis of fishing vessel propulsion systems, with an acceptable level of accuracy following comparison using the CFD numerical method.

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INTRODUCTION

A ship moving through water experiences resistance that acts against the direction of the ship's motion. This drag must be overcome by the thrust generated by the ship's propulsion system so that the ship can move at the desired speed. The magnitude of a ship's resistance is influenced by several key factors, such as frictional resistance, wave resistance, and other additional resistances related to the hull shape and the ship's operational conditions [1] .In conventional ship propulsion systems, thrust is generated by a propeller driven by the ship's main engine. Therefore, the compatibility between the characteristics of the engine and the propeller is a critical factor in determining the performance of the ship's propulsion system. Mismatch between engine and propeller characteristics can lead to various operational problems, such as excessive load on the engine, inefficient fuel consumption, and the ship's inability to reach its design speed [2] .

One method used to evaluate the compatibility between the engine and the propeller is the Engine Propeller Matching (EPM) method. This method is used to match the propeller load characteristics with engine performance characteristics so that the propulsion system can operate under safe and efficient operating conditions. EPM analysis is generally performed by comparing the propeller load curve with the engine characteristic curve to determine the operating point of the ship's propulsion system [3] .

Issues with propulsion system compatibility are frequently encountered on small-scale fishing vessels, which typically use relatively low-power diesel engines with fixed-pitch propellers. In many cases, the selection of engines and propellers is not based on a comprehensive propulsion system analysis, resulting in suboptimal vessel performance. Therefore, analyzing the compatibility of the propulsion system on fishing vessels is crucial for improving operational efficiency and ensuring the engine operates within safe operating limits.

Several previous studies have applied the concept of Engine–Propeller Matching to various types of vessels. Research on the characteristics of the Wageningen B-Series propellers indicates that this series is widely used in propulsion system analysis because it possesses comprehensive hydrodynamic characteristic data and can be applied to various vessel operating conditions [4] . Additionally, studies on engine-propeller characteristic matching have shown that EPM analysis can be used to determine the optimal operating point of the propulsion system based on the vessel's power and speed requirements [5] .

However, most of these studies still focus on a single operating condition or a specific design point. Analyses that account for variations in vessel operating conditions and evaluate the operating range of the propulsion system are still relatively limited, particularly for small-scale fishing vessels. Furthermore, the use of Computational Fluid Dynamics (CFD) as a verification tool for Engine–Propeller Matching calculation results has not yet been widely applied in the analysis of fishing vessel propulsion systems.

Therefore, this study was conducted to analyze the characteristics of engine-propeller matching on a 5 GT fishing vessel based on variations in operating conditions. The analysis was performed using the classical Engine Propeller Matching method, utilizing ship resistance data calculated using the MARIN method and the characteristics of the Wageningen B-Series propeller. Additionally, hydrodynamic verification was performed using the Computational Fluid Dynamics (CFD) method to evaluate the agreement between theoretical calculation results and numerical simulations. The research results are expected to provide insights into the operating range of the propulsion system as well as the safe and optimal operating limits for small-scale fishing vessels.

METHODS

The research method employed in this study is a theoretical and numerical analysis of the propulsion system of a 5 GT fishing vessel using the Engine Propeller Matching (EPM) approach. The analysis is based on vessel resistance and propulsion data calculated using the MARIN method, as well as the characteristics of the engine and propeller used.

This study does not aim to redesign the propeller but rather to analyze the matching characteristics between the engine and propeller based on variations in the vessel's operating conditions. The discrepancy between the classical method and the CFD results was analyzed to evaluate the level of matching and the validity of the EPM method used in this study.

Research Object

The research object in this study is a traditional fishing vessel of 5 GT equipped with a single-screw propeller propulsion system. The vessel is designed to operate at a service speed of approximately 6–8 knots with a small-capacity marine diesel engine.

The vessel geometry data used are from the design plans for a FRP-hulled fishing vessel. The dimensions and visualization of the object are shown in Table 1 and Figure 1, while the propeller data and propeller model are presented in Table 2 and Figure 2.

Table 1. Principal Dimensions of the 5 GT Fishing Vessel

Parameter	Symbol	Full Scale	Unit
Total length	LoA	11.00	Meters
Length Deck	LDK	10.68	Meters
Waterline Length	LWL	9.15	Meters
Overall Width	B	2.60	Meters
Block Coefficient	CB	0.709	-
Deck Height	H	0.95	Meters
Ship's Draft	T	0.60	Meter
Speed	Vs	6–8	Knots

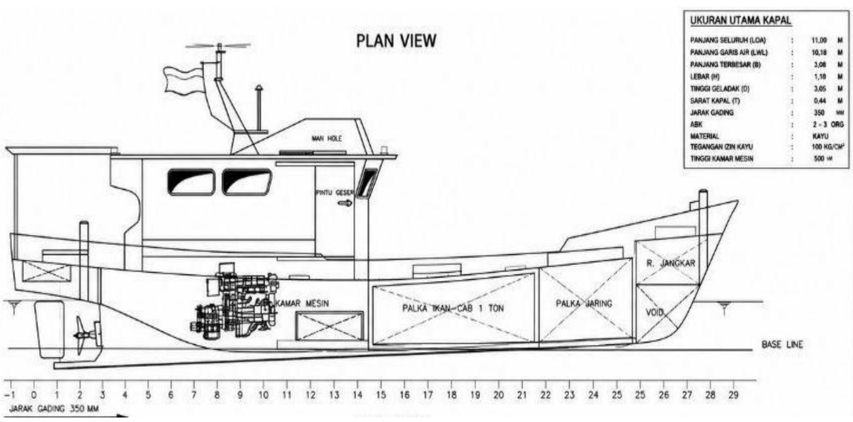


Figure 1. The General Arrangement of 5 GT Fishing Vessel

Table 2. Principal Dimensions of the 5 GT Fishing Vessel Propeller

Parameter	Symbol	Full Scale	Unit
Diameter Pitch to Diameter Ratio	D	0.555	M
Expanded blade area ratio	Ae/A0	0.665	-
Number of blades	Z	4	Blades
Propeller type	B-series		
Minimum revolutions		700	1/min
Maximum revolutions		800	1/min
Design shaft power		34	kW

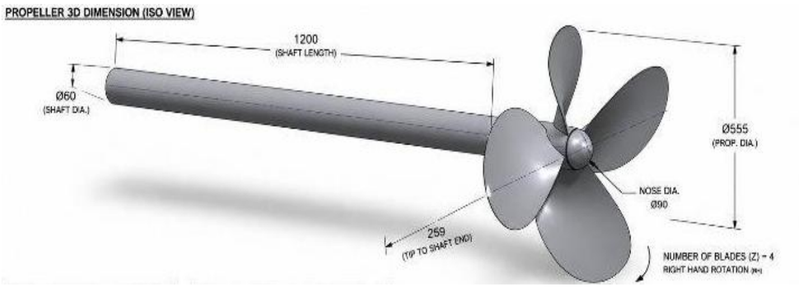


Figure 2. Fishing Vessel Propeller Model 5 GT

Research Data

The data used in this study were obtained from calculations using MARIN (DESPPC 1999 – *Performance Prediction of Displacement Ships*). This method was used to predict the performance of displacement vessels in calm water conditions, including total ship resistance, required thrust, shaft rotation, and shaft power at various operating speeds. The data show that resistance increases non-linearly with ship speed, which in turn affects the required propeller thrust. The geometric data and hydrodynamic parameters of the 5 GT fishing vessel, based on calculations using the MARIN DESPPC 1999 program, are shown in Tables 3, 4, and 5.

Table 3. Comprehensive Hydrodynamic Performance and Propulsion Data (Calm Water)

Vs (knots)	Vs (m/s)	RTOT (N)	EHP (kW)	Treq (N)	w	t	η0	ηD	n (rpm)	Power (kW)
5.00	2.57	600	1.54	690.45	0.134	0.131	0.616	0.624	370	3
6.00	3.08	1200	3.7	1,380.90	-	-	-	-	478	6
7.00	3.6	2200	7.92	2,531.65	-	-	-	-	617	14
8.00	4.11	3,800	15.63	4,372.84	0.132	0.131	0.497	0.502	779	31
9.00	4.63	7,500	34.72	8,630.61	-	-	-	-	901	81
10.00	5.14	11,200	57.61	12,888.38	0.131	0.131	0.394	0.364	1343	160

The machine data and specifications used in this study are shown in Table 4.

Table 4. Main Engine Data

Engine Parameters	Value	Unit	Description
Engine Type	Marine diesel	-	Conventional small ship engine
Maximum Power (MCR)	34–36	kW	(based on Dongfeng ZS series low-speed diesel)
Maximum RPM	800	rpm	According to design data
Operating Power (NCR)	85–90% MCR	kW	Continuous operation
Engine Characteristics	Constant Power		Assumptions for engine envelope
Cooling System	Seawater		General assumptions
Transmission System	Direct drive		No gearbox
Number of Engines	1	unit	Single screw

Engine Propeller Matching Analysis Method

Engin Propeller Matching (EPM) analysis is performed to determine the compatibility between engine and propeller characteristics based on the vessel's operating conditions. This method involves comparing the propeller load curve with the engine characteristic curve to determine the operating point of the propulsion system[6] ,[7] .

The calculation of propeller thrust is performed using the equation below.

$$T = \rho \cdot n^2 \cdot D^4 \cdot K_T \quad (1)$$

where T is the thrust (N), ρ is the density of seawater (kg/m³), n is the propeller speed (rps), D is the propeller diameter (m), and KTis the thrust coefficient[1] .

The propeller torque is calculated using the following equation:

$$Q = \rho \cdot n^2 \cdot D^5 \cdot K_Q \quad (2)$$

where Q is the torque (Nm) and KQis the propeller torque coefficient[1] .

Propeller shaft power is calculated using the equation:

$$P = 2\pi \cdot n \cdot Q \quad (3)$$

where P is shaft power (W), n is propeller speed (rps), and Q is torque (Nm)[1] .

The propeller load curve is plotted based on the relationship between propeller torque and rotational speed, while the engine characteristic curve is plotted using the constant power characteristic approach, which is commonly used in small marine diesel engines[3] ,[2] .

In this approach, engine power is considered constant, so the relationship between engine torque and rotational speed is expressed as:

$$Q_{engine} = P_{engine} / (2\pi \cdot n) \quad (4)$$

The operating point of the propulsion system is determined under conditions where engine torque equals propeller torque.

$$Q_{engine} = Q_{propeller} \quad (5)$$

To obtain a more accurate operating RPM value, a linear interpolation method is used based on the difference in torque (ΔQ) versus RPM. The operating point is obtained under the condition ΔQ = 0, which indicates a balance between engine capacity and propeller load requirements[6] .

CFD Simulation

The simulation method for the Engine Propeller Matching results employs a Computational Fluid Dynamics (CFD) approach using ANSYS Fluent software. This simulation aims to numerically evaluate the propeller's hydrodynamic performance under open-water conditions, where the propeller is modeled without interaction with the hull.

The computational domain was modeled as a cylindrical domain surrounding the propeller, with the domain size adjusted to the propeller diameter to minimize boundary effects on the flow[1] ,[8] .

The boundary conditions used are defined by a cylindrical domain corresponding to an open-water test, with a velocity inlet boundary condition and a specified pressure at the outlet. Propeller rotation is modeled using the rotating reference frame (RRF) approach, with the rotational speed based on the results of the Engine Propeller Matching[9] . A visualization of the simulation computational domain is shown in Figure 3.

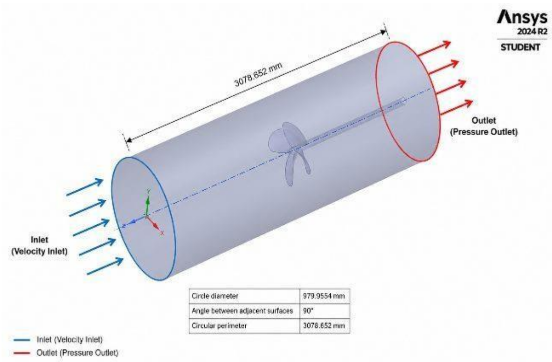


Figure 3. Domain Visualization

The initial meshing process uses tetrahedral elements with an increasing number of mesh cells at each refinement level. Subsequently, a conversion and optimization process is performed to create a polyhedral mesh using the polyhedral conversion feature in ANSYS Fluent. The use of polyhedral meshes is known to improve convergence stability and computational efficiency compared to conventional mesh methods[9] . The initial number of meshes and the optimization results used in this study are shown in Table 5.

Table 5. Mesh Optimization

Mesh Type	First Mesh (Tetrahedral)	Optimation Mesh (Polyhedral)
Coarse	1.5856.387	327.343
Medium	2.476.546	431.867
Fine	2.598.987	506.803

A visualization of the polyhedral optimization mesh distribution on the propeller model with refinement in the blade area is shown in Figure 4.

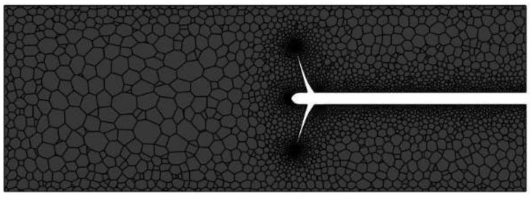


Figure 4. Visualization of Polyhedral Mesh

The turbulence model used is k-ω SST (Shear Stress Transport), which is effective in predicting flow with high pressure gradients as well as flow separation phenomena on propellers[1] .

RESULTS AND DISCUSSION

Ship Resistance Analysis

Total ship resistance is one of the primary parameters in determining power requirements and propulsion system performance. The magnitude of this total resistance is the sum of all drag force components acting on the ship, including frictional resistance, wave resistance, appendage resistance, air resistance, and so on[10] . Total ship resistance can be obtained from Equation 6.

$$R_T = R_F + R_W + R_{AP} \tag{6}$$

In this study, the total resistance values were obtained using a numerical method based on MARIN data (Delft Systematic Series) under calm water conditions. The results of the resistance calculations for varying ship speeds are shown in Table 3.

Based on Table 3, the ship's total resistance increases significantly as speed increases. At 5 knots, the resistance is 600 N, then increases to 3,800 N at the design speed of 8 knots, and reaches 11,200 N at 10 knots.

Thus, at the design speed of 8 knots, the ship has entered an operating condition requiring a large thrust, so the propulsion system must be capable of generating thrust greater than the total resistance to ensure the ship's operational performance.

Thrust Requirement Analysis

Propeller thrust is the force generated to overcome ship resistance so that the ship can move at a certain speed. Based on the principle of force balance in the propulsion system, the ship's operating conditions must satisfy $T \geq RT$[1] .

The thrust requirement is obtained from the MARIN calculation results, as expressed in Equation (1).

Based on the calculation results in Table 3, the thrust requirement increases as the ship's speed increases. At a speed of 5 knots, the required thrust is 690.45 N to overcome a resistance of 600 N. At the design speed of 8 knots, the thrust increases to approximately 4372.84 N to overcome a resistance of 3800 N.

A comparison between thrust and resistance shows that across the entire speed range, the thrust value is greater than the ship's resistance. This indicates that the propulsion system has the capability to overcome the ship's resistance and maintain operational speed. A situation where thrust exceeds resistance indicates that the propeller can generate sufficient thrust; however, an excessively large difference may indicate inefficient energy loss within the propulsion system. Therefore, further analysis using the Engine-Propeller Matching method is required to determine the optimal operating point between the engine and the propeller[6] . The relationship between thrust and ship resistance versus speed is shown in Figure 5.

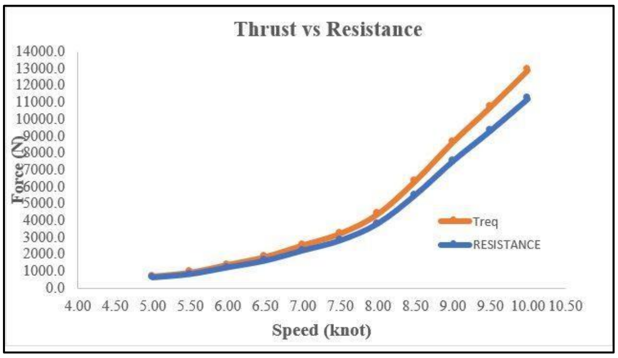


Figure 5. Relationship between thrust and ship resistance

Propeller Characteristic Analysis

Generally, the characteristics of ship propellers under open-water test conditions are as represented in the KT–KQ–J diagram. Each type of ship propeller has distinct performance curve characteristics.[11] .

The hydrodynamic characteristics of the propeller were analyzed using data from the Wageningen B-Series type B4-30 with a blade area ratio AE/A0 = 0.30. The primary parameters used in this analysis are the thrust coefficient (KT), the torque coefficient (KQ), and the advance coefficient (J), which were obtained based on the ship's operating conditions[4] ,[6] .

Theoretically, propeller performance parameters can be expressed in the form of non-dimensional coefficients formulated using equations (7), (8), (9), and (10).

$$KT = \frac{T}{\rho \times n^2 \times D^4} \tag{7}$$

$$KQ = \frac{Q}{\rho \times n^2 \times D^5} \tag{8}$$

$$\eta = \frac{J \times K_T}{2 \times \pi \times K_Q} \tag{9}$$

Where KT is the thrust coefficient, KQ is the propeller torque coefficient, η is the propeller efficiency, J is the propeller advance coefficient, Va is the advance velocity, D is the propeller diameter, n is the propeller rotational speed, T is the propeller thrust, Q is the propeller torque, and ρ is the fluid density[7] .

The values of KT and KQ are obtained through interpolation of the Wageningen B-Series graph shown in Figure 6, based on the calculated value of J. The diagram illustrates the relationship between the propeller performance coefficient and the advance coefficient for various pitch ratios, allowing it to be used to determine propeller characteristics under specific operating conditions[12] .

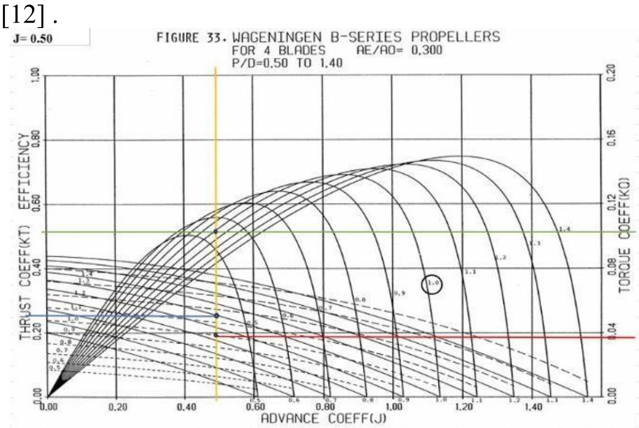


Figure 6. Line interpolation on the Wageningen B-Series graph

Based on the results of the graphical method (interpolation) from the Wageningen B-Series open-water graph, the results are presented in Table 6.

These values indicate that the propeller is operating under near-optimal conditions, where the generated thrust is relatively high compared to the ship's resistance requirements. This aligns with the general characteristics of propellers, where an increase in the J value tends to decrease KT and increase KQ, indicating the energy distribution between thrust and torque[1] ,[4] . These results are then used as the basis for calculating propeller thrust and torque using equations (1) and (2) in Table 6.

Table 6. Calculation Results for Thrust and Torque of the Wageningen B-Series Propeller

VS	J	KT	KQ	η_o	Tp [N]	Qp[N.m]
5.00	0.66	0.17	0.040	0.62	604.71	78.26
6.00	0.61	0.21	0.036	0.65	1249.06	117.77
7.00	0.55	0.27	0.039	0.58	2676.23	212.61
8.00	0.50	0.25	0.039	0.53	3950.60	338.96
9.00	0.42	0.20	0.041	0.49	5645.73	714.18
10.00	0.36	0.18	0.039	0.40	9871.26	1628.76

Based on the calculation results at the ship's design speed of 8 knots, the values obtained are J ≈ 0.50, KT ≈ 0.25, and KQ ≈ 0.039.

The calculation results in Table 9 show that at a speed of 8 knots, the propeller generates a thrust of 3950.61 N and a torque of 338.96 Nm. When compared to the ship's thrust requirement of 3800 N (Table 3), the propeller is capable of generating a thrust slightly greater than the ship's resistance. This condition indicates that the propulsion system is in a safe condition and capable of reaching the design speed.

Engine Load Curve (Engine Envelope)

Engine characteristics are one of the primary parameters in propulsion system analysis, as they determine the engine's operational limits in generating power and torque at various RPMs. In Engine–Propeller Matching analysis, engine characteristics are expressed as the relationship between torque and RPM (Q-n)[7] .

In this study, the engine is modeled using the constant power approach, which is commonly used for small-scale marine diesel engines[11] . The relationship between power, torque, and rotational speed is expressed in Equation (3). Thus, engine torque can be written as in Equation (4).

Where P is engine power (W), n is rotational speed (rps), and Q is torque (Nm).

Based on the engine data used, it is known that the engine's maximum power (MCR) is 34 kW at a maximum speed of 800 rpm. Using this equation, the relationship between engine speed and torque is obtained as shown in Table 7.

Table 7. Engine Torque Characteristics

n (rpm)	n (rps)	P (kW)	P (W)	Qengine (N·m)
600	10.00	34	34,000	541.1
700	11.67	34	34,000	463.8
800	13.33	34	34,000	405.8
850	14.17	34	34,000	382.0
900	15.00	34	34,000	360.8

These results show that engine torque decreases as RPM increases, which is a common characteristic of engines with constant power, Q $\propto^{1}$ [11] .

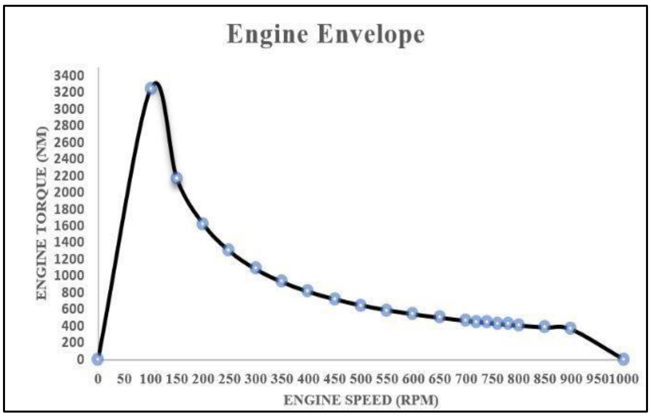


Figure 7. Engine Characteristic Curve (Engine Envelope)

For analysis purposes, the engine speed range was expanded from 600 rpm to 900 rpm, divided into three categories: 600–700 rpm (underload), 700–800 rpm (operating range), and 800–900 rpm (overload). The overload range above 800 rpm is used as an evaluation threshold to assess the propeller load's relationship to the engine's capacity, even though it falls outside the engine's normal operating conditions.

The engine characteristic curve is then plotted as shown in Figure 7, which illustrates the relationship between torque and engine speed. This curve is subsequently used as the basis for the Engine–Propeller Matching process by comparing it to the propeller load curve.

Propeller Load Curve

The propeller load curve is plotted to illustrate the relationship between the torque required by the propeller and shaft speed. This curve represents the actual load that the engine must meet and serves as the basis for Engine–Propeller Matching analysis[11] .

The RPM range used in this analysis refers to the engine characteristics discussed in the previous subsection, namely 600–900 rpm. This range covers underload, normal operation, and overload conditions, allowing for a comprehensive evaluation of the compatibility between the engine and the propeller.

Propeller torque is calculated using the propeller hydrodynamic equation in Equation (11).

$$Q_{propeller} = \rho \times n^2 \times D^5 \times K_Q \quad (10)$$

Where Qpropeller is the propeller torque (Nm), is the fluid density (kg/m³), n is the propeller rotational speed (rps), D is the propeller diameter (m), and KQ is the torque coefficient obtained from the Wageningen B-Series diagram[13].

In this study, a B-Series propeller with a diameter of 0.55 m and a value of KQ of 0.039 was used under operating conditions close to the design point, obtained from interpolation results on the Wageningen diagram in Table 9.

Based on the calculation results, the relationship between propeller speed and torque is shown in Table 8.

Table 8. Propeller Load Curve

n (rps)	Qprop (Nm)
10.00	201.0
10.83	235.9
11.67	273.6
12.00	289.4
12.33	305.7
12.67	322.5
13.00	339.7
13.33	357.3
14.17	403.4
15.00	452.3

The calculation results show that propeller torque increases significantly with increasing rotational speed. Theoretically, this relationship follows the basic propeller characteristic where torque is directly proportional to the square of the rotational speed ($Q \propto n^2$)[11] .

The propeller load curve is plotted in Figure 8, showing a nonlinear increasing trend between rotational speed and torque. At low rotational speeds (600–700 rpm), the torque value is relatively small, so the propeller is underloaded relative to the engine. In the 700–800 rpm range, the propeller load increases, and the propeller operates within normal conditions consistent with the engine's characteristics.

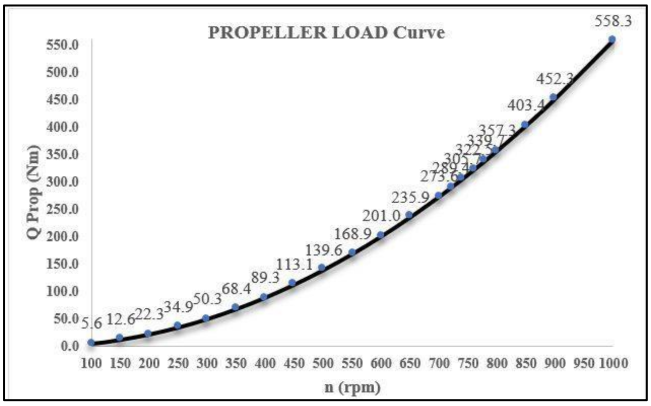


Figure 8. Propeller Load Curve vs. RPM

However, at RPMs above 800, there is a significant increase in torque, reaching 452.3 Nm at 900 RPM. This condition indicates a tendency toward overload, where the propeller load approaches or has the potential to exceed the engine's capacity.

Engine Matching Analysis

Engine-Propeller Matching (EPM) analysis was conducted to determine the optimal operating point of the propulsion system by comparing the engine characteristic curve with the propeller load curve. The operating point is identified at the condition where the torque generated by the engine equals the torque required by the propeller[7] ,[14] .
Determination of the Operating Point.

The comparison between engine torque and propeller torque is performed within the 600–900 rpm range. The torque difference (ΔQ) is calculated using Equation (12).

$$\Delta Q = Q_{engine} - Q_{propeller} \quad (11)$$

The calculation results show that the sign ofΔQ changes between 800 rpm and 850 rpm, specifically at 800 rpmΔQ = +48.51 Nm and at 850 rpm $\Delta Q = -21.42\ Nm$

At that speed, the values of Qengine and Qpropeller are both 389 Nm.
Matching Curve Analysis

The comparison curve between engine torque and propeller torque is shown in Figure 10. The engine curve shows a decreasing trend with increasing RPM, while the propeller curve shows an increasing trend. The intersection point between the two curves represents the actual operating conditions of the propulsion system.
In this study, the operating point is at 834 rpm, which is slightly higher than the design speed of 778 rpm. This difference of approximately 7% indicates that the propeller load is relatively light for the engine, so the engine is still capable of operating the propeller without experiencing overload. A visualization of the analysis is shown in Figure 9.

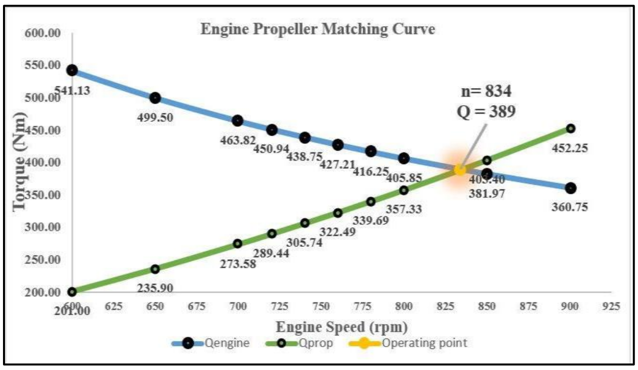


Figure 9. Engine-Propeller Matching Curve

Analysis of Operating Condition Variations

An analysis of operating condition variations was conducted to evaluate the performance of the ship's propulsion system at several operating speeds: 6, 7, 8, and 9 knots. The analyzed parameters include thrust, torque, propeller power, and the ratio to ship resistance. The calculation of propulsion parameters includes propeller thrust, which is calculated using Equation (1), while torque and propeller power are calculated using Equations
(2) and (3), respectively.

Where KTand KQare propeller coefficients obtained from the Wageningen B-Series diagram. The calculation results based on equations (1), (2), and (3) are shown in Table 9.

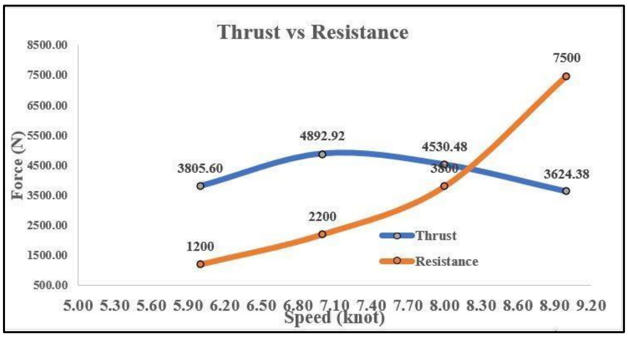
Table 9. Propulsion system performance calculations

Vs (knot)	J	KT	KQ	Thrust (N)	Torque (Nm)	Power (kW)	Rt (N)
6.00	0.61	0.21	0.036	3805.60	358.814	31.337	1200
7.00	0.55	0.27	0.039	4892.92	388.715	33.948	2200
8.00	0.50	0.25	0.039	4530.48	388.715	33.948	3800
9.00	0.42	0.20	0.046	3624.38	458.485	40.041	7500

The calculation results show that at a speed of 6 knots, a thrust of 3805.60 N is obtained with a power requirement of 31.34 kW, while the ship's resistance is 1200 N. This condition indicates that the thrust is significantly greater than the resistance, so the system is in an underload condition.

At a speed of 7 knots, the thrust increases to 4,892.92 N with a power requirement of 33.95 kW, while the ship's resistance is 2,200 N. The system remains in an underload condition but is approaching the engine's operating capacity.

The relationship between propeller thrust and ship resistance at each speed variation is shown in Figure 10. The graph indicates that the condition of force equilibrium (T = RT) occurs at a speed of 8 knots, where the propeller thrust is able to overcome the ship's resistance optimally.



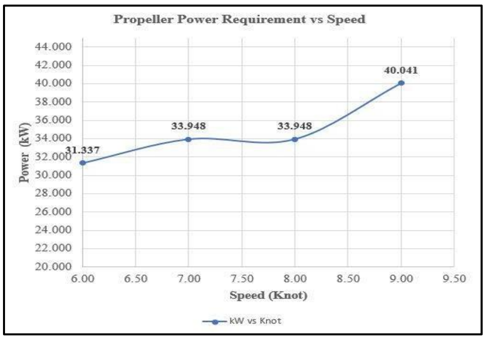


Figure 10. Thrust vs. Resistance Graph under Operating Conditions, Relationship Between Propeller Thrust and Vessel Speed

The relationship between propeller thrust and ship speed is shown in Figure 12. The graph shows that thrust reaches its maximum value at medium speeds, then decreases at higher speeds due to changes in flow conditions and propeller efficiency.

CFD Verification and Validation

Verification and validation were performed on the Wageningen B-Series open-water propeller, type B4-30, with a pitch ratio P/D ranging from 0.50 to 1.40 and a blade area ratio (AE/A0) of 0.30.

Verification was performed to assess simulation uncertainties resulting from discretization errors, particularly regarding the number of mesh cells, using the Grid Convergence Index (GCI) method to systematically evaluate numerical convergence as the mesh is refined. The GCI approach was used to evaluate numerical uncertainty. Equations (14) through (17) were used to obtain the GCI values as described by Celik et al.[15] .

The Grid Convergence Index results are shown in Table 10.

Table 10.. Grid Convergence Index

Parameter	Value
Fine Configuration	*Total Cells* = 506,803 K_T = 0.230
Medium Configuration	*Total Cells* = 431,867 K_T = 0.232
Coarse Configuration	*Total Cells* = 327,343 K_T = 0.226
Fine solution, S_1	0.230
Medium solution, S_2	0.232
Coarse solution, S_3	0.226
Medium-Fine, ε_{21}	-0.00862
Coarse-Medium, ε_{32}	0.02655
e^{21}	0.00870
Convergence ratio, R	3.08
Order of accuracy, p	21.04
GCI Method (%)	1.1 %

Based on the GCI calculation results in Table 13, a GCI value of 1.1% was obtained, indicating that the numerical error due to the influence of mesh size is within acceptable limits. The changes in parameter values between variations are relatively small, so the simulation results have shown convergence.

A grid independence test was conducted to determine whether a model maintains stable values during the meshing process, ensuring that the number of meshes does not affect the analysis results. The results of the grid independence test are shown in Table 11.

Table 11. Grid Independence Study

Tipe Mesh	cells	KT	Difference (%)	KQ	Difference (%)	ηo	Difference (%)
Coarse	327.343	0,226	-	0,0384	-	0,468	-
Medium	431.867	0,230	2,58%	0,0394	2,54%	0,469	0,21%
Fine	506.803	0,23	0,87%	0,0391	0,77%	0,468	0,21%

The validation stage was conducted by comparing the nondimensional parameters obtained from the Wageningen B-Series empirical method with the CFD simulation results under design operating conditions with an advance ratio of J = 0.50. The parameters compared included KT, KQ, and ηo, using simulation results from the medium mesh selected as the optimal mesh based on Grid Independence and GCI. The comparison diagram is shown in Figure 11.

Based on the comparison results in Figure 11, the KT value from the Wageningen B-Series method is 0.250 (as shown in Table 8), while the CFD result is 0.232. The KQ values are relatively close, at 0.039 for the empirical method and 0.0394 for the CFD simulation. Meanwhile, the efficiency of the open-water-ηo shows a value of 0.530 for the Wageningen B-Series method and 0.469 for the CFD.

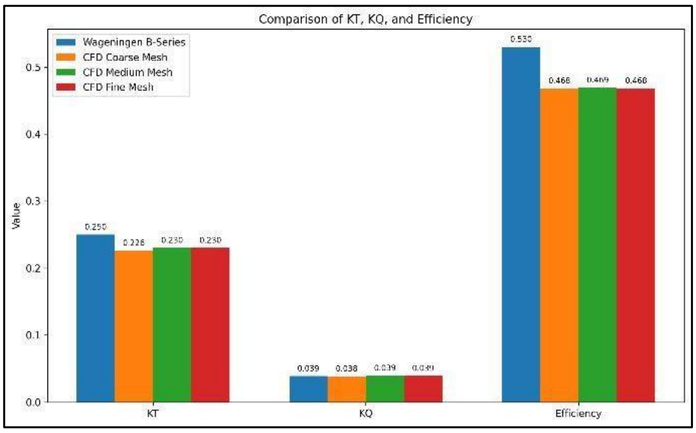


Figure 11. Comparison of KT, KQ, and ηo

CFD Hydrodynamic Visualization Results

The flow velocity distribution in Figure 12 shows fluid acceleration around the propeller blades and the formation of a wake region behind the propeller. This phenomenon indicates energy transfer from the propeller to the fluid, which generates thrust.

The flow pattern formed corresponds to the propeller flow characteristics, where the velocity increases around the blades and decreases in the wake region due to energy loss ([11]).

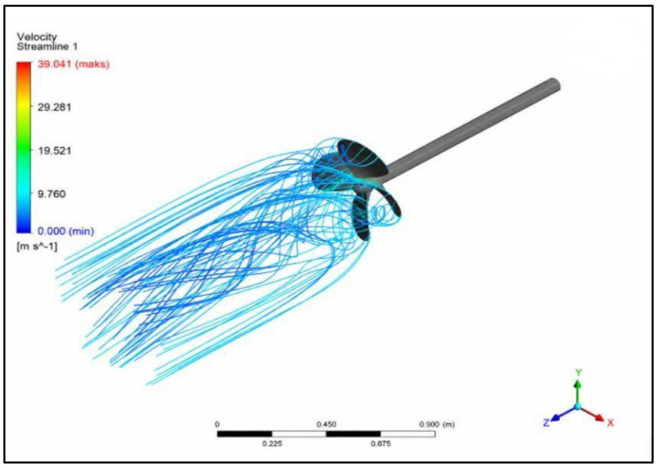


Figure 12.. Flow velocity distribution around the propeller

The pressure distribution shown in Figure 13 indicates a pressure difference between the pressure side and the suction side of the propeller blades. High pressure occurs on the front side of the blade, while low pressure occurs on the rear side. This pressure difference is the primary mechanism for generating propeller thrust in accordance with the principle of fluid momentum[11] . The pressure distribution on the propeller is illustrated in Figure 15.

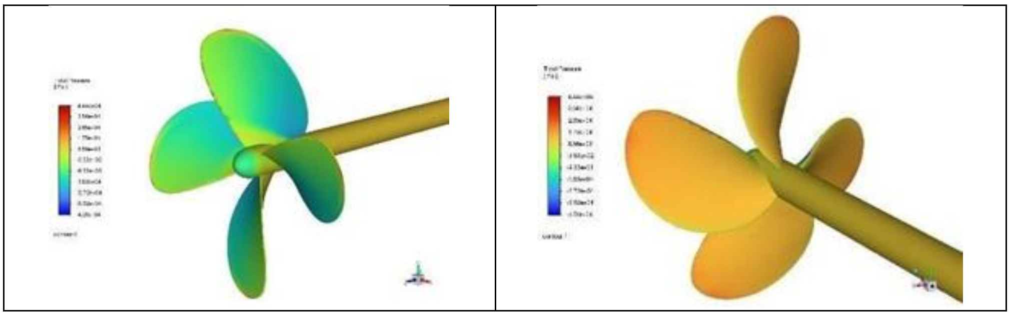


Figure 13. Pressure distribution on the propeller, front and rear views

Force Results and CFD Thrust Calculations

The computational results show the forces on each axis, as shown in Table 12.

Table 12. Forces on each axis

Axis	Pressure (N)	Viscous (N)	Total (N)
X	-0.781	0.054	-0.727
Y	-0.342	-0.031	-0.373
Z	3710.25	-44.08	3666.17

The propeller thrust is derived from the axial force component (Z-axis). The total force consists of pressure and viscous contributions as expressed in equation (18).

$$T = Fpressure + Fviscouse \quad (12)$$

Based on the simulation results, the propeller thrust ($TCFD$) is 3666.17 N.

Comparison Analysis of EPM and Simulation

A comparative analysis was conducted to evaluate the consistency between the results of the Engine–Propeller Matching (EPM) method and the results of Computational Fluid Dynamics (CFD) numerical simulations. The comparison was performed at a single operating point, which is the propeller's optimal operating point.

Thrust Comparison;

Based on the analysis results, the EPM method yielded a thrust value of 3,950.61 N, as shown in Table 8, while the CFD results yielded a thrust value of 3,666.17 N, as shown in Table 12.
The difference between the two methods is calculated using equation (15).

$$Error = \frac{|T_{CFD} - T_{EPM}|}{T_{EPM}} \times 100\% \quad (13)$$

$$= \frac{|3666{,}17 - 3950.61|}{3950.61} \times 100\% \approx 7{,}2\%$$

$$Error = \frac{|Q_{CFD} - Q_{EPM}|}{Q_{EPM}} \times 100\% \quad (14)$$

Based on the calculations in equations (19) and (20), the results show a$Error \approx 7{,}2\%$ t for thrust and 1.13% for torque.

The differences between the results of the EPM and CFD methods are due to several key factors. The EPM method uses an empirical approach based on the Wageningen B-Series open-water data, which assumes ideal and uniform flow conditions; thus, it does not fully account for the effects of viscosity and complex fluid interactions. In contrast, the CFD method models flow phenomena more realistically by considering viscosity effects, turbulence, and three-dimensional distributions of fluid pressure and velocity[16] . Additionally, limitations in the computational domain and the number of mesh elements in the CFD simulation can also affect the accuracy of the results[17].

The bar chart in Figure 16 shows the range of differences between the EPM and CFD results in this study, which is 7.2% for thrust and 1.13% for torque. This illustrates the deviation between the empirical and numerical approaches. The comparison of thrust and torque between the methods at the operating point is shown in Figure 14.

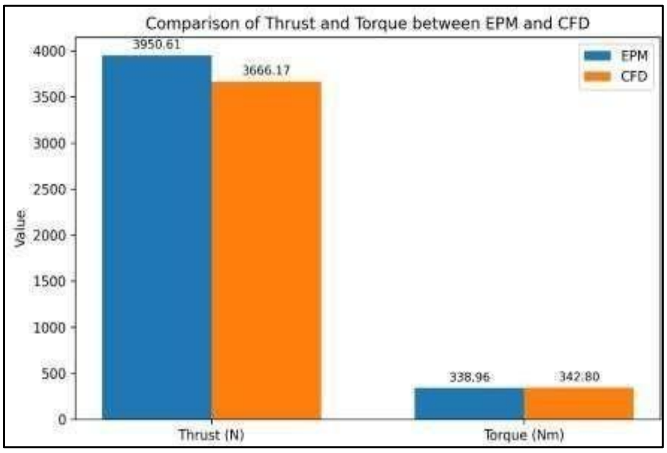


Figure 14. Comparison of Thrust and Torque between EPM and CFD

CONCLUSION

Based on the results of the analysis conducted, it can be concluded that the Engine–Propeller Matching (EPM) method is capable of effectively determining the matching characteristics between the engine and propeller on a 5 GT fishing vessel. The optimal operating point of the propulsion system is achieved at 834 rpm with a torque of 389 Nm, where the propeller generates a thrust of 3950.61 N and the required ship resistance is 3800 N at a design speed of 8 knots; thus, the propulsion system is deemed capable of meeting the vessel's operational requirements. Analysis of operating condition variations shows that at speeds of 6–7 knots, the system is in an underload condition, while at 8 knots, the system operates under optimal conditions, and at speeds above 8 knots, it tends to enter an overload condition due to increased power demand exceeding the engine's capacity. The CFD simulation results yielded a thrust of 3666.17 N. A comparison between the empirical method (EPM) and the CFD simulation shows that the thrust value from the CFD, 3666.17 N, is lower than the EPM calculation result of 3950.61 N. The difference is 284.44 N, or approximately 7.2%. This difference is due to the differing methodological approaches, where EPM uses idealized conditions, while CFD accounts for hydrodynamic effects more realistically. Thus, the Engine–Propeller Matching method can be used as a reasonably good initial approach in the analysis of fishing vessel propulsion systems, with results showing acceptable agreement after CFD validation using the CFD numerical method.

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