Study on the Graph Final Design of a Ship CPP

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Abstract. Aiming at the problems that a ship’s Controllable Pitch Propeller (CPP) of insufficient load power and low power-weight ratio did not fully utilized the rated power of the main engine (M/E), we redesigned the CPP by the graph method. Firstly, introduced the structure and function of the main components of the CPP, and analyzed the theoretical basis of the CPP graph design, then selected the CPP type according to the main parameters of the M/E and hull. Secondly, with the graph method, calculated the maximum ship speed and determined the optimal elements of the CPP driven by the M/E rated power and other M/E power, and converted the best factor of other power CPP according to the design propeller diameter. Then, check the strength and cavitation performance of CPP. Thirdly, calculated the navigation characteristics of refrigerated ships to verify the graph design results. Finally, summarized the graph design and drawn the blade extension profile. The Research results show that the CPP can make full use of the M/E rated power after the graph final design, and the CPP power-weight ratio can be improved by retaining the original hub and increasing the blade diameter and area ratio. The method of the graph final design is usable and effective.

1. Introduction

The CPP can adapt to the revs speed of the M/E, it can optimize the matching characteristics of the machine and the propeller to the maximum extent, can make the M/E to works at rated power and rated speed, and it is widely used in the multi-working condition ships with high maneuverability and controllability. Because the CPP efficiency is higher than the fixed pitch propeller, the CPP application is the development trend of ship industry. There are three kinds of the CPP design: the experience design, the graph design and the theoretical design, among which the graph design is simple and reliable, and the ideal scheme can be obtained quickly and accurately. For the CPP design, the graph design is an effective method and it is used extensively at present.

The rated bearing power of the original CPP is only 67.5% of the M/E rated power, and the some M/E power is wasted when the generator is not or partially in operation. And the power weight ratio of the original CPP is lower than that of other CPP with similar size. In order to make full use of the M/E rated power and improve the power-to-weight ratio of the CPP, the final design of the CPP is carried out by using the graph method under the condition of retaining the original propeller hub.

Based on the JDC4-55 CPP graph, we carried out the final design of the CPP driven by each power of the M/E, and determined the optimal scale elements of the CPP. According to the regression polynomial formula of the open water characteristic curve, we predicted the hydrodynamic performance of the design propeller. Checked the blade strength according to the rated working condition, calculated the navigation characteristic to verify the graph final design requirements, and finally summarized the design results of the graph.
2. The CPP Structure and the Function of Main Components

2.1. The Characteristics of the CPP
The ship's CPP adopts plane-tight structure, which is mainly composed of blade and hub. The blade is fastened to the crank pin disc through the root bolt and can be rotated on the hub. There are plane bearings on the hub shell, crank-sliding shoe transmission mechanism in the hub, and servo cylinder behind the hub. The pitch propeller structure is shown in figure 1.

![Figure 1. Schematic diagram of CPP’ structure](image)
1) Servo Cylinder  2) Plane Bearing  3) Crank Pin  
4) Sliding frame 5) Hub Shell  6) Blade  7) Sliding shoe

2.2. The Function of the CPP Main Components
The blade is composed of blade and leaf root flange, as one by casting, it is the main component that chang the M/E power to the force of push ship.

Four plane bearings are arranged in the circumferential direction of the hub shell, to support the blade root flange and crank pin disc, bearing external loads such as blade thrust, tangential force and centrifugal force.

The crank pin is inserted in the sliding shoe that is installed in the transverse groove of the sliding frame, and the sliding frame shaft is rigidly connected with the servo piston. The crank-sliding shoe converts the axial movement of the servo piston into the circumferential rotation of the blade to change pitch.

3. The Graph Design Basis, the Parameters of Main Engine and Hull, the Ropulsion Factors
The CPP final design is known as the M/E power, speed and hull effective power curve, according to the graph to determine the optimal diameter, pitch ratio, efficiency and the maximum ship speed.

3.1. The Graph Design Basis
The graph is the figure, which is drawn according to the test results of the CPP model series in the open water, using it to calculate the hydrodynamic properties of the real propeller, the propeller model and the real propeller should meet the dynamic similarity conditions:1) geometric similarity, any corresponding linear size is in the same proportion, 2) movement similarity, the advance coefficient J is equal; (3) The gravity similarity, the Froude number is equal, when the propeller shaft sunk depth h_s≥1.1D, the effect of wave formation is negligible;4)the viscosity similarity, which the Reynolds number is equal.

In theory, the CPP open water test satisfy all similarity conditions, then the hydrodynamic parameters of the propeller model and the real propeller are equal. In fact, when the advance coefficient and Reynolds number are equal at the same time, the thrust of the propeller model is equal to that of the real propeller, and the thrust of the propeller model is too large to measure. The revs and
advance speed of propeller model are too high to realize the test of propeller model. The Reynolds number of real propeller is higher than that of propeller model, the resistance caused by viscosity is smaller than that of propeller model; the surface of real propeller is rougher than that of propeller die, the increase of resistance caused by roughness basically offsets the small scale effect. Therefore, in practice, the propeller model test only needs to satisfy the equal of advance coefficient, to ensure that the boundary flow of the propeller model is turbulent, and the Reynolds number is not lower than the critical value $R_n = 3.0 \times 10^5$, then the results of the propeller model test can be directly used in the real propeller.

3.2. Main Engine Parameters, Hull Parameters and Thrust Factors

The M/E rated power 5664 KW, rated speed 520 r/min, drives three generators and a CPP through the gearbox. The generator power is one 1200 KW, two 320 KW. and the rated bearing power of the original CPP is 3824 KW, the max bearing power is 4205 KW, the CPP revs is 160 r/min.

The refrigerated vessel is a multi-purpose ocean carrier with single propeller, ball nose head and square tail. The values of the full load effective power curve were calculated by the Ayer method[1]-[4] are shown in Table 1.

| The ship speed(kn) | 15  | 16  | 17  | 18  | 19  |
|-------------------|-----|-----|-----|-----|-----|
| Effective hull power $P_e$(KW) | 1669.62 | 2297.33 | 3028.63 | 4447.62 | 6111.62 |
| Effective hull power $P_e$(hp) | 2271.59 | 3125.62 | 4120.59 | 6051.18 | 8315.13 |

According to the hull dimensions, the propulsion factors were calculated as follows: the side flow fraction $w=0.258$; the thrust reduction fraction $t=0.155$; the shafting transfer efficiency $\eta_s = 0.93$; relative rotation efficiency $\eta_r = 1.0$; hull efficiency $\eta_h = 1.139$.

4. The Graph Design of CPP Drived by Each Power of M/E

The horizontal ordinate of the JDC4-55 CPP graph is the power coefficient and the ordinate is the pitch ratio. There are the diameter coefficient curve, the open water efficiency curve and the optimum efficiency curve in the graph, see figure 2. When the CPP was designed by graph, the power coefficient was calculated first, then the vertical line was made on the graph. The data of the pitch ratio, the open water efficiency and the diameter coefficient, corresponding to the intersection point of the vertical line and the best efficiency line were the best parameters of the CPP under a certain working condition.

Figure 2. Schematic diagram

4.1. Calculation of the Max Designing Ship Speed and Optimal Elements under Every M/E Power

The diesel main engine drive three generators and one CPP, according to the refrigeration needs, the main engine can supply 5664 KW, 5344KW, 5024KW, 4644KW, 41444KW, 3824KW to drive CPP.
Under the drive of the main engine power, the open water received power $P_d$, the power coefficient $B_P$, the effective thrust power $P_{te}$ and the diameter coefficient are given:

$$P_d = P_b \cdot \eta_s \cdot \eta_T \quad \text{KW} \quad (1)$$

$$B_P = 1.166 \frac{n \cdot P_d^{1/2}}{V_a^{1.25}} \quad (2)$$

$$P_{te} = Z \cdot P_d \cdot \eta_0 \cdot \eta_h \quad \text{KW} \quad (3)$$

$$\delta = \frac{n \cdot D}{V_a} \quad (4)$$

where $P_b$ is the rated power of the M/E, KW; $V_a$ is the advance speed of CPP, kn; $Z$ is the number of CPP; $\eta_0$ is the open water efficiency of the CPP;

When CPP was drive by the rated power of M/E, the ship speed was assumed to be 16 kn, 17 kn, 18 kn, 19 kn, the calculation results of the CPP graph design are shown in Table 2.

| Element | Unit | Value  |
|---------|------|--------|
| $V_a$   | kn   | 11.872 |
| $V_a^{1.25}$ | kn | 22.04 |
| $\sqrt{B_P}$ |     | 5.28  |
| JDC4- |  $\delta$ | 61.4  |
| $P_{te}$ | kW  | 3365.8 |

When CPP was driven by the other power of M/E, the ship speed was assumed to be 15 kn, 16 kn, 17 kn, 18 kn, the calculation results of the CPP graph design are shown in Table 3.
### Table 3. Design and calculation table for other power CPP

| Element | Unit | Value | Value | Value | Value |
|---------|------|-------|-------|-------|-------|
| $V$ | $kn$ | 15    | 16    | 17    | 18    |
| $V_a = V(1 - \omega)$ | $kn$ | 11.13 | 11.872| 12.614| 13.356|
| $V_a^{1.25}$ |      | 20.33 | 22.04 | 23.77 | 25.532|
| $\sqrt{B_p}$ |       | 5.64  | 5.20  | 4.82  | 4.49  |
| $\delta$ |       | 65.11 | 60.44 | 56.44 | 52.44 |
| $P_b = 5344\, KW$ | $\eta_0$ | 0.544 | 0.565 | 0.587 | 0.597 |
| $P/D$ | $m$ | 0.772 | 0.808 | 0.848 | 0.874 |
| $D$ |       | 4.529 | 4.485 | 4.45  | 4.377 |
| $P_{te} = 3079.44\, KW$ |       | 3198.32 | 3322.85 | 3379.46 |
| $\sqrt{B_p}$ |       | 5.55  | 5.12  | 4.75  | 4.42  |
| $\delta$ |       | 64    | 59.9  | 56    | 52.4  |
| $P_b = 5024\, KW$ | $\eta_0$ | 0.548 | 0.567 | 0.585 | 0.60  |
| $P/D$ | $m$ | 0.776 | 0.808 | 0.848 | 0.874 |
| $D$ |       | 4.452 | 4.445 | 4.415 | 4.374 |
| $P_{te} = 2958.90\, KW$ |       | 3017.44 | 3113.23 | 3193.06 |
| $\sqrt{B_p}$ |       | 5.39  | 4.97  | 4.61  | 4.29  |
| $\delta$ |       | 62    | 58.4  | 54.8  | 50.8  |
| $P_b = 4464\, KW$ | $\eta_0$ | 0.556 | 0.575 | 0.593 | 0.606 |
| $P/D$ | $m$ | 0.79  | 0.82  | 0.848 | 0.886 |
| $D$ |       | 4.313 | 4.333 | 4.32  | 4.24  |
| $P_{te} = 2629.09\, KW$ |       | 2718.93 | 2804.05 | 2865.52 |
| $\sqrt{B_p}$ |       | 5.29  | 4.88  | 4.53  | 4.22  |
| $\delta$ |       | 61.4  | 57.2  | 53.8  | 50.44 |
| $P_b = 4144\, KW$ | $\eta_0$ | 0.56  | 0.581 | 0.595 | 0.609 |
| $P/D$ | $m$ | 0.79  | 0.83  | 0.86  | 0.89  |
| $D$ |       | 4.271 | 4.244 | 4.241 | 4.21  |
| $P_{te} = 2458.18\, KW$ |       | 2550.36 | 2611.81 | 2673.27 |
| $\sqrt{B_p}$ |       | 5.19  | 4.79  | 4.44  | 4.13  |
| $\delta$ |       | 60    | 56.2  | 52.8  | 49.1  |
| $P_b = 3824\, KW$ | $\eta_0$ | 0.566 | 0.585 | 0.60  | 0.615 |
| $P/D$ | $m$ | 0.803 | 0.84  | 0.87  | 0.91  |
| $D$ |       | 4.174 | 4.169 | 4.163 | 4.099 |
| $P_{te} = 2288.61\, KW$ |       | 2369.63 | 2430.39 | 2491.15 |

Taking the ship speed as the horizontal coordinate, the curves of the effective thrust power, the diameter, the pitch ratio, the open water efficiency and the effective hull power of every power CPP were drawn from 5664kW to 3824kW in turn, as shown in figure 3.
Figure 3. Design ship Speed and Optimal Elements of other power CPP

In figure 3, when $P_{te} > P_e$, the refrigerated ship accelerates forward under the push of CPP; when $P_{te} = P_e$, the effective thrust power of CPP is balanced with the effective hull power, and the refrigerated ship will go forward at the constant speed at the intersection point; when $P_{te} < P_e$, the refrigerated ship is blocked and the ship speed drops. Therefore, the intersection speed is the maximum ship speed that the refrigerated ship can achieve. The corresponding values of the intersection point between the vertical line of the maximum ship speed and the curves of diameter, pitch ratio and open water efficiency are the best element value of each power CPP, as shown in Table 4.

### Table 4. Max design ship speed and CPP Optimal Elements for every Power

| Element | Unit | Design of Optimal Element Value of Ship Speed and Pitch |
|---------|------|------------------------------------------------------|
| $P_b$   | KW   | 5664 5344 5024 4464 4144 3824                        |
| $V_{max}$ | kn  | 17.30 17.22 17.06 16.65 16.37 16.09                  |
| $\delta$ |      | 55.834 55.576 55.776 56.072 57.08 55.636            |
| $\eta_0$ |      | 0.584 0.589 0.588 0.579 0.586 0.567                 |
| $P/D$   |      | 0.8391 0.842 0.8421 0.838 0.841 0.823               |
| $D_i$   | m    | 4.51 4.434 4.412 4.325 4.243 4.168                  |
| $P_{te}$ | KW  | 3496.5 3335.05 3118.3 2774.4 2572.9 2375.15        |

4.2. Determination of the CPP Diameter and the Pitch Conversion of Every Power

The optimum diameter, pitch ratio and open water efficiency of different power CPP are different. The diameter of CPP driven by rated power is the largest, and it is also the design diameter of the refrigerated ship CPP. The optimum diameter and pitch of other power CPP need to be converted into the design diameter, different pitch but same hydrodynamic performance.

When the CPP revs speed is constant, the power that CPP received under a certain power is same, so the diameter increases and the pitch must be reduced, and vice versa. When the CPP diameter varies in the range of -5%~10% of the optimum diameter, the sum of diameter and pitch can be considered as
constant, and the hydrodynamic performance of CPP is basically the same[5]. To take the design propeller diameter as the standard, the change amplitude of other power CPP diameter is less than 10, see Table 5; therefore, the best elements of other power CPP can be converted according to the design diameter, see Table 6.

Table 5. CPP diameter change table for other power

| Element | Unit | the change amplitude of other power CPP diameter |
|---------|------|-----------------------------------------------|
| \( P_b \) | KW  | 5344 5024 4464 4144 3824                      |
| \( P_d \) | KW  | 4969.92 4672.32 4151.52 3853.92 3556.32      |
| \( D_i \) | m   | 4.434 4.412 4.325 4.243 4.168                |
| \( D \)   | m   | 4.51 4.51 4.51 4.51 4.51                      |
| \( \Delta \) | %  | +1.7 +2.2 +4.3 +6.3 +8.2                       |

Table 6. The adjustment of the CPP optimum elements

| Element | Unit | the optimum pitch after adjusting |
|---------|------|----------------------------------|
| \( P_b \) | KW  | 5344 5024 4464 4144 3824          |
| Before adjusting \( D_i \) | m   | 4.434 4.412 4.325 4.243 4.168    |
| adjusting \( P_i \) | m   | 3.733 3.715 3.624 3.568 3.430    |
| After adjusting \( D \) | m   | 4.51 4.51 4.51 4.51 4.51         |
| Adjusting \( P \) | m   | 3.657 3.617 3.439 3.301 3.088    |
| \( P/D \)  |     | 0.8109 0.802 0.7625 0.7319 0.6847 |

4.3. Calculation of Open Water Characteristics of CPP

The regression expressions of thrust coefficient, torque coefficient and open water efficiency of the combination of initial pitch ratio and advance speed coefficient of JDC4-55CPP model series are given by \( K_T, K_Q \) and \( \eta_0 \) [6]:

\[
K_T = \sum_i \sum_j B_T (P/D)^i (J)^j
\]

\[
K_Q = \sum_i \sum_j B_Q (P/D)^i (J)^j
\]

\[
\eta_0 = \frac{K_T}{K_Q} \cdot \frac{J}{\pi}
\]

Where \( B_T, B_Q \) is coefficient; \( i, j \) is index and \( J \) is advance coefficient.

During the calculation, the advance coefficient \( J \) was respectively taken 0.1~0.8, and the open water characteristic curves of CPP were drawn according to the calculation result of the open water performance, as shown in figure 4.
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Figure 4. The open water characteristic curve of CPP

4.4. The Correction of the CPP Pitch
The thickness ratio of design propeller and series blades is equal and the ratio of hub diameter is different, in order to make their hydrodynamic performance equal, the design pitch need to be corrected. JDC4-55 series hub diameter ratio $d_h/D = 0.28$, design hub diameter ratio $(d_h/D)' = 0.277$, pitch ratio correction value is $\Delta(P/D) = 0.0003$, the modified pitch ratio for each power CPP were shown in Table 7.

Table 7. Adjusting elements of CPP

| Element | Unit | The modified pitch of each power CPP |
|---------|------|------------------------------------|
| $P_b$   | KW   | 5664 5344 5024 4464 4144 3824 |
| P/D     |      | 0.8388 0.8106 0.8017 0.7622 0.7316 0.6844 |

4.5. The Vacuole Check of the CPP
Vacuole checking was carried out for CPP by using the Burrell boundary line. The horizontal coordinate of the Burrell boundary line is the vacuoles number at 0.7R section, and the vertical coordinate is the average thrust coefficient on unit projected area, as shown in figure 5.

Figure 5. Pollier bubble limit Chart
When the maximum ship speed was 17.3 kn, the minimum area ratio of the CPP was 0.476, the design propeller area ratio was 0.55, so the design propeller meets the upper limit requirement of the merchant ship CPP without cavitation.

5. Strength Check of CPP

According to the specification, the blade strength is checked to meet the requirements of the CPP entry classification, and then the blade stress is checked to confirm that the maximum stress does not exceed the allowable range.

5.1. Check Strength According to Specifications

The Code for the Entry and Construction of Steel Ships requires that the maximum thickness of the blade at R0.35 and R0.60 section of the CPP must not be less than the calculated values of the following formula [7]:

\[
t = \left( \frac{Y}{K - X} \right)^{1/2}
\]

\[
Y = \frac{1.36A_1P_d}{2bne}
\]

\[
X = \frac{A_2\rho A_1n^2D^3}{10^{10}Zb}
\]

\[
A_1 = \frac{D}{P} \left( K_1 - K_2 \frac{D}{P_{0.7R}} + K_3 \frac{D}{P_{0.7R}} - K_4 \right)
\]

\[
A_2 = \frac{D}{P} (K_5 + K_6\epsilon) + K_7\epsilon + K_8
\]

Where \(Y\) is power factor; \(X\) is torque factor; \(K\) is material factor; \(D\) is the CPP diameter, m; \(P\) is the pitch of cross section, m;

After calculating, the minimum thickness of the blade required by the entry classification is as follows: \(t_{0.35R} = 132.79\ mm, t_{0.6R} = 62.584\ mm\); actual thickness of the CPP is: \(t_{0.35R} = 151.31\ mm; t_{0.6R} = 98.32\ mm\). The design thickness is greater than the minimum thickness required by the classification, and the CPP meets the requirements of entry the classification.

5.2. Root Section Stress Check

When the pitch propeller works, the root section is most stressed, and the blade fracture occurs at the root. In order to ensure that the blade has sufficient strength, the maximum stress of the root section should be within the allowable range. The load on the blade is thrust, tangential force, centrifugal force, bending moment, torque, rotating moment, etc. The distribution of three forces and three moments is shown in figure 6.

![Figure 6. Schematic diagram of the force radius on the blade](image)

1) \(R_g\) is radius of gravity  2) \(R_f\) is tangential force radius  3) \(R_t\) is thrust radius  4) \(R_h\) is hub radius  5) \(C_r\) is leaf root section centroid  6) \(R_o\) is the distance from blade surface to blade axis  7) \(L_x\) and \(L_y\) are centrifugal force arm
5.2.1. The calculation of each force radius. The blade weight of the CPP is composed of the weight of each blade element. The blade was divided into 12 stations along the radial direction, including 6 stations from 0.277R to 0.4R and 6 stations from 0.4R to 1.0R. The blade radius of gravity center, which was calculated according to the gravity center radius of each station, was $R_g = 1.306 \text{ m}$.

The thrust is distributed linearly along the radial direction, and the thrust moment of the blade equal to the thrust moment sum of each blade section, then the thrust radius was calculated by $R_t = \frac{2}{3} \cdot \frac{R^2 + Rr_h + r_h^2}{R + r_h} = 1.594 \text{ m}$.

The tangential force is uniformly distributed along the radial direction, and the tangential moment equal to the tangential moment sum of each blade section. The tangential force radius is obtained from $R_t = \frac{R + r_h}{2} = 1.44 \text{ m}$.

5.2.2. The calculation of each moment. Thrust and tangential force bend the blade and centrifugal force stretch the blade. The blade has no posterior oblique but slightly oblique, the centrifugal force action line does not pass through the leaf root section, and the centrifugal force action line does not coincide with the rotating blade axis. According to the right hand rule, in the x direction, the thrust moment $M_T$ and the centrifugal force moment $M_C$ are in the same direction; in the y direction, the rotating resistance moment $M_F$ and the centrifugal force moment $M_S$ are in the same direction, see Figure 7.

A single blade thrust force $T_1 = 116.245 \text{ KN}$, produced a bending moment $M_T = 1.13 \times 10^5 \text{ N} \cdot \text{m}$ at the root of the blade. A single blade was subjected to tangential force $F_{C1} = 54.61 \text{ KN}$, resulted in bending moment $M_F = 4.451 \times 10^4 \text{ N} \cdot \text{m}$ at the root of the blade. The weight of a single blade is 1207.465 Kg, which produces centrifugal force $F_{C1} = 442256.76 \text{ N}$; Centrifugal arm $L_x = 0.112\text{m}$, $L_y = 0.06 \text{m}$, then x direction centrifugal torque $M_C = 4.953 \times 10^4 \text{ N} \cdot \text{m}$, y direction centrifugal torque $M_S = 2.654 \times 10^4 \text{ N} \cdot \text{m}$.

5.2.3. Analysis and calculation of root section stress. The minimum inertia moment axis 1-1 and the maximum inertia moment axis 2-2 were established on the leaf root section. The inertia moment axis passes through the gravity center of the leaf root section. According to the right hand rule, the bending moment directions are shown in figure 7.

**Figure 7.** Schematic illustration of bending moment of blade root section

The synthetic moment of thrust, tangential force and centrifugal force on the inertia axis of the blade root section were given by $M_1, M_2$:

\[
M_1 = (M_T + M_C) \cos \Phi + (M_F + M_S) \sin \Phi
\]

\[
M_2 = (M_T + M_C) \sin \Phi - (M_F + M_S) \cos \Phi
\]

where $\Phi$ was Leaf root pitch angle, $\Phi = 15.5^\circ$. 

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The synthetic moment calculation results were $M_1 = 1.756 \times 10^5 \text{ N} \cdot \text{m}$, $M_2 = -2.504 \times 10^4 \text{ N} \cdot \text{m}$. The synthetic moment “+” meant that the direction of moment was consistent with the direction of moment axis, and “-” meant that the direction of moment was opposite to that of moment axis. The leaf root section was divided into four regions I, II, III and IV by the moment of inertia axis 1 and 2. The stress properties of bending moment and centrifugal force in each region were shown in Table 8.

| Regional | I   | II  | III | IV  |
|----------|-----|-----|-----|-----|
| Stress by $M_1$ | compressive | compressive | tensile | tensile |
| Stress by $M_2$ | tensile | compressive | compressive | tensile |
| Stress by $F_{C1}$ | tensile | tensile | tensile | tensile |

On the leaf root section, the farthest coordinate points from the inertia axis were taken, and the mutual positions of each coordinate point were shown in figure 8.

Figure 8. Drawing of maximum coordinate point’s site of root section

The stress of bending moment $M_1$ and moment $M_2$ acting on the maximum coordinate point were given by $\sigma_1(X), \sigma_2(X)$:

$$\sigma_1(X) = \frac{M_1}{W_{1X}}, \quad \sigma_2(X) = \frac{M_2}{W_{2X}}$$

(15)

Where $W_{1X}$ was the bending section modulus for inertia axis 1 at maximum coordinate point, m$^3$; $W_{2X}$ was the bending section modulus for inertia axis 2 at maximum coordinate point, m$^3$.

Under the action of thrust, tangential force and centrifugal force, the total stress of the maximum coordinate point were given by $\sigma(A), \sigma(B), \sigma(C)$ and $\sigma(D)$:

$$\sigma(A) = \sigma_1(A) + \sigma_2(A) + \sigma_C$$

(16)

$$\sigma(B) = \sigma_1(B) - \sigma_2(B) + \sigma_C$$

(17)

$$\sigma(C) = -\sigma_1(C) + \sigma_2(C) + \sigma_C$$

(18)

$$\sigma(D) = \sigma_1(C) + \sigma_2(C) + \sigma_C$$

(19)

Where $\sigma_C$ was the stress of centrifugal force on the cut surface of the leaf root, Pa.

The partial stress and total stress of the maximum coordinate point were shown in Table 9.

| Element | Unit | A    | B    | C    | D    |
|---------|------|------|------|------|------|
| $\sigma_1(X)$ | MPa  | 37.8 | 23.4 | 54.5 | 69.96 |
| $\sigma_2(X)$ | MPa  | 1.15 | 1.49 | 0.31 | 1.15 |
| $\sigma(X)$   | MPa  | 43.72| 26.68| -49.42 a| 75.88 |

a The “-” Symbol represented the compressive stress.
From Table 9, it can be seen that the total stress at each maximum coordinate point were less than that of the leaf root Permissible stresses $[\sigma] = 110 \text{ MPa}$, the leaf root section met the strength requirement.

5.3. Root Section Stress Check

The plane bearing of the hub shell is not only subjected to the centrifugal force of the blade-leaf root flange-crank pin disc, but also by the combined bending moment of the blade thrust and tangential force. The plane bearing outer diameter $D = \Phi 676 \text{ mm}$, inner diameter $d = \Phi 560 \text{ mm}$, and the stress caused by the centrifugal force in the lower plane of bearing is $\sigma_1 = 4.59 \text{ MPa}$. The synthetic moment of thrust, tangential force is $M_{sm} = 1.371 \times 10^5 \text{ N\cdotm}$, the anti flexural section modulus of plane bearing is $W = 0.016 \text{ m}^3$, the maximum stress of synthetic moment in plane bearing is $\sigma_2 = 8.55 \text{ MPa}$.

When the pitch propeller works, the plane stress under the plane bearing is $\sigma = 13.14 \text{ MPa}$. Allowable stress $ZQCuAL9-4-4-2$ propeller hub material is $[\sigma] = 98 \text{ MPa}$. ∵ $\sigma < [\sigma]$, ∴ the propeller hub plane bearing strength meet the requirements.

6. The Calculation of CPP Navigational Performance and CPP Performance Analysis

6.1. Calculation of CPP Navigational Characteristics

Different pitch CPP produces different effective thrust power, at the same time it consumes different power of M/E. The calculation[8] of navigation characteristics is to calculate the required M/E power according to the open water characteristic curve, which is the inverse calculation of the graph design. The curve of the CPP effective thrust power, the hull effective power and the power of M/E were drawn according to the calculation results of the navigation characteristics. The ship speeds at intersection of curves were the maximum ship speed, the powers at the intersection point of the maximum ship speed vertical line and the main engine power curve were the required M/E powers, which were shown in Table 10.

| Element | The value of navigational characteristics |
|---------|------------------------------------------|
| P/D     | 0.8388 0.8188 0.8017 0.7622 0.7316 0.6844 |
| $V_{max}$ | 17.71 17.45 17.38 17.09 16.77 16.11 |
| $P_s$    | 5685.6 5340.8 5077.9 4455.7 4042 3499.4 |
| $P_{te}$ | 4041.8 3720.7 3618.2 3179.2 2826.4 2357.3 |

The deviations between the calculation results of navigation characteristics and the requirements of the graph design were shown in Table 11. It can be seen that the ship speed error and power error of the other pitch ratio were less than 3% except the power error of the pitch ratio 0.6847.

| Element | Value |
|---------|-------|
| P/D     | 0.8391 0.8191 0.802 0.7625 0.7319 0.6847 |
| Navigation ship speed / kn | 17.71 17.45 17.38 17.09 16.77 16.11 |
| Design ship speed / kn | 17.30 17.22 17.06 16.65 16.37 16.09 |
| Error % | 2.37 1.34 1.88 2.64 2.44 0.12 |
| Navigation power / kW | 5685.6 5340.8 5077.9 4455.7 4042 3499.4 |
| Design power / kW | 5664 5344 5024 4464 4144 3824 |
| Error % | 0.38 -0.06 1.07 -0.19 -0.02 -8.48 |
6.2. CPP Performance Analysis

The normal operation of the shaft generator required that the M/E speed was basically constant, but the M/E power supplying the CPP might be vary. Under the same advance speed, the torque coefficient was changed by changing the pitch, and the CPP could absorb different M/E power. Under the different advance speed, the torque coefficient could be kept unchanged by changing the CPP pitch, and the CPP could absorb all M/E power. Both conditions made the M/E always works at rated power and rated speed, which could reduce the fuel consumption and improve the economy of the M/E operation. The CPP performance was shown in figure 9.

![CPP performance curve](image)

**Figure 9.** Working performance curve of CPP

7. Summary of the CPP Design

In order to make full use of the M/E power, the final design of each power drive CPP were carried out by graph method, which was shown in Table 12, and the CPP best elements driven by each power were shown in Table 13.

| Table 12. Summary of CPP design |
|--------------------------------|
| **Element** | **Content** |
| CPP diameter | D = 4.51m |
| Type | JDC |
| Number of blades | Z = 4 |
| Area ratio | $A_e/A_0 = 0.55$ |
| Rake | $\varepsilon = 0^\circ$ |
| Hub ratio | $d_h/D = 0.277$ |
| Turning direction | Right handed |
| Leaf material | ZQAL12-8-3-2 |
| Hub Material | ZCuAL9-4-4-2 |
| Weight | 12995.86 Kg |
| Inertia Moment | 9451.58 Kgf·m² |

| Table 13. The list optimal elements of CPP design |
|--------------------------------|
| **$P_b$ (kw)** | **$V_{max}$ (kn)** | **$P/D$** | **$\eta_0$** |
| 5664 | 17.30 | 0.8388 | 0.584 |
| 5344 | 17.22 | 0.8188 | 0.589 |
| 5024 | 17.06 | 0.8017 | 0.588 |
| 4464 | 16.65 | 0.7622 | 0.579 |
| 4144 | 16.37 | 0.7316 | 0.586 |
| 3824 | 16.09 | 0.6844 | 0.567 |
According to the geometric scale table of JDC series pitch propeller, to calculate the chord length and maximum thickness of each section of CPP, the extension profile and pitch distribution of blade was shown in figure 10.

![Figure 10. Stretch profile and pitch distribution of the CPP blade](image)

8. Conclusion

- The graph method can meet the requirements of the CPP final design driven by each M/E power, in any circumstance CPP can fully utilized the rated power of M/E.
- By replacing the blades, the power-weight ratio of CPP is increased from 0.302 kW/kg to 0.412 kW/kg, to save the material cost of CPP.
- According to the design propeller diameter, we converted the best elements of other power CPP, to solve the pitch problem of the design propeller at other power, and creates conditions for calculating the hydrodynamic characteristics of other power CPP.
- The reson, which the deviation is slightly larger between the Navigation calculation power and the required power of the graph design at pitch ratio 0.6847, is that the Uniformity flow field behind the ship has an effect on the thrust and torque of the CPP, and the change of hydrodynamic performance of the CPP in the non-uniform flow field behind the ship has the value of further research.
- On the basis of the graph design, the theoretical design optimization of the CPP can be carried out in order to further improve the hydrodynamic performance of the CPP. The main geometric parameters of CPP are determined after synthetically considering the efficiency of CPP, cavitation and hub mechanism.

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