Imperial College London

MECHANICAL TRANSMISSIONS TECHNOLOGY

IMPERIAL COLLEGE LONDON

DEPARTMENT OF MECHANICAL ENGINEERING

Transmission System for a High Speed Yacht

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Word Count:

5794

22nd March 2019

Abstract

The luxury, high speed 46m long yacht requires an extremely reliable transmission to connect the two gas turbine inputs to the output water jet pump. The specification provided by the client includes a maximum yacht speed of 46 knots with superior heat generation and dissipation efficiency, without failure during the life of the yacht.

This can be achieved with a two stage helical gear configuration with a 99.5% reliability. A two stage design with a gear ratio of 9.30 was possible as weight was not a major consideration so larger gears could be used. The gears are made of 14NiCr18 carburised steel with a high toughness to prevent tooth fracture through bending stresses.

Bearings include a mixture of rolling element and hydrodynamic journals to fully support the shafts at the required loads and speeds. The bearings selected have excellent life estimates, all of which are longer than the yacht life, estimated at 21840 hours. The shafts are made from 440A stainless steel due to its excellent corrosion resistance making it perfectly suited to marine applications.

Lubrication with a synthetic base oil, polyalphaolefin was chosen for the best efficiency and excellent water resistance whilst providing adequate lubrication to the transmission. The lubricant is manufactured by internationally recognised Mobil Industrial and can be quickly changed at any port around the world. The cooler operates with a seawater heat exchanger, where the sea water is routed into a heat exchanger inside the hull for minimum drag.

The total material cost of the transmission system was found to be £ 8480.55. With an efficiency of 96.5%, this transmission meets all the requirements laid out by the client.

Contents

1 Introduction

The bespoke transmission designed for a new 46m long high speed luxury yacht connects two gas turbine engines, which run at 15400 rev/min with a combined power of 11 MW, to a single water jet pump situated at the back of the boat and running at 1664 rev/min. Due to the nature of this application, the ability to shift up and down gears as well as having a reverse gear are not required within the gear box since the direction of the water jet can be adjusted accordingly. Therefore, the client requires a single speed transmission which is capable of propelling the vessel at speeds up to 45 knots.

For this application it is critical that the transmission is designed so as to not fail since this would leave the user in a potentially dangerous situation stranded at sea. For this reason, the gearing and other features of the system have been designed to ensure they have a life which sufficiently exceeds the expected life of a luxury yacht, while health and usage monitoring (HUM) systems have been proposed to forewarn the customer of any potential failure. In addition to the life requirements, heat generation and dissipation are of paramount importance to the client so the developed transmission will be designed with excellent lubrication and cooling systems.

The client's specification has been quantified further in Table 1 which provided the basis for the design of the transmission. Within this report an overview of the design will be presented before detailing the individual component analyses which justify our design choices and the monitoring systems that will be incorporated. Considerations regarding the manufacture and associated testing are presented, concluding with an evaluation of the costing for this project.

Table 1: Product Design Specification

	ASPECT	**OBJECTIVE**
Performance	Overall Ratio	Must achieve a reduction of $9.255 : 1 \pm 10\%$
	Reliability	Must have a reliability of at least 99.5%
	Life	Must have a life of at least 20 years, with $20,000$ hrs runtime
	Efficiency	Must have an efficiency of over 85%
	Power	Must have a power of 11 MW
	Oil Change	Every 6 months
	Operating Temperatures	-10°C to $+50$°C
	Lubricant Temperature	Lubricant must not exceed 80 °C while in continued use
	Shock Loading	Must be able to withstand medium shocks
	Running Time (Cruising)	Must be able to cruise for 1 week uninterrupted
	Running Time (Top Speed)	Must be able to run at top speed for 6 hours
Packaging	Weight	Maximum weight of 2500 kg
	Dimensions	less than 4 m x 2 m x 2 m
	Input/Output	2 inputs, 1.27m apart; 1 output on centreline
Safety	Meet Safety Standards	Must comply with Directive 2013/53/EU
	Contamination Resistance	Must not leak any lubricant
Resources	Corrosion Resistance	Can last at least 20 years with minimal damage due to corrosion
	Lubricant Resupply	Replacement lubricant must be readily available to enable servicing in most locations worldwide

2 Design Overview

The proposed design is made up of a two-stage helical gear configuration as shown in fig. 1.

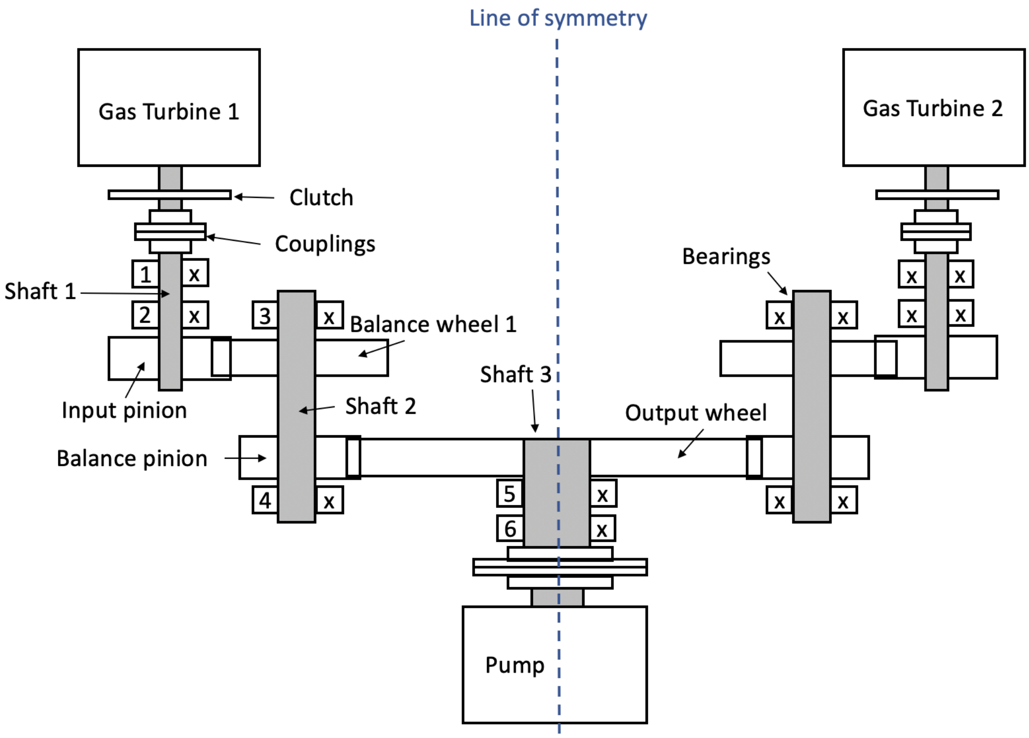

Figure 1: Diagram of proposed design - Not to Scale

From fig. 1 the connections between each gas turbine and the gearbox inputs are seen to have clutches to allow individual turbines to be disconnected if only half power is required, for example in cruise. This also serves as an important safety feature enabling the yacht to run on one engine in an emergency. A load balancer, as frequently seen in helicopter transmissions, is used to account for any differences between the two turbines, allowing all produced power to be used for driving the boat.

The design has been kept simple by using only two stages to reduce the risk of misalignment and minimise energy losses. However, there is an increased risk of failure as a result of the larger torque transmission and hence stress induced by having fewer ratios.

Once the parts had been specified they were produced within CAD, from which a render was produced, shown in fig. 2.

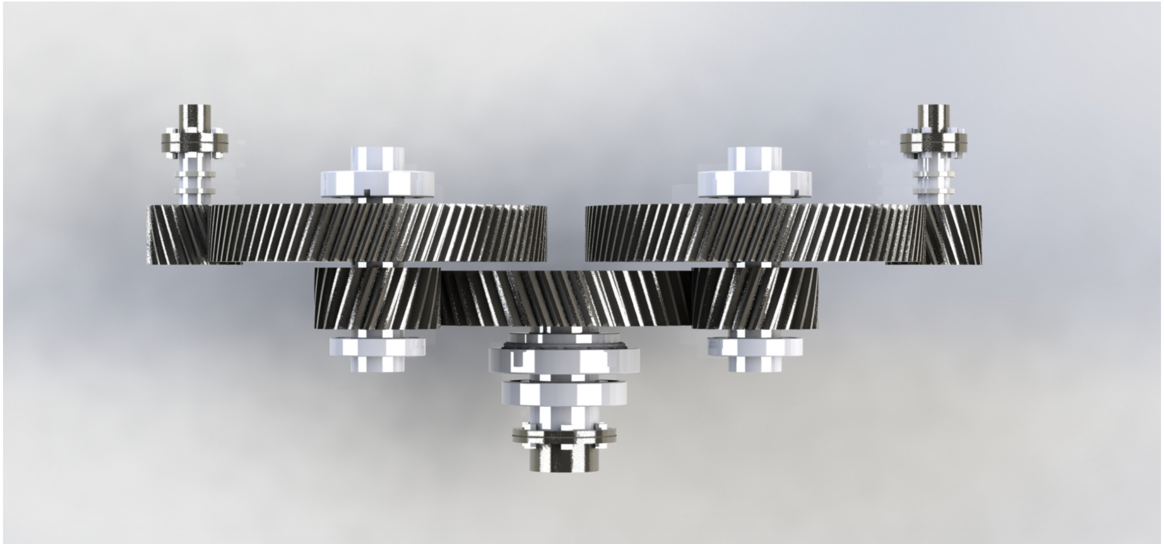

Figure 2: Digital mock-up of the design

3 Key Features

The key components of our design were the gears, bearings, shafts, couplings, lubricant, cooling and ingress protection all of which have undergone more detailed design.

3.1 Gears

The gear design was found in accordance with the method in the British Gear Association (BGA) Module 4 [1], which specifies that the following data must either have been given, selected or calculated: gear ratio, design torque, gear life, safety factor for contact stress, safety factor for strength, spur or helical gears, gear material and gear accuracy.
From the initial specification of the input speed of 15400 rpm and the output speed of 1664 rpm, the target gear ratio was found to be 9.25 ± 0.93. An numerical analysis was conducted to find the gear ratio for a 2-stage transmission that would minimise the mass of the gears, following the method outlined in [2]. Numerical integration of the curve in fig. 3 found the optimum gear ratio to be 3.61 and 2.56 for the first and second stages respectively.

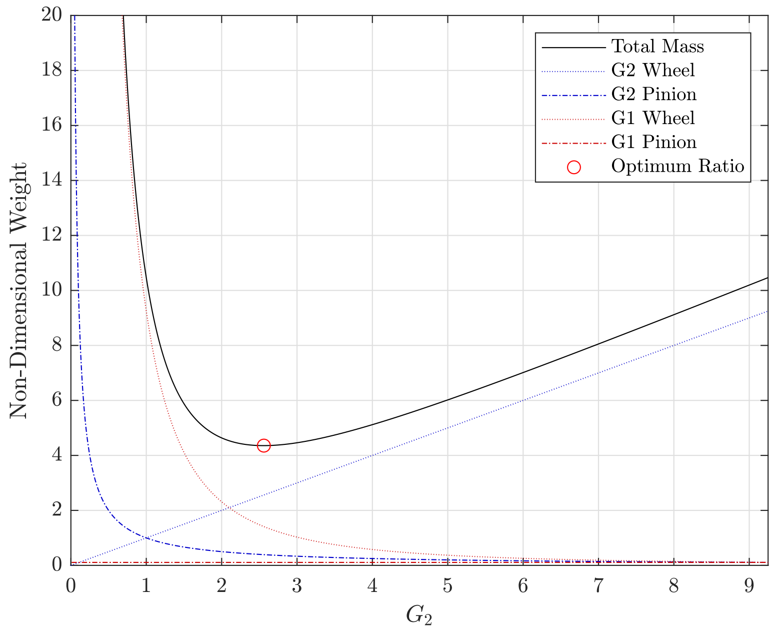

Figure 3: Mass of each gear as a function of gear ratio

Values for design torque were calculated based on a estimated life cycle data given by table 2, where all values are relative to the maximum life and peak torque. Case carburised 14NiCr18 was chosen as the gear material based on the high surface fatigue limits which are required by the very long life span of this gearbox. For this material the endurance limit, N_∞ was taken to be 5×10^7 cycles and the exponent, and the exponent q' was 6.6. Thereby the design torque for each stage was found and is presented in table 3. These are calculated based on pitting resistance since they are more limiting than bending failure resistance.

Table 2: Estimated Load Data

Top Speed	Relative Load	100	%
	Relative Life	15	%
Cruising	Relative Load	50	%
	Relative Life	80	%
Mooring	Relative Load	10	%
	Relative Life	5	%

Table 3: Calculated Design Torque

Gear	Design Torque	
Input Pinions	3.42×10^3	kN m
Balance Wheels	1.14×10^4	kN m
Balance Pinions	1.05×10^4	kN m
Output Wheel	5.37×10^4	kN m

Preliminary gear sizing was conducted by the method outlined by [1], finding permissible stresses using tabulated constants and material properties for the chosen material of case carburised 14NiCr18. The facewidth ratio was chosen to be 0.6 given due to relaxed weight requirements on a marine gear box permitting larger, less compliant shafts. The pinion pitch diameter could be roughly sized based on limiting contact stress by eq. (1). Minimum module required to resist bending failure was found by eq. (2). Once helix angles of $24.19°$ and $19.72°$ had been chosen for the first and second stage respectively in order to set the face contact ratio, ϵ_β, to an integer value, the minimum number of teeth could be calculated. The nearest, next largest standard module was chosen for each gear pair resulting in the prelinary gear specification reported in table 4. The optimum gear ratio was not achievable so the total gear ratio is 9.30 which is within the desired tolerance. This is split by 3.625 and 2.57 on the first and second stages respectively.

$$d_1 = 700\sqrt{\left[\frac{S_H}{\sigma_{HP}}\right] \cdot \frac{T_d}{c} \cdot \frac{G+1}{G} \cdot K_v \cdot K_{H\beta}} \tag{1}$$

$$m_n = \frac{2T_D \cdot 10^3}{cd_1^2} \cdot \frac{S_F}{\sigma_{FP}} \cdot 2.9 \cdot K_{F\beta} K_v \tag{2}$$

Table 4: Preliminary Gear Specification

Gear	Module (m) mm	Teeth (z)	Nominal Centre Distance (a) mm
Input Pinions	6	24	365.055
Balance Wheels	6	87	
Balance Pinions	8	23	348.435
Output Wheel	8	59	

Detailed stress analysis was conducted using the gears as specified by table 4. The method used was a version of that presented by BS 436 and DIN 3990 that has been simplified to permit the calculation to be conducted long-hand without the aid of specialist gear design software. Reference stresses for the contact and bending stress, denoted as the nominal bending and contact stresses, σ_{F0} and σ_{H0}, were calculated from the specified geometry, without resorting to graphical methods. The nominal stresses are then modified by factors to introduce the effects of inertial forces and manufacturing inaccuracies in order to give a good approximation of the real bending and contact stresses. These values are compared against the maximum permissible stress limited by the chosen material to 1500 MPa and 920 MPa for the contact and bending stresses respectively. The safety factors are shown in table 5 but the minimum safety factors for the first and second stages are 2.445 and 1.530 respectively. Both of these minima occur for the bending stress which is expected given that, while contact stress was used to initially specify the pitch diameter of the pinion, the specification of the gear was done based on the minimum standard module and the integer number of teeth that could be cut into the blank. Marginal reductions in the safety factor for the contact stress could be gained by reducing the facewidth however this was not felt necessary since the large safety factor can be justified by the requirement for no pitting to maintain the long lifespan required for an expensive yacht. The potential saving in weight is not important given the overall mass of the yacht and that fluid drag due to increased weight of the boat causes negligible losses.

Table 5: Detailed Stress Analysis

	1st Stage	2nd Stage	
σ_{H0}	−66.95	−85.22	MPa
σ_{F0}	192.90	381.80	MPa
σ_H	−95.55	−109.41	MPa
σ_F	376.33	601.44	MPa
S_H	15.70	2.44	
S_F	13.71	1.53	

3.2 Bearings

For the transmission design shown in Figure 1 there were 6 bearings that needed to be selected.

Due to the high speeds (15 400 rpm) and loads (≈20 kN) exerted on the first shaft rolling element bearings were unsuitable since they would generate too much heat and therefore prematurely fail. To overcome this, hydrodynamic tilting thrust pad bearings (as shown in Figure 4) were chosen to support the input shafts. Hydrodynamic bearings operate by separating the bearing surface from the journal surface with a lubricant film. The journal takes the radial loads, while the tilting pads support the axial loads. Compared to rolling element bearings they can struggle during start-up due to high initial friction but this downside is offset by the superior life and efficiency that can be achieved.

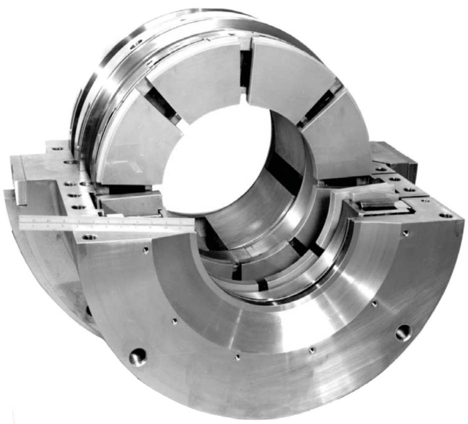

Figure 4: Hydrodynamic Tilting Pad Thrust and Journal Bearing [3]

As specified in the product design specification shown in table 1, the reliability of the gearbox was required to be 99.5% over a lifetime of 20 000 h. Consequently, for a rolling element bearing the relevant lifetime is given by eq. (3). Where: $L_{0.5}$ is the bearing life at 99.5% reliability; a_1 is the life adjustment factor as specified in ISO 281 ($a_1 = 0.18$ obtained from [4]); a_{mod} is the bearing life modification factor (a function of the lubrication and cleanliness); C is basic dynamic load rating; P is the equivalent dynamic bearing load; and the index p is 3 for ball bearings and $\frac{10}{3}$ for rollers.

$$L_{0.5} = a_1 a_{mod} \left(\frac{C}{P}\right)^p \tag{3}$$

Since the lubrication and cleanliness of the gearbox were dictated by the gear design the bearings had to be specified so as to ensure that they provided sufficient life under these operating conditions. An iterative process between bearing selection and the shaft design (presented in section 3.3) was used to match suitable bearings with the shafts. This involved determining the axial and radial loads exerted on the bearing as part of the shaft calculations, consulting the SKF bearing catalogue to select the bearing with appropriate load carrying capacity and limiting speed and then finally calculating the bearing life. If this life exceeded the criteria given in the PDS (table 1) then the bearing was deemed to be suitable, else a new bearing was identified. The resulting bearing selection is presented in table 6 and gives the calculated $L_{0.5}$ life for each of the bearings.

Table 6: Bearing Selection

Bearing Number	Type	Product Code	Lifetime
1	Hydrodynamic Journal	-	-
2	Hydrodynamic Journal	-	-
3	4-way Angular Contact Ball Bearing	QJ 324 N2MA	23400
4	Deep Groove Ball Bearing	6218	69840
5	Deep Groove Ball bearing	6224	80300
6	Taper Roller Bearing	31324 X	33120

3.2.1 Justification of bearing type

Bearing 3 was particularly challenging to specify due to the large radial (33 kN) and axial (20.6 kN) loads as well as the high speeds (4270 rpm). The 4-way angular contact bearing specified in table 6 was the only bearing manufactured by SKF that was capable of operating at such high speeds and loads, while providing adequate life. Deep groove ball bearings were selected for bearings 4 and 5 due to their high efficiency, low cost and sufficient load carrying capability. Finally, a taper roller bearing was selected for bearing 6 since the rollers line contact splits the load over a larger area and therefore can support the higher loads caused by the large axial forces (76.9 kN) from the helix angle of the gear. In addition, a taper roller bearing is suitable because the shaft speed is low and so overheating of the bearing due to excessive speed is not an issue.

3.3 Shafts

3.3.1 Maximum Bending Moment

For each of the three different shafts, a diagram (such as the one shown for the input shaft in fig. 5) of the loading was constructed and the corresponding bending moment along the shaft was calculated. From the bending moment profile along the shaft the maximum value has been extracted and were found to be 3053, 10054 and 5495 Nm for the input, intermediate and output shafts respectively.

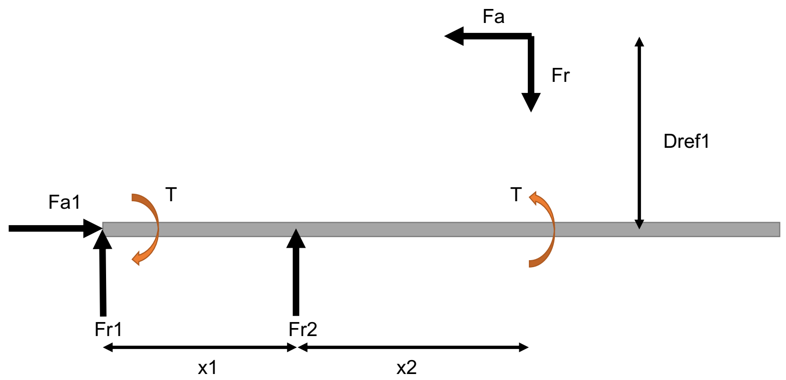

Figure 5: Free Body Diagram for the Input Shaft

3.3.2 Shaft Sizing

With both the torque and bending moment of the shafts known, the Westinghouse formula (eq. (4)) [2], has been used to determine the shaft diameter. Where: n is the safety factor ($n = 2$); T_a is the alternating torque ($T_a = 0$); T_m is the mean torque; M_a is the alternating bending moment (maximum bending moment as found from free body diagram); M_m is the mean bending moment ($M_m = 0$); S_e is the allowable fatigue stress ($S_e = 465$ MPa) and S_y is the tensile yield stress ($S_y = 755$ MPa).

$$d = \left(\frac{32n}{\pi} \left[\left(\frac{T_a}{S_e} + \frac{T_m}{S_y} \right)^2 + \left(\frac{M_a}{S_e} + \frac{M_m}{S_y} \right)^2 \right]^{\frac{1}{2}} \right)^{\frac{1}{3}} \tag{4}$$

3.3.3 Vibrational Analysis

During the design process it was crucial to ensure that the rotating components of the gearbox operate at a speed far away from their critical speed, since this would cause additional stresses to be exerted on the components leading to premature failure. The critical speed for the shafts have been estimated by assuming that the distributed shaft mass is negligible in relation to the gear mass and therefore the system can be analysed as a 1-DoF spring-mass system of equivalent stiffness and mass. With this assumption, the natural frequency of the system is given by eq. (5) (where k is the shaft stiffness and m is the mass). The shaft stiffness is given by eq. (6) with $\alpha = 3$ for the first and last cantilevered shafts and 48 for the simply supported intermediate shaft [5]. The values obtained are shown in table 7.

$$\omega_n = \sqrt{\frac{k}{m}} \tag{5}$$

$$k = \alpha \frac{EI}{l^3} \tag{6}$$

Similarly the torsional modes of vibration have been considered by modelling the gears as solid steel discs and then using eq. (7) (where k_t is the torsional stiffness given by eq. (8) and I_t is the total inertia of the shaft). The parameters in eq. (8) are G, the shear modulus taken to be 76 GPa, J, the polar moment of area and L, the length of the shaft. The values for the torsional vibration modes are presented in table 7.

$$\omega_t = \sqrt{\frac{k_t}{I_t}} \tag{7}$$

$$k_t = \frac{GJ}{L} \tag{8}$$

Table 7: Critical speeds of the gearbox shafts

Shaft	k N m^{-1}	m kg	k_t N m rad^{-1}	I_t kg m^2	ω_n rpm	ω_t rpm
1	2.87E+9	14.0	1.48E+6	0.0875	136000	39300
2	3.77E+9	208	2.40E+6	15.3	40600	3780
3	8.11E+9	142	1.68E+7	8.91	72200	13100

Comparing the critical speeds for the vibrational modes of the shafts to the running speeds at which they operate it can be seen that all of the shafts operate at speeds differing by more than 15%.

3.3.4 Shaft Specification

After following the design process detailed, the shafts were specified as in table 8. The shaft diameters were found to have a safety factor greater than 2 when using the Westinghouse formula. As a result the shafts have sufficient resistance to fatigue over the lifetime of the gearbox and so will not fail, as required by the PDS. The shaft diameters were not increased further since this would increase the weight of the transmission unnecessarily.

Table 8: Shaft Design Parameter Summary

Shaft	Units	1	2	3
Length	mm	200	297	338
Diameter	mm	55	90	120
Mass	kg	3.7	14.7	29.8
Critical Speed	rpm	136000	40600	72200
Torsional natural frequency	rpm	39300	3780	13100

3.4 Couplings

As the gearbox will not be taken apart during the life of the yacht, removable couplings are not needed. All the shafts are horizontal, and a misalignment accuracy of less than 1° can be achieved during assembly, therefore a rigid coupling is appropriate. A coupling consisting of two flanges bolted together (as shown in fig. 6) has been selected to allow for two shafts of different diameters to be attached together in order to transmit power.

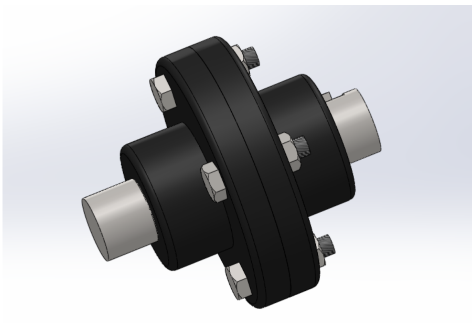

Figure 6: Standard Flange Coupling

3.5 Lubricant

Suitable lubrication is crucial not only to the gear contacts but also the bearings and cooling the gearbox by dissipating the heat generated due to frictional heating both from the contact and also churning and windage losses. Due to the gear tip velocity of both stages, calculated by eq. (9) and found for the pinions to be 137.9 m s^{-1} and 47.5 m s^{-1} for the first and second stages respectively, an oil is required, despite the very high loads which might be more suited to a grease. It is well known that the oil viscosity is highly dependent on temperature hence all viscosity used for the lubricant specification are based off the ASTM interpolation method (eq. (10)), with viscosity specified at 40 °C and 100 °C.

$$U_a = \frac{\pi N d_a}{60} \tag{9}$$

$$\log\log(\nu + 0.7) = A - B\log T \tag{10}$$

It was decided that the gear oil should enter the mesh at a temperature of 40 °C which is approximately the maximum ambient air temperature in the hottest cruising locations worldwide so even under the worst conditions it will have negligible heating effects from the ambient conditions. Furthermore the sea temperatures at these locations are generally about 20 °C to 25 °C[6]. An active oil cooler will be required to cool the hot oil and the simplest cooling method is to feed the hot oil through a heat exchanger with the ambient sea water. [7] suggests that a temperature difference of 15 K is required for a heat exchanger so a design inlet temperature of 40 °C will avoid the requirement for any intermediary cooling stage which would add complexity and cost. It was desired to keep the tooth temperature below 100 °C due to the limited ability of the ATSM method to extrapolate accurately. However the intial client requirement that the maximum oil temperature would be less the 80 °C could not be met without very viscous oils, leading to large churning losses, or a complete redesign of the gearbox to reduce tooth loading. Therefore the criteria was relaxed for the final design. From these requirements an initial specification of the gear oil could be conducted.

AGMA 925-A03 suggests a semi-empirical method for calculating the tooth temperature. Firstly an estimate for the coefficient of friction is made by eq. (11) where C_w is an empirical constant with value 29.7. U_1 and U_2 are the pinion and wheel velocities at the radius of contact while $\overline{u}$ is the entrainment velocity.

$$\mu = 0.0127\log_{10}\left[\frac{C_w}{\eta|U_1 - U_2|\overline{u}}\right] \tag{11}$$

The flash temperature rise can the be calculated at every point in the mesh by eq. (12) where W_L is the load per unit length, E^* is the reduced Young's modulus and R'_x is the reduced radius of contact. Due to the periphral speed of fastest gears exceeding 14 m/s, it is required that the lubricant is sprayed into the contact. Due to the high loads and high speed it will be sprayed as a mist under compressed air which will also increase the cooling rate of the oil. Under these conditions, the mean temperature of the teeth is suggested by AGMA to be given by eq. (13).

$$\Delta T_f = \frac{0.75 W_L^{\frac{3}{4}} \mu \left| \sqrt{u_1} - \sqrt{u_2} \right| E^{*\frac{1}{4}}}{R_x'^{\frac{1}{4}} \sqrt{\rho \sigma k}} \tag{12}$$

$$T_{\text{tooth}} = 0.56 \Delta T_{f,\,\text{max}} + 0.2 T_{\text{ambient}} \tag{13}$$

Once a lubricant has been chosen which gives acceptable cooling to the gear contact, oil film thickness and hence the lambda ratio must be found. To calculate the minimum film thickness, h_{min}, which occurs at the first point of contact of the gear tooth, the regression equation proposed by Dowson and Higginson[8] was used as presented in eq. (14). The pressure viscosity coefficient, α_l, was calculated by the AGMA equation using constants dependent on the type of base oil. The lambda ratio, which determines the lubricant operating regime of the contact, is then found by eq. (17), where the RMS surface roughness, R_q, was assumed to be 1.63 µm, the poorest surface finish achievable with finish grinding[9]. For the chosen lubricant specified in table 10, the lambda ratios of the first and second stages are 6.47 and 2.5 respectively. The lambda ratio of the second stage is acceptable, although should not be much lower to avoid operating in the boundary lubrication regime ($\Lambda = 1.0$). The lambda ratio of the first stage is limited by the second stage since it would add significant cost and weight to have multiple oils in the gearbox due to the required coolers and seals, however it is acceptably close to 3 so as to have a negligible effect on efficiency due to increased churning or windage.

$$h_{min} = 1.6 \alpha_l^{0.6} \left(\eta_0 \overline{u}\right)^{0.7} \left(E'\right)^{0.03} R_x'^{0.43} W_L^{-0.13} \tag{14}$$

$$\alpha_l = k \eta_0^s \tag{15}$$

$$R_{qc} = \sqrt{R_{q1}^2 + R_{q2}^2} \tag{16}$$

$$\Lambda = \frac{h_{min}}{R_{qc}} \tag{17}$$

The flash temperature of the chosen lubricant is significantly greater than the calculated

maximum temperature rise of eq. (12). While the pour point of this oil is fairly high compared to other common gear box oils, this should not be an issue for this application given the gearbox will be in the bottom of the hull, surrounded by sea water, which even in the most severe conditions, in which the yacht is not expected to be commonly used, will never drop below −2 °C[6]. The synthetic base oil, polyalphaolefin (PAO), was chosen on a balance of cost, due to being cheaper than polyalkylene glycol (PAG) based lubricants, and lambda ratio, since it gives a much lower lambda ratio than mineral based oils. Additionally it provides better water resistance than PAG, and while this should not be a problem due to the required sealing performance of the gearbox, it may be advantageous for a marine gearbox. The manufacturer claims that has exceptional corrosion resistance, even in marine conditions, and can offer good scuffing resistance which is beneficial due to the high loads in this gearbox. Furthermore the oil has been specially formulated to provide improved performance for rolling element bearings compared to conventional gear oil chemistries[10]. Additionally the manufacture claims that oil has a long life even in extreme conditions so the client requirement of an oil change less frequently than every 6 months should be fulfilled.
The lubricant is manufactured by an internationally recognised company, Mobil Industrial. As such the lubricant should be available, either in stock or quick to order, to any dealer worldwide. This fulfills the client requirement that the oil can be quickly changed at any port around the world.

Table 9: Gear Pair Properties

	Input Pair		Output Pair		
R_x	23.1	125.2	19.9	113.56	mm
U_i	17.2	25.8	4.2	9.38	m s^{-1}
U	21.5		6.8		m s^{-1}
W_L	0.55		1.05		MN m^{-1}
h_{min}	14.9		5.76		µm
Λ	6.47		2.5		
β	748.0		306.0		
ΔT_f	15.2		81.1		°C

Table 10: Lubricant Properties

Property	Value
Brand	Mobil SHC Gear
ISO VG	680
Pour point	−27 °C
Flash point	234 °C
Density	0.86 g/cm^3
Viscosity @40 °C	680 cSt
Viscosity @100 °C	75.5 cSt
Base Oil	Polyalphaolefin

3.6 Cooling

As previously mentioned, as well as lubricating the gears the oil fulfills a vital role of cooling the gearbox. Exact values of gearbox efficiency are difficult to calculate due the complexity of the interacting effects of friction in teeth contacts, friction in bearings and churning or windage losses. However, [11] suggests that for an overall gear ratio of 9.30, the total gearbox losses are approximately 2%. Due to the high peripheral speeds in our gearbox, which will increase churning losses, it is expected that our gearbox will have a lower efficiency than this. A safety factor of 1.5 was applied to account for this so our total gearbox efficiency is 96.5%. Therefore 385 kW must be removed by the cooling system.

An approximation for the amount of energy dissipated by natural convection and radiation from the casing can be made via eqs. (18) to (19). The surface area of the casing is calculated from the assumption of a rectangular box around the transmission elements, which equals 4.04 m^2. The assumption that the ambient temperature is 20 °C is conservative given that, as mentioned previously, the hottest oceans the yacht is likely to be used in only ever reach 20 °C or sometimes 25 °C. The temperature of the wall is assumed to be equal to the bulk temperature of the teeth, although the actual value may be slightly less. Therefore the amount of energy dissipated by natural convection and radiation is 2.36 kW and 4.53 kW respectively. A powerful cooler is then required to dissipate the remaining 378.1 kW. Given the rule of thumb from [11] that 1 $dm^3\,min^{-1}$ at 1 bar is sufficient to dissipate 75 kW, it is estimated that lubricant will be required from the cooling system at a flow rate of approximately 5.0 $dm^3\,min^{-1}$

$$q_{conv} = [0.15 + 0.013(T_a - 60)]\,A \tag{18}$$

$$q_{rad} = [0.45 + 0.02(T_a - 60)]\,A \tag{19}$$

The cooler will operate with a seawater heat exchanger, with sea water routed into a heat exchanger inside the hull from the intake to the impeller. This will cause least drag on the outside of the yacht and the high flow-rate coolant will cool the oil more efficiently and will not require an additional pump. Additional pumps would either have to be directly run off the gearbox, which would complicate the design, or require an additional motor just for pumping coolant water. Either method will decrease the available peak power and should be avoided

if unnecessary.
The oil from the gearbox will be pumped from the sump, through 2 oil filters, each of 50 μm due to the high gear speeds, through the heat exchanger before being sprayed into a gear contact. The two oil filters, which are mounted in parallel, are required to provide a backup system should one filter become blocked while at sea. It also permits the oil filter to be easily changed while the system is active. The oil filters will also extend the service life of the oil.

3.7 Ingress Protection

Standard radial lip seals have been selected from literature [12] as the seals for the shafts. The yacht is likely to experience contamination, in the form of particulates or sea water so dust lips have been added. The conditions should not be especially extreme therefore fluoro rubber (FKM) has been selected as a material, which is compatible with the PAO synthetic oil based lubricant. The seals are capable of working in temperatures between -40 and 200°C, therefore are perfectly suited to this transmission.

4 Assumptions

The assumptions made for this transmission design were as follows:

- The gas turbine engines are two-shaft engines, which is reasonable since they are a common form of gas turbine [2].
- The propeller will slip to allow the boat to get moving, this is reasonable since the resistance to motion of the propeller is proportional to its rotational speed [13].
- As shown in Appendix A, the original specification dictated that the input and output shafts needed to be 1.27m apart, we took this to mean the distance between the two Gas Turbine shafts.
- Total gearbox efficiency, including all friction losses in bearing and gear contacts and churning losses in the oil, is 96.5%.
- The shaft mass is negligible in relation to the gear mass and therefore the system can be analysed as a 1-DoF spring-mass system of equivalent stiffness and mass.

5 Heath and Usage Monitoring

Health and Usage Monitoring (HUM) is especially important for a luxury yacht as it can help to predict failures before they occur, reducing maintenance costs and informing the remaining useful life value for the transmission.

Both electromagnetic sensors and accelerometers were selected to monitor the transmission to increase the chances of predicting a failure correctly. Electromagnetic sensors were chosen to give information on debris over magnetic plugs because they can be used on-line rather than the samples having to be sent off to be analysed. For the most reliable and effective functionality, the sensor should be placed upstream of the filter, meaning all the oil flow is monitored.

Vibration analysis is excellent for finding fatigue pits and broken teeth. Accelerometers were specifically chosen because they have a high natural frequency, small mass, wide measurement range, high temperature range and are robust, making them ideal for use in transmission systems. The positioning of the accelerometers should be considered carefully. Multiple positions should be used, with locations that are near bearings being preferable as they generally have the shortest path of vibration. The Fast Fourier Transform (FFT) of the collected data can then be analysed for the shaft frequencies, mesh frequencies, sidebands and harmonics.

The lubricant itself should be implemented to ensure that it is changed when necessary. This should be done using electro-chemical oil analysis, which determines the remaining antioxidant concentration to estimate the remaining useful life. This can be done whilst out at sea, reducing the need to send oil samples to a laboratory for analysis. To enable this, there must be a point in the gearbox casing where the lubricant can be accessed to be sampled. Having this access means that ferrography and filter monitoring can be carried out to allow for a more in depth analysis of the wear debris, where the size of particles can be used to predict the failure mode by comparing to a wear debris atlas.

6 Manufacture and Suggested Testing

The assembled system should be tested by running at an accelerated rate to test that it has a life of 20 years (20,000 hours) as required by the specification. This testing will highlight any unexpected wear modes experienced by the gears, which should be designed against through a material change or modification to the overall design.
Furthermore, vibration testing should be carried out on the system to identify any natural frequencies that may be excited during normal running speeds. These modes would cause large displacement that could result in issues from minor noise all the way to catastrophic failure if not resolved.

6.1 Gears

The material chosen for the gears was 14NiCr18 carburised steel, conforming to grade EN39a, because it has high toughness, meaning it can withstand gear tooth bending stresses without fracture. The carburisation process results in compressive stresses near the surface region, which resist surface crack growth to improve the contact fatigue life of the gears.
To manufacture the gears, the process of hobbing would be used. To avoid unnecessary expense and due to the small batch size for luxury yachts, a standard module size was chosen so no bespoke tooling would be required. This gives a roughly machined gear, with a hardness of 250 Hv, and so the next step is to case harden, or carburise, the material. This is done by subjecting the gear to a temperature of 920°C and holding this temperature for a few hours to give a case depth of around 2 mm and increase the hardness by 50 Hv to 300 Hv. The gear is then oil quenched and tempered at 150°C to give a hardness of 700 Hv. This process causes distortion and so further processing is required to achieve a suitable level of quality. For this reason, finish grinding was utilised to allow a quality factor of Q5 to be reached.
Finite element analysis (FEA) or another analytical method could be used to model the stress distributions within the gears and validate the calculations. FEA can give an indication of the stresses experienced by the system, but testing on the manufactured gears should also be carried out as assumptions in the modelling may cause errors in the values obtained through this method. Therefore, a back-to-back rig can be used to test the efficiency of the

gears and the assembly testing would show any failure modes that occur from stress concentrations, or other features that should be designed against.

6.2 Shafts

The shafts will be manufactured by turning round 440A stainless steel stock to the required shaft dimensions, cutting the splines for the gears and then heat treating to improve strength. This material was selected due to it's balance between corrosion resistance, high yield and allowable fatigue stresses, as well as it's availability enabling easy sourcing and manufacture of the shafts.

6.3 Sub-contracting

All bearings and seals would be sourced from SKF and Kingsbury. It should be noted that the engine must be run with the full assembly until the seals have completed their bedding in period and then tested before use on the ocean.
The externally sourced lubrication system should be tested using the gear scuffing test, also known as FZG testing. This would allow for the scuffing resistance of the oil to be assessed because it involves running the gears while progressively increasing the load until scuffing occurs. The cooling system should be tested to check that it is able to cool the lubricant to the specified inlet temperature of 40 °C, i.e. to remove 378.1 kW of energy from the gearbox. If the cooling system is unable to meet this requirement, the sump temperature will increase until the anti-wear additives in oil cease to function. At the same time the oil viscosity will reduce until the lambda ratio falls below unity. Heat dissipation is of especial importance to the client so this stage of testing should be thorough.

7 Costing

The cost of material for the components of the transmission is summarised in the table below. This estimate assumes the cost of shafts and gears based on weight, which is likely to be an overestimate since bulk purchasing of these materials would probably reduce this cost significantly. It has also been assumed that these material costs make up 10% of the total cost to produce this transmission system, with labour costs making up the remaining 90%.

Table 11: Material Costs

Item		Unit Cost	Quantity	Total
Shafts	1	£ 7.05	2	£14.10
	2	£36.64	2	£73.28
	3	£43.51	1	£43.51
Gears	1	21.75	2	£43.50
	2	£285.79	2	£571.58
	3	£ 41.68	2	£83.36
	4	£219.41	1	£219.41
Bearings	1	352.22	2	£704.44
	2	£352.22	2	£704.44
	3	£467.88	2	£935.76
	4	£123.77	2	£247.54
	5	£342.45	1	£342.45
	6	£1289.87	1	£1289.87
Couplings		£122.64	3	£367.92
Seals		£ 2.17	10	£21.70
Lubricant		£ 14.52	150	£2178.00
Casing (steel)		£118.33	1	£118.33
Cooling		£520.53	1	£520.53
Total				£8480.55

8 Conclusion

This report has presented the design of a two stage transmission system for a new 46 metre luxury yacht based on the specification provided by the client. The transmission was required to transmit 11 MW from two gas turbines rotating at 15 400 rpm to a water jet pump rotating at 1664 rpm without failing before the life of the yacht, estimated at 21840 hours.

The proposed system uses a helical gear train with an overall ratio of 9.30 to fully meet the specifications given. Our transmission had an estimated efficiency of 96.5% and an approximate lifetime of 23 400 h at 99.5% reliability. Analysis on the part selection has been given, presenting the equations and calculation results for the development of the gears, shafts, bearings, lubrication, couplings, cooling and seals.

As mentioned by the client, heat generation and dissipation are seen as extremely important so a particular emphasis was given to designing the lubrication and cooling system for the transmission. A cooler that utilises seawater was chosen with a high flow rate to cool the oil efficiently without requiring an additional pump.

Implementation of health monitoring is discussed, explaining how accelerometers will be used to monitor the transmission in real time enabling instant detection of damage to the gearbox. After detailing the design, the manufacturing and necessary testing were discussed, explaining how the transmission would be put through a sped up version of an entire transmission life cycle. Finally the costs of the developed gearbox are calculated with a total material cost of £8480.55.

To conclude, the final transmission meets all the client's requirements that have been quantified in the PDS in Table 1.

References

[1] D. A. Hofmann. Design and stress analysis of spur and helical gears. In *BGA Gear Technology Teaching Pack*, chapter 4, page 119.

[2] Amir Kadiric. *Mechanical Transmissions Technology Course Notes*. Imperial College London, London, 2018.

[3] Kingsbury Inc. *A General Guide to the Principals, Operation and Troubleshooting of Hydrodynamic Bearings*. Kingsbury Inc, 2018.

[4] Life modification factor for reliability. URL `http://eb-cat.ds-navi.co.jp/enu/jtekt/tech/eb/info/03/3_1_33-3.htm`.

[5] Frederic Cegla. *Machine System Dynamics Lecture Notes*. Imperial College London, London, 2017.

[6] World Water Temperature Sea Temperatures. URL `https://www.seatemperature.org/`.

[7] Matthew Eaton. *ME2 Thermodynamics*. Imperial College London, 2016.

[8] D Dowson and G.R. Higginson. *Elastohydrodynamic lubrication*. Pergamon, 1966.

[9] Complete Surface Finish Chart, Symbols & Roughness Conversion Tables. URL `https://www.cnccookbook.com/surface-finish-chart-symbols-measure-calculators/`.

[10] Mobil Industrial. *Mobil SHC Gear Series*. 2019.

[11] H.K Kohler. BGA Teaching Pack Module 5. In *BGA Gear Technology Teaching Pack1*. British Gear Association, 1991.

[12] SKF. *CR Seals Handbook*. SKF USA, 2019.

[13] Richard Thiel. Everything you need to know about your boats transmission, 2019. URL `https://www.powerandmotoryacht.com/maintenance/everything-you-need-know-about-your-boats-transmission`.

A Original Specification

The original specification given was as follows:
"You are required to design a transmission system for a 46 m, high speed (45 knots) yacht, powered by two gas turbine engines running at 15400 rev/min with a total power of 11 MW. The transmission is to drive a single water jet pump running at 1664 rev/min. Both input and output shafts are horizontal 1.27 m apart, and at the aft end of the assembly. Failure of the gearbox would be completely unacceptable as the affected boat would have to be completely dismantled to remove the transmission. Heat generation and dissipation is seen by the customer as especially important. "
1.0