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Effect of dressing parameters when external cylindrical grinding 90CrSi on wheel life time

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ABSTRACT

This paper studies the influence of the dressing parameters on the external cylindrical grinding process on wheel life time to achieve the maximum grinding wheel life time. In this study, six dressing parameters were investigated, including depth of rough dressing, rough dressing time, depth of fine dressing, fine dressing time, non-feeding dressing time, and dressing speed. The study used the response surface method to design the experiment and ANOVA method to analyze the effect of input parameters on response. This study propose the regression equation describes the relationship between the dressing parameters and grinding wheel life time and the optimal dressing condition to achieve the maximum wheel life time was determined.

Keywords—*external grinding; dressing condition; surface roughness;*

I. INTRODUCTION

Grinding process is one of the most important final process in manufacturing. This process could processing the high accuracy and high surface quality of workpiece. But this is also a high-cost manufacturing process. Therefore, there are many researches to reduce the processing cost and increase the grinding performance.

The previous researches presented there are many processing parameter affect on the performance of grinding process as grinding condition, cooling condition and dressing condition… these processing parameters may affect on the quality of workpiece and grinding[1]. In the study of Sandeep Kumar presented the workpiece speed, grinding wheel speed and feed rate has more significant effect for surface roughness in external cylindrical grinding when grinding EN15 AM steel[2]. The study of Ja-Seob Kwak in grinding hardened SCM440 steel indicated the depth of cut has more effect on surface roughness than traverser speed [3].The grinding processing parameter on material removal rate in external cylindrical grinding EN45 steel also is mentioned in the research of Dersse [4]. The effect of the different cutting fluid on grinding performance have been study by Talon [5], and the minimum quantity lubricant (MQL) method also has been applied to improve the grinding quality and reduce grinding cost [6-8].

In the grinding process, not only improving the surface quality of the workpiece, but also improving productivity and reducing costs is also focused on research. The different study method have been applied to optimum the grinding condition as Taguchi method, response surface method, neural network and genetic algorithm[9, 10]. The exchanged grinding wheel to obtain the lowest grinding cost have been studied also [11, 12].

In addition to the effect of grinding condition, the dressing condition also has a great influence on the productivity and quality of the grinding process. Because the dressing condition determine the profile of the grinding wheel[13, 14]. But very few researchers investigate the effect of dressing condition on the grinding wheel life time [15]. Therefore, this study focus on the influence of dressing parameter on wheel life time when external cylindrical grinding the 90CrSi harden steel. Moreover, by applying the ANOVA method to find out the optimum dressing condition to maximum the wheel life time.

II. EXPERIMENTAL DESIGN

In this study, an experiment design using response surface Box-Behnken method was applied to investigate the effect of six dressing parameters including depth of rough dressing, rough dressing time, depth of fine dressing, fine dressing time, non-feeding dressing time, and dressing speed on surface roughness when external cylindrical grinding 90CrSi. The investigated levels of input factors were shown in table 1.

TABLE 1.INPUT PARAMETERS AND INVESTIGATED LEVELS

No.	**Input parameters**	**Symbol**	**Unit**	**Level of investigating**	
				Low	High
1	Depth of rough dressing	a_r	mm	0.02	0.04
2	Rough dressing time	n_r	-	1	5
3	Depth of fine dressing	a_f	mm	0.005	0.015
4	Fine dressing time	n_f	-	0	4
5	Non-feeding dressing time	n_0	-	0	4
6	Feed speed	S_d	m/min	1	2

The experiments were conducted using the following fixed grinding conditions and equipment: The Cantext Aquatex 3810 with a concentration of 3% and a flow-rate of 10 l/min was used for coolant condition; the grinding condition is: feed rate at 1.8 m/min, depth per cut is 0.005 per single stroke, total depth of cut is 0.05 mm and the speed of grinding wheel is 29.3 m/s; Grinding machine: CONDO-Hi-5 HTS (Japan origin); grinding wheel: Ct80MV1-G 400x40x203 35m/2 (Vietnam origin); dressing tool: 3908-0088C type 2 (Russian origin); cutting force measurement: Kistler 5233A (German). The workpiece properties are described in table 2.

TABLE 2.90CrSi STEEL PROPERTIES

Steel grade: 90CrSi						
C	Si	Mn	Cr	P	S	Co
0.85-0.95	12-1.6	0.3-0.6	0.95-1.25	≤0.03	≤0.03	≤1

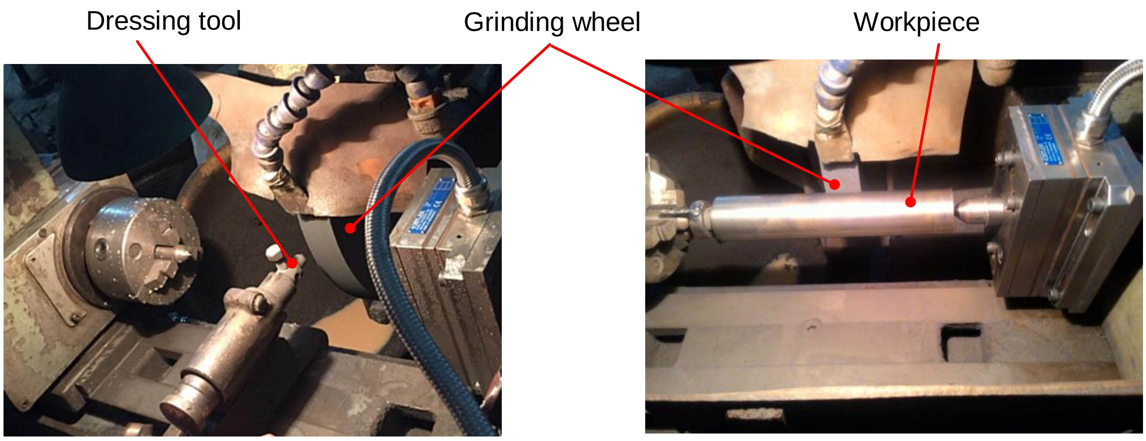

Fig 1. Experiment setup

According to the response surface method, the experiment design is calculated based on the input data in table 1. By using Minitab R19, the experiment design table with 54 experiments is listed in table 3. The wheel life time is determined by cutting force measurement. The experiment result is described in table3 either.

TABLE 3.EXPERIMENT DESIGN AND MEASUREMENT RESULTS

StdOrder	RunOrder	PtType	Blocks	a_r	n_r	a_f	n_f	n_0	S_d	Wheel life time T(min)
16	1	2	1	0.03	5	0.015	2	4	1.5	39.82
28	2	2	1	0.04	3	0.010	4	0	1.5	44.54
24	3	2	1	0.03	3	0.015	4	2	2.0	39.39
44	4	2	1	0.04	3	0.015	2	2	1.0	44.15
26	5	2	1	0.04	3	0.010	0	0	1.5	43.56
21	6	2	1	0.03	3	0.005	0	2	2.0	38.95
27	7	2	1	0.02	3	0.010	4	0	1.5	38.44
49	8	0	1	0.03	3	0.010	2	2	1.5	45.89
10	9	2	1	0.03	5	0.005	2	0	1.5	40.03
2	10	2	1	0.04	1	0.010	0	2	1.5	43.05
32	11	2	1	0.04	3	0.010	4	4	1.5	44.97
35	12	2	1	0.03	1	0.010	2	4	1.0	38.55
20	13	2	1	0.03	3	0.015	4	2	1.0	38.63
30	14	2	1	0.04	3	0.010	0	4	1.5	43.05
42	15	2	1	0.04	3	0.005	2	2	1.0	45.23
8	16	2	1	0.04	5	0.010	4	2	1.5	44.94
3	17	2	1	0.02	5	0.010	0	2	1.5	41.62
53	18	0	1	0.03	3	0.010	2	2	1.5	45.92
40	19	2	1	0.03	5	0.010	2	4	2.0	39.73
6	20	2	1	0.04	1	0.010	4	2	1.5	44.04
51	21	0	1	0.03	3	0.010	2	2	1.5	45.88
54	22	0	1	0.03	3	0.010	2	2	1.5	45.74
29	23	2	1	0.02	3	0.010	0	4	1.5	40.88
43	24	2	1	0.02	3	0.015	2	2	1.0	41.79
50	25	0	1	0.03	3	0.010	2	2	1.5	45.95
12	26	2	1	0.03	5	0.015	2	0	1.5	39.62
48	27	2	1	0.04	3	0.015	2	2	2.0	44.26
15	28	2	1	0.03	1	0.015	2	4	1.5	38.97
5	29	2	1	0.02	1	0.010	4	2	1.5	40.86
38	30	2	1	0.03	5	0.010	2	0	2.0	39.04
17	31	2	1	0.03	3	0.005	0	2	1.0	39.79
31	32	2	1	0.02	3	0.010	4	4	1.5	42.14
36	33	2	1	0.03	5	0.010	2	4	1.0	42.68
9	34	2	1	0.03	1	0.005	2	0	1.5	39.15
33	35	2	1	0.03	1	0.010	2	0	1.0	39.26
22	36	2	1	0.03	3	0.015	0	2	2.0	39.22
14	37	2	1	0.03	5	0.005	2	4	1.5	40.15
4	38	2	1	0.04	5	0.010	0	2	1.5	43.88
37	39	2	1	0.03	1	0.010	2	0	2.0	38.81
45	40	2	1	0.02	3	0.005	2	2	2.0	41.54
1	41	2	1	0.02	1	0.010	0	2	1.5	40.88
34	42	2	1	0.03	5	0.010	2	0	1.0	39.23
18	43	2	1	0.03	3	0.015	0	2	1.0	40.37
52	44	0	1	0.03	3	0.010	2	2	1.5	45.66
19	45	2	1	0.03	3	0.005	4	2	1.0	40.84
23	46	2	1	0.03	3	0.005	4	2	2.0	40.49
7	47	2	1	0.02	5	0.010	4	2	1.5	42.25
41	48	2	1	0.02	3	0.005	2	2	1.0	43.26
25	49	2	1	0.02	3	0.010	0	0	1.5	42.13
47	50	2	1	0.02	3	0.015	2	2	2.0	41.96
39	51	2	1	0.03	1	0.010	2	4	2.0	38.96
13	52	2	1	0.03	1	0.005	2	4	1.5	39.27
46	53	2	1	0.04	3	0.005	2	2	2.0	44.68
11	54	2	1	0.03	1	0.015	2	0	1.5	39.15

III. RESULT AND DISCUSSIONS

In order to investigate the effect of dressing parameters on wheel life time, the ANOVA method is conducted. The effect of input factor are described in the figure 2.

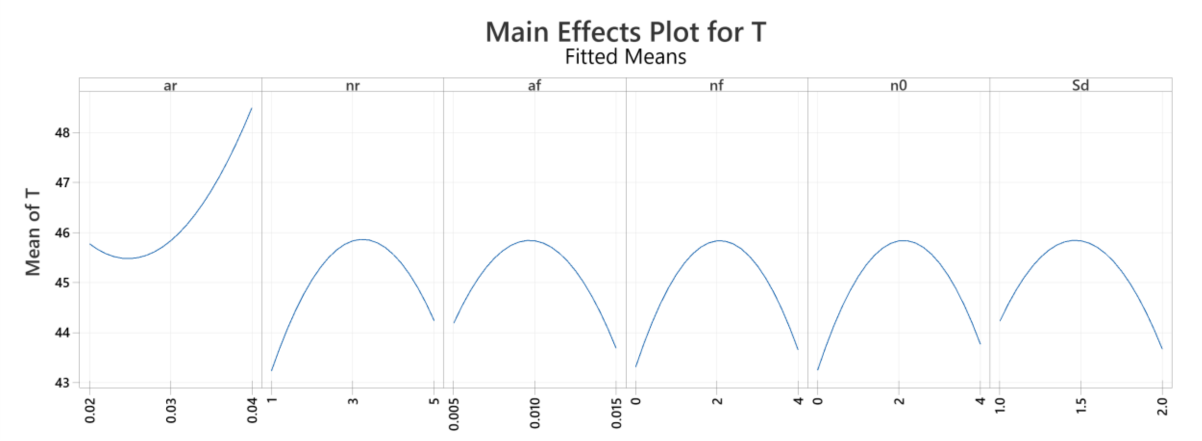

Fig 2.Effect of input factors on surface roughness

Observing the analysis result in figure 2 shows all of the input factors affect the output response in the quadratic form. This result indicated that investigated area has extreme value and the optimal value of the input parameter can be determined.

The influence degree of investigated parameter is described in the Pareto chart in figure 3. The factors which cross over the red reference line are significant factors.

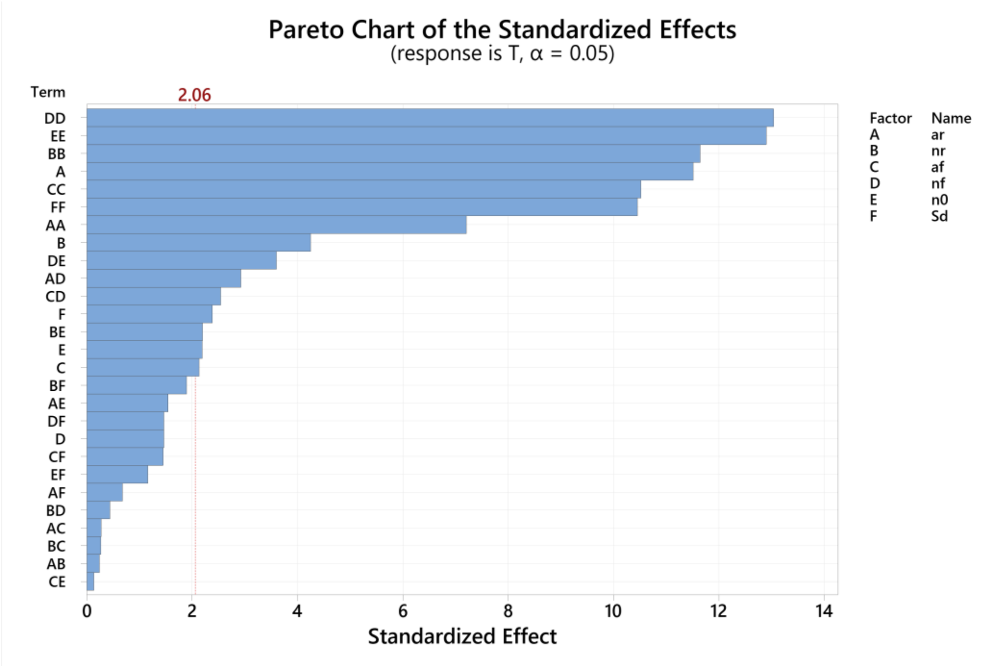

Fig 3. The influence degree of input factor on surface roughness

The figure 3shows clearly that the quadratic effect of DD (n_f*n_f) is largest, then followed by the effect of EE (n_0*n_0), BB (n_r*n_r), A (a_r), CC (a_f*a_f), FF (S_d*S_d), AA (a_r*a_r), B (n_r), DE (n_f*n_0), AD (a_r*n_f), CD (a_f*n_f), F (S_d), BE (n_r*n_0), E (n_0) and C (a_f).

The influence trend of the input factors is also determined and described in the figure 4.

Fig 3. The normal plot of the standardized effects of input factors on wheel life time

The figure 4 indicated the following factors: A (a_r),AA (a_r*a_r),B (n_r),DE (n_f*n_0),AD (a_r*n_f),BE (n_r*n_0)E (n_0) have positive effect on the wheel life time. Otherwise, the factors DD (n_f*n_f),EE (n_0*n_0),BB (n_r*n_r), CC (a_f*a_f), FF (S_d*S_d), CD (a_f*n_f), F (S_d),C (a_f) have negative effect on the wheel life time.

The ANOVA method was applied to calculate the effect of input factors to output response. After remove the negligible influence factors, the analysis result is described in the table 4.

TABLE 4.ANALYSIS OF VARIANCE

Analysis of Variance

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Model	16	311.649	19.4781	56.02	0.000
Linear	6	56.070	9.3450	26.88	0.000
ar	1	44.282	44.2817	127.36	0.000
nr	1	6.040	6.0401	17.37	0.000
af	1	1.525	1.5251	4.39	0.043
nf	1	0.718	0.7176	2.06	0.159
n0	1	1.607	1.6068	4.62	0.038
Sd	1	1.898	1.8984	5.46	0.025
Square	6	244.603	40.7671	117.25	0.000
ar*ar	1	17.349	17.3494	49.90	0.000
nr*nr	1	45.306	45.3060	130.31	0.000
af*af	1	36.985	36.9850	106.38	0.000
nf*nf	1	56.803	56.8029	163.37	0.000
n0*n0	1	55.601	55.6007	159.92	0.000
Sd*Sd	1	36.499	36.4990	104.98	0.000
2-Way Interaction	4	10.977	2.7443	7.89	0.000
ar*nf	1	2.865	2.8646	8.24	0.007
nr*n0	1	1.613	1.6129	4.64	0.038
af*nf	1	2.163	2.1632	6.22	0.017
nf*n0	1	4.337	4.3365	12.47	0.001
Error	37	12.864	0.3477		
Lack-of-Fit	32	12.799	0.4000	30.77	0.001
Pure Error	5	0.065	0.0130		
Total	53	324.514			

Model Summary

S	R-sq	R-sq(adj)	R-sq(pred)
0.589648	96.04%	94.32%	90.41%

The regression equation of wheel life time is described by the following model:

Regression Equation in Uncoded Units

T = 21.12 - 686 ar + 3.240 nr + 1571 af + 1.954 nf + 1.848 n0 + 22.04 Sd + 12988 ar*ar- 0.5247 nr*nr - 75850 af*af - 0.5875 nf*nf - 0.5812 n0*n0 - 7.535 Sd*Sd + 21.16 ar*nf+ 0.0794 nr*n0 - 52.0 af*nf + 0.1841 nf*n0

the optimal value of the input parameters to achieve the maximum wheel life time is determined by using Minitab R19 software base on the regression equation above.

TABLE 5.THE CALCULATED OF THE OPTIMAL VALUE OF THE INPUT PARAMETERS

Solution

Solution	ar	nr	af	nf	n0	Sd	T Fit	95% CI	Composite Desirability
1	0.04	3.2222	0.0094	2.2626	2.0202	1.4747	48.5840	(48.058,49.110)	1

From the optimal calculation results in table 5, an optimal set of dressing condition parameters can be obtained to achieve the longest wheel life time based on the adjustment range of the machine are a_r = 0.04 mm, n_r = 3 times, a_f = 0.01 mm, n_f = 2 times, n_0 = 2 times and S_d = 1.5 m/min. The predicted value of wheel life time corresponding to optimum dressing condition is 48.5840minutes. By the confident interval 95% the wheel life time is distributed in the range from 48.058 to 49.110 minutes.

IV. CONCLUSION

This study presents an investigation to determined the effect of dressing condition to wheel life time in external cylindrical grinding. The response surface method and ANOVA are applied to find the orthogonal array for the experimental plan. The following conclusions can be made:

- The effect of quadratic of DD (n_f*n_f) is largest, then followed by the effect of EE (n_0*n_0), BB (n_r*n_r), A (a_r), CC (a_f*a_f), FF (S_d*S_d), AA (a_r*a_r), B (n_r), DE (n_f*n_0), AD (a_r*n_f), CD (a_f*n_f), F (S_d), BE (n_r*n_0), E (n_0) and C (a_f).

- The influence trend of each parameter on the wheel life time has been determined also. The analysis result indicated that the parameters A (a_r),AA (a_r*a_r),B (n_r),DE (n_f*n_0),AD (a_r*n_f),BE (n_r*n_0)E (n_0) have positive effect on the wheel life time. Otherwise, the parameters DD (n_f*n_f),EE (n_0*n_0),BB (n_r*n_r), CC (a_f*a_f), FF (S_d*S_d), CD (a_f*n_f), F (S_d),C (a_f) have negative effect on the wheel life time.

- The regression equation for the relationship between dressing condition and wheel life time is determined as follows:

T = 21.12 - 686 ar + 3.240 nr + 1571 af + 1.954 nf + 1.848 n0 + 22.04 Sd + 12988 ar*ar- 0.5247 nr*nr - 75850 af*af - 0.5875 nf*nf - 0.5812 n0*n0 - 7.535 Sd*Sd + 21.16 ar*nf+ 0.0794 nr*n0 - 52.0 af*nf + 0.1841 nf*n0

- The optimum value of wheel life time is predicted in the range from 48.058 to 49.110 minutes with the confident interval up to 95% correspond to the optimal dressing condition isa_r = 0.04 mm, n_r = 3 times, a_f = 0.01 mm, n_f = 2 times, n_0 = 2 times and S_d = 1.5 m/min

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REFERENCES

[1.] El-Hofy, H., *Fundamentals of machining processes: conventional and nonconventional processes*. 2018: CRC press.
[2.] Kumar, S., O.S.J.I.J.o.M. Bhatia, and C. Engineering, *Review of analysis & optimization of cylindrical grinding process parameters on material removal rate of En15AM steel.* 2015. **12**(4): p. 35-43.
[3.] Kwak, J.-S., et al., *An analysis of grinding power and surface roughness in external cylindrical grinding of hardened SCM440 steel using the response surface method.* 2006. **46**(3-4): p. 304-312.
[4.] Deresse, N.C., et al., *Experimental investigation of the effects of process parameters on material removal rate using Taguchi method in external cylindrical grinding operation.* 2020. **23**(2): p. 405-420.
[5.] Talon, A.G., et al., *Grinding performance of hardened steel: a study about the application of different cutting fluids with corrosion inhibitor.* 2020. **108**(9): p. 2741-2754.
[6.] Bianchi, E.C., et al., *Application of the auxiliary wheel cleaning jet in the plunge cylindrical grinding with minimum quantity lubrication technique under various flow rates.* 2019. **233**(4): p. 1144-1156.
[7.] Rodriguez, R.L., et al., *Evaluation of grinding process using simultaneously MQL technique and cleaning jet on grinding wheel surface.* 2019. **271**: p. 357-367.

[8.] Smith, E.D. and D.J.P.C. Curtis, *An investigation of additively manufactured coolant nozzles for cylindrical grinding applications and progression to high stock removal.* 2021. **101**: p. 162-165.
[9.] CHOUDHARY, D.K., D.M.J.R.J.o.E.T. GANGIL, and Management, *A Review of Optimization of Cylindrical Grinding Process.* 2020. **3**(02).
[10.]Rudrapati, R., A. Bandyopadhyay, and P.J.I.J.o.M.E.T.o.t.I. Pal, *Parametric optimization of cylindrical grinding process through hybrid Taguchi method and RSM approach using genetic algorithm.* 2018. **19**(1): p. 34-62.
[11.]Tu, H.X., et al., *Calculation of optimum exchanged grinding wheel diameter when external grinding tool steel 9CrSi.* 2019. **8**(1): p. 59-64.
[12.]Tu, H.X., et al., *Determination of optimum exchanged grinding wheel diameter when external grinding stainless steels.* 2019. **18**: p. 2345-2352.
[13.]Cao, H., X. Chen, and H.J.T.I.J.o.A.M.T. Li, *Dressing strategy and grinding control for cylindrical microstructural surface.* 2018. **99**(1): p. 707-727.
[14.]Kubo, A., et al., *Determining Surface Topography of a Dressed Grinding Wheel Using Bio-Inspired DNA-Based Computing.* 2021. **14**(8): p. 1899.
[15.]Manikandan, M., S. Prabagaran, and M.S.J.i.-M.s.J.o.M.E. Dennison, *A Review on Effect of Dressing and Grinding Fluid in Cylindrical Traverse Rough and Finish Cut Grinding Processes.* 2020. **10**(2): p. 35.

Design and Construction of the Propulsion System of a Modelled Offshore Service Vessel

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ABSTRACT

The paper focuses on the design, construction and testing of a propeller system for a modelled offshore vessel. It will discuss the construction of the hull of a vessel using steel, design calculations and selection of the propeller system which was powered by battery and electric motor. The experimental testing of the vessel propulsion system was conducted in a 3-meter by 3-meter water reservoir with a depth of about 5-meter to mimic a real-life environment. The results of the tests are satisfactory. The contribution of this work is the practically demonstrating of ship movements in water using a propeller system powered by battery and electric motor in order to enhance pedagogy teachings in Nigeria/West Africa. Further work will focus on the development of a controller system to aid maneuverability for the modelled vessel.

Keywords*—Propulsion/propeller system, offshore vessel, ship movement*

I. Introduction

Ship propulsion is of importance to ship movement and maneuverability. In the last century much work has been done and published in the field of propulsion [1, 2] with focus ranging from control of propulsion system [3, 4], to propulsion performance [5], and more recently, on performance optimization of propulsion [6] and development of innovative propellers [7].

In this paper, our focus is to develop a propeller system for a modeled vessel. This will involve applying basic rules/principle of design (i.e., which is to ensure safety, simplicity and functional ability of the system). The work also includes fabrication and construction of the ship hull structure, shaft, and propeller so that the system can be experimentally tested in a lab environment. This will be useful in practically demonstration of ships movement in water.

The paper is organized into 6 sections comprising of the introduction, design, construction and experimental testing and conclusion sections. The introduction section provides an overview on propulsion system and basic concepts is provided in the methodology section. This is followed by the design section which describes the design process and calculations with the aid of charts to generate propeller characteristics/curve which was modified to match the prime mover. The next section described the construction of the vessel from steel sheet, while the experimental testing followed it. The final section in the paper is the conclusion section.

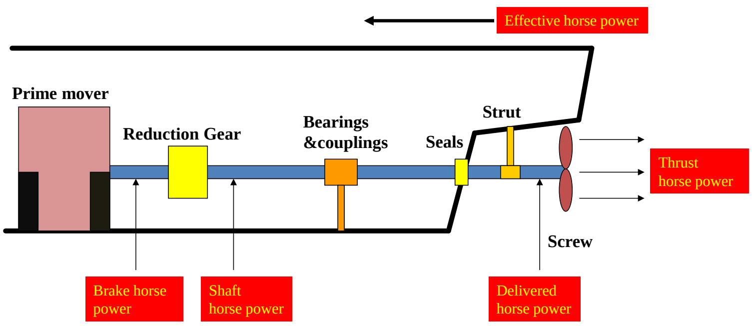
II. Methodoology

A propulsion system is used to provide thrust to a ship/flight system to overcome resistance. There are various types of propulsion system, such as vertical-axis (or cycloidal), nozzle, jet, fixed pitch, controllable-pitch, contra-rotating propellers and they all have their individual advantages and disadvantages. Our focus is on controllable-pitch propeller (with a 3-blade screw propeller). For a detailed discussion on Marine propulsion, the reader is referred to the literature [1, 2].

There are two main methods used in the design of a propeller. The first method is based upon Charts giving the results of open-water test on a series of model propellers such as pitch ratio, blade area ratio, number of blades and section shapes. The second method is based on the Circulation theory. It leads to a detailed design of the propeller and is specifically useful for the design of heavily loaded propeller, liable to cavitation.

We used the design based on Charts for this work. It is important to state that Computer design approach using Computation Fluid dynamics could also be used for propeller design.
A transmission system consists of the shaft, the bearings and the propeller and it transmits power from the prime mover (engine) to the propeller to thrust the ship. The propeller comprises of a boss securing the tail shaft with blades of helical form attached to it. The propeller in operation screws or thrust its way via the water by giving momentum to the column of water passing through it. Fig. 1 illustrates the schematics of a propulsion system.

Fig. 2 and 3 show the fabricated screw propeller and propeller shaft coupling (with bearings) respectively.



Fig, 1: Propulsion system

Fig. 2: Picture of the propeller

III. Design

The design and construction of a ship is in general expected to satisfy safety, economic, technical and operational requirements. The structural and architectural arrangement of the hull results from the proper consideration of these factors.

In the design and construction of a ship, the most fundamental requirement is to achieve adequate floatation of the ship under the desired loading conditions and the possibility to be moved from one location to another.

A secondary consideration is to achieve efficient operational characteristics such as having minimal resistance to the motion of the ship under varying sea conditions. Other factors such as durability, habitability and maintainability etc. are also considered and the final dimension is expected to conform to the numerous criteria that governed its design and construction.

Fig. 3: Propeller shaft coupling

For this work, the Taylor's series B charts SNAME [2] are used. The charts for the Propeller power coefficient, BP, – Propeller pitch/Diameter ratio coordinates are presented in [8]. Using the charts, the contours of propeller efficiency circulation, Γ and Propeller speed coefficients, δ Parameters were obtained. The design calculations are also reported extensively in [8].

From the calculations, the values of Speed, n in RPM were used to get the corresponding values of Propeller power coefficient, BP, that were used in estimating the values of the advance co-efficient (δ) and the Pitch diameter (P/D) ratio for a given blade number and blade area ratio. With this, we generate a table by varying the speed, n from 300 RPM to 410 RPM at an increment of 10RPM so as to generate propeller characteristics/curves (Table 1 and Fig. 4).

Based on the variation of n we obtain Table 1:

TABLE 1: PROPELLER CHARACTERISTICS

Propeller RPM	n	300	310	320	330	340	350	360	370	380	390	400	410
BP		20.94	21.638	22.336	23.034	23.732	24.43	25.128	25.826	26.524	27.222	27.92	28.618
δ		220	222	223	223	227.5	230	232.5	234	236	239.5	241	243
Corrrected δ (6%)		206.80	208.68	209.62	209.62	213.85	216.20	218.55	219.96	221.84	225.13	226.54	228.42
D		6.03	5.89	5.73	5.56	5.50	5.40	5.31	5.20	5.11	5.05	4.95	4.87
ηo		0.61	0.6	0.59	0.58	0.579	0.575	0.573	0.57	0.565	0.564	0.56	0.56
ηh		1.13	1.13	1.13	1.13	1.13	1.13	1.13	1.13	1.13	1.13	1.13	1.13
ηr		1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
P/D		0.855	0.840	0.836	0.833	0.831	0.829	0.828	0.825	0.822	0.820	0.818	0.815
Ps	BHP	218.87	222.52	226.29	230.19	230.59	232.19	233.00	240.09	242.21	242.64	244.37	244.37
Ps	kW	163.21	165.93	168.74	171.65	171.95	173.15	173.75	179.03	180.62	180.94	182.23	182.23
ηD		0.689	0.678	0.667	0.655	0.654	0.650	0.647	0.644	0.638	0.637	0.633	0.633
D	M	1.838	1.795	1.747	1.694	1.677	1.647	1.619	1.585	1.557	1.539	1.510	1.486

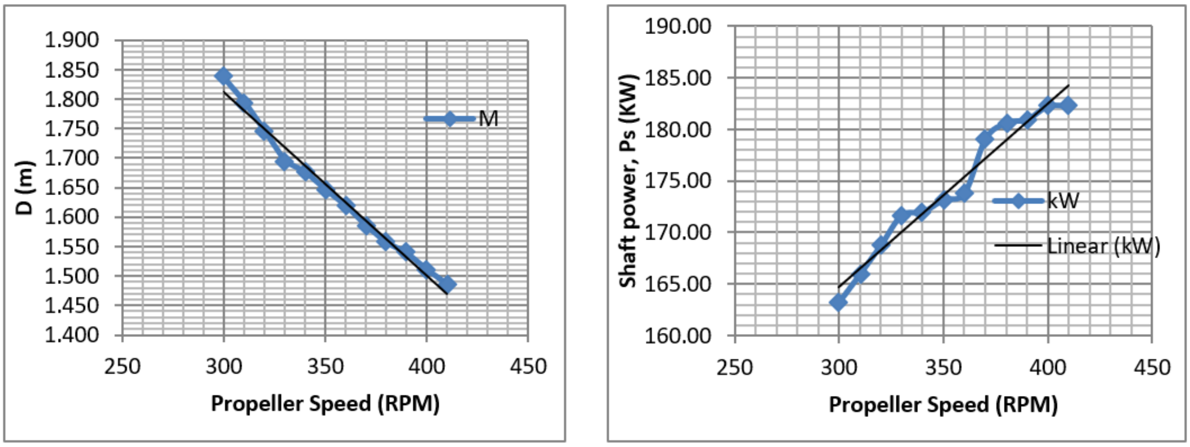


Figure 4: Engine/Propeller Curve

A. Selection of Propeller

The aim of the propeller selection is to choose the propeller with maximum possible diameter. For optimum propeller efficiency, a maximum propeller diameter of 1.48 is chosen and set as a limit.

Results deduced from the two graphs above indicates that the main propulsion engine will supply power output of 166kw (222 B.H.P) to propel the vessel at the given service speed, (corresponding speed is 310 rpm) - for it to operate at its optimum efficiency. The design calculation procedures take the same form as in the previous case are documented in [8].

B. Propeller - Main Engine Matching

For an engine that best suits the propeller to be selected it is often convenient to present the layout diagram on a logarithmic co-ordinate such as log power/log R.P.M. This enables us to obtain the nominal curve on a linearized form, thereby making the work easier. With the relationship that exists between power and speed, we obtained the power for our speed range of speed, n, which is between 250 – 350 rpm. Thus, the propeller curve was determined.

The powers obtained were in the range of 80 – 110% of the design speed (310 R.P.M), i.e., 248 R.P.M to 341 R.P.M is required to obtain the propeller curve.

We obtained the propeller curve (Table 2 and Figure 5) based on the following speed range in RPM:248, 257, 266, 275, 284, 293, 302, 311, 320, 329, 338.

C. Selection of Main Engine

The selection of the propelling unit (main engine) is based principally on its agreement with the propeller other factors that are considered include the engine's reliability in service, its operational and maintenance cost.

The main factors that are going to be considered in this selection are:

- Specific fuel oil consumption for service power on diesel
- Power required at specific speed
- Weight of machinery
- Reliability and cost of maintenance
- Noise and vibration

TABLE 2: MODIFIED PROPELLER CHARACTERISTICS DATA

PROPELLER MAIN ENGINE MATCHING											
RPM	248	257	266	275	284	293	302	311	320	329	338
Log n	2.394	2.410	2.425	2.439	2.453	2.467	2.480	2.493	2.505	2.517	2.529
P (bhp)	113.664	126.493	140.253	154.977	170.696	187.444	205.253	224.155	244.184	265.372	287.752
P (kW)	84.759	94.326	104.587	115.566	127.288	139.777	153.057	167.153	182.088	197.888	214.576
Log P	1.928	1.975	2.019	2.063	2.105	2.145	2.185	2.223	2.260	2.296	2.332
FIRST ENGINE 3304T (4 CYLINDERS) LOG C = -0.3009											
RPM	248	257	266	275	284	293	302	311	320	329	338
P (kW)	92.494	95.851	99.208	102.564	105.921	109.278	112.634	115.991	119.348	122.704	126.061
P (bhp)	124.037	128.538	133.040	137.541	142.043	146.544	151.045	155.547	160.048	164.549	169.051
Log n	2.394	2.410	2.425	2.439	2.453	2.467	2.480	2.493	2.505	2.517	2.529
Log P	1.966	1.982	1.997	2.011	2.025	2.039	2.052	2.064	2.077	2.089	2.101
SECOND ENGINE 3306T (4 CYLINDERS) LOG C = -0.1627											
RPM	248	257	266	275	284	293	302	311	320	329	338
P (kW)	127.150	131.764	136.378	140.993	145.607	150.221	154.836	159.450	164.064	168.679	173.293
P (bhp)	170.511	176.699	182.886	189.074	195.262	201.450	207.638	213.826	220.014	226.202	232.390
Log n	2.394	2.410	2.425	2.439	2.453	2.467	2.480	2.493	2.505	2.517	2.529
Log P	2.104	2.120	2.135	2.149	2.163	2.177	2.190	2.203	2.215	2.227	2.239
THIRD ENGINE 3306TA (6 CYLINDERS) LOG C = -0.052											
RPM	248	257	266	275	284	293	302	311	320	329	338
P (kW)	164.065	170.019	175.973	181.927	187.881	193.835	199.789	205.743	211.697	217.651	223.605
P (bhp)	220.015	227.999	235.983	243.968	251.952	259.937	267.921	275.906	283.890	291.874	299.859
Log n	2.394	2.410	2.425	2.439	2.453	2.467	2.480	2.493	2.505	2.517	2.529
Log P	2.215	2.230	2.245	2.260	2.274	2.287	2.301	2.313	2.326	2.338	2.349

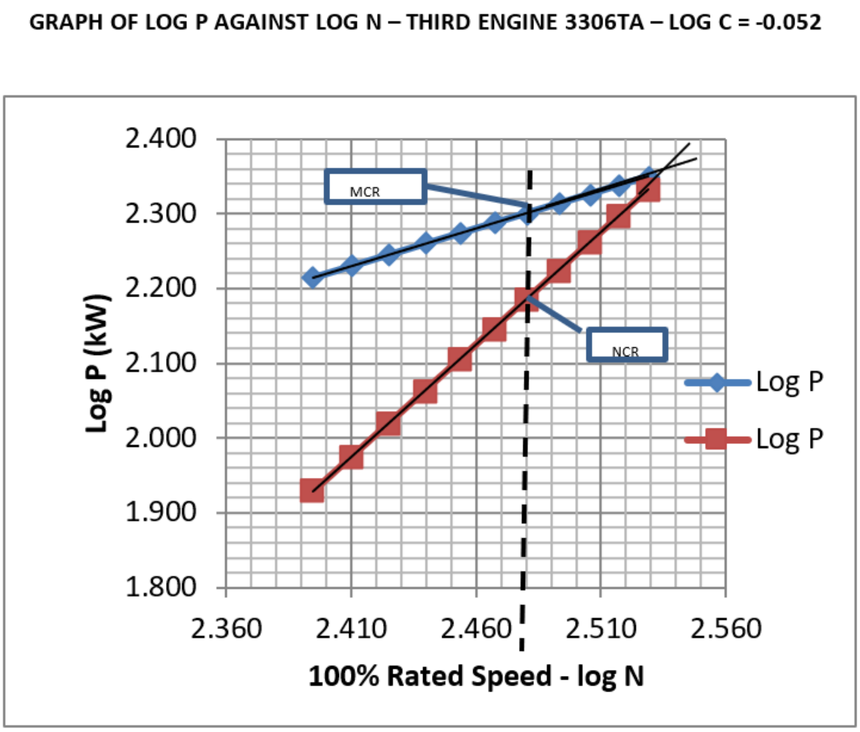


Fig. 4: Engine-Propeller Match for selected Engine

And above all the engine whose data gives a curve at about 10 – 15% below point of maximum power, (i.e. acceptable sea margin).

Based on these factors, and after critical analysis using tables and values read from graphs, for three different engines, the third engines having power capabilities of 162 – 262 kw and R.P.M was found to be having acceptable margin for a successful engine propeller matching.

Therefore, 3306TA caterpillar engine [9] was chosen, and its main/relevant data is given in Table 3.

TABLE 3: PRELIMINARY ENGINE DATA [9]

Engine No.	CAT 3306TA
Cycle	4
No. of Cylinders	6
Bore (mm)	121
Stoke (mm)	152
Mean Piston Speed (m/s)	10.1
Speed (RPM)	2000 – 2200
B.MEP (Bar)	10.7
Output (KW/Cyl)	160 – 265
Specific Fuel Oil Consumption, SFOC (g/kw/h)	214 – 223

IV. CONSTRUCTION

The propulsion system consists of the hull, and the mechanisms for propulsion. The hull is basically the body structure of the ship/vessel that is in contact with water.

The hull was constructed by first, itemizing the required dimensions of the vessel (model) from normalizing the final design parameters data based on the available portable prime mover in Nigeria.

This also necessitated a need for it to be electrically powered by a battery. At the end of the iteration process, bearing in mind of the basic design principle of safety, simplicity and function-ability, the data of the model vessel is given in Table 4.

After specifying the dimensions, a 2mm thick mild steel sheet, was cut into the desired pieces using oxy-acetylene torch. A frame (structure) that divides the vessel into compartments (transverse and longitudinal) was first welded in place. This frame serve as beams and pillars that supports the sheets when they are welded together to withstand all types of load. The keel of the vessel was welded in place, with its location at the basement of the ship.

TABLE 4: MODELLED OFFSHORE SERVICE VESSEL DATA

Parameters	Dimensions (mm)
Length	930
Breadth	470
Draft	335
Metal Sheet Thickness	2
Propeller Shaft Diameter	20
Propeller Blade Thickness	2
Propeller Blade Radius	42

The frame takes the curvature (shape) of the bow (forward) and stern (back side) of the ship. The metal sheets were tagged in position to the frames, and with the aid of clamps, the required angles were gotten. The sheets were fully welded to its respective frame. The heights and curvatures at the bow and stern were checked, the bottom of the vessel was checked for leakage. The vessel was then taken to a pool and tested for stability.

The propulsion system consists of 24-volts powered D.C. motor connected to a shaft (propeller shaft)through a system flywheel to reduce the speed that is eventually delivered to the propeller. The propeller shaft, which is a 20mm, solid shaft, was turned out on a lathe machine, which the front-end fitting into the flywheel, and the end section, having a coupling for the propeller blade (3 – blades). Fig. 6 and 7 show the propeller blade and shaft diagram respectively.

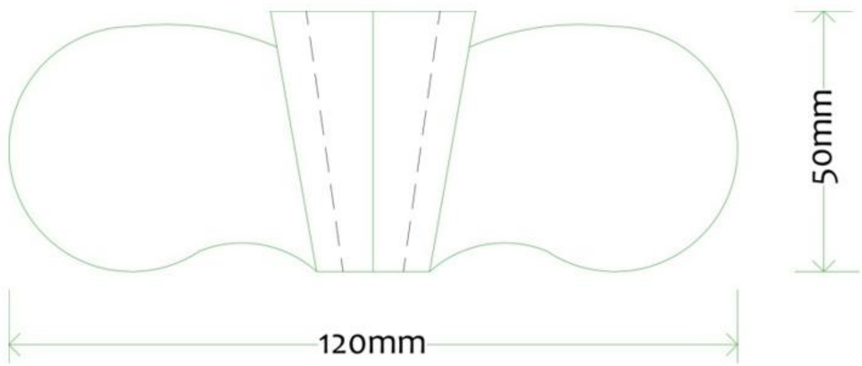


Fig. 6: Propeller Blade

Fig. 7: Propeller Shaft

A bedding (support) was constructed to carry the propulsion system from amidship (middle of the ship) where the engine (motor) is located to the aft of the ship. A 27.5mm hole was bored at the bottom end of the ship where the propeller shaft protrudes out to link the propeller blade via a hollow shaft that has oil seals at both ends to prevent leakage of water into the vessel when the propeller is moving.

The electric motor is connected to two dc batteries (12V, 26amps each), one located at port side, the other at starboard side, amidship. The other end of the motor is connected to a control mini-panel and a switch with three bottoms, one for forward, backward movement and stop.

V. Experimental Testing

The vessel was lowered into a pool of water, 5 m deep with length and breadth of 3m by 3m, for experimental testing (Fig. 8).

The switch was engaged in the forward direction, and the motor started, propelled the shaft, and eventually rotated the propeller, which eventually moved the vessel forward.

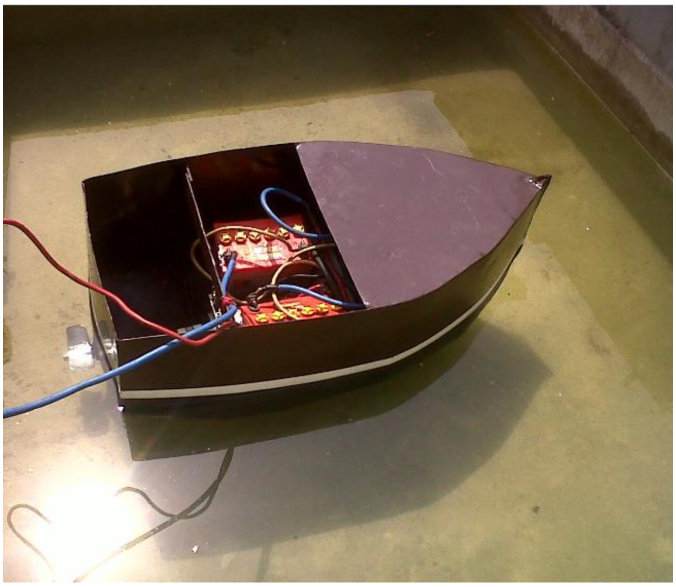


Fig. 8: Modelled Vessel during Experimental Testing

The vessel moved backward when the reverse switch was actuated, and it was stopped when the stop button was pressed. Thus, the experimental testing of the designed and constructed propulsion system of a modelled offshore service vessel achieved the desired results.

VI. Conclusion

The results of the experimental tests are satisfactory. Based on the results, we have practically demonstrated ship movements in water using a low-cost propeller system powered by battery and electric motor. This contribution to knowledge will enhance pedagogy teachings in the field of marine propulsion and control in Nigeria/West Africa countries.

Further work will focus on the development of a controller system to aid maneuverability for the modelled vessel.

References

[1] J. SCarlton, "Marine Propellers and Propulsion", Butterworth-Heinemann, Amsterdam, 2007

[2] SNAME, The Society of Naval Architects and Marine Engineers, Principles of Naval Architecture, Vol. I, II and III

[3] E.C. Kerrigan and J. M. Maciejowski, "Fault-Tolerant Control of a Ship Propulsion System Using Model Predictive Control" in Proc. European Control Conference, Karlsruhe, Germany, August 1999.

[4] M. Altosole, G. Benvenuto, M, Figari, and U. Campora, "Real-time simulation of a COGAG naval ship propulsion system" in Proceedings of the Institution of Mechanical Engineers, Part M: Journal of Engineering for the Maritime Environment 2009. 223: 47-61

[5] A, Brient, J. P. Martineau, and J. Y, Hascoet, "Influence of machining paths on hydro propeller performance" in Proceedings of the Institution of Mechanical Engineers, Part B: Journal of Engineering Manufacture 2003 217:1757-1762

[6] C. Mundt, and J. Lieser, "Performance improvement of propulsion systems by optimization of the mixing efficiency and pressure loss of forced mixers", Paper presented at the 8th European Propulsion Forum Affordability and the Environment – Key Challenges for Propulsion in the 21st Century, 26-28 March 2001 in Nottingham

[7] W. Han, and L. J. Tao, "A new marine propulsion system" in Journal of Marine Science and Application, 2003, Vol. 2, No. 1, 30-34

[8] B. Ipaa, "Design and construction of the Propulsion System of a Modelled Offshore Service Vessel" M.Sc. Thesis, Dept. of Mechanical Eng., University of Lagos, Akoka, Lagos, Nigeria, 2011.

[9] Caterpillar Engine Catalogue for 3306TA caterpillar engine