ME-465: Design Project Progress Report #1

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Disclaimer

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1. Background

This project involves the design of a proposed winch system which must meet some given minimum requirements. From the provided project description:

> *A winch, schematically shown in the figure below* (Figure 1)*, is operated by a worm-gear mesh. The input torque to the worm is provided by an electric motor that rotates at 1,500 rpm. The expected speed of the worm gear is between 30 and 35 rpm. The peak torque requirement for the winch is about 4,000 lbf·in, the operating ambient temperature is 120 °F, and the average output power doesn't have to exceed 1.2 hp. Knowing that the winch drum radius is 8 inch and that the winch operates 4 to 5 hours a day, perform the following tasks, which combined lead to the overall design project.*

2: Problem Statement

The first component of designing the winch is to select an appropriate worm and gear for the gearset which will transmit power from the electric motor to the winch drum. From the provided project description:

> *Design the worm-gear mesh that meets the requirements for the operation of the winch (i.e. perform force, bending and wear analyses for the worm-gear mesh). Assume a design factor of 1.2 for the gear teeth and make sure that the mesh is self-locking. Using a gear manufacturer's catalog, select a readily available worm-gear mesh that meets the design requirements. Your report should incorporate… A computer code that allows the design of the worm-gear mesh based on the given inputs, meeting the imposed bending and wear requirements for the worm-gear mesh.*

Upon completion of the initial gearset design, work may begin on an initial shaft design which will be explored in a future progress report.

3. Objectives

Our objective is to design a worm gearset which is sufficient for the required task within some factor of safety. Specifically, the gearset must be able to:

- Convert the 1500 rpm, 1.2 hp output of the motor to the required 4,000 lbf·in torque to the winch drum
- Rotate the worm gear (and therefore the winch drum) at between 30 and 35 rpm
- Operate in the 120°F environment without producing excessive heat
- Operate under given load 4-5 hours per day without excessive wear or damage

In addition, the design should minimize cost as much as possible. The size of the assembly and the power required from the motor should also be kept as low as feasible.

4. Approach

First, known values will be used to calculate other values which are implicitly dependent on the input values. Next, these values will be used to identify possible gears from the supplier which meet the required parameters. These gears will be checked for satisfactory performance using an Excel spreadsheet to simplify and expedite calculations. Should issues arise with the available gears not meeting required criteria, the objectives of the problem statement will be evaluated to identify where they can be adjusted. This process would ordinarily involve meeting with the project's client to address design issues and work out a functional solution. However, lacking a client for this exercise, assumptions will be made about the relative importance of design constraints. These assumptions and their rationale will be outlined below. Once design constraints can be met by available gears, the gears with superior performance will be selected for moving forward with the project.

5. Assumptions

As shown in the Calculations section below, the output torque and speed are not able to be accomplished using the provided motor input power. At this point in an actual design process, the client would be informed of the issue and a strategy for moving forward would be established. The client may select a motor with greater output power, or they may choose to prioritize the output torque, the output speed, or some combination thereof. Without a client to contact, *a posteriori* assumptions must be made based on the project objectives to complete this project in the best possible way. This report will continue under the assumption that the 4,000 lbf·in torque requirement is critical, and that the 30–35 rpm output speed is noncritical. This is because it is assumed that, if a client has requested a particular torque output from the winch (4,000 in·lbf) with a known winch drum diameter (8 in), they have done so with lifting a particular maximum load in mind (1000 lbf). The output speed is assumed to be noncritical since, even at lower speeds, the winch will be capable of lifting the necessary load, albeit more slowly.

The previous assumption results in a gear ratio larger than 80:1. The gear selected gear supplier has gearsets of ratios 80:1, 96:1, and 100:1. No gearsets are available in ratios exceeding 100:1. As shown below, once losses in the gearset are taken into account, no available gears can provide the 4,000 lbf·in peak torque output required at the maximum motor speed of 1500 rpm. As such, this report will move forward under the assumption that the motor can be driven at lower speeds, increasing the output torque at the cost of further decreasing the output speed of the winch drum.

The chosen gear supplier offers worm gears in diametral pitches between 3 and 16 inches. Of these pitches, those between 6 and 16 inches are available in the gear ratios needed. In addition, all possible gearsets use a pressure angle of 14.5°, which removes this as a variable in our design.

As shown in section 6.2.1, the design constraints require further changes to meet listed requirements due to the limited selection of gears from the supplier. The highest possible output torque attainable with the given design constraints and stock gears from the required gear supplier is 3968.5 in·lbf, just shy of the 4,000 in·lbf which was deemed critical earlier in this section. Other gear combinations come close to this value, with six others exceeding 3,500 in·lbf. This would ideally be solved by using a higher gear ratio, but the chosen supplier does not produce worm/gear pairs with ratios higher than 100:1. Given this, the 1,500 rpm motor speed listed in the design objectives will be assumed to be the *maximum* motor speed, with lower speeds being achievable through some motor speed control mechanism. This lower speed will increase the torque input to the gearset, allowing the winch to operate at its required 4,000 in·lbf torque output. When loads do not exceed approximately 875 lbf (requiring output torque exceeding 3,500 in·lbf), the winch can be operated at its full 1,500 rpm. While a motor speed of as high as 1,485 rpm can operate the winch at full load given the best available gearset, the reduction of this design to one possible gearset eliminates the possibility of further refining the design for cost or space. Since the design parameters are already being adjusted to accommodate the gear supplier's limited selection, it would be advantageous to choose a lower motor speed for this high-load operating mode so that multiple gearsets can be evaluated for their stresses, maximum possible output power, cost, and size. Thus, it will be assumed that the motor speed while lifting the maximum 1,000 lbf load will be approximately 1,200 rpm, a reduction of 20%.

Because the gearset should self-lock, the lead angle should be kept below 10°. While a higher lead angle (as close to 10° as possible) would improve the efficiency of the gearset, the only gears available from the supplier in appropriate gear ratios have lead angles of under 5°, thus removing this as a variable during the design process.

The worm is assumed to be case-hardened steel, and the gear is assumed to be made of chill-cast bronze. The coefficient of friction is calculated as dependent on sliding velocity, but is generally between 0.02 and 0.04.

The application factor K_a is assumed to be 1.25, given uniform loading from the driving machine (the electric motor) and moderate shock from the driven machine (the winch drum). The case is approximated as a box with side lengths:

$$L = 1.2d_G$$

$$W = 2d_W$$

$$H = d_G + 2.5d_W$$

This approximates lateral case area as:

$$A \approx 2\,(L \cdot W + L \cdot H + W \cdot H)$$

This approximation is motivated by Figure 2.

6. Calculations

6.1 Ideal calculations

The calculations in this section are used to calculate gearset parameters under the assumption that *all* of the available power from the motor is transmitted through the gearset to the output of the winch. In reality, the worm applies some force to the gear radially and axially, which do not contribute to the transmitted torque and rob the system of output power. Additionally, frictional losses dissipate input power as heat, further reducing the amount of input power which is transmitted to the output. Calculations including these factors are performed after the ideal conditions are used to set some bounds on the design of the gearset.

6.1.1 Output power

For a peak torque output of 4000 lbf·in at a minimum output speed of 30 rpm, we can calculate the required output power as:

$$P_{out} = T_{out}n_{out} = 4000\ in \cdot lbf * (30rpm * \frac{2\pi}{1\ rev} * \frac{1\ min}{60\ s}) = 12566.4\ \frac{in \cdot lbf}{s} * \frac{1\ hp}{6600\frac{in \cdot lbf}{s}} \approx 1.9\ \text{hp}$$

This required 1.9 hp output power exceeds the maximum 1.2 hp input power from the provided motor specifications, even before considering frictional losses. As such, the project objectives cannot be completed as requested. These calculations will continue under the assumption that the 4,000 lbf·in torque requirement is critical, and that the 30–35 rpm output speed is noncritical and can be changed to meet other requirements. Reducing the output speed will allow the required output power to remain below that which can be output from the gearset, given the 1.2 hp input to the gearset from the motor.

6.1.2 Gear ratio

The gear ratio m can be determined by comparing the input and output torque:

$$T_{out} \geq 4000\ in \cdot lbf$$

$$T_{in} = (1.2\ hp * \frac{6600\ \frac{in \cdot lbf}{s}}{1\ hp}) \div (1500\ rpm * \frac{2\pi}{1\ rev} * \frac{1\ min}{60\ s}) \approx 50\ in \cdot lbf$$

$$m \geq \frac{4000}{50} = 80$$

Thus, the minimum possible gear ratio must be at least 80:1. However, the actual output torque from the gearset will be lower than predicted, as some of the torque from the worm will be applied to the gear axially or radially which do not contribute to output torque. There will additionally be power loss in the gearset due to sliding friction between the gear teeth. With this in mind, the gear ratio needed to achieve required output torque will exceed 80:1 by a potentially substantial amount.

6.1.3 Output speed

With an ideal gear ratio of 80:1, the maximum possible output speed can be determined:

$$n_{out} \le \frac{n_{in}}{m} = \frac{1500\ rpm}{80} = 18.75\ rpm$$

Because the design must already necessarily operate below the output speed objective (by a difference of 37.5%), the gear ratio should remain as close to 80:1 as possible to minimize the difference between desired and actual output speeds.

6.2 Non-ideal calculations

The previous calculations provide helpful insight to designing a successful gearset. Where the initial design constraints from the problem statement leave a vast array of possible gears to check for performance, these additional constraints help reduce the number of possibilities to a manageable level. Critically, a gear ratio greater than or equal to 80:1 eliminates the vast majority of possible gear combinations from the gear supplier. There are only twelve possible worm gears remaining, which are listed in Table 1. Each of these twelve gears can be procured in a left- or right-handed variant, but as this difference plays no role in this stage of design, these variants have been ignored. The twelve options were each analyzed to find their resultant forces and output torque. This process is demonstrated for one possible gearset combination below, while the results for all twelve are provided in Table 2.

6.2.1 Initial force, output torque calculations

This section will cover the force and torque analysis of the worm gear WB6100 and its associated worm LWG6R. This combination yields the highest output torque of the twelve possibilities, which will become important as the calculations proceed. Critical parameters for WB6100 are listed in Table 3.

Output speed is found using input speed and gear ratio *m*:

$$n_G = \frac{n_W}{m} = \frac{1500\ rpm}{100} = 15\ rpm$$

Worm and gear pitch-line velocities are respectively given by:

$$V_W = \frac{\pi d_W n_W}{12} = \frac{2\ in \cdot 1500\ rpm \cdot \pi}{12\ \frac{in}{ft}} = 785.4\ \frac{ft}{min}$$

$$V_G = \frac{\pi d_G n_G}{12} = \frac{16.667\ in \cdot 15\ rpm \cdot \pi}{12\ \frac{in}{ft}} = 65.45\ \frac{ft}{min}$$

Sliding velocity is given by:

$$V_S = \frac{V_W}{cos(\lambda)} = \frac{785.4\ \frac{ft}{s}}{cos(4.6667^o)} = 788.0\ \frac{ft}{min}$$

The coefficient of friction is computed by equation:

$$V_S > 10\ \frac{ft}{min} \ \Rightarrow\ f = 0.103\ exp(-0.110 V_S{}^{0.450}) + 0.012 = 0.02327$$

Tangential force on the worm is given by:

$$W_{Wt} = W^x = \frac{33000\ H}{V_W} = \frac{33000 \cdot 1.2\ hp}{785.4\ \frac{ft}{s}} = 50.42\ lbf$$

This is used to compute the total force applied to the gear from the worm:

$$W = \frac{W^x}{cos(\Phi_n)sin(\lambda) + f\ cos(\lambda)} = \frac{50.42\ lbf}{cos(14.5^o)sin(4.6667^o) + 0.02327 \cdot cos(4.6667^o)} = 494.5\ lbf$$

Finally, the output torque is derived from the force tangential to the gear:

$$|W_{Gt}| = W^z = W[cos(\Phi_n)cos(\lambda) - f\ sin(\lambda)] = 476.2\ lbf$$

$$T_{out} = \frac{W_{Gt} d_G}{2} = \frac{476.2\ lbf \cdot 16.667\ in}{2} = 3968.7\ in \cdot lbf$$

Despite the fact that this setup has a gear ratio of 100:1 and generates an output speed of only 15 rpm, it is still insufficient to produce the required output torque described in the problem statement. The choice must be made to either check if this slightly lower torque output is acceptable to the client, or make further adjustments to the design constraints. As before, having no client to consult in this hypothetical exercise, assumptions must be made about the usage requirements of the winch. This report will continue under the assumption that the listed 1,500 rpm motor speed is the *maximum* speed of the motor, and that its speed can be controlled to be lower than 1,500 rpm when the maximum output torque is required for winch operation.

6.2.2 Force, output torque calculations for reduced input speed

With the 1,500 rpm motor input speed found to be incompatible with other design constraints, it will be assumed that the motor can operate at a slower 1,200 rpm when lifting loads approaching the required load limit of the winch. At this speed, the torque input to the gearset is:

$$T_{in} = (1.2\ hp * \frac{6600\ \frac{in \cdot lbf}{s}}{1\ hp}) \div (1200\ rpm * \frac{2\pi}{1\ rev} * \frac{1\ min}{60\ s}) \approx 63\ in \cdot lbf$$

This additional input torque should provide sufficient additional output torque such that multiple combinations of available gearsets can meet the design requirements. The calculations performed in the previous section can be repeated for this increased input torque. These results will be summarized below for the same combination of WB6100 and LWG6R, and results for each of the twelve gearset combinations can be found in Table 4.

Output speed is found by:

$$n_G = \frac{n_W}{m} = \frac{1200\ rpm}{100} = 12\ rpm$$

Sliding velocity is given by:

$$V_S = \frac{V_W}{cos(\lambda)} = \frac{628.3 \frac{ft}{s}}{cos(4.6667^o)} = 630.4 \frac{ft}{min}$$

The total force applied to gear from the worm is:

$$W = \frac{W^x}{cos(\Phi_n)sin(\lambda)+f\ cos(\lambda)} = \frac{63.03\ lbf}{cos(14.5^o)sin(4.6667^o)+0.025927 \cdot cos(4.6667^o)} = 602.5\ lbf$$

Lastly, the output torque is given by:

$$T_{out} = \frac{W_{Gt}d_G}{2} = \frac{580.09\ lbf \cdot 16.667\ in}{2} = 4834.2\ in \cdot lbf$$

As predicted, the 20% reduction in motor speed results in an approximately 20% increase in output torque. This allows seven of the twelve gearset combinations to produce sufficient output torque to meet the design requirement.

6.2.3 Stress calculations, check for sufficiency

With seven gearsets capable of producing the necessary output torque, these gears may now be analyzed for failure and fatigue due to contact and bending stresses. Calculations will be demonstrated below for the combination of WB6100 and LWG6R as before. Results for all seven combinations can be found in Table 5.

Using a design factor $n_d = 1.2$, application factor $K_a = 1.25$, output power $H_0 = 0.920425$ hp, gear pitch-line velocity $V_G = 52.361$ ft/min, and efficiency $e = 0.767$, the actual tangential force applied to the gear teeth is:

$$W_G{}^t = \frac{33000\ n_d H_0 K_a}{V_G e} = \frac{33000 \cdot 1.2 \cdot 0.920425 \cdot 1.25}{52.361 \cdot 0.767} = 1134.4\ lbf$$

To compute the allowable tangential force on the gear teeth, the following correction factors are used:

$$D_m = d_G = 16.667\ in$$

$$F_e = 1\ in$$

$$C_s = 1412 - 456\ log(D_m) = 854.8 \quad \text{(Chilled-cast, } C > 3\text{, } D_m > 8\ in\text{)}$$

$$C_m = 1.1483 - 0.00658\ m_g = 0.4903 \qquad (m_G > 76)$$

$$C_v = 0.659\ exp(-0.0011\ V_S) = 0.3294 \qquad (V_S < 700 \frac{ft}{min})$$

Computing the allowable tangential force:

$$W^t{}_{allow} = C_s D_m{}^{0.8} F_e C_m C_v = 854.8 \cdot 16.667^{0.8} \cdot 1 \cdot 0.4903 \cdot 0.3294 = 1310.8\ lbf$$

The actual tangential force on the gear teeth is within the allowable force indicated by the AGMA method.

The gears will now be checked for sufficient power to drive the output. The frictional force is:

$$W_f = \frac{f\,W_G{}^t}{f\,sin(\lambda) - cos(\Phi_n)cos(\lambda)} = 30.5\ lbf$$

From this, we can compute the frictional power:

$$H_f = \frac{|W_f|\,V_S}{33000} = \frac{30.548\ lbf \cdot 630.4084\ \frac{ft}{min}}{33000} = 0.584\ hp$$

The power transmitted by the worm and gear is, respectively:

$$H_W = \frac{W_W{}^t V_W}{33000} = \frac{123.3\ lbf \cdot 628.3\ \frac{ft}{min}}{33000} = 2.35\ hp$$

$$H_G = \frac{W_G{}^t V_G}{33000} = \frac{1134.4\ lbf \cdot 52.4\ \frac{ft}{min}}{33000} = 1.8\ hp$$

The power needed at the output is given by:

$$H_{out} = \frac{H_0 K_a}{e} = \frac{0.920425\ hp \cdot 1.25}{0.767021} = 1.50\ hp$$

This illustrates that the gears are able to transmit the required power to the output.

To estimate the gear's ability to withstand the required load, the following factors are used in the Buckingham wear load equation:

$K_w = 90$ (Hardened worm, chilled bronze gear, 14.5°)

$$d_G = 16.667\ in$$

$$F_e = 1\ in$$

$$(W_G{}^t)_{allow} = K_w d_G F_e = 90 \cdot 16.667 \cdot 1 = 1500.3\ lbf$$

The 1,134.4 lbf actual tangential force on the gear is within this limit.

From these calculations, we can conclude that this combination of worm and gear provides sufficient output power to operate the winch and can do so without failure due to wear or stress.

6.2.4 Heat management

The heat loss rate of the gearset is given by:

$$H_{loss} = 33000(1 - e)H_{in} = 33000\ (1 - 0.767)\ 2.35\ hp = 18,042\ \frac{ft \cdot lbf}{min}$$

Assuming there is no fan on the worm shaft, the overall radiative and convective heat transfer coefficient is:

$$\hbar_{CR} = \frac{n_W}{6494} + 0.13 = \frac{1200}{6494} + 0.13 = 0.31479\ \frac{ft\cdot lbf}{min\cdot in^2\cdot {}^oF}$$

The lateral case area is approximately 1080 in^2 and the ambient temperature is 120°F. The oil sump temperature is therefore:

$$t_s = t_a + \frac{H_{loss}}{\hbar_{CR}A} = 120^oF + \frac{18042}{0.31479\cdot 1080} = 173^oF$$

This is a reasonable oil sump temperature, as it is low enough to avoid damaging grease or oil in the gearbox. The results of these calculations for the other six gear combinations are listed in Table 6.

7. Results

Initial results showed that the proposed input to output rpm would be impossible to achieve while still meeting all other constraints. In particular, the stated maximum motor power (1.2 hp) was not enough to deliver the torque and speed output requirements. To adjust for this, we proposed a reduced output speed, which meant the torque output always failed to meet the minimum requirement of 4000 in-lbf given the maximum gear ratio from the supplier of 100:1, with the maximum output being found on the WB6100, which reached a torque output of 3968.5 in-lbf (Pitch 6, Candidate #1 on attached Excel spreadsheet). To resolve this, another analysis was performed, with the input speed reduced to 1,200 rpm and the reduced output speed of 15–20 rpm. Using this proposed solution, two gear candidates passed all requirements, with the maximum output torque produced by the WB6100 at 4834.2 in-lbf. The worm for each gear was kept the same, selecting the maximum available length from the supplier while still matching the pitch of the worm gear.

8. Discussion

To resolve the issue of being unable to meet all requirements, we proposed reducing the motor speed, allowing for an increase in torque. As a result, all other requirements can be met, and the output rpm can remain the same. The only two gear combinations to transmit the required torque and power, have sufficiently low tooth stress and wear, and prevent excessive sump heat are the WB6100 and WB696. All gears of higher diametral pitch experienced tangential forces higher than allowable by the AGMA method and the Buckingham wear load equation. There are no real significant advantages to one or the other—the WB6100 exceeds requirements by a wider margin and has a lower sump temperature, while the WB696 is marginally smaller, less expensive, and has a slightly higher output speed. Given the design objective to reduce price and size, and that the sump temperature is still quite low, the WB696 gear and associated WHG6 worm are the preferred gearset combination for this winch design. The analysis also showed that there are very few possible combinations of gears that can fully accomplish the task set, which means that a computer program or Excel sheet can reduce the vast number of gears to just a few suitable options with relative ease. In this case, an Excel spreadsheet was used which can be found attached to this document.

Custom-ordering parts would allow for more efficient designs, which mostly closely matched the requirements of the project, however that would significantly increase cost, and introduce issues that come with custom parts, such as more complex tolerancing.

9. Conclusions

The initial problem stated that we should convert the 1500 rpm, 1.2 hp output of the motor to the required 4,000 in·lbf torque to the winch drum, which should have an output speed of 30–35 rpm. The design should be able to operate in 120°F temperatures without producing excessive heat and operate under a given load 4–5 hours per day without excessive wear or damage. A focus on minimizing cost and size was emphasised when approaching the design of this system. However, after an initial analysis, it became clear that the input speed, output speed, torque, and power requirements could not all be met simultaneously. If the first three were met, then the power requirements became too high, and similarly if the power or speed requirements were met, then the torque fell too low. To resolve this, we proposed reducing the input and output speeds to 1,200 rpm and 15–20 rpm, respectively. This allowed for the torque, stress, and sump temperature requirements to be met by two worm gears: WB6100 and WB696. The WB696 gear was selected, along with the matching WHG6 worm, as they were slightly smaller and cheaper than the other option and still maintained a low sump temperature.

10. Appendices

A1. Tables

Table 1: Available worm gear options which meet initial design constraints

	m **= 100**	***m*** **= 96**	***m*** **= 80**
P **= 6 in.**	WB6100	WB696	WB680
P **= 8 in.**	WB8100	WB896	WB880
P **= 10 in.**	WB10100	*N/A*	WB1080
P **= 12 in.**	WB12100	*N/A*	WB1280
P **= 16 in.**	WB16100	*N/A*	WB1680

Table 2: Output torque and resultant forces from the twelve possible gearsets

	T_{out} **(in·lbf)**	***W*** **(lbf)**	W^z **(lbf)**	**Efficiency** ***e***
WB6100	3968.5	494.5	476.2	78.7%
WB696	3809.7			
WB680	3174.7			
WB8100	3837.0	637.7	613.9	76.1%
WB896	3683.5			
WB880	3069.6			
WB10100	3657.4	759.8	731.5	72.5%
WB1080	2925.9			
WB12100	3585.8	894.5	860.6	71.1%
WB1280	2868.9			
WB16100	3557.2	1186.5	1138.3	70.6%
WB1680	2845.8			

Table 3: Critical parameters of WB6100

Number of teeth (N_G)	100
Pressure angle (ϕ_n)	14.5°
Pitch diameter (d_G)	16.667 in.
Worm pitch diameter (d_W)	2 in.
Worm lead angle (λ)	4.6667°
Resulting gear ratio (m)	100:1

Table 4: Output torque and forces with 1,200 rpm input speed

	T_{out} (in·lbf)	W (lbf)	W^z (lbf)	Efficiency e
WB6100	4834.2*	602.5	580.1	76.7
WB696	4640.7*			
WB680	3867.2			
WB8100	4662.4*	775.0	746.0	74.0%
WB896	4475.9*			
WB880	3729.9			
WB10100	4436.9*	922.0	887.4	70.4%
WB1080	3549.5			
WB12100	4349.7*	1085.4	1044.0	69.0%
WB1280	3480.1			
WB16100	4327.6*	1444.0	1384.8	68.7%
WB1680	3462.0			

** Options which produce the required 4,000 in·lbf output torque*

Table 5: Stress calculations for seven gearset combinations

	Actual tangential gear force W_G^t (lbf)	Allowable tangential gear force $(W^t)_{all}$, AGMA (lbf)	Allowable tangential gear force $(W_G^t)_{all}$, Buckingham wear load (lbf)
WB6100	1158.1*	1310.9	1500.0
WB696	1158.1*	1373.3	1485.0
WB8100	1544.2	990.8	843.8
WB896	1544.2	1019.4	810
WB10100	1893.8	796.1	567.0
WB12100	2267.5	617.9	375.0
WB16100	3022.0	366.4	174.4

* *Options for which* W_G^t *remains below* $(W^t)_{all}$ *and* $(W_g^t)_{all}$

Table 6: Heat calculations for gearset combinations

	Approximate lateral case area A (in^2)	Heat loss from case H_{loss} ($\frac{ft \cdot lbf}{min}$)	Sump temperature t_s (°F)
WB6100	1080	19,660	177.8
WB696	1063	19,660	178.8
WB8100	630	22,572	233.8
WB896	591	22,572	241.3
WB10100	423	25,112	308.8
WB12100	300	26,610	401.8
WB16100	172	27,017	619.4

A2. Figures

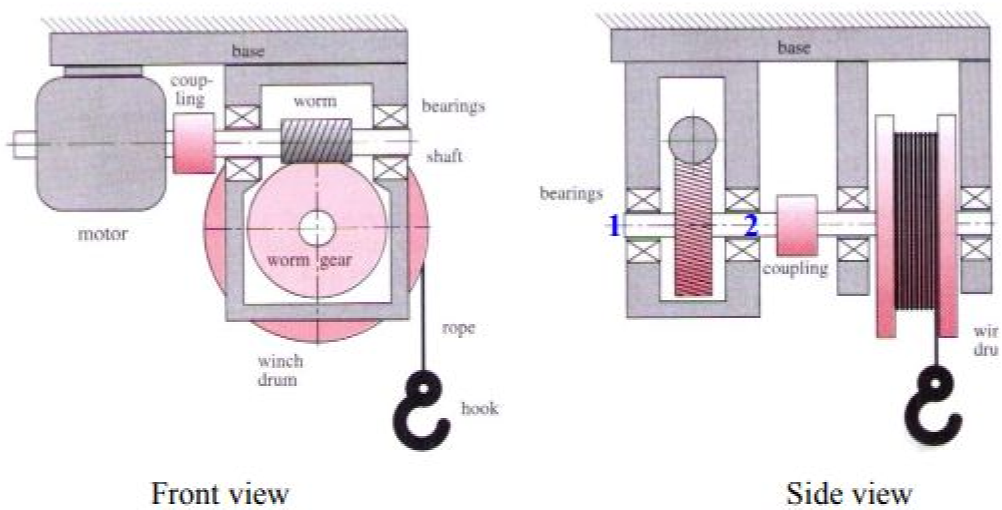

Figure 1: Diagram of the basic winch setup, provided in the project description

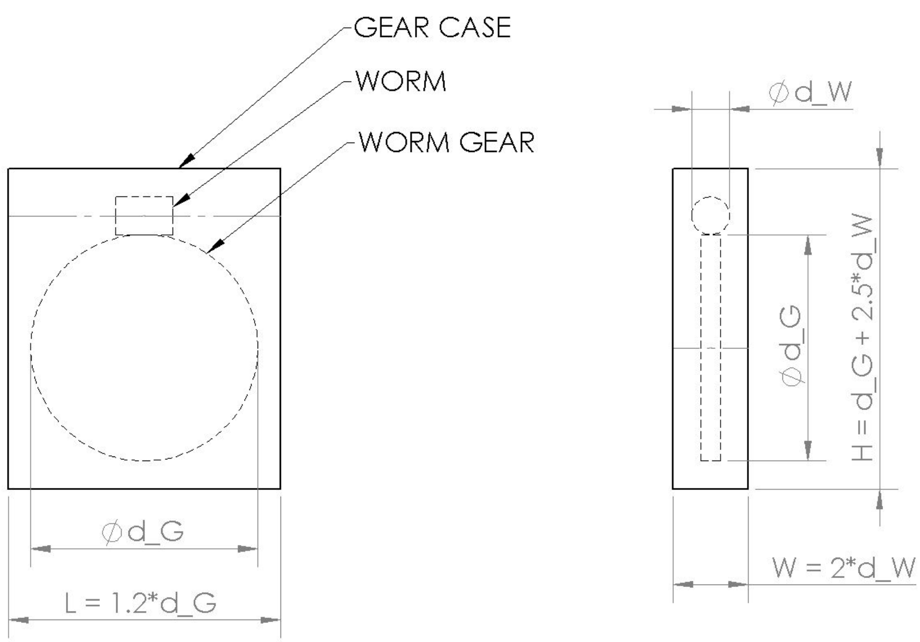

Figure 2: Case dimensions approximated by d_W *and* d_G