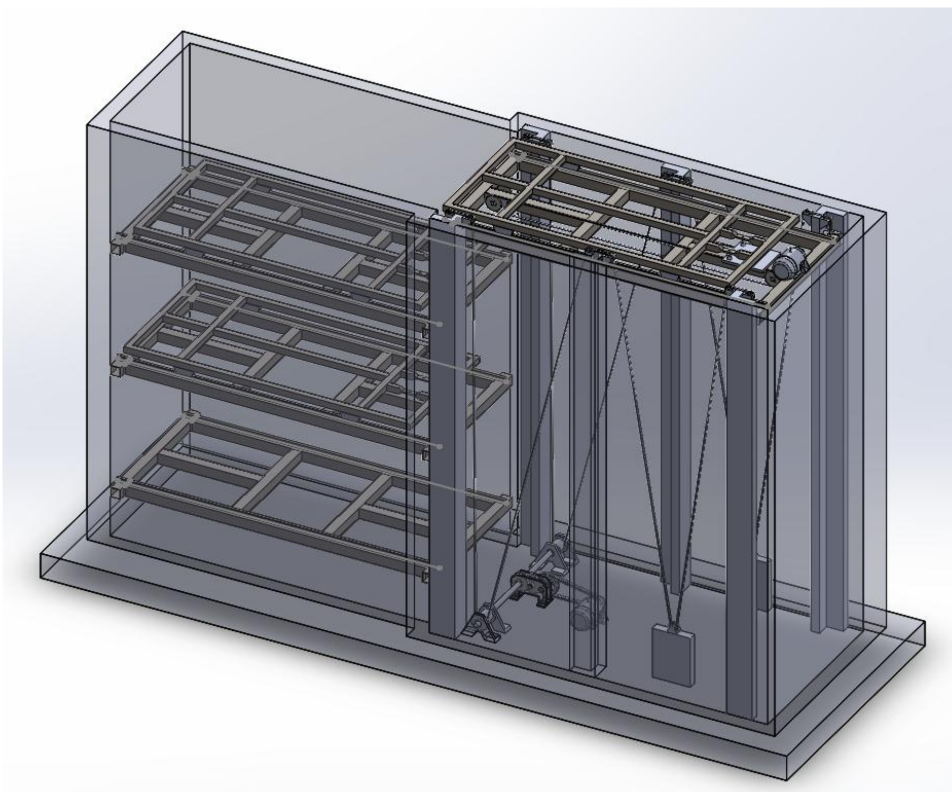
Mechatronic Systems Engineering
School of Engineering Science
SIMON FRASER UNIVERSITY

MSE 320 – Machine Design
- Project Report Part 2-

Roadside Underground Parking System

To the Attention of Dr. Krishna Vijayaraghavan



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Executive Summary

This report acts as a sequel and conclusion to the design of the Speed-Park underground parking system conferred by *"Roadside Underground Parking System"*. Proceeding from the subsequent report, this report conveys the design process and analysis taken to implement a cost-effective yet practical means for the power transmission required by the Speed-Park parking system. This report will systematically communicate details involved in the comparison of concept designs in order to arrive at the superior concept, followed by the mechanical design of the recommended power transmission system along with respective CAD models of the individual components and integrated system. Upon determining the particulars of the power drive system, analysis of the overall cost; overall safety; and ease of maintainability and installation. Lastly, information is given into recommended housings and frames, and installation practises for the power transmission system.

To develop such a power transmission system, market research is conducted on common industry practises for related applications from which concepts are drawn and analyzed to arrive at the optimal power transmission system for implementation in the Speed-park system which requires individual systems for vertical and horizontal actuation. Upon establishing the optimal concept, the system is designed for the power transmission requirements and ensuing loads to tailor the system for performance and reliability. These calculations are to be based on theoretical knowledge and governing relationships of the individual components of the mechanical drive and their bonds. Solidworks is then implemented to provide visualizations of the individual components, such as V-belts, Spur gear drives, Shafts and the integration of the aforementioned; which also serve to verify the dimensioned system. Industry contacts are then to be consulted, along with previous design calculations, to encapsulate and review the overall cost, safety and maintainability and installation of the entire system.

Doing so, the optimal design was found to consist of a vertical actuation system which consisted of a step down from the AC motor conducted by a V-belt followed by a gear train, which transmits power to the input shaft. The input shaft then further transfers the power through steel wire ropes, wound about drums on either end of the shaft, to engage the vertical translation of the elevator platform and subsequently the counterweight. The horizontal actuation system consisted of much lower power requirements, and consists of the moving platform being driven by chains and a motor mounted on the elevator platform.

Integration of the power transmission not only follows the cost effective, practical and safe nature of the Speed-Park structure but also has been done such to provide a total retrieval time of 1.5 minutes. A typical installation of the Speed-Park system is expected to run a total cost of $150,000 without consideration to the cost of land. The system has incorporated proven techniques and processes utilized commonly in industry on a simplistic design, which gives proof to high reliability expectations. Possible improvements to the system would include addition of a backup generator, as under power loss the retrieval of vehicles is currently prohibited.

Introduction

In the attempt to increase density of parking available in metropolitan cores, implementation of underground parking systems such as the speed-park is expected to disseminate across the globe due to wide spread recognition as an optimal resolution to the problem at hand.

This report addresses the design of a power transmission system for the speed-park underground parking structure which provides an optimal relationship between performance, capital cost, and safety. In order to devise such a system, initial market research is conducted by studying key features and drawbacks of possible design concepts in order to arrive at a consolidated optimal design. Design constraints such as power transmission and allocated volume are then integrated into the mechanical designed to ensure the capability and practicality of the actuation system.

Objectives and Scope

The Objective of this report is to exercise and apply engineering fundamentals and knowledge obtained throughout the latter portion of MSE 320 towards the mechanical design and analysis of a power transmission system comprising of V-belts, Spur gears and shafts. This is accomplished by way of this report as it addresses the systematic kinematic and mechanical design for the transmission of power over various components through application of governing imperial formulas followed by verification through MDesign.

This report shall focus on the design and analysis of an optimal power transmission system, and its individual components, for the speed-park parking structure by implementing the power requirements and physical constraints which are extracted from blueprint drawings. Upon weighing the Pro's and Con's of various concepts, an optimal design is reached suffice to economic feasibility, feasibility and practicality requirements for the driven structures. Motor selection and housing design are to be followed by mechanical design specific to individual components to derive dimensional and material specifications of the mechanical drive components such to withstand and suffice to safety requirements for the stresses they will be subject to. CAD tools and analytical software such as Mdesign, are to be implemented in order to illustrate the holistic mechanical drive and provide a means of confirmation to calculated values. Furthermore, our analysis will conclude with providing measures of safety and cost of the recommended system along with an introduction to installation. Detailed electronic power and control system design is out of the scope of this analysis along with tolerances, bolts, nuts, welds, clutches and brakes.

Preliminary Research Findings

Common Industry Practises

The actuation system required for the speed-park structure may be divided into two subsystems, which provide vertical actuation of the elevator platform and horizontal actuation of the moving platform respectively. Thus, our design shall commence with industry research for proven and practical approaches of power transmission systems for both the vertical and horizontal actuation requirements.

Description of Concept Alternatives

In this section we examined three candidates for vertical actuation and deduced the optimal concept for integration into speed-park.

Hydraulic Actuation

Hydraulic lifting mechanisms provide for an effective of actuation for high load applications such as the vertical actuation of the elevator platform in the Speed-park parking system. Hydraulics may be integrated by way of boom actuation or scissor lift achieving a very high power density in comparison with other forms of actuation.

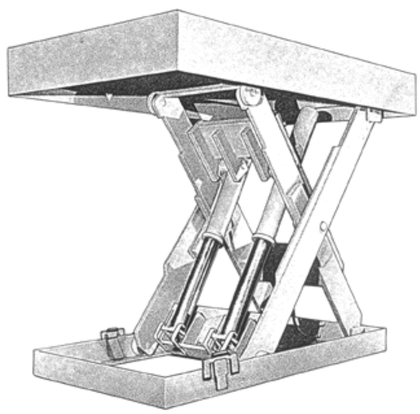


Figure xx: Hydraulic scissor lift schematic

For the application in question key drawbacks from a hydraulic actuation would be the requirement for a significant initial investment, along with the necessity to implement regular maintenance and upkeep of the hydraulics. For the required stroke of 7.5m, a hydraulic based actuation system would require capital of magnitudes higher than that allocated for alternative approaches, as such strokes would require specialty hydraulics or the acquisition of overly high load specified system. In addition to aspects of cost related to the high stroke, considerations into the control, power and structural requirements results in limited application for high stroke precision applications. Furthermore, hydraulics by nature are inefficient as they produce losses whilst transmitting power from fluidic to mechanical domains and the performance of hydraulic lifts also becomes erratic as fluid properties vary with temperature which will result in issues between winter and summer seasons. Upon consideration of the above factors and costs induced by the requirement of pumps and motors along with the needs of long term maintenance, this hydraulics are not considered to be a viable option for the Speed-park System

Cable Elevator Actuation

Another concept that was considered and analyzed as a candidate for vertical actuation was a cable elevator system, consisting of 2 steel ropes at either end of the elevator platform which would be routed from the platform, through pulleys to drums placed at the base of the parking structure. Torque to power the rotation of the drums, and subsequently provide vertical actuation for the platform, may be provided by an AC motor which would follow with cascading speed reduction drives to arrive at a shaft-drum assembly. This method would provide for a much more cost efficient means of actuation, as the relative cost of AC motors, mechanical power transmission components and wire ropes is far below that of hydraulics. Utilization of steel wire ropes allows for reliability and safety in the design, as properties for wire ropes are well defined as they are utilized commonly in engineering practise. The maintenance requirements for such a system also consist of replacement of sheaves, belts, chains and gears after predetermined lifespans which often occur at several thousand hours. This system offers great simplicity, which will ensure greater reliability and cost effectiveness but the motor must be capable of handling large loads and must be able to provide constant holding torques while the platform is stationary. For an application such as that required by the Speed-Park system this method may be acceptable, but offers room for improvement.

Evolving from the basic cable elevator actuation system, we may integrate counterweights into the system in order to decrease the holding torque required by the motor while the system is stationary and reduce the load on the motor when vertical actuation is required. This is approach is commonly implemented in elevators, where the counterweight is designed to balance the average weight from the platform to save the motor from holding and moving large weights. The incremental cost of implementing such a counterweight is well accepted for as it allows for greater life expectancy in the motor as well as decreased power requirement of the motor. Efficiency in either cable systems is much higher than that achieved by hydraulic actuation as mechanical drives such as spur gears and V-belts provide transmission efficiency of above 90%

Concept for Horizontal Actuation

Due to the limitation in space allocated for the horizontal actuation system, a dense power transmission system must be adopted with the goals to continue the simplistic approach of the Speed-Park system whilst providing a means of translating from the elevator platform to the designated parking stall within 15 seconds. The two options for available would be either a V-belt drive or Chain transmission, where the V-belt is better suited for high speed applications and areas where slip is permissible. In the operation of the horizontal translation, the moving platform must be accurately lined up to docking points which gives the need to accurate and precise displacement. For the reasons above and to avoid slippage a chain drive is the most prominent option.

Concept Integration and Recommendation

Through integration of the key features of the respective concepts we arrived at an optimal design based on the cable elevator actuation with counterweights, which is modeled by figure xx below:

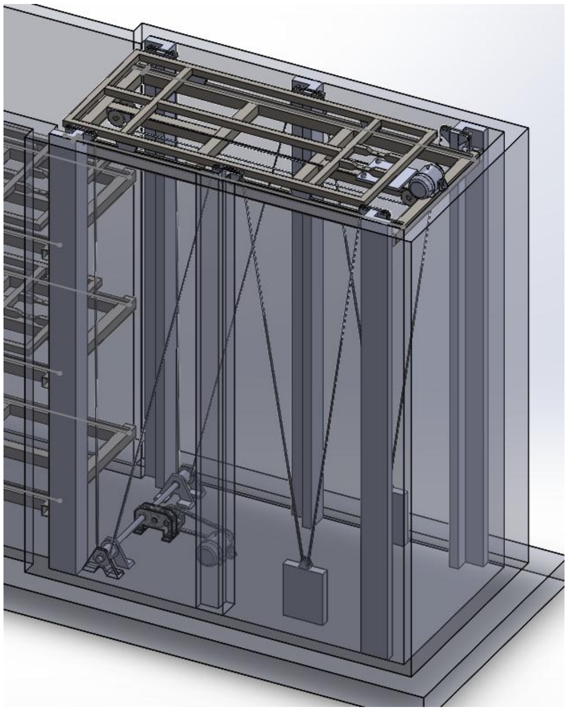


Figure xx: Power Transmission system for Speed-Park parking system

Transmission System Design

For detailed calculations please refer to Appendix 1, while the main results are summarized in the following sections (figure of Layout at the bottom):

AC Motor

To minimize the force due to acceleration, a triangular velocity profile is assumed for the elevator at maximum load (5.55 Tons). If we allow 30 seconds for the platform to move the vehicle up all the way (7.5m), the acceleration and maximum velocity becomes:

$$v_{max} = \frac{d}{\frac{t}{2}} = 0.5\frac{m}{s}, \qquad a = \frac{d}{\left(\frac{t}{2}\right)^2} = 0.033\frac{m}{s^2}$$

An estimate of power required to drive the system is calculated as:

$$\begin{aligned} P &= f.v = [M_{total}(g+a)].v_{\max} \\ &= 27.3\ KW = 36.6\ hp \end{aligned}$$

The EM2540T-CI a Baldor product, shown in Figure xx, will be selected as the main AC motor for vertical translation of the elevator, offering 40hp and 1185 rpm [1].

Figure xx: Main AC Motor for Vertical Translation

Steel Wire Ropes

6x19 IWRC wire ropes as shown in Figure xx will be used to hold the weight of the elevator platforms and the car. A single wire rope of this type has a breaking strength of 6.0 tons, while the total maximum load is 5.55tons distributed between the four corners of the platform [2]. Taking safety factors into account, double wire ropes will be used in each corner, resulting in a maximum of roughly 0.7 tons of load on each wire.

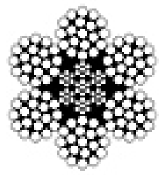
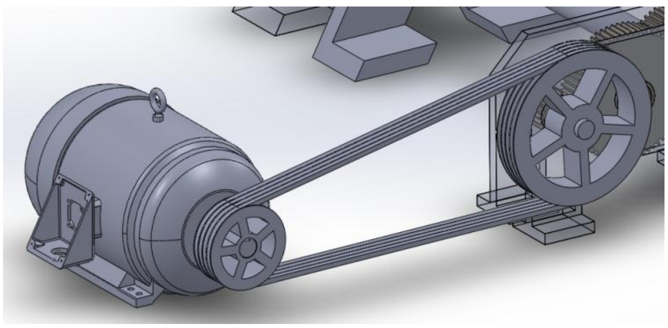


Figure Xx: 6x19 IWRC rope having 15 through and 26 wires per strand

V-Belt

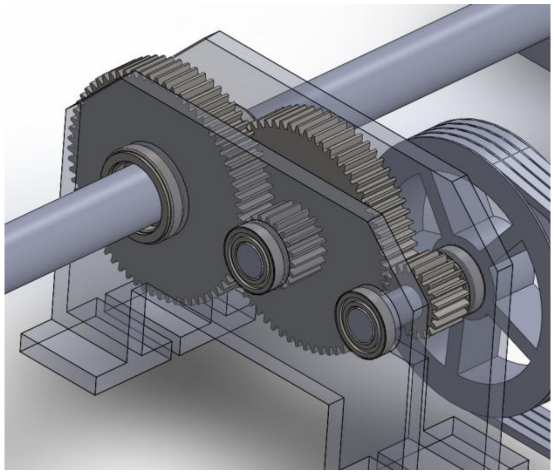


The AC motor is directly connected to the belt drive, where the speed was aimed to be reduced by a factor of 2. The V-Belt serves as a mode of damping from torque shocks between the motor and system, this not only ensures smooth operation of the elevator platform, but protects the motor from possible damages as slight slippage is expected under abrupt changes in operation of the belt. Because there is about of meter of free space at the bottom of the elevator column, 5V belts were used to transmit power to the main shaft. This ensures the sheaves remain at limited sizes, which serve to ensure satisfaction of size constraints as well as reducing the inertia of the transmission system itself. Through belt design analysis the diameters of the sheaves, which satisfy the power requirements, provide the actual speed reduction of:

$$\frac{D_2}{D_1} = \frac{15.90\ in}{8.40\ in} = 1.89$$

With 38hp transmitted, and assuming the motor provides a speed of 1000rpm for the maximum load, 4 V-belts will be required.

Gear Train



Given the motor speed of 1000rmp and desired shaft speed of 47rmp, the required speed reduction ratio of the gear train is about 11. Two pairs of the same spur gears connected through a shaft are used in this manner:

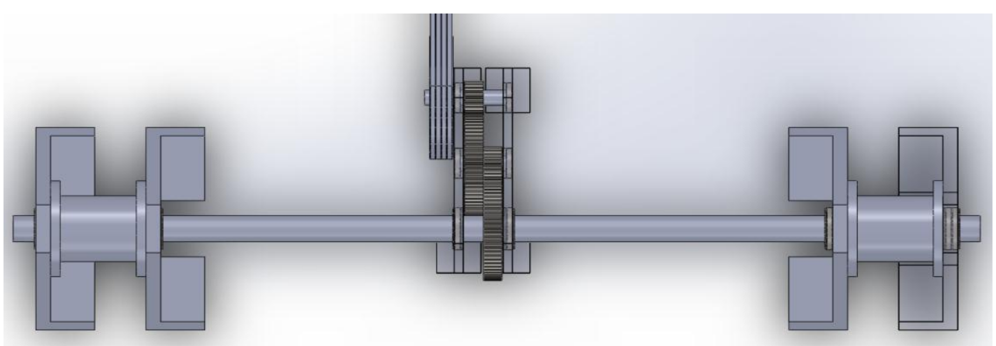
$$\left(\frac{67}{20}\right)\left(\frac{67}{20}\right) = 11.2$$

With a diametral pitch of P_d=5, the gear and pinion diameters are:

$$D_G = 13.40\ in, \qquad D_P = 4.00\ in$$

Through bending and contact stress analysis, a brittle hardness of 478.5 was required; therefore, AISI 1040 steel, water-quenched and tempered at 550°C, was the selected material for the gears.

Shaft

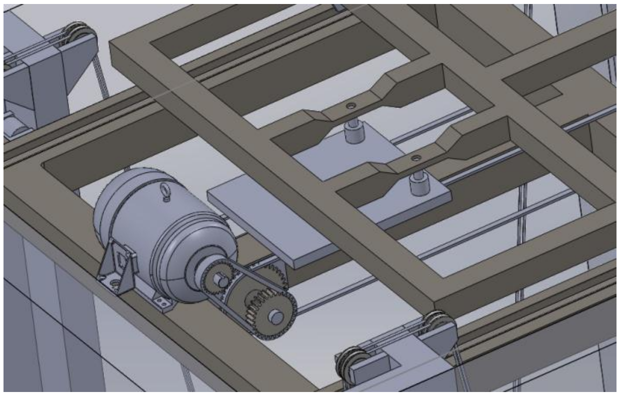


Due to symmetry about where the gear is mounted, design analysis is done only for the right half of the main shaft. Given the vertical loadings from the gears, bearings, and the drums, the point of maximum moment was evaluated to be just before the drum connection. 440C stainless steel was picked to offer high tensile and yield strength (S_u=260ksi, S_Y=240ksi). Using this type of stainless steel, the shaft can operate with a diameter of 3.5 inches.

Counterweight

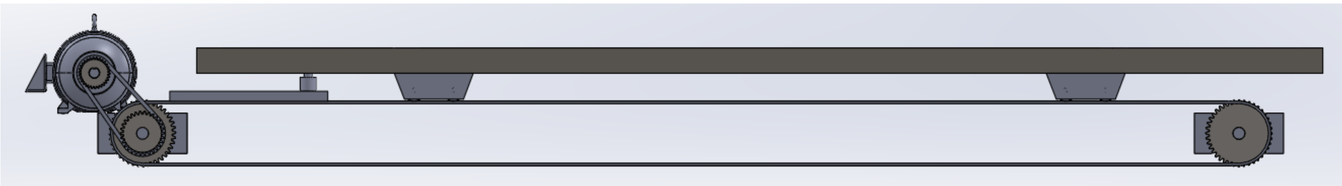
The counterweight pair for the mechanical lift is designed to have the same weight as an unloaded elevator platform. This ensures that the system is in static equilibrium when the system is inactive, so the system will be relatively safe as compared to an unbalanced system even if the brakes fail. Alternatively, a heavier pair of counterweights can be used to further lessen the load on the actuation system when moving the vehicle vertically, reducing power consumption and extending the lifespan of the components.

Horizontal Actuation



The horizontal actuation system consists of a hydraulic latch slider driven by chains and a motor mounted on the elevator platform. The chains convert rotational motion of the rotary actuator to translational motion of the latch slider as in a rack-pinion configuration, powering the moving platform to move onto or back from the stationary platform. The latching mechanism has two hydraulic piston pins that engage the moving platform from beneath, and the slider platform on which it is mounted is able to extend beyond the elevator platform to grip onto the moving platform on a stationary platform.

The platform moves at a maximum speed of 0.8 m/s and a maximum acceleration of 0.107 m/s^2. When loaded, the platform and vehicle weigh at max 3500 kg. Our calculation shows that the power requirement to drive the moving platform is less than one half horsepower, therefore a lower-power AC motor may be used for horizontal actuation. The selected motor is the L3507 from Baldor, which costs $477 and provides 0.75hp [3].



Parts List and Sourcing

1) **Steel Cables** – [4]
2) **Main AC Motor** –[1].
3) **Sheaves –** The
4) **Gears** – In
5) **Belt –** The
6) **Translating AC Motors –** the [3].

Performance Summary

Machine Safety

The design and specifications of the concrete foundation can be obtained by contracting or consulting industry professionals. The rest of the system is made mostly of structural steel and is designed with a safety factor of four. In the stress and/or deformation analysis for each component, dynamic loads are assumed whenever applicable to obtain more conservative figures for the specifications. In practice, enclosures will be used to cover most of the transmission components to ensure safety and to prevent damage to the vehicles.

Cost Analysis

The bare-bone

Maintainability and Installation

The modular design of the Speed-Park system makes it easy to install and deploy. The number of moving components is minimized to reduce the complexity of fabrication and assembly as well as to extend the lifespan of system components. The choice of actuation systems also takes into account the manufacturability and maintainability of each alternative. By using wire ropes and counterweights, the load on the vertical transmission actuator is greatly reduced.

Conclusions

To conclude, the Speed-Park system is a suitable and competitive product given the design requirements. The modular design makes the system highly scalable and flexible in arrangement as well as easy to manufacture and assemble. The design principle of Speed-Park is centred on minimalism. By using proven technologies and materials whose mechanical strengths and properties are known and well-defined, we are able to achieve an efficient configuration that is also safe and practical. Despite being oriented towards simplicity, the system is able to complete a retrieving-parking cycle in one and a half minutes, which is well below the 2.5-minute requirement.

References

[1] "*Product Overview: EM2540T-CI*" – URL: http://www.baldor.com/products/detail.asp?1=1&catalog=EM2540T-CI&product=AC+Motors&family=General+Purpose%7Cvw_ACMotors_GeneralPurpose
[2] "*Standard Wire Rope*" – URL: http://www.stren-flex.com/wire-rope-standard.aspx
[3] "*Product Overview: L3507*" – URL: http://www.baldor.com/products/detail.asp?1=1&catalog=L3507&product=AC+Motors&family=Single+Phase%7Cvw_ACMotors_SinglePhase&winding=34WGY094&rating=40CMB-CONT
[4] "*Wire Co US and Canada Pricing*" – URL: http://unionrope.com/Resource_/PageResource/General%20Purpose-GP-313.pdf

Appendix 1 (Detailed Calculations)

BELT DESIGN

The aim is to reduce the speed of the motor speed by a factor of 2. With 38hp transmitted, the design power using a service factor of 1.4 becomes:

$$P_{des} = (1.4)(38) = 53.2\ hp$$

Because of size constraints for the bottom of the parking (about a meter in height available), the larger diameter (driven sheave) is to be minimized. Therefore, 5V belts are selected to transmit sufficient power, and also reduce size. MDESIGN Mott software was used for iteration, resulting in the following sheave diameters and center distance:

$$D_1 = 8.40\ in$$
$$D_2 = 15.90\ in$$
$$C = 39.75\ in$$

Assuming a motor speed of 1000rpm, the rated power is about 14.3hp given the above diameter. A standard belt length, belt speed, along with the angle of wrap (for small sheave) is then evaluated:

$$v_b = \frac{\pi D_1 n_1}{12} = 2199.1\ \frac{ft}{min}$$

$$L = 2C + 1.57(D_2 + D_1) + \frac{(D_2 - D_1)^2}{4C} \approx 118.0\ in$$

$$\theta_1 = 180° - 2\sin^{-}\left[\frac{D_2 - D_1}{2C}\right] = 169.2°$$

With the above information, correction factors are determined to be C_θ=C_L=0.98. Finally, the corrected power and the number of belts required are:

$$P_{corr} = (14.3)(0.98)(0.98) = 13.7\ hp$$
$$\#Belts = \frac{53.2}{13.7} \approx 4$$

GEAR DESIGN

The actual speed reduction from the belt drive is 1.89; therefore, a speed reduction of roughly 11 is required using a gear train. Two pairs of spur gears, connected through a shaft, were used in this manner:

$$\left(\frac{67}{20}\right)\left(\frac{67}{20}\right) = 11.2$$

Please refer to APPENDIX-2 for tables of detailed information on each gear. For the first set of gears, the pinion will have a speed of n_p=528.3rmp, while the output speed is n_G=157.57rpm.

An overload factor of K_0=1.4 is used for this application, resulting in a design power of:

$$P_{des} = K_0 P \approx 50HP$$

We start by using a diametral pitch of P_d=5, and a pinion diameter of D_p=4.0in. The nominal face width will then be:

$$F = \frac{12}{P_d} = 2.4$$

The above information was used to evaluate the required parameters for material selection:

$J = 0.339$
$SF = 1.3$
$K_R = 1$
$Y_N = 1.315$
$Z_N = 1.213$
$K_0 = 1.4$
$K_s = 1$
$K_m = 1.339$
$K_B = 1$
$K_v = 1.315$
$C_p = 230$

The tangential force and the line speed are:

$$W_t = \frac{126000P}{nD} = 2265.76\ lbf$$
$$v_t = \frac{\pi Dn}{12} = 553.23\ \frac{ft}{\min}$$

The bending and contact stress will then be:

$$s'_t = \frac{SF.K_R}{Y_N}.\frac{W_tP_d}{FJ}K_0K_SK_mK_BK_v = 33975.97\ psi$$
$$s'_c = \frac{SF.K_R}{Z_N}.C_p\sqrt{\frac{W_tK_0K_sK_mK_v}{FD_pI}} = 183188.1\ psi$$

From the contact stress, the brittle hardness is calculated as:

$$HB = \frac{183188.1 - 29100}{322} = 478.5$$

Heat treated AISI 1040, water-quenched and tempered at 550°C, is used to withstand the contact stress in the gears.

SHAFT DESIGN

The forces acting on the x-z axis, applied on the main shaft, are shown in Figure A1, where R_1 and R_2 are the reaction forces due to bearings, W_t is the tangential force from the gear train. W_d is the distributed force experienced along the length of drums, evaluated through The forces were calculated to be:

$$W_t = 7596.625\ lbf$$
$$W_d = 520\frac{lbf}{in}$$

The reaction forces are calculated as:

$$R_2 = \frac{1}{61}[5(W_t) + 12W_d(52)] = 5993.17\ lbf$$
$$R_1 = R_2 + W_t - 12W_d = 7289.8\ lbf$$

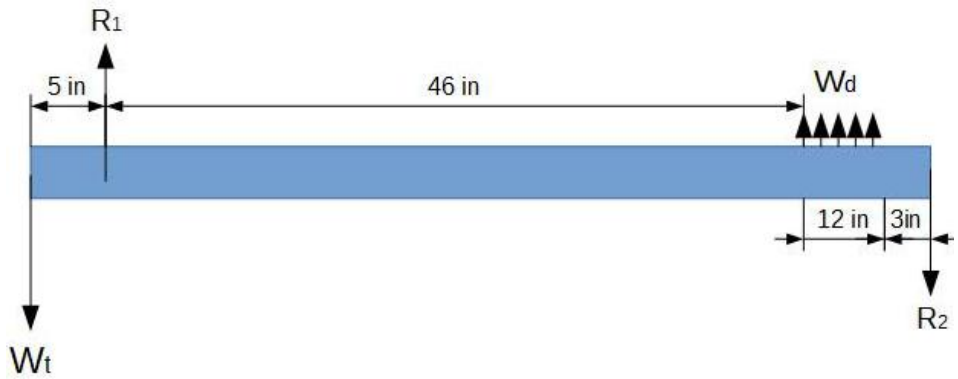


Figure A1: Vertical loading (x-z axis) on the main shaft.

Due to symmetry, the moment is zero at the point where W_t is applied; therefore by observation, the maximum moment will be to the left of the drum (at x=51in), calculated as:

$$M_{y,max} = 5(W_t) + 46(W_t - R_1) = 52.1\ ksi$$

Figure A2 shows the forces on the shaft with respect to y-z axis, where W_r is the radial force from the gear train. The reaction forces are calculated as:

$$W_r = 2764.945\ lbf$$
$$R_2 = \frac{1}{61}[5(W_t)] = 226.64\ lbf$$
$$R_1 = W_t + R_2 = 2991.585\ lbf$$

As the above forces are much lower than the vertical loadings, the overall maximum moment will occur at x=51in. The moment due to horizontal loadings at this point is:

$$M_{x=51} = 5(W_r) + 46(W_r - R_1) = 3.40\ ksi$$

Figure A2: Horizontal loading (y-z axis) on the main shaft.

The overall moment is then:

$$M = \sqrt{M_y^2 + M_x^2} = 52.21\ ksi$$

Because of the high loadings, 440C stainless steel will be used as the material for the main shaft to minimize the diameter. This type of stainless steel offers an ultimate strength of S_u=260 ksi, and the yield strength of S_Y=240 ksi. The endurance strength (S_n) will be about 67 ksi, with 99% reliability (C_R=0.81), and a size factor of C_S=0.8, the actual endurance strength becomes:

$$S_n' = (67)(0.81)(0.8) = 43.42\ ksi$$

The torque in the shaft can be calculated as:

$$T = 63000W_t * \frac{D_G}{2} = 50.9\ ksi$$

Finally, the diameter of the shaft is calculated for a safety factor of N=3.5:

$$D = \left[\frac{32N}{\pi}\sqrt{\left(\frac{M}{S_n'}\right)^2 + \frac{3}{4}\left(\frac{T}{S_Y}\right)^2}\ \right]^{\frac{1}{3}}$$
$$\approx 3.5\ inch$$

Therefore, a standard value of 3.5 inch will be used as the diameter of the main shaft.

Appendix 2 (Table of results)

V-Belt

Input data:

V-Belts, US Standards

Transmitted power	P =	38		hp
Speed of input and output sheave	n =	1000	500	rpm
Driver type		AC motor: High torque		
Driven machine type		Bucket elevator		
Duration of service per day		Less than 6 hours		

Results:

Service factor	C_s =	1.400	
Design power	P_d =	53.200	hp
Actual rated power	=	14.304	hp
Corrected rated power	=	13.773	hp
Center distance	=	39.748	in
Belt speed	V_b =	2199.115	ft/min

Cross section	Length, in	Number of belts
5V or 5VX	118.00	4

		Input	Output	
Sheave diameter	D =	8.400	15.900	in
Angle of wrap	θ =	169.173	190.827	°

		Design	Actual	
Speed ratio	S_r =	2.000	1.893	
Output speed	n =	500.000	528.302	rpm

Gears (connected to shaft)

Input data:

Spur Gear, US Standards

Pressure angle		20°	
Diametral pitch		Pd = 5	
Face width		F = 2.4	in
Transmitted power		P = 38	hp
Rotational speed of pinion		np = 157.57	rpm
Number of pinion teeth		Np = 20	
Desired output speed		ng = 47	rpm
Design life		L = 10	h
Number of load applications per revolution		q = 1	
Rim thickness of pinion and gear	tr = 1	2	in
Gear application	Open gearing		
Elastic coefficient		Cp = 2300	
Overload factor		Ko = 1.4	
Factor of safety		SF = 1.3	
Reliability factor		Kr = 1	

Results:

Actual output speed	ng	=	47.036	rpm
Actual number of gear teeth	Ng	=	67	
Gear ratio	mg	=	3.350	
Quality number	Av	=	11.000	

Geometry parameters

			Pinion	Gear	
Pitch diameter	D	=	4.000	13.400	in
Outside diameter	Do	=	4.400	13.800	in
Root diameters	Dr	=	3.500	12.900	in
Base circle diameter	Db	=	3.759	12.592	in

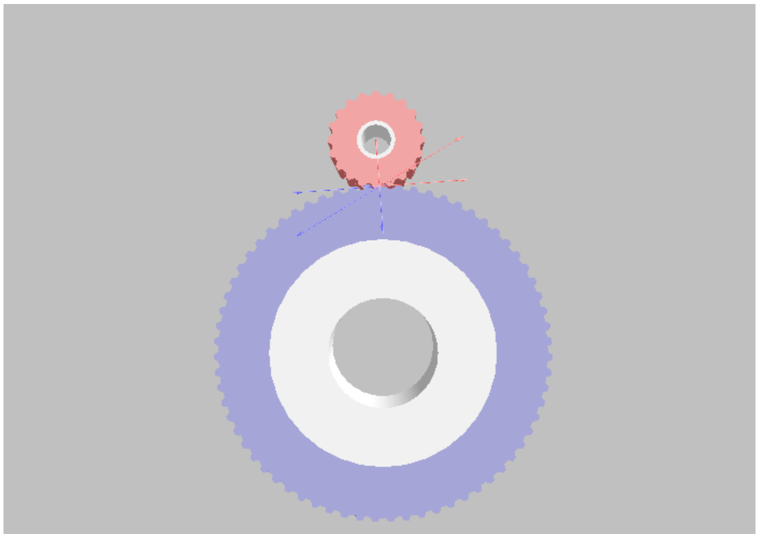
Addendum	a	=	0.200	in
Dedendum	b	=	0.250	in
Clearance	c	=	0.050	in
Circular pitch	p	=	0.628	in
Whole depth	ht	=	0.450	in
Working depth	hk	=	0.400	in
Tooth thickness	t	=	0.314	in
Center distance	C	=	8.700	in
Fillet radius in basic rack	rf	=	0.060	in

			Pinion	Gear	
Bending geometry factor	J	=	0.339	0.429	
Pitting geometry factor	I	=	0.105		

Force and speed factors

Pitch line speed	vt	=	165.007	ft/min
Tangential force	Wt	=	7596.624	lbf
Normal force	Wn	=	8084.158	lbf
Radial force	Wr	=	2764.945	lbf
Size factor	Ks	=	1.000	
Load distribution factor	Km	=	1.339	
Dynamic factor	Kv	=	1.175	

			Pinion	Gear	
Rim thickness factor	Kb	=	1.000	1.000	
Number of load cycle	Nc	=	94542.0	28221.5	
Bending stress cycle factor	Yn	=	1.492	1.693	
Pitting stress cycle factor	Zn	=	1.298	1.389	
Expected bending stress	St	=	102883.166	81279.017	psi
Expected contact stress	Sc	=	295739.572	295739.572	psi
Allowable bending stress number	Sat	=	89642.407	62413.775	psi
Allowable contact stress number	Sac	=	296138.388	276752.912	psi



Gear (Connected to Belt)

<u>Input data:</u>

Spur Gear, US Standards

Pressure angle		20°	
Diametral pitch		Pd = 5	
Face width		F = 2.4	in
Transmitted power		P = 38	hp
Rotational speed of pinion		np = 528.3	rpm
Number of pinion teeth		Np = 20	
Desired output speed		ng = 157.57	rpm
Design life		L = 10	h
Number of load applications per revolution		q = 1	
Rim thickness of pinion and gear	tr = 1	2	in
Gear application	Open gearing		
Elastic coefficient		Cp = 2300	
Overload factor		Ko = 1.4	
Factor of safety		SF = 1.3	
Reliability factor		Kr = 1	

<u>Results:</u>

Actual output speed	ng	=	157.701	rpm
Actual number of gear teeth	Ng	=	67	
Gear ratio	mg	=	3.350	
Quality number	Av	=	11.000	

<u>Geometry parameters</u>

			Pinion	Gear	
Pitch diameter	D	=	4.000	13.400	in
Outside diameter	Do	=	4.400	13.800	in
Root diameters	Dr	=	3.500	12.900	in
Base circle diameter	Db	=	3.759	12.592	in

Addendum	a	=	0.200	in
Dedendum	b	=	0.250	in
Clearance	c	=	0.050	in
Circular pitch	p	=	0.628	in
Whole depth	ht	=	0.450	in
Working depth	hk	=	0.400	in
Tooth thickness	t	=	0.314	in
Center distance	C	=	8.700	in
Fillet radius in basic rack	rf	=	0.060	in

			Pinion	Gear	
Bending geometry factor	J	=	0.339	0.429	
Pitting geometry factor	I	=	0.105		

Force and speed factors

Pitch line speed	vt	=	553.234	ft/min
Tangential force	Wt	=	2265.758	lbf
Normal force	Wn	=	2411.169	lbf
Radial force	Wr	=	824.669	lbf
Size factor	Ks	=	1.000	
Load distribution factor	Km	=	1.339	
Dynamic factor	Kv	=	1.315	

			Pinion	Gear	
Rim thickness factor	Kb	=	1.000	1.000	
Number of load cycle	Nc	=	316980.0	94620.9	
Bending stress cycle factor	Yn	=	1.315	1.492	
Pitting stress cycle factor	Zn	=	1.213	1.298	
Expected bending stress	St	=	34368.045	27151.196	psi
Expected contact stress	Sc	=	170928.585	170928.585	psi
Allowable bending stress number	Sat	=	33980.445	23658.979	psi
Allowable contact stress number	Sac	=	183156.680	171167.084	psi

Chain

The calculation module is based upon DIN ISO 10823 "Guidance on the selection of roller chain drives". This international standard refers to roller chains and chain wheels in accordance with ISO 606.

The selection procedures and the chain ratings in ISO 10823 provide for roller chain drives with a life expectancy of approximately 15,000 hours under the following conditions:

- Utilisation of a suitable method of lubrication (as shown on the results page),
- Use of a lubricating oil with a suitable viscosity class (table can be selected to be shown on results page)
- Centre distance measures between 30 and 50 times the chain pitch,
- Arc of contact of not less than 120° on the drive sprocket,
- Usage of chain adjustment.

In ISO 10823 it is specifically recommended to consult with the supplier of the equipment intended to be used to ensure its suitability.

Input data:

Roller Chains ISO 10823

Method of calculation	Selection and calculation of one chain	
Type of roller chain to be used	American Type (DIN 8188)	
Input power	P = 0.31439	kW
Input speed	n1 = 47.7465	rpm
Number of teeth on driver sprocket	z1 = 40	
Number of teeth on driven sprocket	z2 = 40	
Drive ratio	i = 1	
Approximate centre distance	a0 = 5.5	m

Define minimum and maximum centre distance?	No
Characteristics of driver machine	Slight shocks
Characteristics of driven machine	Smooth running
Application factor to allow for the operating conditions	f1 = 1.1
Factor for number of teeth on drive sprocket	f2 = 0.58
Corrected power	Pc = 0.2006 kW
Allow adjustment of factors f1 and f2?	No
Display intermediate results on output page?	No
Display listing of viscosity classes of lubricating oil on output page?	Yes

Results:

Selected chain type	=	40A	
Number of strands	=	1	
Chain pitch	p =	63.500	mm
Number of links in chain	X =	214	
Chain speed	v =	2.021	m/s
Maximum centre distance	a =	5524.500	mm

Recommended method of lubrication: Oil bath or disc lubrication

Viscosity classes of chain drive lubricating oils

Ambient temperature	**Viscosity class of oil**
-5°C < T < +5°C	VG 68 (SAE 20)
+5°C < T < +25°C	VG 100 (SAE 30)
+25°C < T < +45°C	VG 150 (SAE 40)
+45°C < T < +70°C	VG 220 (SAE 50)

he Moving Platform.