DESIGN OF AUTOMATIC GATE MECHANISM

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We certify that the project entitled "*Design of Automatic Gate Mechanism*" Is written by *Chai Fook Siang* .We have examined the final copy of this project and in our opinion; it is fully adequate in terms of scope and quality for the award of the degree of Bachelor of Engineering. We herewith recommend that it be accepted in partial fulfillment of the requirements for the degree of Bachelor of Mechanical Engineering.

(   )

Examiner   Signature

SUPERVISOR'S DECLARATION

I hereby declare that I have checked this project and in my opinion, this project is adequate in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering

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STUDENT'S DECLARATION

I hereby declare that the work in this project is my own except for quotations and summaries which have been duly acknowledged. The project has not been accepted for any degree and is not concurrently submitted for award of other degree.

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Dedicated to my parents

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I am heartily thankful to my supervisor, Mr. Zamri bin Mohamed, whose encouragement, guidance and support from the initial to the final level enabled me to develop an understanding for this design of automatic gate mechanism project. His wide knowledge and his logical way of thinking have been of great value for me and provided a good basis for the present thesis.

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ABSTRACT

Automatic gate is one of the most preferable domestic intended to provide easy access to gated home. There are three types of automatic gate mechanism, such as sliding, screw drive piston and swing cubic underground. Designs available today are limited only to the three types mentioned. Products available in the market are quite pricy, even when installation and maintenance cost is not yet been considered. Most of the products available in our country are imported from foreign country. The objectives of this project is to study, analyze, and develop a new mechanism that concern with the cost reduction and the mechanism produce should be safe and reliable as well. Here, different type of analysis method was in used in order to develop a proper automatic gate mechanism. Those methods are finite element modeling and mechanical design concept and theories. Stress analysis is done by applying variable maximum loaded stress to ensure the product life service is sustainable. Analysis done also helps in order to select proper material and component specification or sizes for the product development. Therefore, the durability assessment results are significant to reduced the cost and improve the product reliability so as to gain customer confidence. In order to improve the designed mechanism, vibration factor should be take into consideration and more features should be provided.

ABSTRAK

Pintu pagar automatik adalah salah satu barang domestiK pilihan ramai untuk tujuan mudah kawal pintu pagar. Disini, terdapat tiga jenis mekanisme pintu pagar automatik iaitu jenis menggelungsur, pandu skru omboh dan ayunan bawah tanah. Reka bentuk yang terdapat pada masa kini adalah terhad. Produk yang terdapat di pasaran agak mahal,kos tersebut tidak merangkumi pemasangan dan penyengaraan. Kebanyakan produk yang terdapat di negara ini adalah diimport dari negara-negara asing. Matlamat repot ini adalah untuk mempelajari, menganalisis, dan menghasilkan mekanisme baru yang menekankan terhadap pengurangan kos. Mekanisme yang dihasilkan juga haruslah baik dari segi kualiti dan keselamatan. Disini, pelbagai kaedah analisis digunakan untuk tunjuan penghasilan pintu pagar automatik mekanisme yang sempurna. Keadah yang digunakan merangkumi analisis unsur terhingga, rake bentuk mekanikal konsep dan teori. Analsis tersebut adalah berdasarkan tegasan apabila pelbagai daya maksimum dikenakan dan ini adalah untuk memastikan jangka hayat produk dapat dipanjangkan. Analisis tersebut juga membantu dalam process pemilihan bahan dan komponen spesifikasi atau saiz untuk penghasilan produk. Keputusan penilaian kebolehtahanan amat bermakna, ini dapat dibuktikan dari segi pengurangan kos, memperbaiki kepercayaan produk dan keyakinan pelanggan. Sebagai langkah untuk mempertingkatan lagi reka bentuk mekanisme, ciri ciri getaran harus diambil kira dan dilengkapkan dengan perbagai kemudahan sampingan.

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LIST OF SYMBOLS

Z	Total mechanism length
dR	Displacement
N_t	Number of thread
P	Pitch
L	Lead, piston extend length
$d\theta$	Revolution
ω	Angular velocity
v	Linear velocity, passion ratio
dt	Time required for gate opening
F_N	Normal force
m	Mass
a	Acceleration
F_f	Friction force
μ_s	Static coefficient
μ_k	Kinetic coefficient
T	Total torque
$Jrotor$	Rotor inertia
$J\ screw$	Screw inertia
$J\ reflected$	Reflected inertia
α	Angular acceleration
$T\ friction$	Friction torque
$Tscrew$	Screw torque
$Treflected$	Reflected torque

$Trotor$	Rotor torque
Dp	Diameter pitch
M (Load)	Piston arm weight in kilogram
V	Total volume of piston arm, shearing force
ρ	Density
e	Efficiency
N	Number of teeth or factor of safety
n	Speed in Revolution per minute
P	Power, load
Wt	Tangential force
r	Gear pitch radius
Wr	Radial force
ϕ	Pressure angle
$\sum F x$	Total force in X direction
$\sum M$	Total moment
$V1$	Shear force
R	Reaction force
S_y	Yield strength
S_u	Ultimate strength
S_n	Endurance limit
$S'n$	Modified endurance limit
Cs	Size factor
Cr	Reliability factor
D	Diameter shaft
Kt	Stress concentration factor

M	Moment
Ld	Design life
C	Dynamic load rating
E	Modulus of elascity
Cp	Elastic coefficient
V_t	Tangential velocity
K_v	Velocity factor
σc	Compressive stress
σ_{all}	Stress allowable
A	Area

LIST OF ABBREVIATIONS

AISI	American Iron and Steel Institute
ASTM	American Society for Testing and Materials
PID	proportional integral- derivative
NNC	neural network control
ADC	Analog–to–Digital converter
PIO	Programmable Input/output Controller

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CHAPTER 1

INTRODUCTION

1.1 BACKGROUND

Gates are commonly used nowadays at residential area. A gate is a point of entry to a space enclosed by walls, or an opening in a fence. Gates may prevent or control entry or exit, or they may be merely decorative. Today many gate doors are opened by an automated gate operator. Those gates come with many special features. The need for automatic gates has been on the increase in recent times. The system described here incorporates the use of actuator to control the movement of the gate automatically. The automatic gate described here automates the entrances to parking lots of residential homes, organizations, automobile terminus, and public car parks. It uses a remote control convenience to avoid the stress of manually opening and closing the gate. The technology used eliminates gate monitoring and manning by human beings. The gates have to perform gyrations by open, auto reverse, stop, fully close and fully stop. It provides convenient access and intelligent features that makes it distinct from all other gates which bring it so close to a security device. Those gates come with different type of mechanism such as sliding, swing, folding, and barrier gate. Those mechanisms have their own working principle and feature but, automatic gate design seem limited at the local market. Most of the product is imported from outsider supplier. The price of the product also seems expensive. Cost study and new mechanism design, can be marketable toward wider customer at lower cost and new innovation of auto gate mechanism can enhance local design capability.

1.2 PROBLEM STATEMENT

Nowadays, the automatic gate mechanisms have been improved and developed with different kind of features. These features have increased the product cost and this cost does not include the installation cost. Many people especially with low income could not afford to purchase the gate mechanism. The gate mechanism needs a very skillful or trained person to install the mechanism to the gate. Some gate mechanisms also need to be attached with rail on the ground, this seems to be inconvenience and need a lot of work force to install the track. Development of automatic gate mechanism should help in term of cost reduction and ease of installation.

1.3 OBJECTIVES

Objectives for this project refer to the mission, purpose, or standard that can be reasonably achieved within the expected timeframe and with the available resources. The objective of this project is to design an automatic gate mechanism for residential home with double gate leaf with weight of 100 kg for each side of the gate. Cost reduction and ease of installation are also considered for this mechanism.

1.4 PROJECT SCOPES

The scopes for this project are to study about several types of automatic gate mechanism and to understand the working principle in term of movement. Those mechanisms include swing gate type mechanism, sliding gate type mechanism and folding swing type mechanism. Design and sketches in rough view, are compared between those designs and the best is chosen. The design should consider about the portability and cost. Based on the design, a prototype is constructed for mechanism rough view. Finite element analysis using ALGOR software is to determine the critical failure part of the mechanism. This is to ensure the mechanism can withstand high torque.

1.5 EXPECTED OUTCOME

Designed and fabrication of automatic gate mechanism should be reliable, easy to maintain, safe to operate and less in cost compared to other types of automatic gates. The automatic gate mechanism should also be able to function properly when installed on normal gate with weight of 100kg.

1.6 LIMITATION

The limitation for this project is hard to collect data about the automatic gate mechanism. Automatic gate are normally for commercial purpose and it is impossible for the product company to expose their own design and working principle of the product.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

This chapter will provide the detail description literature review done according to title of design of automatic gate mechanism. Automatic gate mechanisms act as the actuator part for the gate system and provide motion in order to open or closed the gate. The on-board circuit board receives a signal from an access control (like a keypad or a control) and tells the automatic gate opener to open or close the gate, reverse it if an obstacle is in its path. The automatic gate mechanism available type includes arm type for swing gate, rack and pinion sliding type, and cubic underground automatic systems for swing gates.

2.2 GATE DESIGN CONSIDERATION

A gate is a potential traffic hazard, so it is important that you locate the gate far enough away from the road to eliminate the potential of traffic getting backed up. This distance is affected by the size of the gate, how often it is used, and how fast the gate operates. The operator you choose to install on your gate must be designed for the type and size of your gate and for the frequency with which you use the operator. Gate must be properly installed and must work freely in both directions before the automatic operator is installed. An automatic operator should be installed on the inside of the property/fence line. Do not install the operator on the public side of the property/fence line. Pedestrians should not use a vehicular gate system. Prevent such inappropriate use by installing separate gates for pedestrians. Exposed, reachable pinch points on a gate are potentially hazardous and must be eliminated or guarded. Outward swinging gates with automatic operators should not open into a public area. The operating controls for an automatic gate must be secured to prevent the unauthorized use of those controls. The controls for an automatic gate should be located far enough from the gate so that a user cannot accidentally touch the gate when operating the controls. An automatic gate operator should not be installed on a gate if people can reach or extend their arms or legs through the gate. Such gates should be guarded or screened to prevent such access (Automated Gates Ltd. 2009).

2.3 AUTOMATIC GATE SYSTEM NORMAL DESIGN

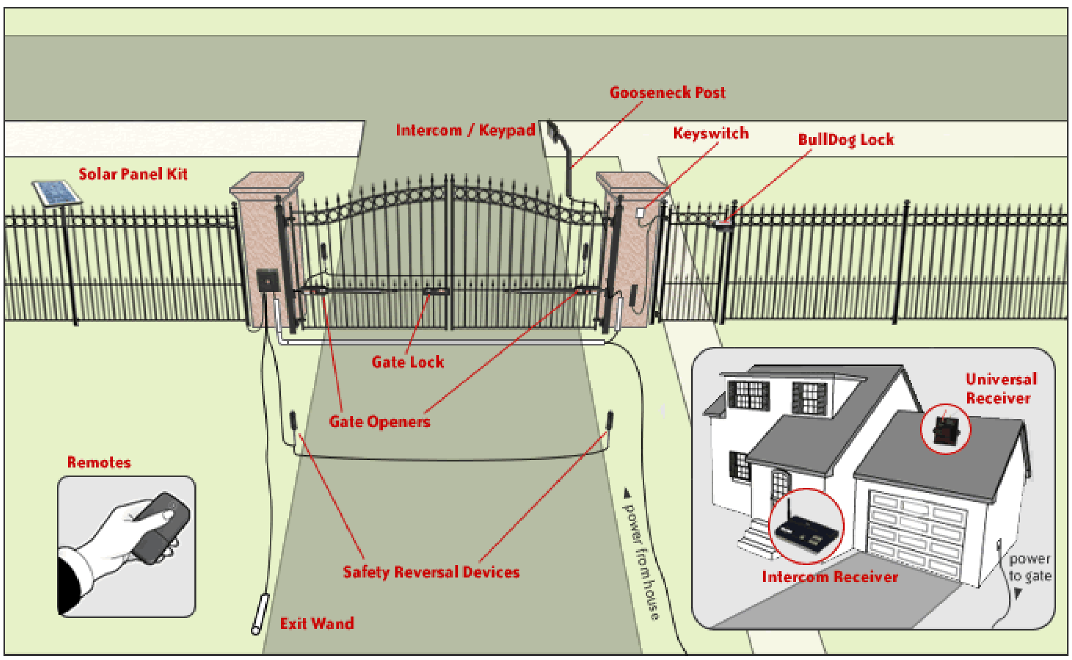


Figure 2.1: Automatic gate component

Source: Prestige Door and Gates Ltd 2008

Hand held remote's, are radio transmitters. Use to control gate opening/closing for a certain range. Gate openers refer to the gate mechanism or actuators that use to provide motion for the gate opening and closing activities. Next automatic gate control to seriously consider is the 'keypad'. It will allow visitors without remotes to enter the property with a numeric code. The codes are easily programmed and may be changed frequently. These units are designed for all weather conditions (Prestige Door and Gates Ltd .2008).

A visitor leaving the property needs to open the gate. One way is to have a keypad on the inside, or a simple push button that will open the gate. More convenient is the 'exit sensor'. An exit sensor is buried alongside the driveway and detects moving metal. A vehicle, motorcycle or bicycle passing the sensor will open the automatic gate. It will also sense a lawn mower and some children's toys and may not be the best choice for every situation. Receive signal form the remote and respond to it. The signal then will be processing for gate opening (Prestige Door and Gates Ltd .2008).

2.4 TYPES OF AUTOMATIC GATE MECHANISM

There are a lot of mechanisms in market. The mechanisms of automatic gate also refer to the actuator. Mechanism is the component for automatic gate that provide movement.

2.4.1 Linear Drive with Screw Drive and Piston Rod

This is the arm type automatic gate mechanism for swing gate. This mechanism using screw driven piston type where actuated by hydraulic cylinder. This type of mechanism produce Accurate path and position control, High force output use for heavy duty automatic gate with extra weight, provided with self locking system, Ideal for level regulation, lifting and other applications with intermittent operations (Parker Hannifin Corporation. 2008).

Installation instructions, Use the threaded holes in the free end cap and a mid-section support close to the motor end for mounting the linear actuator. The linear actuator can be fitted in any position. To prevent contamination such as fluid ingress, the actuator should be fitted with its sealing band facing downwards (Parker Hannifin Corporation. 2008).

Maintenance, all moving parts are long-term lubricated for a normal operational environment. This arm type automatic mechanism product recommends a check and lubrication of the linear drive, and if necessary a change of wear parts, after an operation time of 12 months or 300 km travel of distance. Please refer to the operating instructions supplied with the drive (Parker Hannifin Corporation. 2008).

Figure 2.2: Screw drive and piston rod for swing gate

Source: Parker Hannifin Corporation 2008

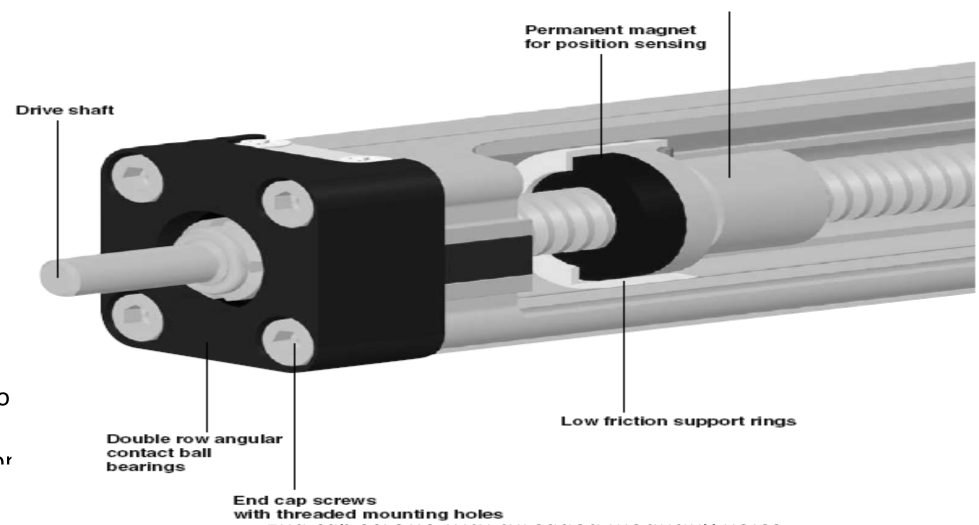


Figure 2.3: Screw drive and piston rod mechanism for the low part

Source: Parker Hannifin Corporation 2008

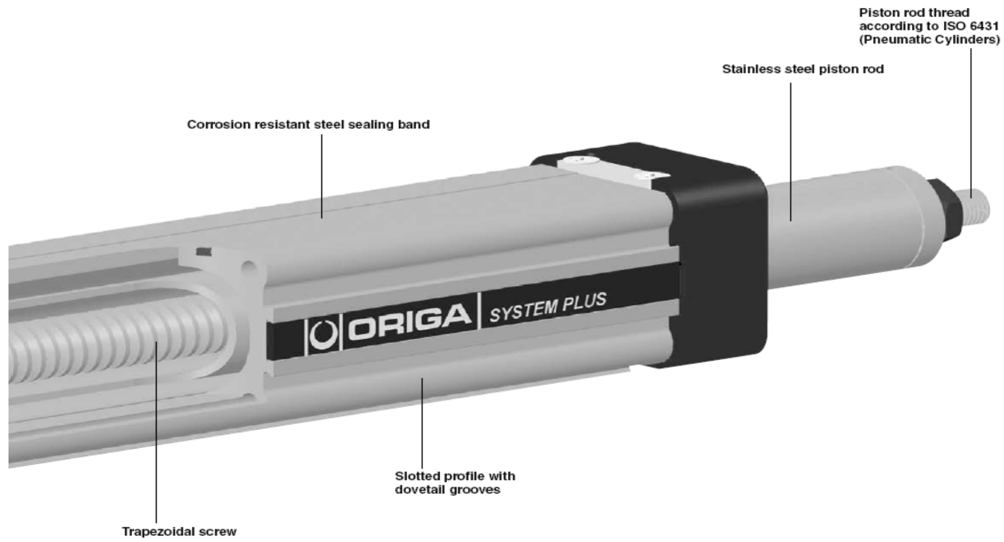


Figure 2.4: Screw drive and piston rod mechanism for the upper part

Source: Parker Hannifin Corporation 2008

2.4.2 Cubic Underground Automatic Systems for Swing Gates

Valuable gates or those of a special design require "invisible" automatic systems in order to avoid altering their aesthetic appearance. The standard underground Cubic range, designed to be positioned so that it is “aligned with the hinge of the leaf”, amply satisfies such requirements because it is easy to insert beneath existing paving, especially when this is of high value. However, sometimes stone or granite paving may heavily condition the positioning of a “hinge aligned” buried automatic system. The alternative Cubic Flexi range has been created to solve this problem, and is extremely flexible in its installation. This type of mechanism was drive by motor to actuate the arm (MAG Malaysia Automatic Gate. 2005).

Features, Good quality with MAG system major components are made of top grade tough, durable and rust resistance material, Fast and silent where the actuator is capable for 90˚ opening within only 8 to 10 seconds and is exceptionally quiet in operation, Last longer. Gear box is submerged in oil bath for more protection. This design significantly improves the actuator's mechanical life span. The gate can only be opened with the remote control transmitter or special key (during power failure. This mechanism comes with Intelligent of Smart microprocessor is used in the control panel, can be upgraded to support standby battery in case of power failure, Gate leaf operation with cushioning and gate will slow down towards the end of opening or closing swing. This is to avoid the gate banging loudly into the stopper and pillar light wall at high speed (MAG Malaysia Automatic Gate. 2005).

Technical specification for product SW200P Automatic Swing Gate System, the technical specification describe the properties and functional of every component that consist in this cubic underground automatic gate system for the product SW200p automatic gate system (MAG Malaysia Automatic Gate. 2005).

Actuator with a rotary drive unit complete with DC motor, speed reducer and electro-mechanical clutch system housed in a waterproof rust resistance cast alloy casing. Actuator drives the gate leaf to move (MAG Malaysia Automatic Gate. 2005).

Remote control & transmitter, Consist of a receiver and transmitters, Allows gate operation to be controlled remotely within the specified range. Modulated RF transmission, Transmission coding (6561 combinations) is set by 8x3 DIP switches in the receiver and transmitter, easily accessible and changed by owner. Coding on both sides must be the same for it to work. Receiver is installed inside the ABS casing together with control panel. Transmitter uses a single 12V small size alkaline battery; normal usage should last 6 months (MAG Malaysia Automatic Gate. 2005).

Control panel, the control panel Consists of an advanced microprocessor based PCB, and transformer housed in weather resistant casing. Use to Controls operation of the whole swing auto gate system (MAG Malaysia Automatic Gate. 2005).

Actuator arm and bracket consist of two arms and two brackets. These are the supporting component that connects the gate to the actuator output shaft. Both are made from good material, Nickel-Chrome plated mild steel (MAG Malaysia Automatic Gate. 2005).

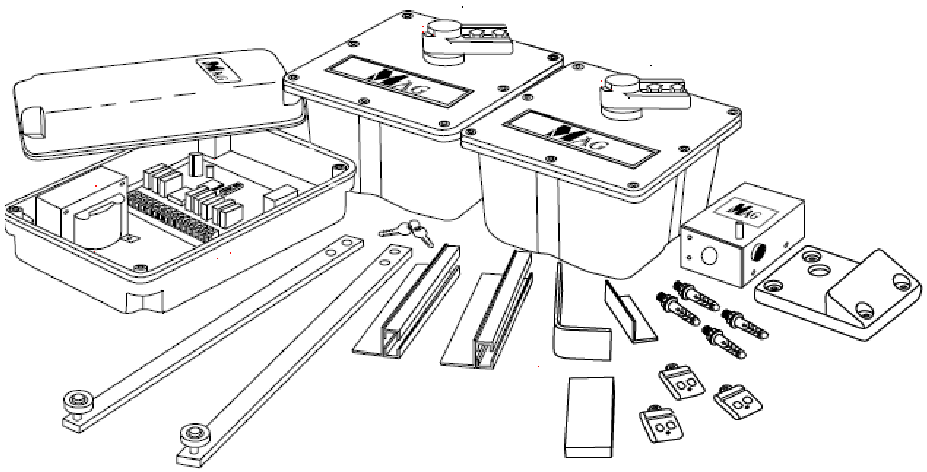


Figure 2.5: Cubic underground automatic system

Source: MAG Malaysia Automatic Gate 2005

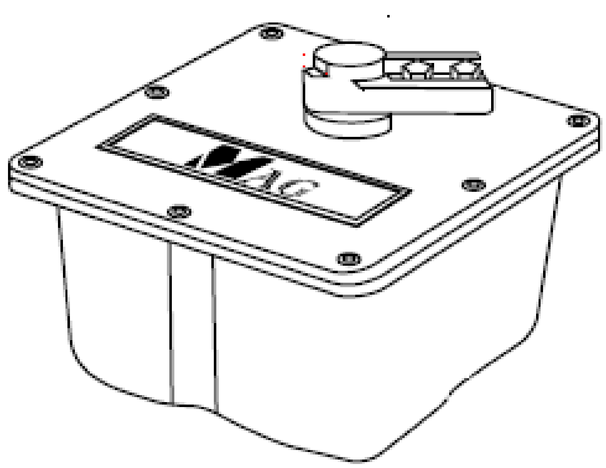


Figure 2.6: Actuator

Source: MAG Malaysia Automatic Gate 2005

Table 2.2: Actuator technical specification

Motor voltage rating	12-24 V DC
Power consumption	25 watt per actuator
Trust force	0 to 60 kg
Max Opening angle	360 o
Max leaf weight allowable	200kg
Motor rated load speed	4700rpm
Actuator output speed	2.2rpm
Traveling speed	8 – 10 sec for 90o
Dimension	240 mm x 240 mm x 140 mm

2.4.3 Wiring and Installation Diagram for Cubic Underground System

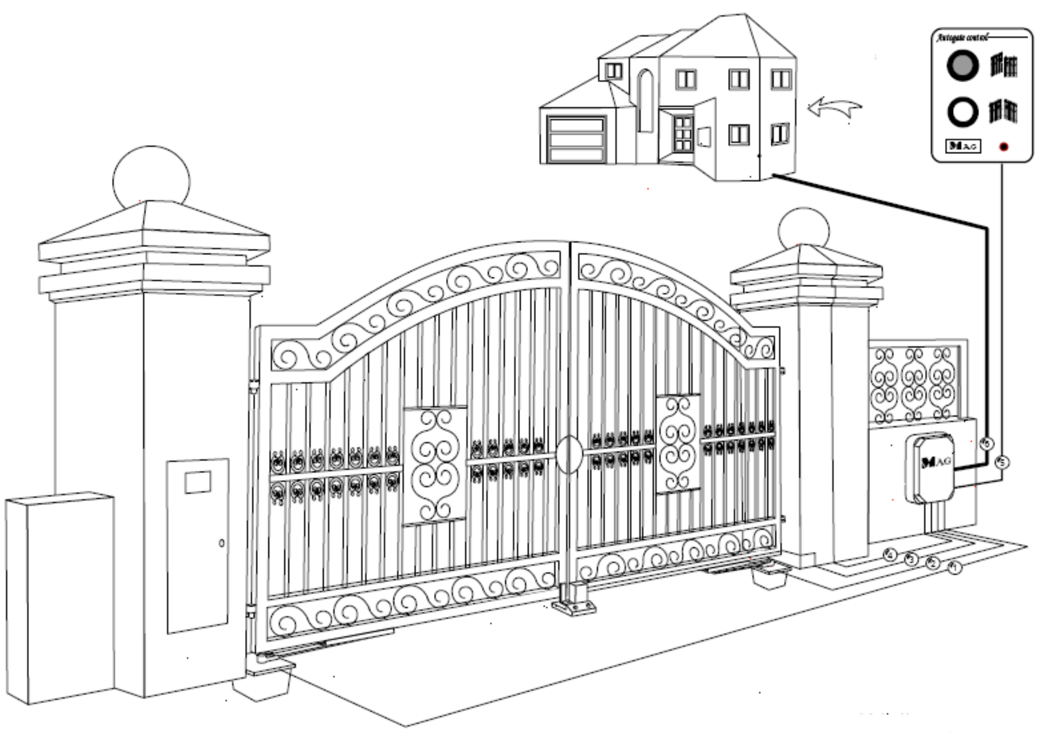


Figure 2.7: Wiring diagram for automatic gate system

Source: MAG Malaysia Automatic Gate 2005

This wiring diagram only shows the correct point-to-point electrical connection. It does not represent the underground wiring path in actual installation. Underground wirings are protected by PVC conduits (MAG Malaysia Automatic Gate. 2005).

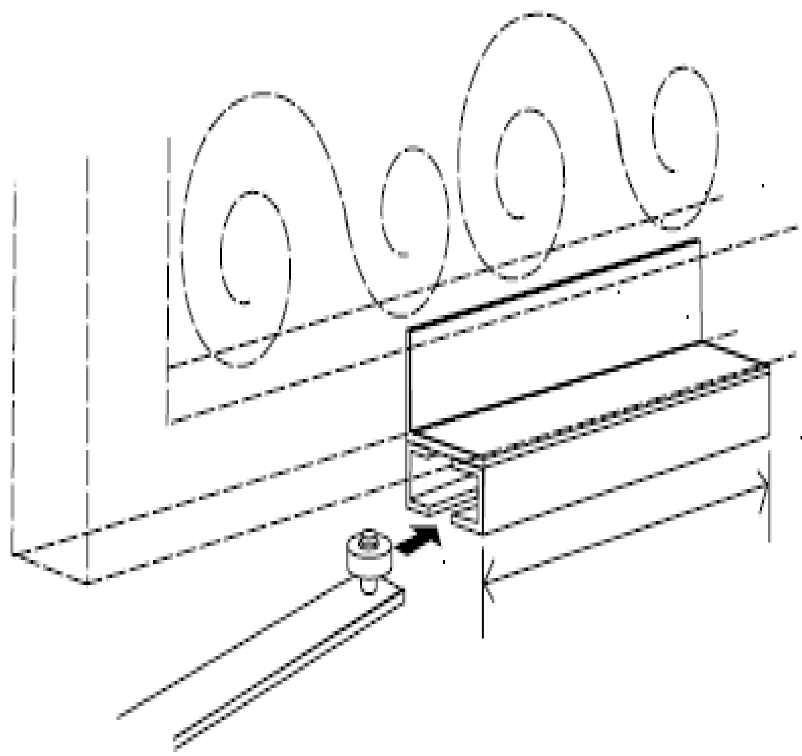


Figure 2.8: Installation diagram for arm and bracket

Source: MAG Malaysia Automatic Gate 2005

The purpose of the bracket is to contain the arm's bearing roller movement as the gate open or close. The bracket mounted should be able to completely include the maximum travelling path of the roller bearing as the gate swing from fully close to fully open position (MAG Malaysia Automatic Gate. 2005).

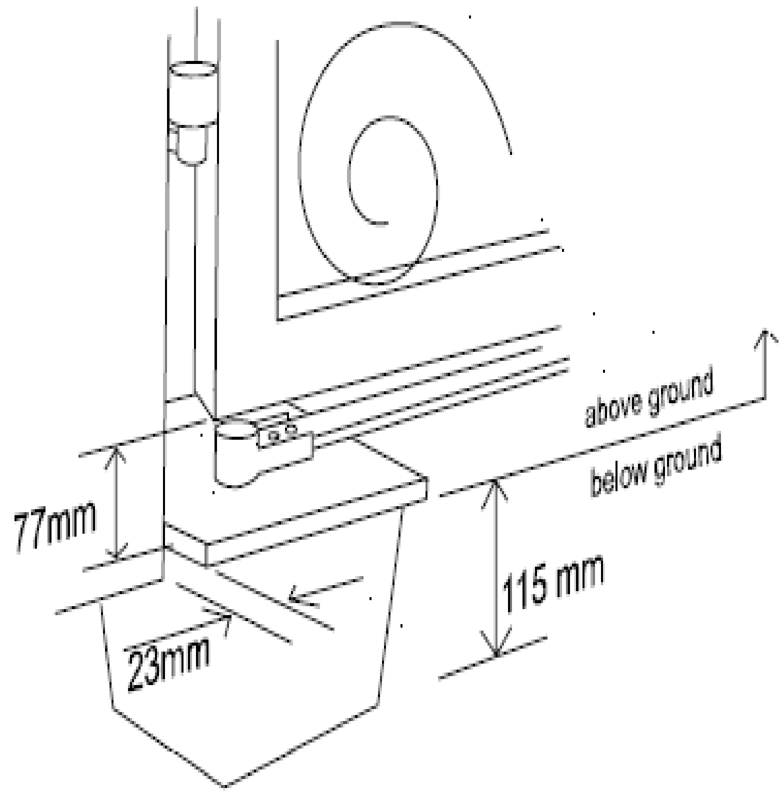


Figure 2.9: Actuator position under the ground

Source: MAG Malaysia Automatic Gate 2005

The final location of the bracket varies depending on the installation site and it has to be measured again for every new installation. Failure to mount the bracket accurately will cause the bearing roller to fall out when gate is fully open or fully close (MAG Malaysia Automatic Gate. 2005).

2.4.4 Sliding Gate Automatic System

The DC Series sliding gate operators are used to drive sliding gates for residential and industrial uses. The operators can be fitted with gear wheel or chain wheel according to your needs, and featured with powerful starting strength, capable of overwork at short time. The operator can be powered be storage battery in case of power failure. An emergency release key allows operate gate manually. Nice appearance, reliable quality and easy installation. Working principle and main structure the device is designed for chain link and includes: a DC motor, worm wheel, worm, tool body, control box, motor base, storage battery and limit switch. The motor rotates the worm and worm wheel, which drive the chain wheel and chain, thus the gate was driven to move (V2 Automatismos. 2007).

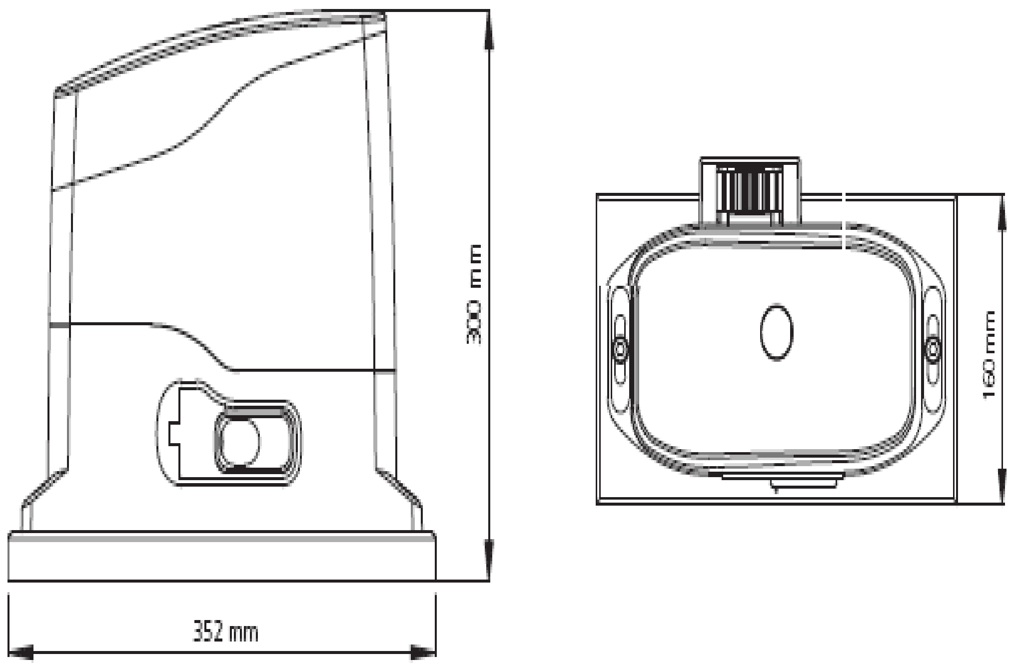


Figure 2.10: Sliding gate automatic mechanism

Source: V2 Automatismos 2007

2.4.5 Guide to Residential Automation

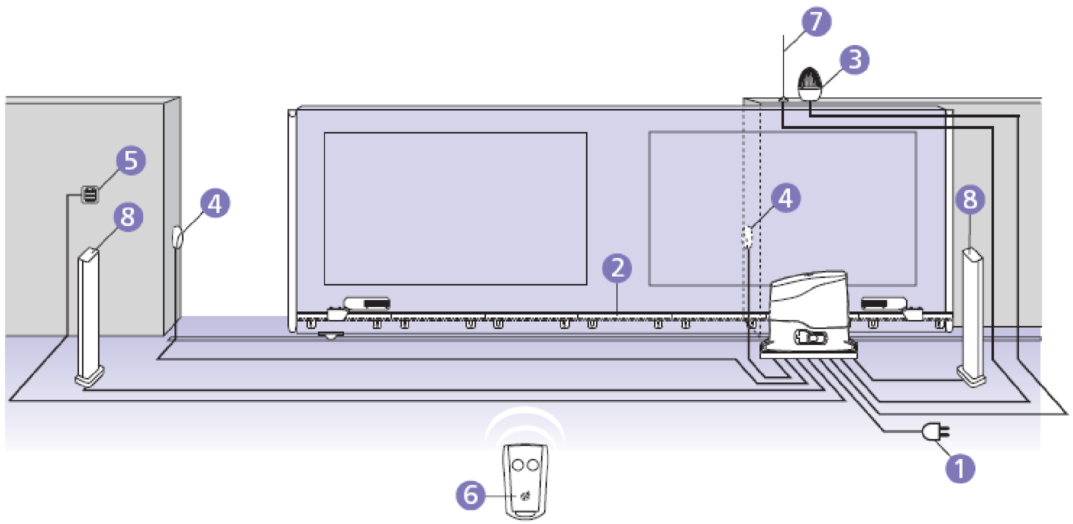


Figure 2.11: Sliding gate automatic system

Source: V2 Automatismos 2007

The transmitter provide signal for the wide range and easy access control. The actuator then will react to the signal from transmitter and the motor will rotate along the rack. The rack is installed along the sliding gate. The gate will stop when touched the limit switch (V2 Automatismos. 2007).

2.5 TORQUE FOR GEARING SYSTEM

Torque is a measurement of twisting ability about an axis. Basically it is a force vector (F) applied at a distance (R) which is measured in ounce-inches, pound-feet, dyne centimeters, Newton-meters or gram (force)-centimeters also can be obtain using Eq. (2.1)

$$\text{Torque (T)} = \text{force (F)} \times \text{radius} \qquad (2.1)$$

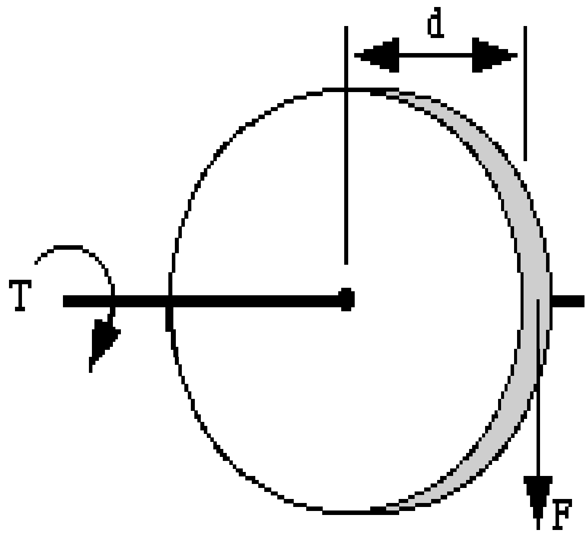


Figure 2.12: In two dimensions torque

Source: Richard Valentine 2006

If the pulley of 1m radius is in use and a cord fixed to it with a 1 Newton weight hanging down on the end, it will produce 1 Newton meter of torque on the pulley shaft (Richard Valentine. 2006).

Torque (T) = force (F) x radius (R)
Torque (T) = 1Newton x 1 meter
Torque (T) = 1Newton- meter

2.6 FRICTION

Frictional resistance to the relative motion of two solid objects is usually proportional to the force which presses the surfaces together as well as the roughness of the surfaces. Since it is the force perpendicular or "normal" to the surfaces which affects the frictional resistance, this force is typically called the "normal force" and designated by N. The frictional resistance force designated by μ . The frictional force is also presumed to be proportional to the coefficient of friction. However, the amount of force required to move an object starting from rest is usually greater than the force required to keep it moving at constant velocity once it is started. Therefore two coefficients of friction are sometimes quoted for a given pair of surfaces, a coefficient of static friction and a coefficient of kinetic friction. The force expression above can be called the standard model of surface friction and is dependent upon several assumptions about friction (R.C.Hibbeler. 2007).

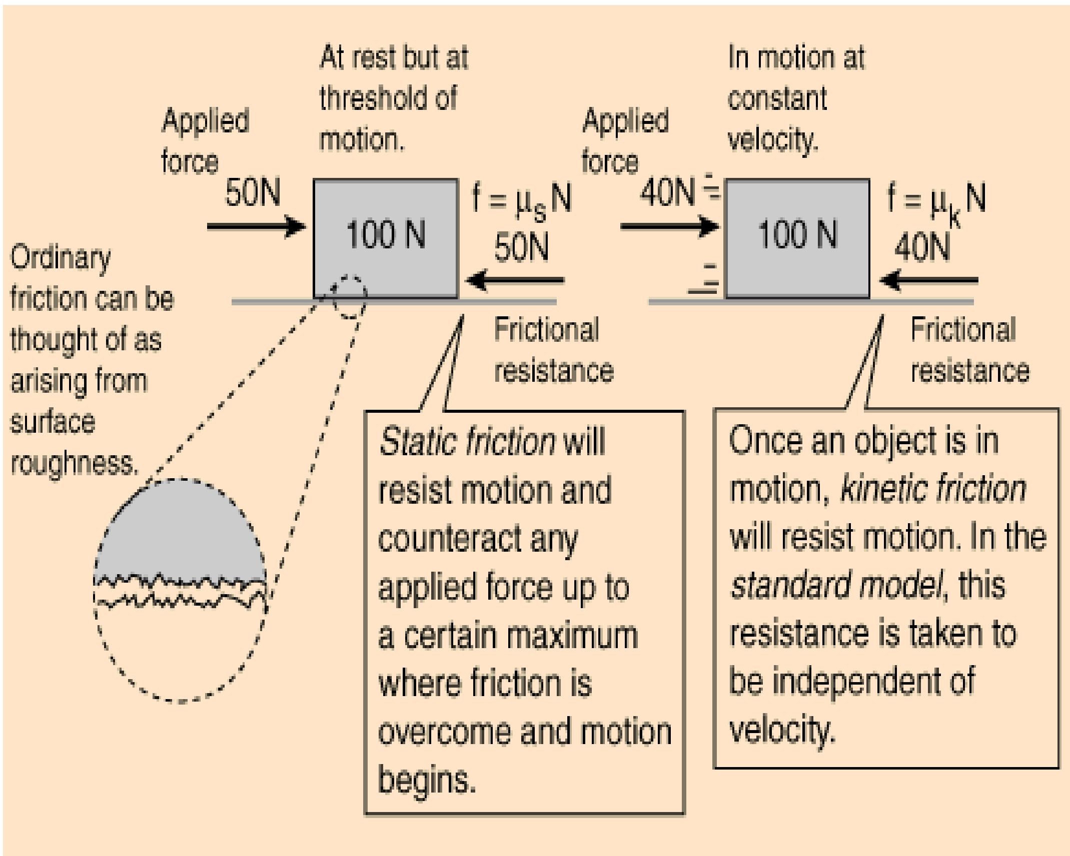


Figure 2.13: Static and kinetic frictional condition

Source: R.C.Hibbeler 2007

2.7 MICROPROCESSOR BASED AUTOMATIC GATE

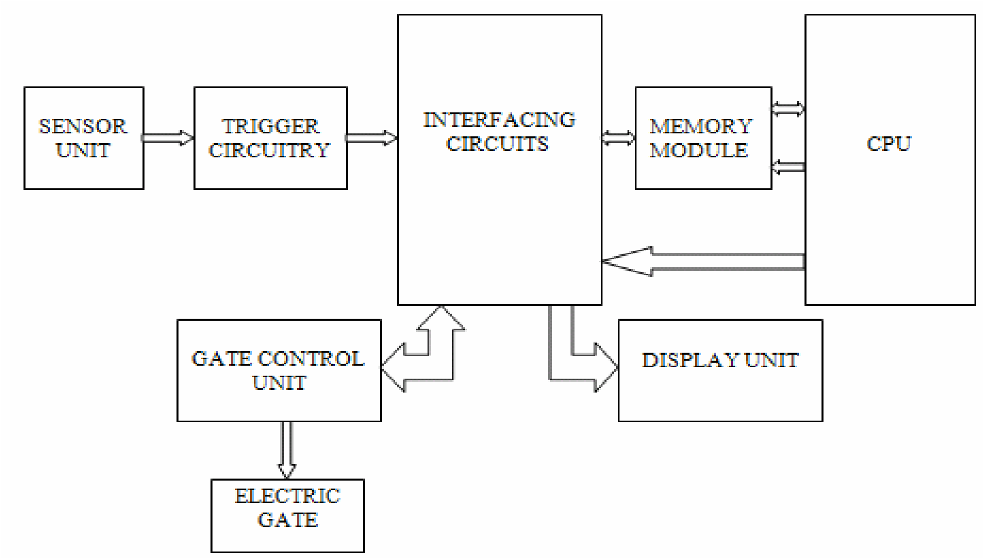


Figure 2.14: Block diagram of the auto gate system

Source: Baruwa 2006

Presented here is the design and development of a microprocessor based automatic gate. As a monitoring and control system, the microprocessor was used to read in data values from the input device and interact with the outside world. The system senses, opens and closes the gate, counts, registers, and displays the number of vehicles crossing the gate (both entrance and exit) and triggers an alarm once the space limit is reached. Once triggered, the gate remains inaccessible until another vehicle leaves the park. The automatic gate system comprises a sensor unit, a trigger circuitry, CPU module, memory module, display unit, gate control unit and the power supply unit as shown in the block diagram figure (2.14). The sensor provides an input signal to the system. It is an optical sensor which, when light rays are focused on it, has a low resistance and hence, causes the input to the trigger circuitry to be held “Low”. But, when a vehicle interrupts the beam, the resistance increases and reaches its dark resistance, thus, the input to the trigger circuitry is held “High”. The trigger circuitry serves as an Analog–to–Digital converter (ADC), which produces a high signal when the beam is interrupted. The trigger circuitry sends a signal to the interface unit, which is made up of Programmable Input/output Controller (PIO). The software causes the microprocessor to be check the input port of the interface unit for the sensor status information (the outputs of the trigger circuitry). A high value causes the microprocessor to send a signal to the output port of the interface unit in order to activate the DC motor to control the gate (open and close). It equally sends a signal to the display unit for counting the number of vehicles. A low value will never activate the gate. False triggering is taken care of by the trigger circuitry. The power supply unit supplies the required DC voltage needed by the entire microcomputer system (Baruwa. 2006).

Direct current (DC) motors have been widely used in many industrial applications such as electric vehicles, steel rolling mills, electric cranes, and robotic manipulators due to precise, wide, simple, and continuous control characteristics. Traditionally rheostat armature control method was widely used for the speed control of low power dc motors. However the controllability, cheapness, higher efficiency, and higher current carrying capabilities of static power converters brought a major change in the performance of electrical drives. The desired torque-speed characteristics could be achieved by the use of conventional proportional integral- derivative (PID) controllers. As PID controllers require exact mathematical modeling, the performance of the system is questionable if there is parameter variation. In recent years neural network controllers (NNC) were effectively introduced to improve the performance of nonlinear systems. The application of NNC is very promising in system identification and control due to learning ability, massive parallelism, fast adaptation, inherent approximation capability, and high degree of tolerance. A constant-power field weakening controller based on load-adaptive multi-input multi-output linearization technique has been proposed to effectively operate a separately excited dc motor in the high-speed regimes. (George. 2008)

CHAPTER 3

METHODOLOGY

3.1 INTRODUCTION

This chapter will provide the detail explanation on the methodology that carries out for this project "Design of automatic gate mechanism" from the beginning till the end. Methodology can properly refer to the theoretical analysis of the methods appropriate to a field of study or to the body of methods and principles particular to a branch of knowledge. The methodology act as the guidance or step that needs to be follow and this will ensure the project done according to the planning. Methodology as an algorithm that finds a solution in the given environment of the multi-layered finite space consisting of literature review, Identifying the suitable mechanism and design, making a Solidwork model of the design, Identifying different parts and making the part list, Fabricating the mechanism, test run and documentation.

3.2 FLOW CHART

In the beginning of this project is receiving title and briefing from supervisor. In order to have better understanding about this project title “Design of Automatic Gate Mechanism, several research on internet and market available product was perform. Designing consideration of automatic gate mechanism include Environmental conditions, location, ground water, elevation, flooding, Reliability, maintenance and cost. Identify the basic requirement or specific specifications of the automatic gate mechanism that can suit the customer for this design.

Define the function of device through explore and understanding the working principles of automatic gate mechanism. The automatic gate could work under static and dynamic condition and required different operating condition.

State the design requirement that possible for this automatic gate mechanism system. The basic requirement should be able to suit the customer need and reliable.

Propose several alternative design concepts, also called the invention of the concepts or concept design. Various schemes must be proposed in order come out with several option design selection.

Evaluate each proposed alternative design. Analysis perform to assess whether the system performance is satisfactory or better, and, if satisfactory, just how well it will perform. Compare and contrast between the advantages and disadvantages. System scheme that do not survive analysis are revised or discarded, those with potential to optimize and determine the best performance of which the scheme is capable. Competing schemes are compared so that the path leading the most competitive product can be chosen.

Complete the design with full dimension and specification so can proceed to the next step for fabrication. The documentation is the final stage by preparing a full report where consist of introduction, literature review, methodology, result and discussion and conclusion.

3.3 SEVERAL EARLY ALTERNATIVE DESIGNS

The design criteria are based on the cost, safety, ease of maintenance, reliability, installation and ease of use. After go through the brainstorming session, few design was constructed with simple sketching or act as the early conceptual design.

3.3.1 Design 1

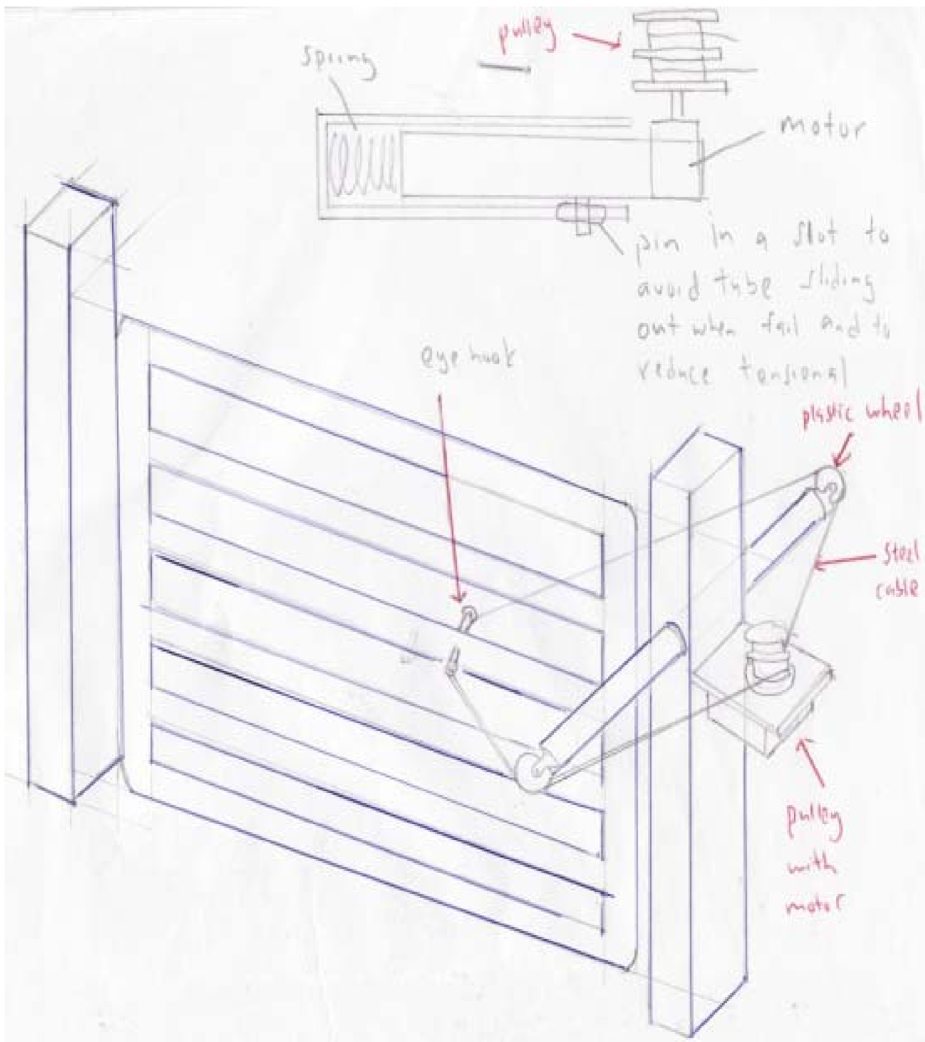


Figure 3.1: Design 1

Design 1 using two way winch system. The motorized pulley come together with the steel cable that attached to the gate, when the motor rotating, the pulley will react by pulling the steel cable and shortening it to provide movement for the gate. This mechanism is simple, with less related component and lower building cost.

3.3.2 Design 2

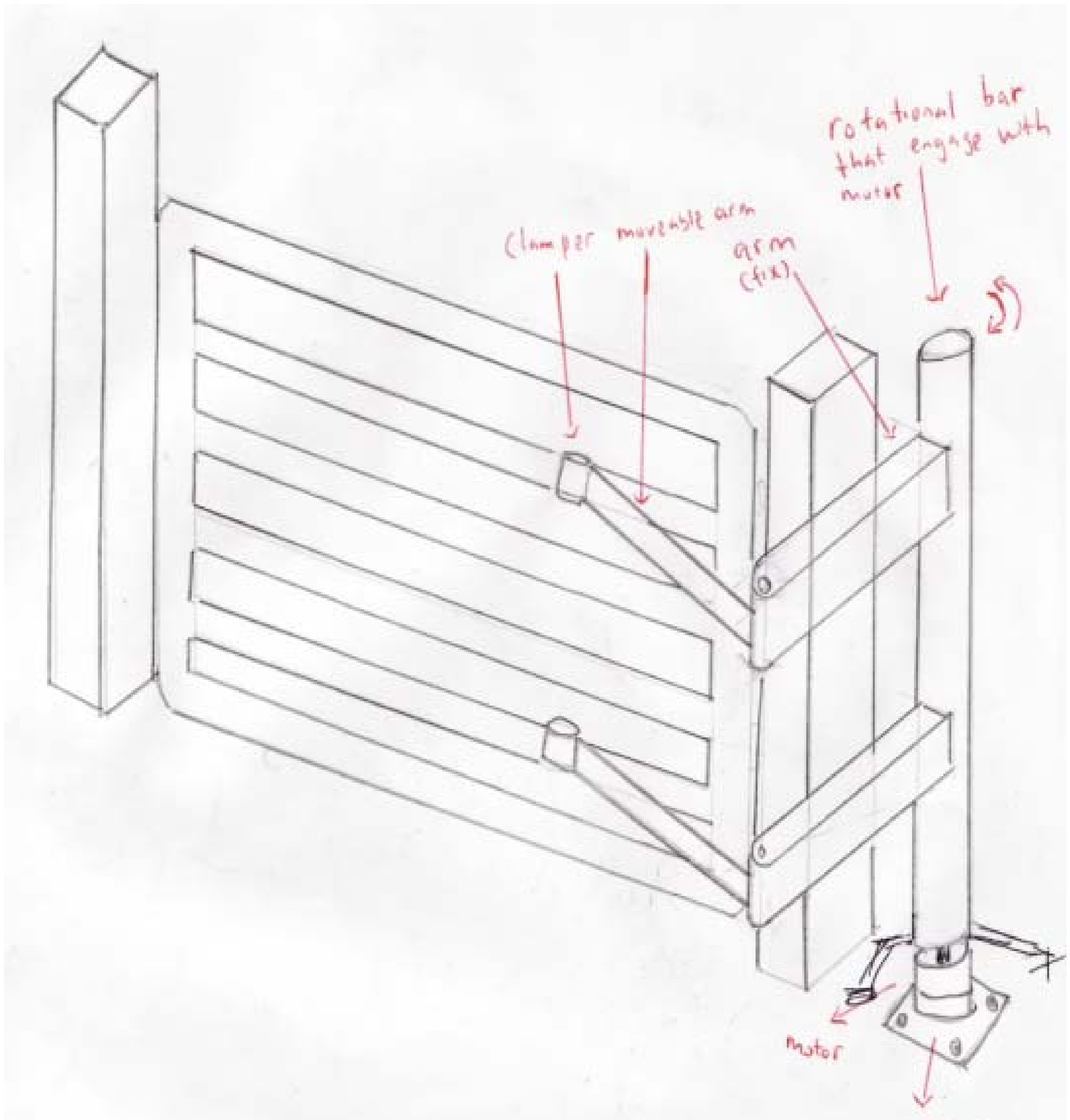


Figure 3.2: Design 2

This is arm type system. This mechanism is simple and easily to operate. The mechanism powered by a motorized pillar. The pulley was attached with fix arm. When the pillar rotating, the fix arm will provide motion to the movable arm and move the gate. This mechanism is easy to operate and not complicated.

3.3.3 Design 3

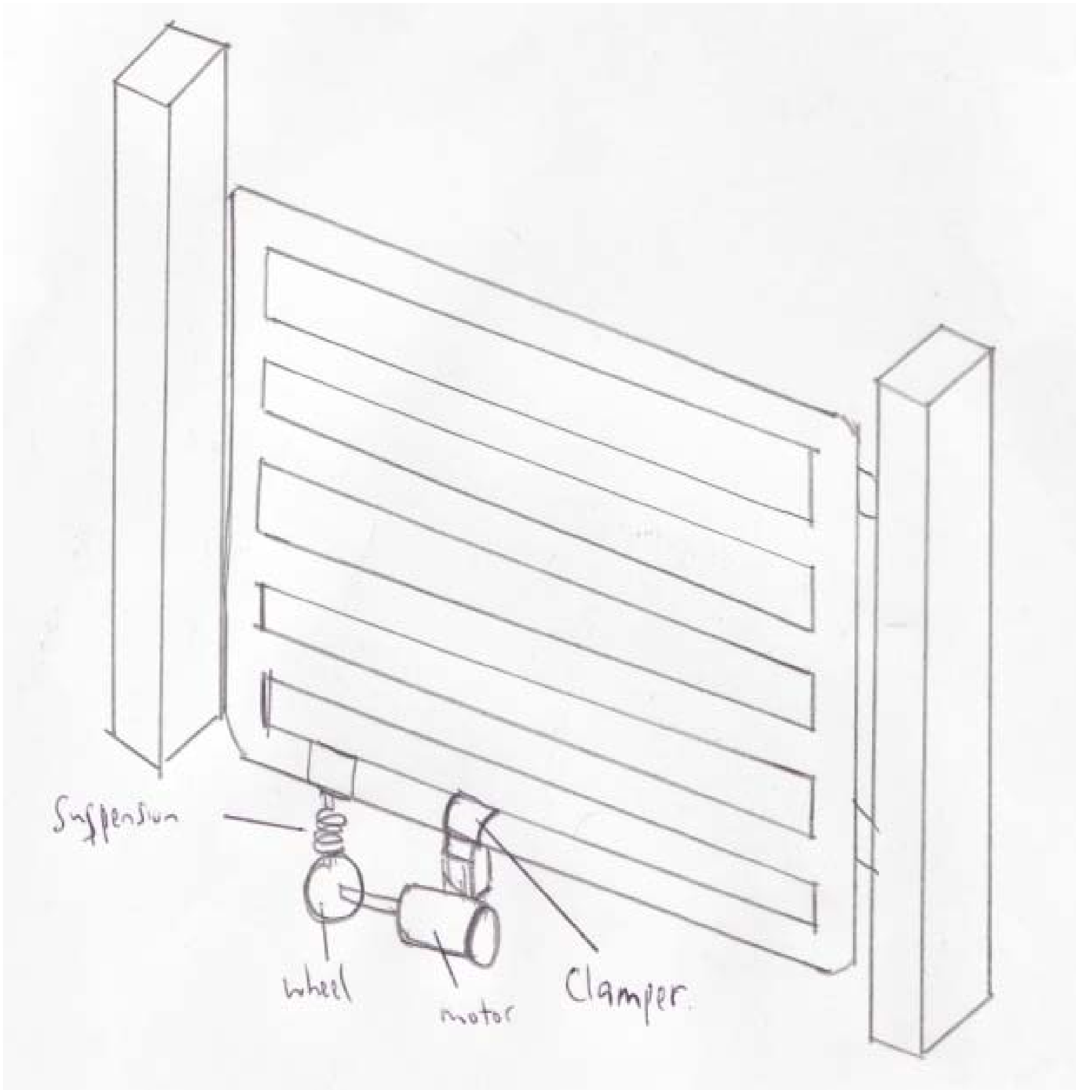


Figure 3.3: Design 3

This is motorized wheel system. The wheel movement was drive by the motor rotation. Motor rotation, clockwise or counter clockwise will decide the gate movement forward or backward. This system was equipped with suspension to ensure the wheel always reach the ground. This system is simple and can be install easily.

3.4 FINAL CONCEPTUAL DESIGNS

From the early alternative design, the option is optimized and selecting two final conceptual designs. Those two design is modeling by using Solidwork. Solidwork is the global standard in 3D mechanical design software. It helps organization to reduce time to market, design better quality product faster, maintain a competitive advantage, and increase the sales. Soliwork delivers powerful 3D design capabilities, unmatched ease of use, and an affordable cost.

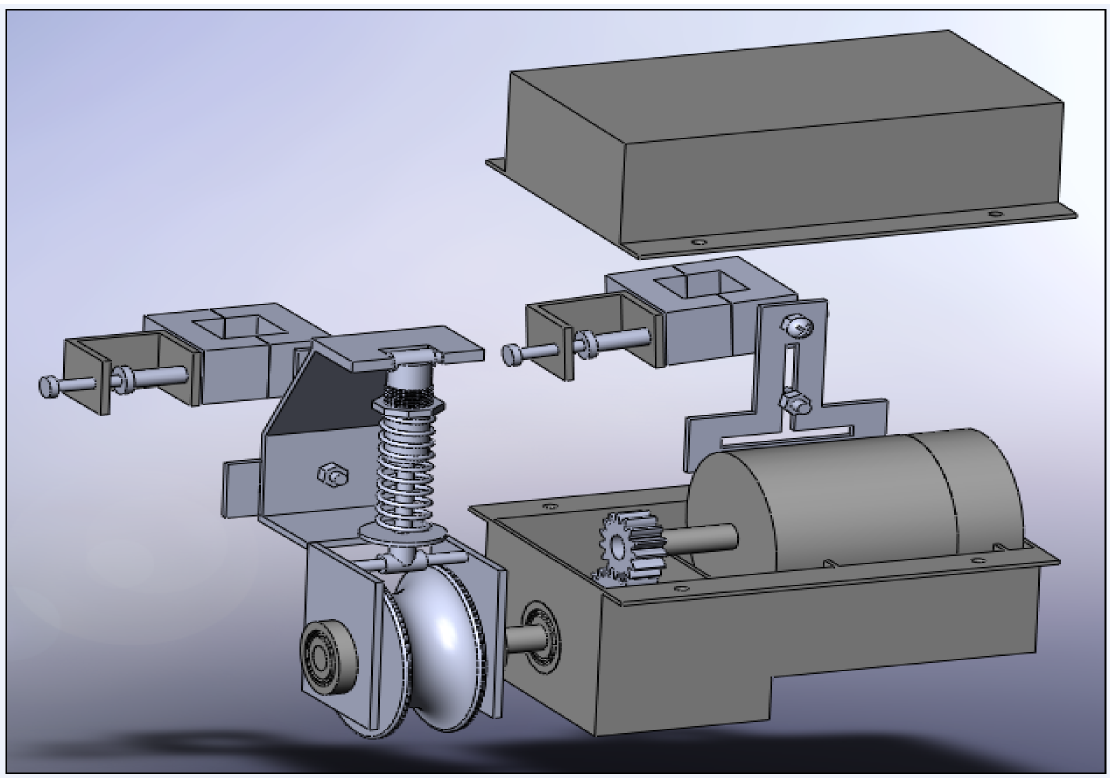


Figure 3.4: Final conceptual design 1

The final conceptual design 1 seems to be complicated and consist of a lot component. The component may easily fail due to the high loaded torque required in the system for this automatic gate mechanism

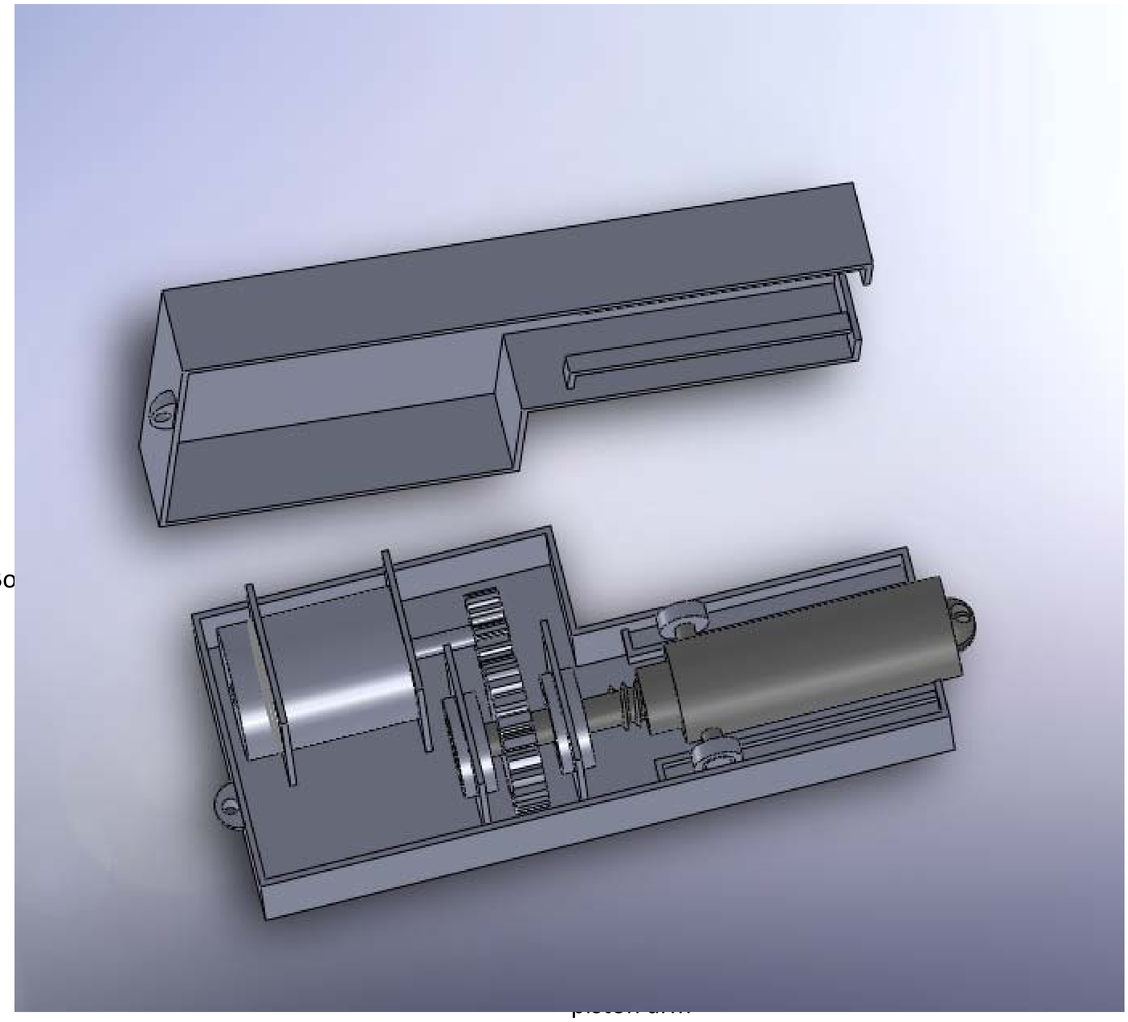
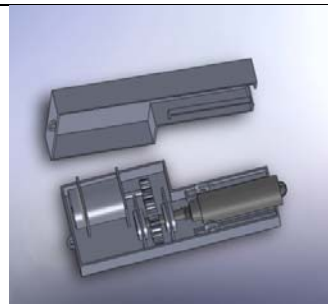
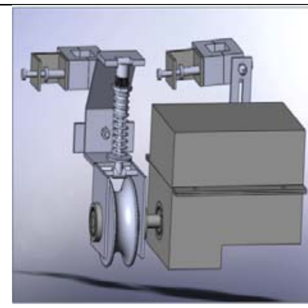


Figure 3.5: Final conceptual design 2

The final conceptual design 2 is actuated by motor. The mechanism form rotary is converting to linear movement. This mechanism is positioned along the gate and pillar. This mechanism is simple, only few components are required for this design.

3.5 EVALUATE EACH PROPOSED ALTERNATIVE

Table 3.1: Compare and contrast conceptual design and market product

Final Design Comparison			
Geometry	Design Compacted	Design not compacted	Design compacted
Simplicity	Design simple (contain few part)	Design complicated (a lot of part)	Design complicated
Usage	Screw drive piston arm convert rotary into linear mechanical movement	Roller using motorized system.	Using screw drive piston
Cost	Low cost	High cost	Very high

Evaluation between the conceptual design and market product is done by comparing in term of geometry, simplicity, usage, and costing. This step is performed in order to select the final stage of complete design for this automatic gate mechanism. Here, final conceptual design 2 seems to be more reliable. The design is simple and can functional as desired.

3.6 COMPLETE DETAILED DESIGN

The complete detailed design is attached with the full dimensional of the automatic gate mechanism. The detailed design is required for fabrication purposed.

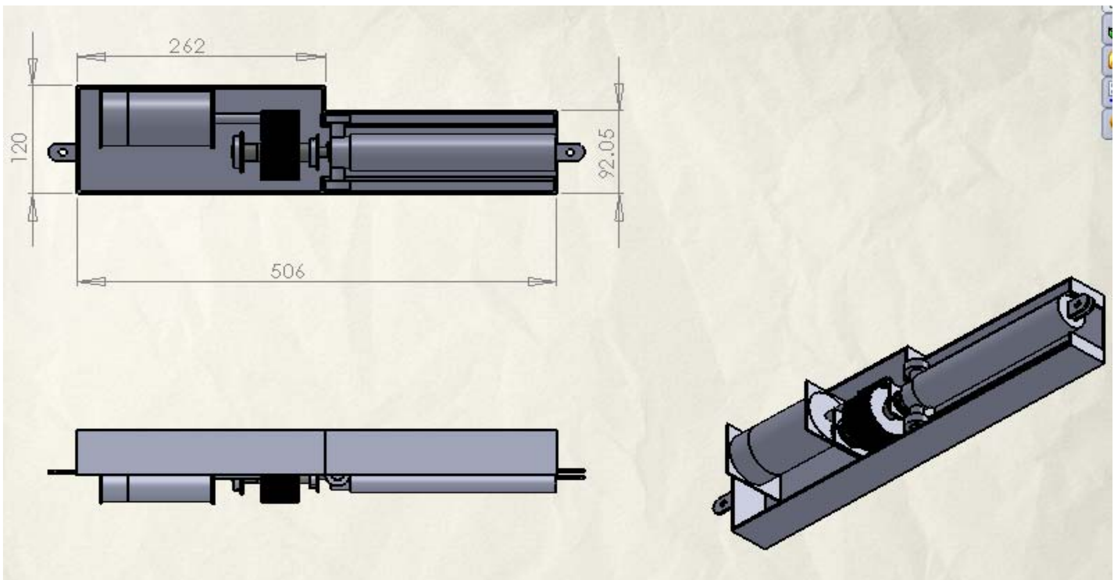


Figure 3.7: third angle viewing of the final complete design

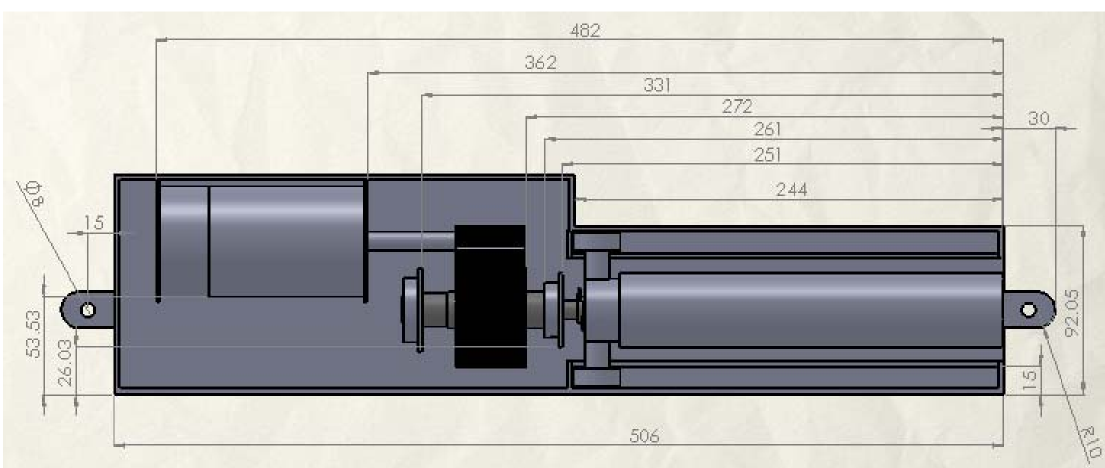


Figure 3.8: Top view with dimension of final complete design

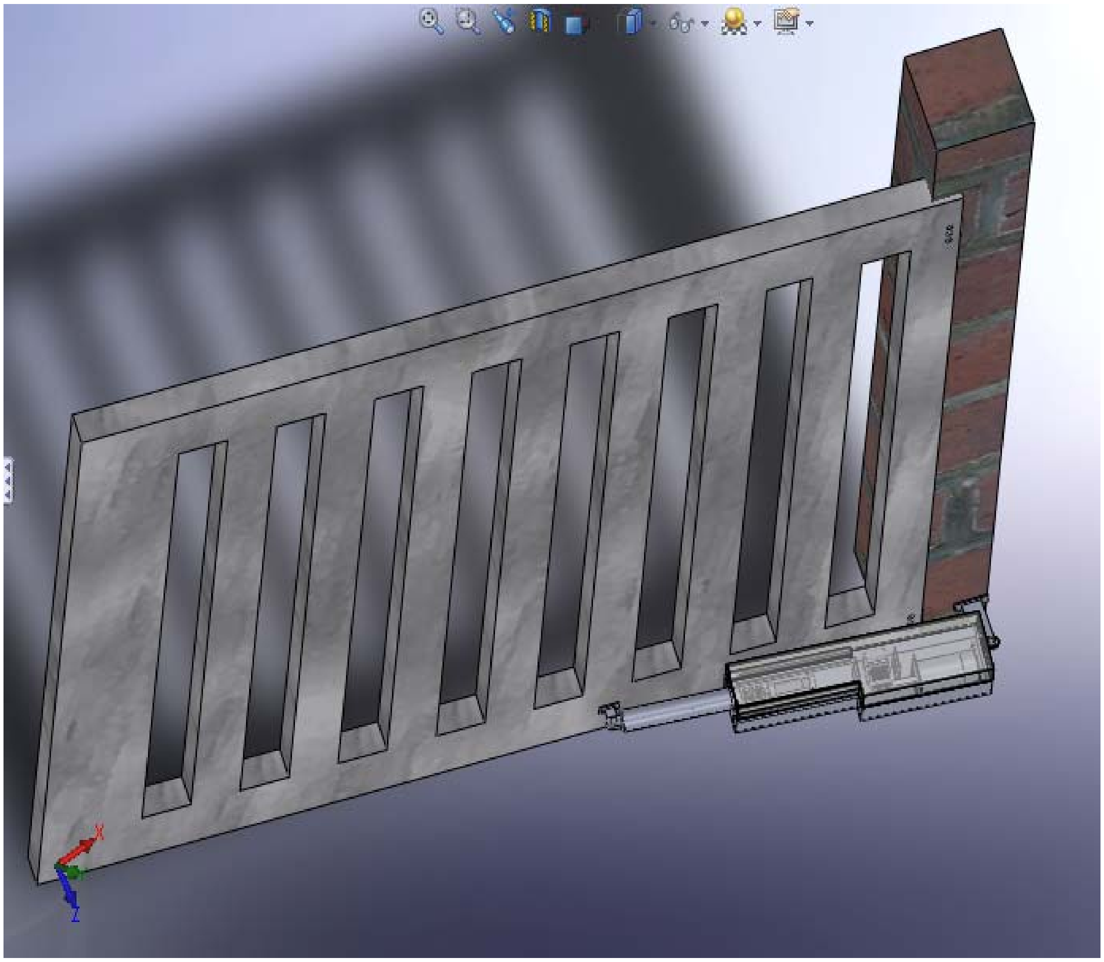


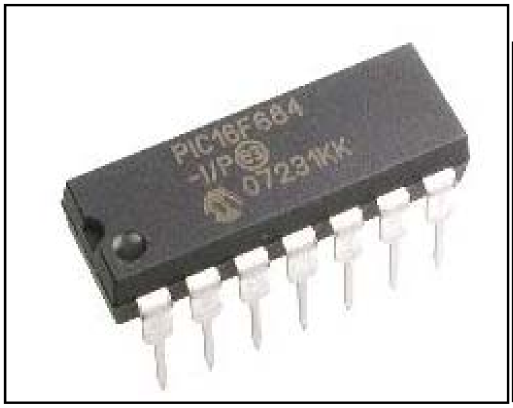
Figure 3.9: Automatic gate mechanism positioned

3.7 MICROCONTROLLER FOR AUTOMATIC GATE MECHANISM

Microcontroller PICAXE14M type is in used because of suitability with the control system that I do. The PICAXE is an easy–to-program microcontroller system that exploits the unique characteristics of the new generation of low-cost ‘FLASH’ memory based microcontroller. PICAXE microcontroller can be programmed over and over again without the need for an expensive programmer. The power of the PICAXE system is its simplicity. No programmer, eraser or complicated electronic system required. PICAXE also can be programmed in a graphical 'flow-chart' environment or in easy to understand basic.

3.7.1 Description and Operation

The PICAXE-14M is considered a step-up from the bare-bones entry-level 08 series. It offers twice the program memory, interrupts, and pulse-width modulation. It is also good for motor controllers system and very suit with my control system. The PICAXE-14M microcontroller provides 5 input and 6 output pins. The coding inside this PICAXE14M that I have done installed is from the visual basic.



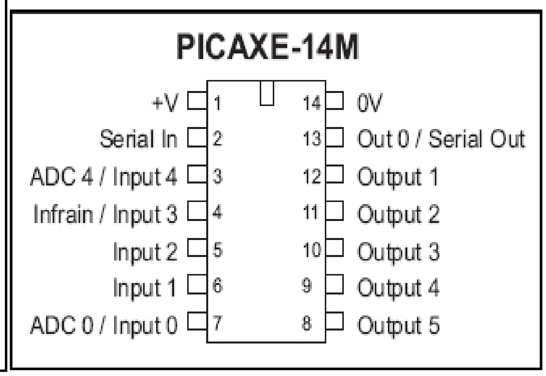


Figure 3.10: PICAXE14M circuit symbol

3.7.2 Infrared Transmitter Circuit

In this circuit contain 3 push button which are ‘forward’ button , ‘reverse’ button and also ‘stop’ button ,when user pressing the first button that is button ‘forward’ it would make microcontroller analyze the data and send data '0' to receiver circuit through Infrared. If user presses the second button that is ‘reverse’ button, Microcontroller will analyze the data and send data'1' to receiver circuit and lastly same subject matter will happen to this system, data’2’ will be sent to receiver circuit if third button is press. Data ‘0’ and data ‘1’ is the data will make the 12volt Motor move bidirectional.

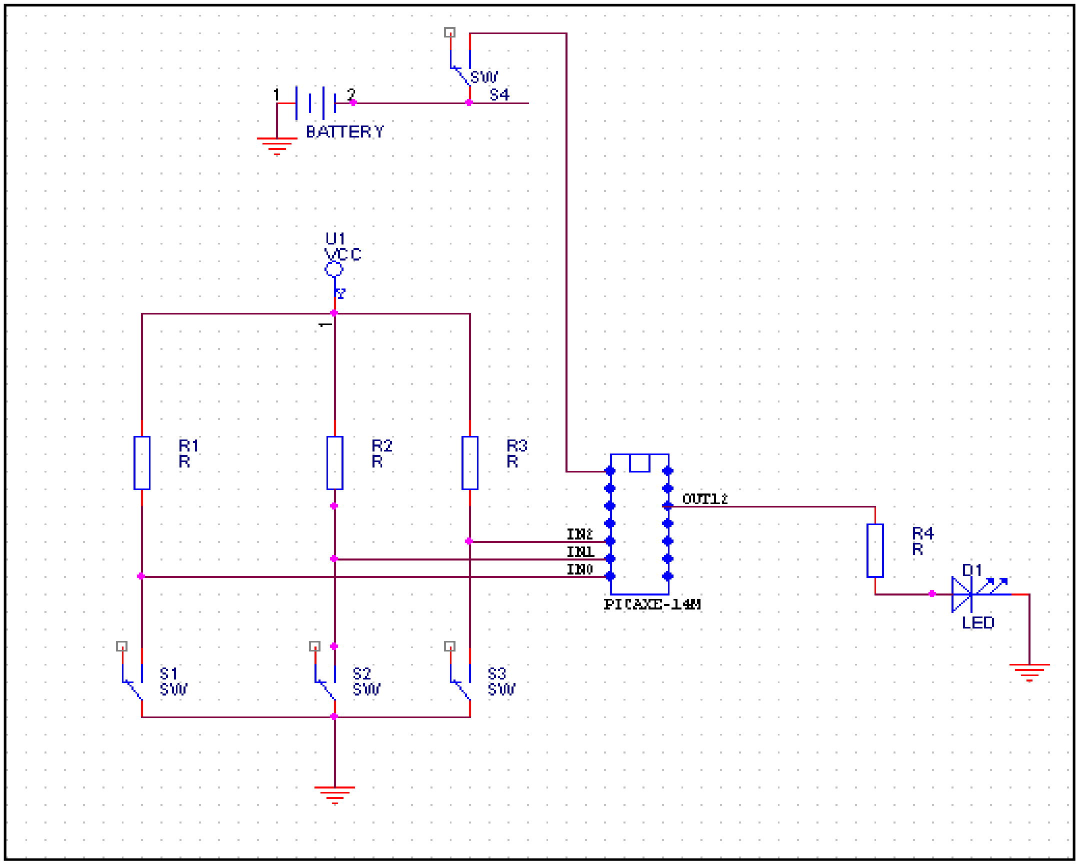


Figure 3.11: Infrared transmitter Circuit

3.7.3 Transmitter Operation

If we press S1 switch, the current will allow current through the wire. In this circuit the resistor position parallel to each other and same with the other two switches; S2 and S3 because of resistor will restrict the current flow and make the amount of current flow same among the tree switches. The value of each resistor is 10KΩ. The first resistor connects to pin IN0, the second Resistor connects to pin IN1 and the last Resistor connects to the pin IN2 at the PICAXE14M Microcontroller. I used 3AA types of Battery to supply the electrical energy to the transmitter circuit. The S4 switch is the switch that active the transmitter circuit. The fourth Resistor is connects with the LEDs in the series way. LEDs is the short form from Light Emitted Diodes, LEDs is a transducer which convert electrical energy to the light. LEDs must be connected the correct way round, because of the leads are labeled '+' for anode and '-' for cathode.

3.7.4 Infrared Receiver Circuit

In this circuit have 2 relays. A relay is an electrically operated switch. Current flowing through the coil of the relay creates a magnetic field which attracts a lever and changes the switch contacts. The coil current can be 'ON' or 'OFF' so relays have two switch positions and they are double throw switches, the coil of a relay passes a relatively large current, typically 30mA for a 12Volt relay like in this circuit. The switches connect with the relay are NPN transistor and a usual switch.NPN transistor is used in other component to make an amplifier transistor or switching circuit. The leads of NPN transistor are labeled base (B), collector (C) and emitter (E).

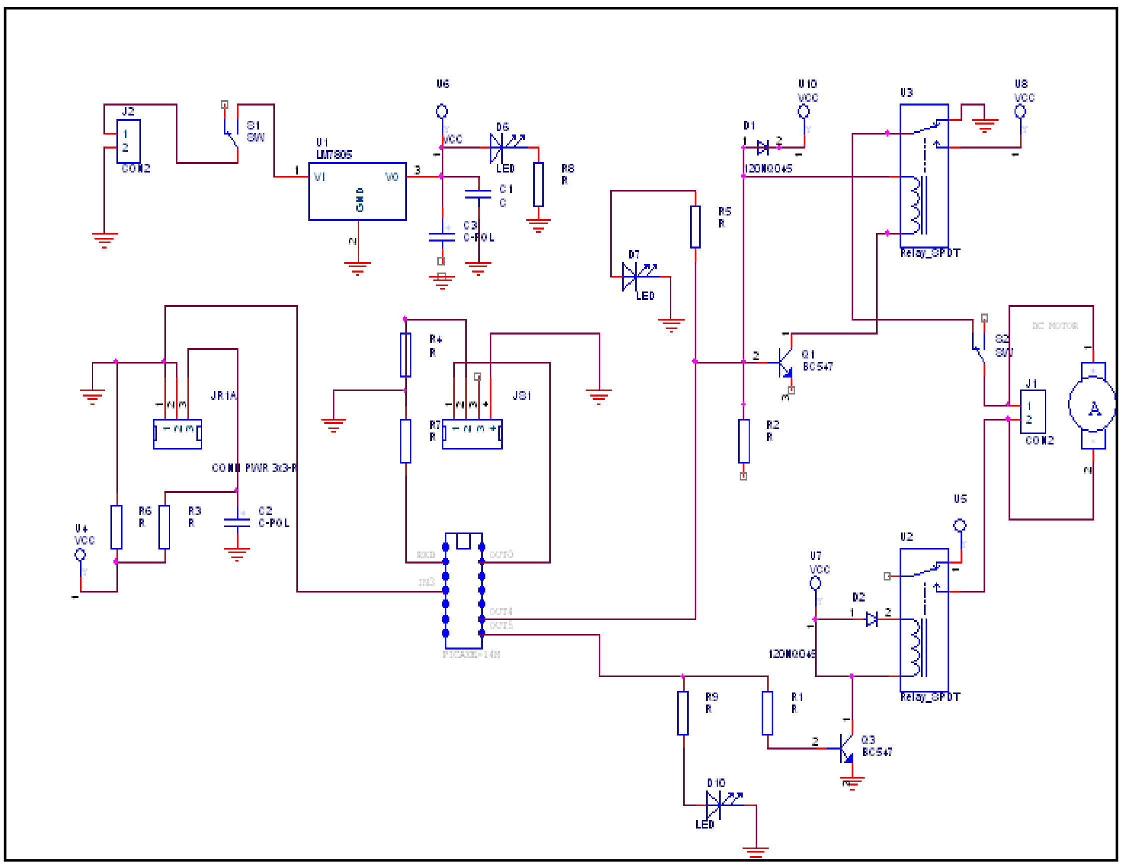


Figure 3.12: Infrared receiver circuit

3.7.5 Receiver Operation

If we press S1 switch at the transmitter circuit, the first relay that connects with the pin OUT4 at the PICAXE14M will active .The first relay will make the 12volt DC motor turn the forward motion and if we press S2 switch at the transmitter circuit, the second relay that connects with the pin OUT5 at the PICAXE14M will active and no active the first relay. The Second relay will make the 12volt DC motor turn opposite position; it will reverse the motion of the DC motor.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 INTRODUCTION

This chapter will discuss on the results acquired through different type of analysis that had been performed in this chapter. The analysis was based on the lead screw and nut analysis, torque analysis, shaft support analysis, bearing, piston arm and gearing system analysis. Those analyses performed on the automatic gate mechanism were to ensure the product designed when loaded with maximum force or stresses would not fail. Those analysis will help to select proper material and part specification by referring to manufacturers' standard table. However, the most important part by performing the analysis was to produce more economical product that is safe and reliable.

The stress analysis on gearing system and piston arm was performed using Finite Element Analysis Algor. The finite element model analysis is a computational technique used to obtain approximate solution of boundary value problems. Here, the finite element analysis was also performed using different meshing, by considering the geometry sizes of the element used for analysis and making a comparison between manual calculation. The results collected will be displayed in graph form.

4.2 LEAD SCREW AND NUT ANALYSIS

In general lead screw mechanism is designed to convert rotary motion to linear motion. The lead screw normally is match with a single nut. The nut is push either extends or retracts into the mechanism. Here, the displacement, linear velocity, revolution to complete the displacement and force required to push the nut is determined also known as the mechanism specifications.

4.2.1 Lead Screw Rotary and Nut Linear Displacement

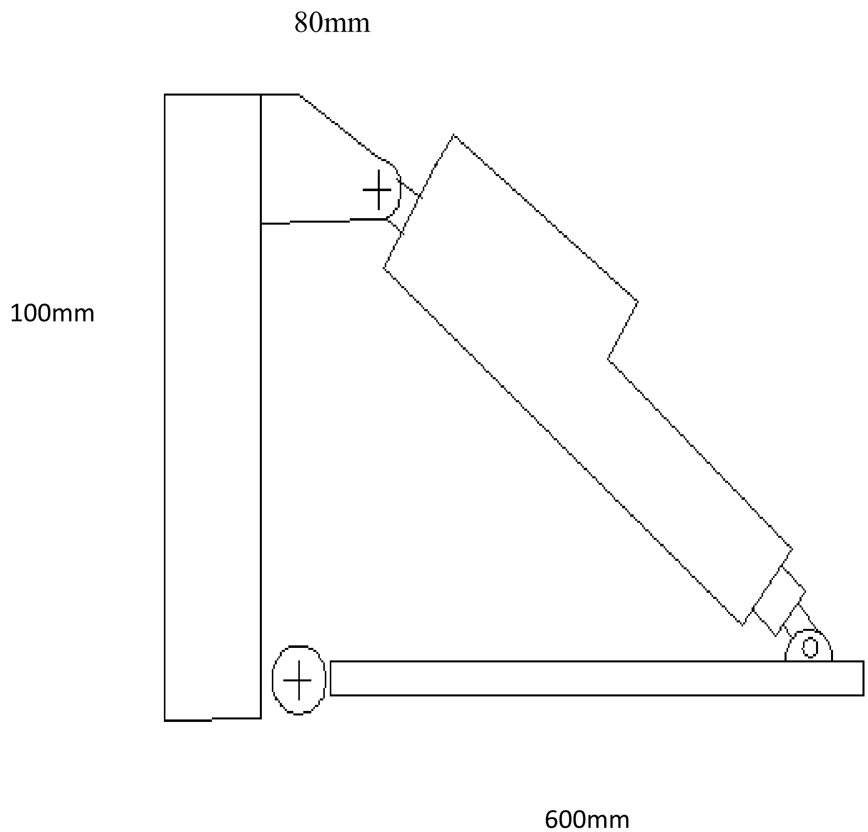


Figure 4.1: Initial position of automatic gate

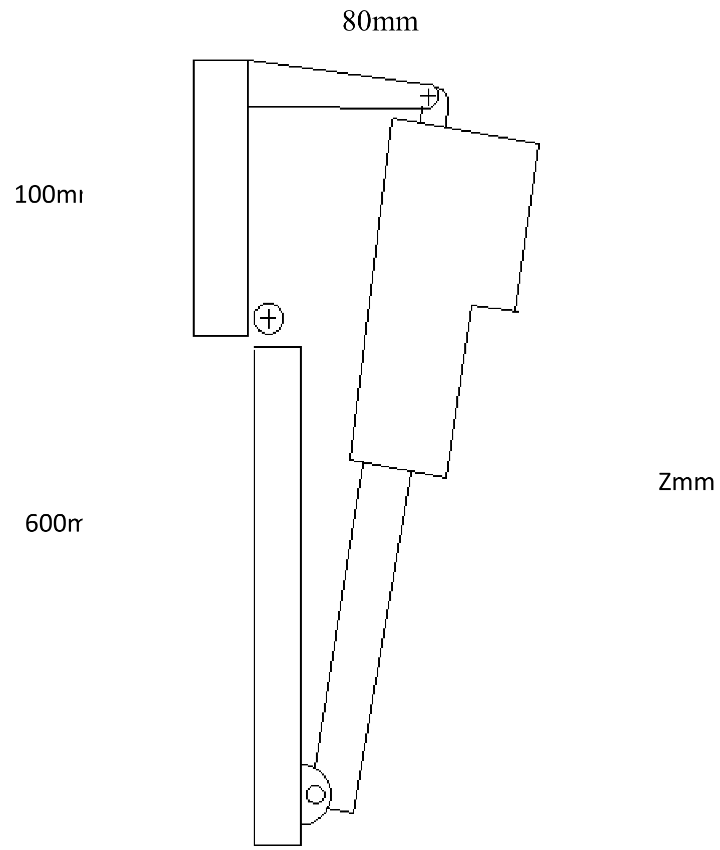


Figure 4.2: Extended position of automatic gate mechanism for 90˚ opening

Using Pythagoras equation to find the total mechanism length when extended in Zmm distance as in Eq. (4.1)

$$Z^2 = \sqrt{X^2 + Y^2} \tag{4.1}$$

$$Z^2 = \sqrt{80^2 + (\,100 + 600)^2}$$

$$Z = 175\text{mm}$$

Linear displacement of the nut required to extend the gate for 90˚ expressed in Eq. (4.2)

$$\text{Displacement, dR} = \text{Final position} - \text{initial position} \tag{4.2}$$

$$\text{Displacement, dR} = 705\text{mm} - 530\text{mm}$$

$$\text{Displacement, dR} = 175\text{mm}$$

In order for the automatic gate angular displacement 90˚ or the arm fully extended position, the nut required to travel along the lead screw for 175mm

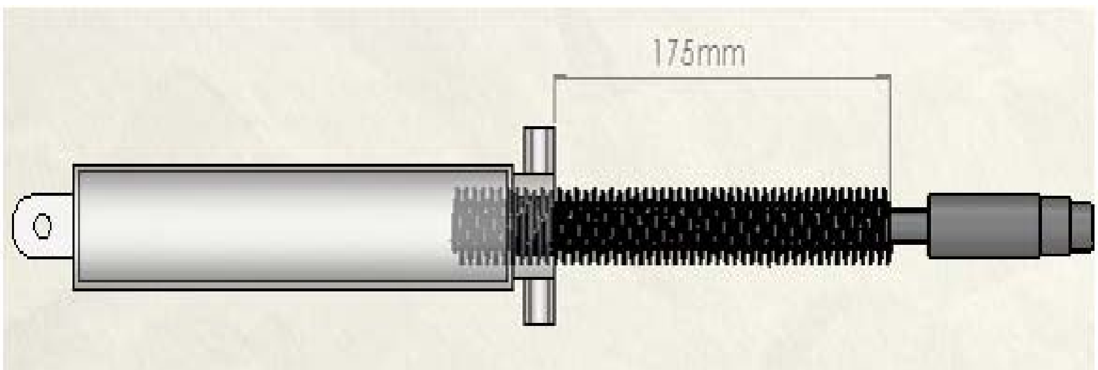


Figure 4.3: Linear displacement of nut along lead screw with 175mm

4.2.2 Revolutions Require by Lead Screw

From the determined linear displacement of nut along the shaft is 175mm. here ACME lead screw in use with following specification (Myszka, 2005)

- Nominal major diameter (in): ¾ in or 19.05mm
- Threads per inch, n: 6
- Pitch (in), p: 0.1667in
- Nominal pitch diameter (in): 0.6424in or 16.31696mm
- Nominal minor diameter (in): 0.5371in or 13.64234mm
- Number of threads, N_t : 1 thread/rev
- Lead, $L = N_t\, p = 0.167$ in/rev or 4.2418 mm/rev

The revolution that required for lead screw for complete gate opening or closing expressed as in Eq. (4.3)

$$d\theta = \frac{dR}{L} = \frac{175\text{mm}}{4.2418\text{mm/rev}} = 41.26\text{rev} \tag{4.3}$$

In order, for the nut to displace linearly 175mm for the selected type ACME lead screw required 41.26 revolutions to complete the gate opening 90˚.

4.2.3 Determine the Arm or Nut Linear Velocity

Estimate the lead screw rotating at angular velocity, ω of 100 revolutions per minute (rpm). The linear velocity of the arm could be determined through Eq. (4.4)

$$v = L\,\omega = \left(0.167\frac{\text{in}}{\text{rev}}\right)\left(100\frac{\text{rev}}{\text{min}}\right) = 16.7\,\frac{\text{in}}{\text{min}} \tag{4.4}$$

$$= 16.7\,\frac{\text{in}}{\text{min}} \times \frac{1\text{min}}{60\text{s}} \times \frac{25.4\text{mm}}{1\text{in}}$$

$$= 7.07\text{mm/s}$$

The nut travels along the lead screw to complete gate opening with constant linear velocity of 7.07mm/s.

4.2.4 Analysis Time Required Completing the Gate Opening

The time required for the automatic gate mechanism arm fully extends with 175mm displacement can be estimate with the following Eq. (4.5)

$$\text{Time required for gate opening or close, } dt = \frac{dR}{v} = \frac{175\text{mm}}{\frac{7.07\text{mm}}{\text{s}}} \tag{4.5}$$

$$= 24.8\text{ second}$$

Time taken for the automatic gate mechanism to complete the nut displacement of 175mm or 90˚ angular displacement of the gate required time of 24.8second.

4.3 TORQUE REQUIRED FOR MOVING THE PISTON ARM OR NUT

The torque analysis that required for this mechanism considers being an important factor. The maximum torque need to be known so the mechanism can functional properly under the operating condition. The torque from motor will transmit power to the shaft through gear.

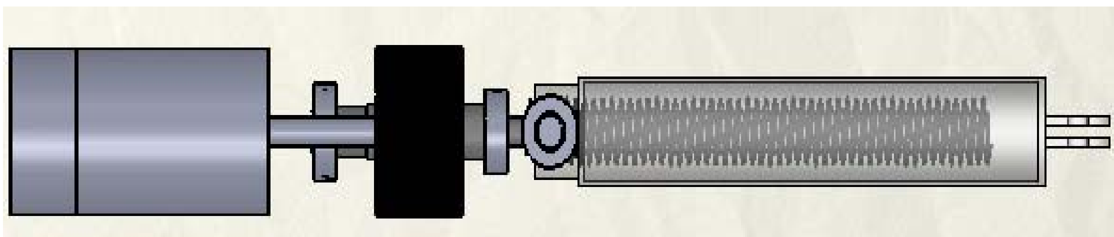


Figure 4.4: Lead screw mechanism overview

Average double gate leaf weight per leaf at Malaysia, m = 100kg
Normal force that acting on the gate can expressed as in Eq. (4.6)

$$F_N = ma \tag{4.6}$$

Static Friction coefficient for steel to steel, $\mu_s = 0.40$
Kinetic friction coefficient for steel to steel, $\mu_k = 0.20$
Acceleration due to gravity, a = 9.819.81 m/s^2

Normal force that react on the gate:

$$F_N = (100kg)\ (9.81\ m/s^{2)}) = 981N$$

Force required moving the gate under static condition can be expressed as in Eq. (4.7)

$$F_f = \mu_s F_N \tag{4.7}$$

F_f= (0.40) (981N) =392.4N

Force required moving the gate under kinetic condition can be expressed as in Eq. (4.8)

$$F_f = \mu_k F_N \tag{4.8}$$

F_f= (0.20) (981) = 196.2 N

Total torque load from lead screw expressed as in Eq. (4.9)

$$\mathrm{T} = (\,\mathrm{Jrotor} + \mathrm{J\ screw} + \mathrm{J\ reflected}\,)\,\alpha + \mathrm{T\ friction} \tag{4.9}$$

$$\mathrm{T} = \mathrm{Trotor} + \mathrm{Tscrew} + \mathrm{Treflected} + \mathrm{T\ friction}$$

Torque due to the effect of the steel screw can be expressed as in Eq. (4.10)

$$\mathrm{T\ screw} = (\mathrm{J\ screw})\alpha \tag{4.10}$$

Inertia of steel screw from the lead screw can be expressed as in Eq. (4.11)

$$\mathrm{J\ screw} = \mathrm{Dp}^4 \times \mathrm{L\ x}\ \frac{\Pi}{32} \times \mathrm{Density\ steel} \tag{4.11}$$

$$\mathrm{T\ screw} = \left(\mathrm{Dp}^4 \times \mathrm{L\ x}\ \frac{\Pi}{32} \times \mathrm{Density\ steel}\right)\alpha$$

Lead screw Diameter pitch, Dp = 3/4 inch or 0.01905m

Displacement of nut along the lead screw, L = 0.175m

Density of lead screw for stainless steel material, AISI 302 = 7920kg/m³ (Beer, 2006).

Assumed the angular velocity, ω of 100rpm and the start up time for the motor before constant velocity, dt=2second.

$$\text{Angular velocity}, \omega = 100\text{rpm} \times \frac{2\Pi \text{ rad}}{\text{rev}} \times \frac{1\text{min}}{60 \text{ sec}} = 10.4720\text{rad/sec}$$

The Angular acceleration for the motor can be estimated using Eq. (4.12)

$$\text{Angular acceleration}, \alpha = \frac{\omega}{\text{dt}} = \frac{10.4720\text{rad/sec}}{2 \text{ sec}} = 5.236\frac{\text{rad}}{\text{sec}^2} \tag{4.12}$$

$$\text{T screw} = \left[(0.01905\text{m})^4 \times 0.175\text{m x } \frac{\Pi}{32} \times \frac{7900\text{kg}}{\text{m}^3}\right] 5.236\text{rad/sec}^2$$

$$\text{T screw} = 9.3604 \times 10^{-05}\text{N.m}$$

Torque due to piston arm loaded weight can be expressed in Eq. (4.12)

$$\text{T reflected} = (\text{J reflected})\alpha \tag{4.12}$$

The reflected inertia from the loaded weight can be expressed in Eq. (4.13)

$$\text{J reflected} = \text{M (Load)} \times \text{L}^2 \times 0.025 \tag{4.13}$$

$$\text{T reflected} = [\text{M (Load)} \times \text{L}^2 \times 0.025]\alpha$$

M (Load) = piston arm weight in kilogram

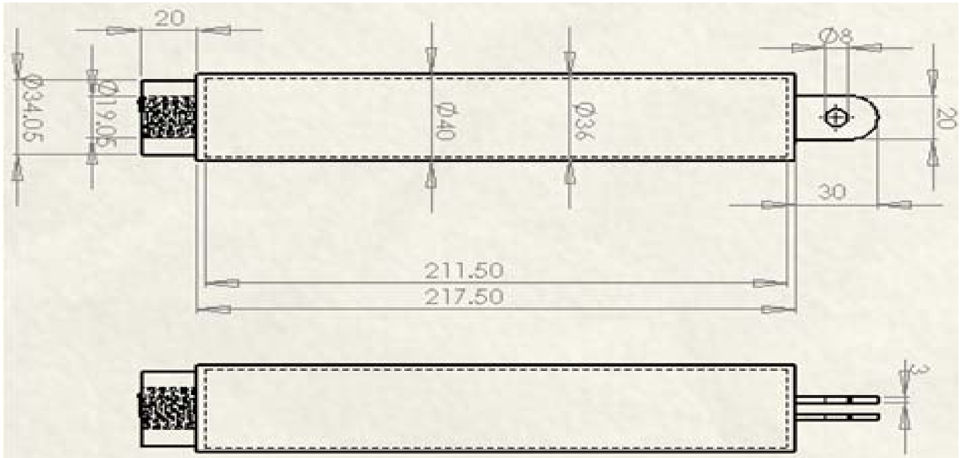


Figure 4.5: Piston arm dimension

Total volume of piston arm,

$$V = [\ (\Pi(17.025\text{mm})^2 20\text{mm}) - (\Pi(19.525\text{mm})^2 20\text{mm})] +$$
$$[\ (\Pi(20\text{mm})^2 217.5\text{mm}) - (\Pi(18\text{mm})^2 211.5\text{mm})] +$$
$$[\ (30\text{mm} \times 20\text{mm} \times 3\text{mm})2] + \ [\Pi(9.525\text{mm})^2 \times 3\text{mm}]$$
$$= 7.3599 \ \times 10^{-05} \text{m}^3$$

Piston arm using stainless steel 302 where density, ρ = 7920 kg/m³. The mass of the piston arm can be acquired using Eq. (4.14)

$$\rho = \frac{\text{M (LOAD)}}{\text{V}}$$

$$\text{M (LOAD)} = \ \rho\ \text{V} = \left(\frac{7920\text{kg}}{\text{m}^3}\right) (\ 7.3599 \ \times 10^{-05} \text{m}^3) = 0.5829 \text{ kg} \qquad (4.14)$$

By referring to equation 4.12 and 4.13:

$$\text{T reflected} = \ [0.5829\,\text{kg} \times (0.175\text{m})^2 \times \ 0.025]\ 5.236\text{rad/sec}^2$$

$$\text{T reflected} = 0.02337\text{N.m}$$

Torque due to the friction of the arm nut and lead screw can be expressed as in Eq. (4.15)

$$\text{T friction} = \left(\frac{1}{2\Pi}\right)\left(\frac{F_f}{pe}\right) \tag{4.15}$$

$$\text{Pitch, p} = 6\ \text{thread/inch} = 6\ \text{rev/inch} \times \frac{1\ \text{inch}}{0.0254\text{m}} = 236.22\ \text{rev/m}$$

Efficiency for ACME lead screw, $e = 0.3$

Screw and nut (lubricated) friction force F_f with friction coefficient of, $\mu = 0.17$

The friction force that occurred between the arm nut and lead screw can be express in Eq. (4.16)

$$F_f = \mu N = (0.17)(5.829\text{N}) = 0.9909\text{N} \tag{4.16}$$

By referring to equation 4.15:

$$\text{T friction} = \left(\frac{1}{2\Pi}\right)\left[\frac{0.9909\text{N}}{(236.22)(0.3)}\right]$$

$$\text{T friction} = 2.225 \times 10^{-3}\text{N.m}$$

By referring to equation 4.9:

$$T = \text{Trotor} + \text{Tscrew} + \text{Treflected} + \text{T friction}$$

$$T = \text{Trotor} + 9.3605 \times 10^{-05}\text{N.m} + 0.02337\text{N.m} + 2.225 \times 10^{-3}\ \text{N.m}$$

$$T = \text{Trotor} + 0.025689\text{N.m}$$

Force that required moving the gate respect to gate friction force can be expressed as in Eq. (4.17)

$$\mathrm{Ff} = \frac{2\Pi \ \times \mathrm{T}}{\mathrm{L}} \times \text{efficiency} \tag{4.17}$$

For static condition by using equation 4.17:

$$392.4\mathrm{N} = \frac{2\Pi\ (\mathrm{Trotor} + 0.025689\mathrm{N.m})}{0.175\mathrm{m}} \times 0.3$$

$$68.67 = \ 1.8850\mathrm{Trotor} + 0.0484$$

$$\mathrm{Trotor} = 36.4040\mathrm{N.m}$$

$\mathrm{Trotor} = 36.4040\mathrm{N.m}$ For static condition is the maximum torque required from motor to generate force for the nut and translate from rotary into linear mechanical movement.

For dynamic condition by using equation 4.17:

$$196.2\mathrm{N} = \frac{2\Pi\ (\mathrm{Trotor} + 0.025689\mathrm{N.m})}{0.175\mathrm{m}} \times 0.3$$

$$34.335 = \ 1.8850\mathrm{Trotor} + 0.0484$$

$$\mathrm{Trotor} = 18.1892\mathrm{N.m}$$

$\mathrm{Trotor} = 18.1892.\mathrm{m}$ For dynamic condition is the minimum torque required from motor to generate force for the nut and translate from rotary into linear mechanical movement.

4.4 GEARING SYSTEM

For this automatic gate design, four time reduction or ratio pinion: gear 1:4 is in used, to increase the torque and to reduce the speed in this mechanism. From the figure below, shaft 1 is drive by the motor and shaft 2 to be supported with 2 bearing and drive by pinion.

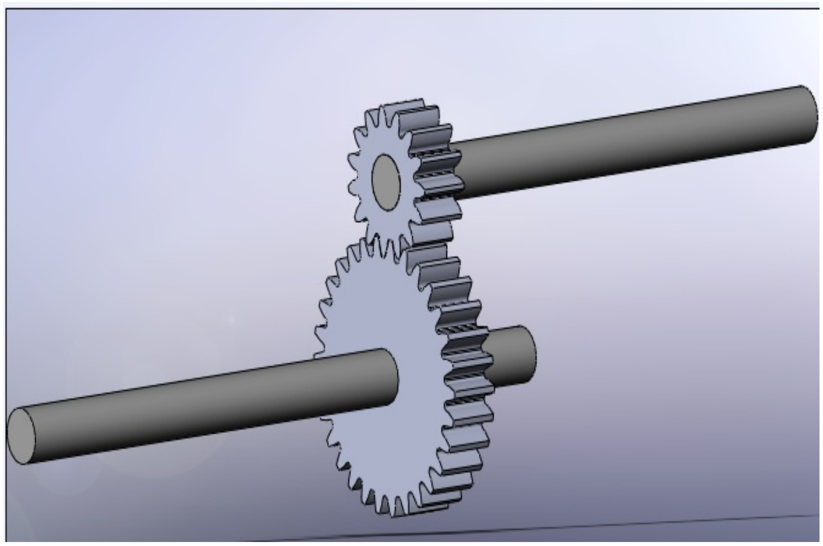


Figure 4.6: Gear train

Gear teeth, N_1= 20 teeth

Pinion teeth, N_2=80 teeth

Shaft 2 rotation speed, n_2 = 100RPM

Gear pitch diameter, d_2 = 60mm

Pinion pitch diameter, d_1 =15mm

Torsion required moving the automatic gate mechanism before the go through gear transmission, $\text{Torsion rotor or } T_2 = 36.4040\text{N.m}$. Only consider the maximum torque required from motor to generate force for the nut and translate from rotary into linear mechanical movement. Once, the motor produce the maximum torque for the system also can drive the minimum condition.

Rotation speed produces by the motor can be obtained using Eq. (4.18)

$$n_1 = \left(\frac{N_2}{N_1}\right) n_2 = \left(\frac{80}{20}\right) 100\text{RPM} = 400\text{rpm} \tag{4.18}$$

Torque required by the motor after going through gearing system using ration 1:4 can be expressed as in Eq. (4.19)

$$T_1 = \left(\frac{r_1}{r_2}\right) T_2 = \left(\frac{7.5\text{mm}}{30\text{mm}}\right) 36.4040\text{N.m} = 9.101\text{N.m} \tag{4.19}$$

Motor operating power for the mechanism can be obtained using Eq. (4.20)

$$\text{power} = T\,(2\Pi f) = T\omega \tag{4.20}$$

$$= (9.101\text{N.m})(400\text{rpm})\left(\frac{2\Pi\text{rad}}{\text{rev}}\right)\left(\frac{1\text{min}}{60\text{sec}}\right)$$

$$= 381\text{watt or } 0.51$$

The motor specification in used for the mechanism to function properly with following data:

$$\text{Torque}, T = 9.101\ \text{N.m}$$
$$\text{Speed}, n = 400\text{rpm}$$
$$\text{Power}, P = 381\ \text{watt or } 0.51\text{hp}$$

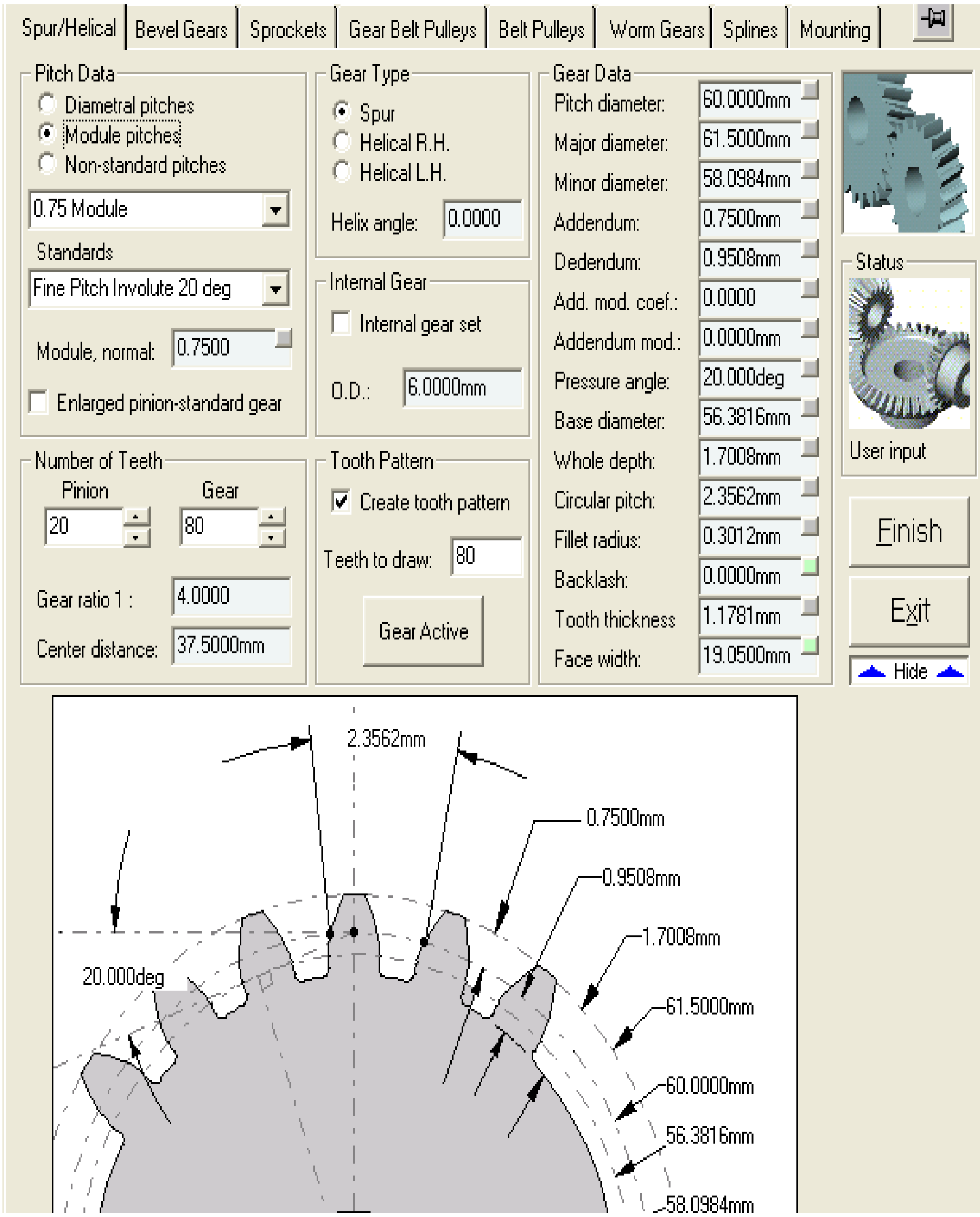


Figure 4.7: Gear in used data

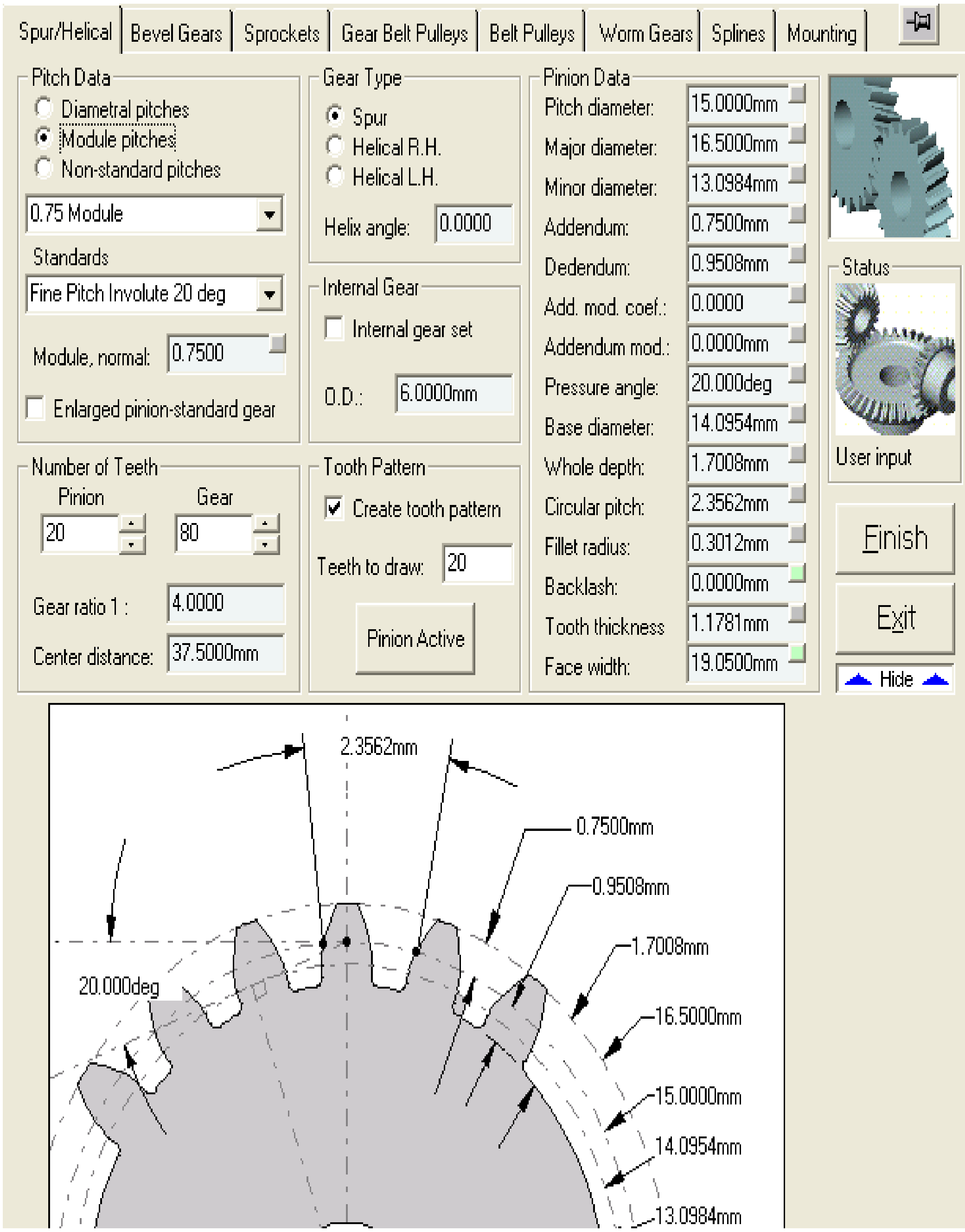


Figure 4.8: Pinion in used data

4.5 SHAFT DESGIN TO SUPPORT GEAR AND BEARINGS

Shaft is a rotating member, in circular cross section used to transmit power or motion. It provides the axis of rotation, or oscillation, of element such as gears. In deciding on an approach to shaft sizing, it is necessary to realize that a stress analysis at a specific point on a shaft can be made using only the shaft geometry in the vicinity of that point. Thus the geometry of the entire shaft not needed. In the design it is usually possible to locate the critical areas, size these meet the strength requirement, and then size the rest of the shaft to meet the requirement of the shaft supported elements. The deflection and slope analyses cannot be made until the geometry of the entire shaft has been defined. Thus deflection is a function of the geometry everywhere, where the stress at a section of interest is a function of local geometry. For this reason, shaft design allows a consideration of stress first. Then, after tentative values for the shaft dimensions have been established, the determination of the deflections and slopes can be made.

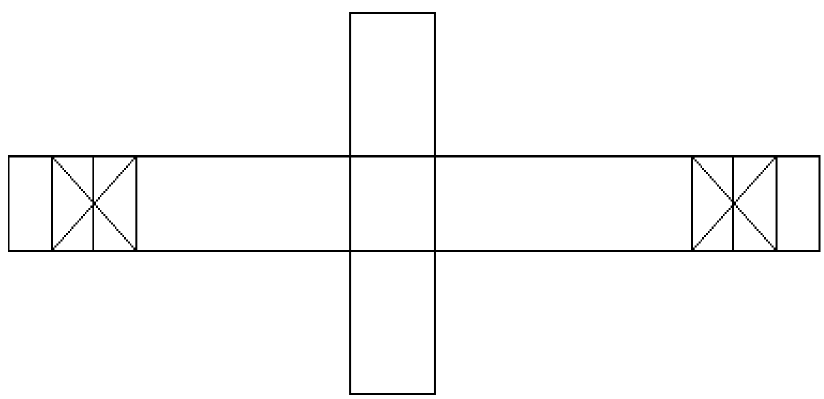


Figure 4.9: Position of bearing and gear on the shaft

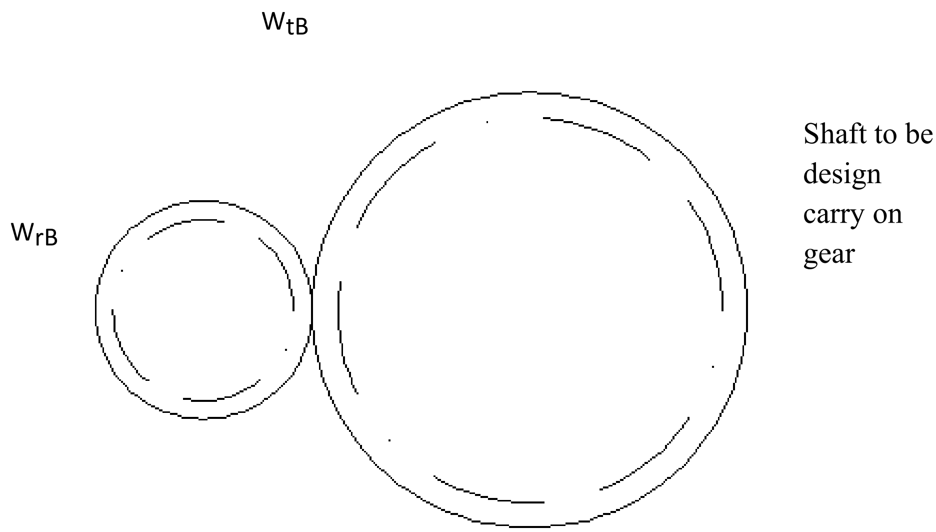


Figure 4.10: Pinion drive gear and show the force exert on the gear.

Tangential force can be obtained using Eq. (4.21)

$$\mathrm{WtB} = \frac{\mathrm{T_B}}{\mathrm{r}}$$

Torque for gear at point B, $\mathrm{T_B} = 36.404\mathrm{N.m}$

Gear picth radius, $\mathrm{r} = 30\mathrm{mm}$ or $0.03\mathrm{m}$

$$\text{tangential force}, \mathrm{Wt_B} = \frac{36.404\mathrm{N.m}}{0.03\mathrm{m}} = 1213\mathrm{N}$$

Radial force can be obtained using Eq. (4.22)

$$\mathrm{Wr_B} = \ \mathrm{Wt_B}\tan(\phi) \qquad (4.22)$$

Pressure angle, $\phi = 20^{\circ}$

$$\text{Radial force}, \mathrm{Wr_B} = (1213\mathrm{N})\tan(20^{\circ}) = 442\mathrm{N}$$

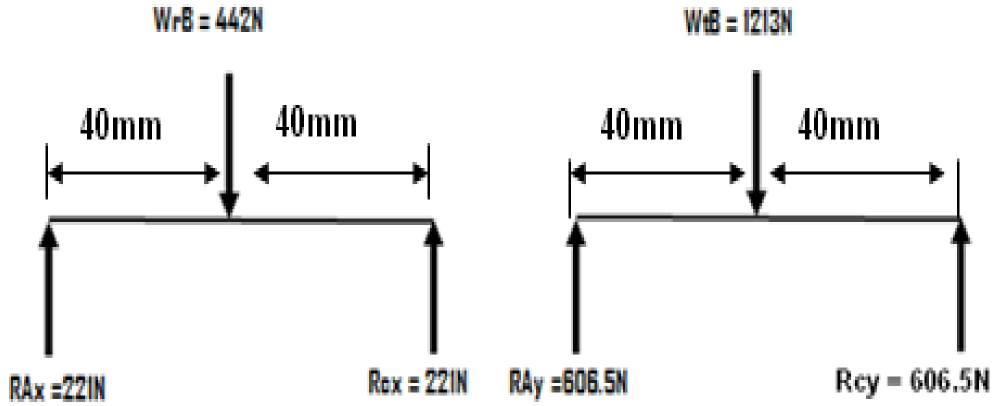


Figure 4.11: Forces react on shaft in X and Y direction

4.5.1 Shearing and Bending Moment X-axis Force

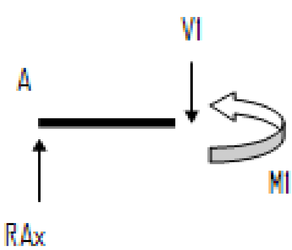


Figure 4.12: Internal forces on shaft in X axis for first part

$$+ \quad \sum Fx = 0$$
$$RAX - V1 = 0$$
$$V1 = 221N$$

Moment under clockwise direction is positive

$$\sum M1 = 0$$
$$M1 - RAx = 0$$
$$M1 = RAx$$

When x = 0.04

$$M1 = 221N \times 0.04m = 8.84N.m$$

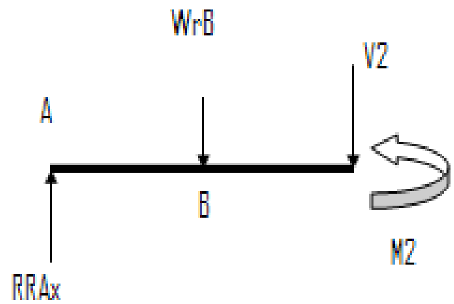


Figure 4.13: Internal forces on shaft in X axis for second part

$$+ \quad \sum Fx = 0$$

$$RAX - WrB - V2 = 0$$

$$V2 = -221N$$

Moment under clockwise direction is positive

$$\sum M2 = 0$$

$$M2 - RAx + Wr_B\left(x - \frac{1}{2}L\right) = 0$$

When x = 0.08m

$$M2 - (221N)(0.08m) + (442N)(0.08m - 0.04m) = 0$$

$$M2 = 0$$

4.5.2 Shearing and Bending Moment Y-axis Force

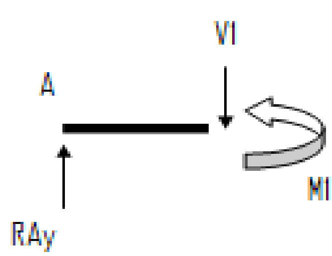


Figure 4.14: Internal forces on shaft in Y axis for first part

$$+ \quad \sum Fy = 0$$

$$RAy - V1 = 0$$

$$V1 = 606.5N$$

Moment under clockwise direction is positive

$$\sum M1 = 0$$

$$M1 - RAy = 0$$

$$M1 = RAy$$

When x = 0.04

$$M1 = 606.5N \times 0.04m = 24.26N.m$$

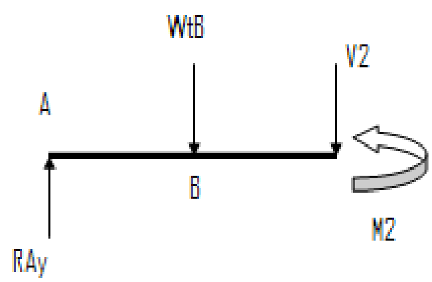


Figure 4.15: Internal forces on shaft in Y axis for second part

$$+ \quad \sum Fx = 0$$

$$RAX - Wt_B - V2 = 0$$

$$V2 = 606.5N - 1213N = -606.5N$$

Moment under clockwise direction is positive

$$\sum M2 = 0$$

$$M2 - RAx + Wr_B\left(x - \frac{1}{2}L\right) = 0$$

When x = 0.08m

$$M2 - (606.5N)(0.08m) + (1213N)(0.08m - 0.04m) = 0$$

$$M2 = 0$$

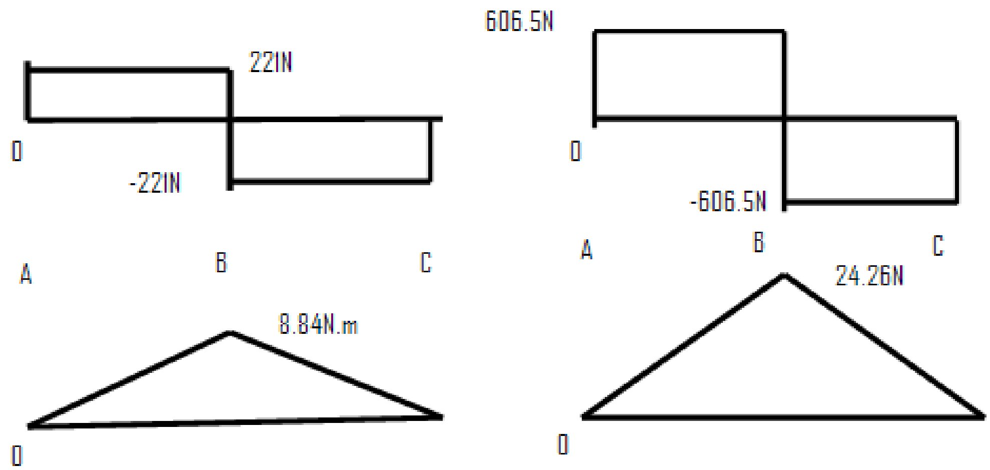


Figure 4.16: Shearing and bending moment diagram

4.5.3 Shaft Sizes to Support Gear and Bearings

The steel shaft material in used is stainless steel 304. (shigley's, 2008)

Yield strength, S_y=276MPa
Ultimate strength, S_u = 568MPa
Endurance limit, S_n = 240MPa

A size factor should be applied to the endurance strength because the shaft will be quite large to be able carrying 0.51hp. Take C_s = 0.75 (Mott, 2004)

Reliability factor also should be consider, Let the design reliability of 0.99 and use C_R = 0.81. (Mott, 2004)

Compute the modified endurance strength using Eq. (4.23)

$$S'n = SnCsCr = (240MPa)(0.75)(0.81) = 145.8MPa \qquad (4.23)$$

The design factor is taken to be N=2. The shaft under loading is not expected to present with any un-usual shock or impact.

Point A: bearing A produces torsion in the shaft from A and to the right. There is no force shearing forces and bending moment in this point. The moment at A is zero because it is a free end of the shaft and diameter for the bearing can be determine using Eq. (4.24)

$$D1 = [\frac{32N}{\Pi}\sqrt{\frac{3}{4}\left(\frac{T}{Sy}\right)^2}]^{\frac{1}{3}} \tag{4.24}$$

$$D1 = [\frac{32(2)}{\Pi}\sqrt{\frac{3}{4}\left(\frac{36.404N.m}{276MPa}\right)^2}]^{\frac{1}{3}}$$

$$D1 = 0.01325m$$

Point B: point B is location of the gear with bearing diameter sharp to the right of B. K_t = 1.5 (well rounded fillet)

The total moment acting on the shaft can be determined using Eq. (4.25)

$$M_B = \sqrt{M_{Bx}^2 + M_{By}^2} = \sqrt{8.84^2 + 24.26^2} = 25.82N.m \tag{4.25}$$

Combine stress condition can be determined using Eq. (4.26)

$$D2 = [\frac{32N}{\Pi}\sqrt{\left(\frac{KtM_B}{S'n}\right)^2 + \frac{3}{4}\left(\frac{T}{Sy}\right)^2}]^{\frac{1}{3}} \tag{4.26}$$

$$D2 = [\frac{32(2)}{\Pi}\sqrt{\left(\frac{(1.5)(25.82N.m)}{145.8MPa}\right)^2 + \frac{3}{4}\left(\frac{36.404N.m}{276MPa}\right)^2}]^{\frac{1}{3}}$$

$$D2 = 0.01806m$$

$$D3 = [\frac{32N}{\Pi} \sqrt{\left(\frac{KtM_B}{S'n}\right)^2 + \frac{3}{4}\left(\frac{T}{Sy}\right)^2}]^{\frac{1}{3}}$$

Kt = 2.0 (with profile keyseat)

$$D3 = [\frac{32(2)}{\Pi} \sqrt{\left(\frac{(2.0)(25.82N.m)}{145.8MPa}\right)^2 + \frac{3}{4}\left(\frac{36.404N.m}{276MPa}\right)^2}]^{\frac{1}{3}}$$

$$D3 = 0.01964m$$

Point C: Point C is the seat for bearing and there is no bending moment here, however there is a vertical shearing force equal to the reaction at the bearing. Using the resultant of X and Y plane reaction, the shearing force is

$$Vc = \sqrt{(221N)^2 + (606.5N)^2} = 645.51N$$

The diameter of the shaft at point C when only considering the shearing force can be determined using Eq. (4.27)

$$D4 = \sqrt{\frac{2.94\ Kt\ (Vc)N}{S'n}} \tag{4.27}$$

$Kt = 2.5$ (sharp fillet)

$$D4 = \sqrt{\frac{(2.94)\ (2.5)\ (645.51N)(2)}{145.8MPa}} = 0.008067m$$

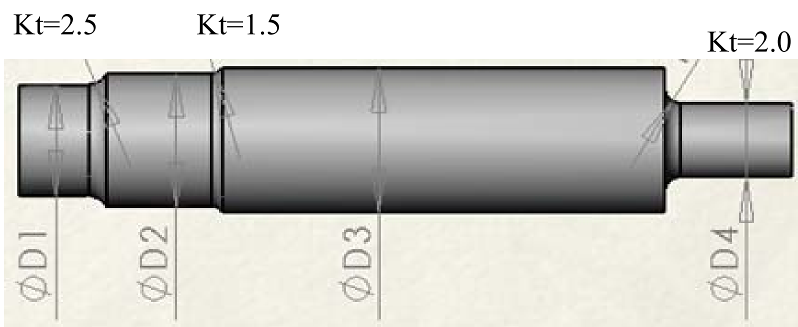


Figure 4.17: Shaft size after designed

The shaft sizes used to support bearing 1, D1 is 0.01325m. To support gear, D3 is 0.01964m and follow by support bearing 2, D4 is 0.008067m. The shaft when used for power transmission under shear and bending moment needed to support with following shaft sizes for this mechanism to avoid failure.

4.6 BEARINGS ANALYSIS

Bearing are all used of bearing in which the main load is transferred through element in rolling contact rather than in sliding contact. In bearing, these elements must be designed to fit into a space whose dimension is specified for the shaft to support the bearing. The bearing also must be designed to receive a load having certain characteristics. Finally, these elements must be designed to have satisfactory life when operated under the specified conditions.

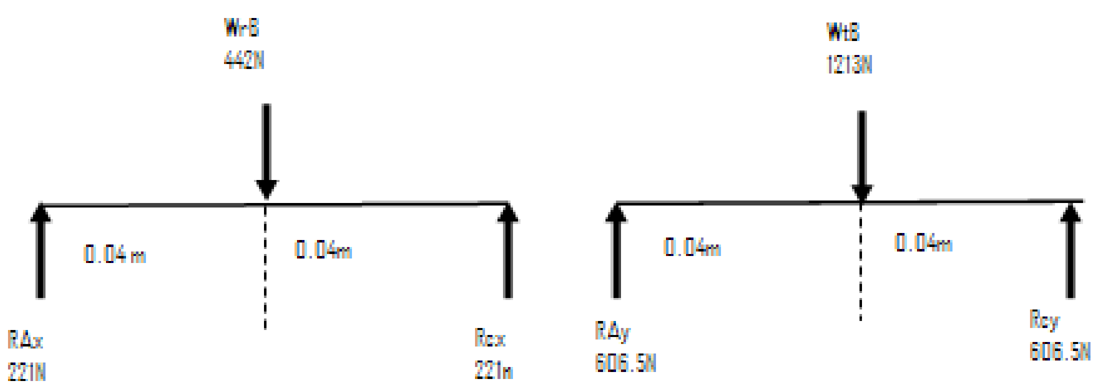


Figure 4.18: Radial bearing load react on point A and C in X ad Y plane

Bearing radial load at A can be expressed as in Eq. (4.28)

$$= \sqrt{RAx^2} + RAy^2 = \sqrt{(221N)^2} + (606.5)^2 = 645.5N \tag{4.28}$$

Bearing radial load at C can be expressed as in Eq. (4.29)

$$= \sqrt{Rcx^2} + Rcy^2 = \sqrt{(221N)^2} + (606.5)^2 = 645.5N \tag{4.29}$$

Single row deep groove ball bearing is more suitable for the automatic gate mechanism. The automatic gate is categorized with the elevators, industrial fans, and multipurpose gearing application with design life, L_{10} of 8000-15000 h (Mott, 2004)

Design life of the bearing on the mechanism can be expressed as in Eq. (4.30)

$$Ld = (h)(rpm)\left(\frac{60min}{h}\right) = (15000)(100rpm)\left(\frac{60min}{h}\right) = 9 \times 10^7 \tag{4.30}$$

Dynamic load rating due to bearing radial load can be expressed as in Eq. (4.31)

$$\text{Dynamic load rating, C} = (\text{bearing radial load})\ (\frac{Ld}{10^6})^{\frac{1}{k}} \tag{4.31}$$

$k = 3$ for ball bearing

$$\text{Dynamic load rating, C} = (645.5N))\ (\frac{9 \times 10^7}{10^6})^{\frac{1}{k}} = 2893N \text{ or } 650.38lb$$

Bearing with bore diameter D1= 13.25mm and dynamic load rating of 650.38lb is suit with bearing No.6202 with nearest bore diameter 15mm.
Bearing with bore diameter D4 = 8.07mm and dynamic load rating of 650.38lb is suit with bearing No.6200 with nearest bore diameter 10mm. (Mott, 2004)

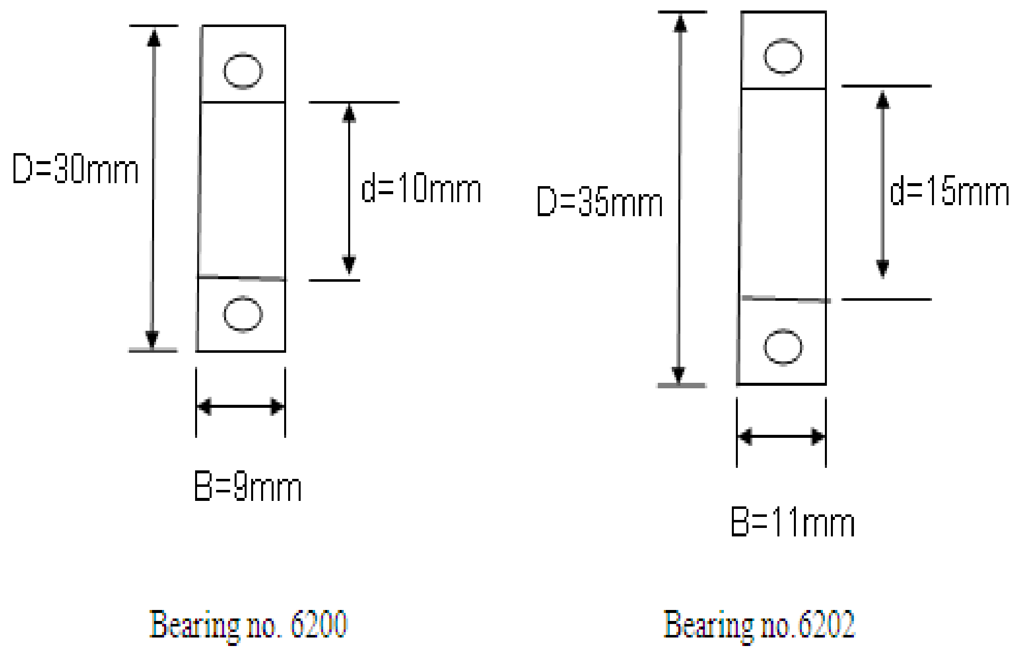


Figure 4.19: Bearing with following dimension selected for design

4.7 GEAR AND PINION STRESS ANALYSIS

Here, stress exert between gear and pinion need to be determined when loaded with high torque application. Fatigue failure of gear and pinion teeth normally due to the many repetitions of high compressive stress. The compressive stress exert on the gear teeth need to compare with the material allowable stress value to avoid the unpleasant failure occurrence and in order to select the proper material in use for the gearing system of the mechanism.

Material of ASTM no.20 is selected for the gear analysis with the following properties (shigley's, 2008)

Modulas of elascity, E_p and E_g = 100GPa
Passion ratio,v_p and v_g = 0.211

Elastic coefficient of the material in used for gear and pinion can be obtained using Eq. (4.32)

$$\text{Elastic coeficient, Cp} = \{\frac{1}{\Pi\left[\frac{1-(\mathrm{Vp})^2}{\mathrm{Ep}}+\frac{1-(\mathrm{Vg})^2}{\mathrm{Eg}}\right]}\}^{\frac{1}{2}} \tag{4.32}$$

$$\text{Elastic coeficient, Cp} = \{\frac{1}{\Pi[\frac{1-(0.211)^2}{100\ \times 10^9}+\frac{1-(0.211)^2}{100\ \times\ 10^9}]}\}^{\frac{1}{2}}$$

$$\text{Elastic coeficient, Cp} = 129062.3\ \text{Pa}$$

The instantaneous values of the radii of curvature on the pinion and gear tooth profiles can be obtained using Eq. (4.33) and (4.34)

Pinion pitch diameter,dp = 0.06m

Gear pitch diameter, dg = 0.015m

Pressure angle, ϕ = 20˚

$$\mathrm{r1} = \frac{\mathrm{dp}\sin\phi}{2} = \frac{(0.06\mathrm{m})\sin 20^\circ}{2} = 0.0103\mathrm{m} \tag{4.33}$$

$$\mathrm{r2} = \frac{\mathrm{dg}\sin\phi}{2} = \frac{(0.015\mathrm{m})\sin 20^\circ}{2} = 0.0026\mathrm{m} \tag{4.34}$$

Tangential velocity can be obtained using Eq. (4.35)

$$\mathrm{V_t} = \omega\mathrm{r} \tag{4.35}$$

gear pitch radius, $\mathrm{r} = 0.06\mathrm{m}/2$

angular velocity, $\omega = 10.47\mathrm{rad/sec}$

$$\mathrm{Vt} = \left(\frac{10.47\mathrm{rad}}{\mathrm{sec}}\right)\left(\frac{0.06\mathrm{m}}{2}\right) = 0.3141\mathrm{m/s}$$

The velocity factor can be obtained using Eq. (4.36)

$$K_v = \frac{600 + V_t}{600} \tag{4.36}$$

$$K_v = \frac{600 + 0.3141\text{m/s}}{600} = 1.0005$$

The Compressive stress exact between the gear and pinion can be obtained using Eq. (4.37)

$$\sigma c = -Cp[\frac{Kv\,Wt}{F\cos\phi}\left(\frac{1}{r1} + \frac{1}{r2}\right)]^{\frac{1}{2}} \tag{4.37}$$

$$\sigma c = -129053.96Pa[\frac{(1.0005)(1213N)}{(0.040)\cos 20^{\circ}}\left(\frac{1}{0.0026m} + \frac{1}{0.0103m}\right)]^{\frac{1}{2}} = -510Mpa$$

ASTM no.20 Compressive strength, σc_{all} is 572 MPa (shigley's, 2008)

Here, $\sigma_c < \sigma_{all}$ the material under the tangential loading obtained a compressive stress value 510Mpa and the obtained stress value still below the allowable compressive limit, so the material of ASTM no.20 is safe to use. The material properties also take in to consideration for the designed process. Where, the cast iron ASTM no.60 offers a reasonable resistance against corrosion. In general, the mechanical properties are lower than those of cast or wrought steels, especially when loaded in tension. In compression high loads can be supported.

4.8 GEAR AND PINION STRESS ANALYSIS USING FINITE ELEMENT ANALYSIS

Finite element method used the lattice of line elements for the solution of stresses in this analysis. In the analysis, maximum torque is applied to the gear using 36.404N.m and pinion using 9.101N.m and using material of ASTM no.20 to obtain the compressive stress value that react on the automatic gate mechanism when under operating condition. Here, the analysis also perform by modified the mesh sizing to differentiate the result obtained using finite element with the manual solution that obtained on previous section 4.7. The result obtained also compare with the material allowable compressive stress value.

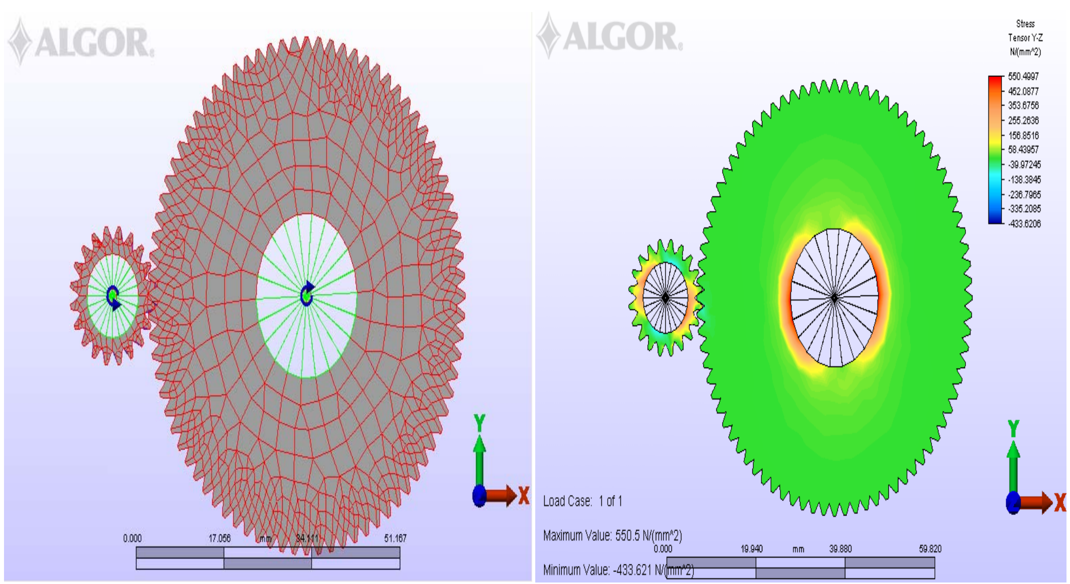


Figure 4.20: Finite element analysis using 100% meshing

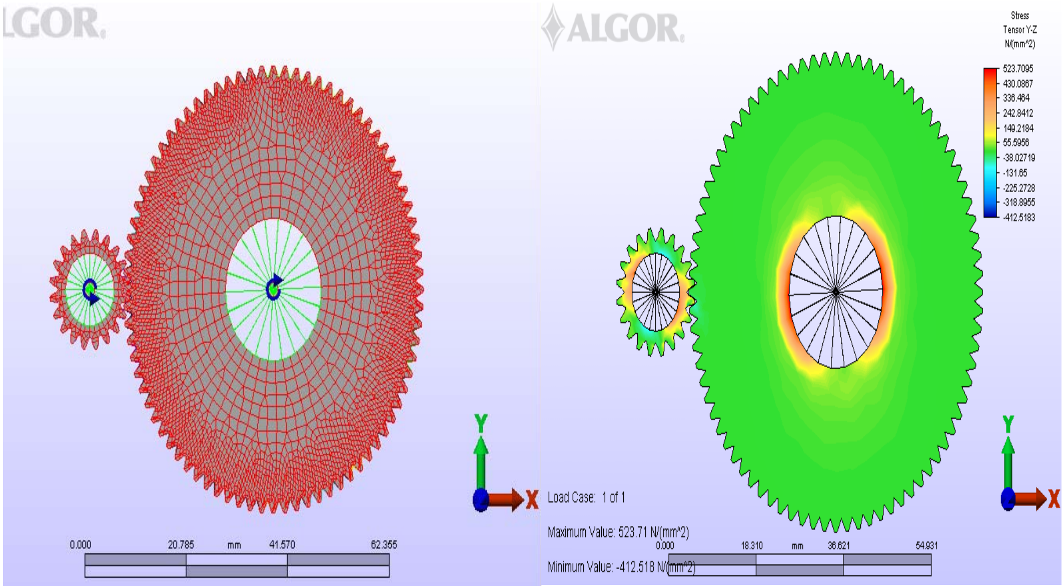


Figure 4.21: Finite element analysis using 70% meshing

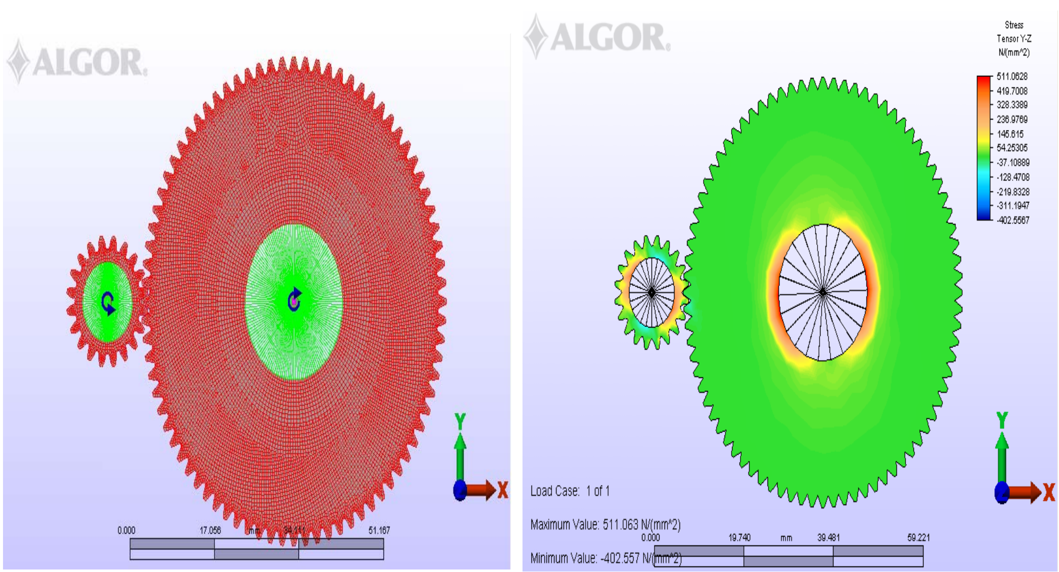
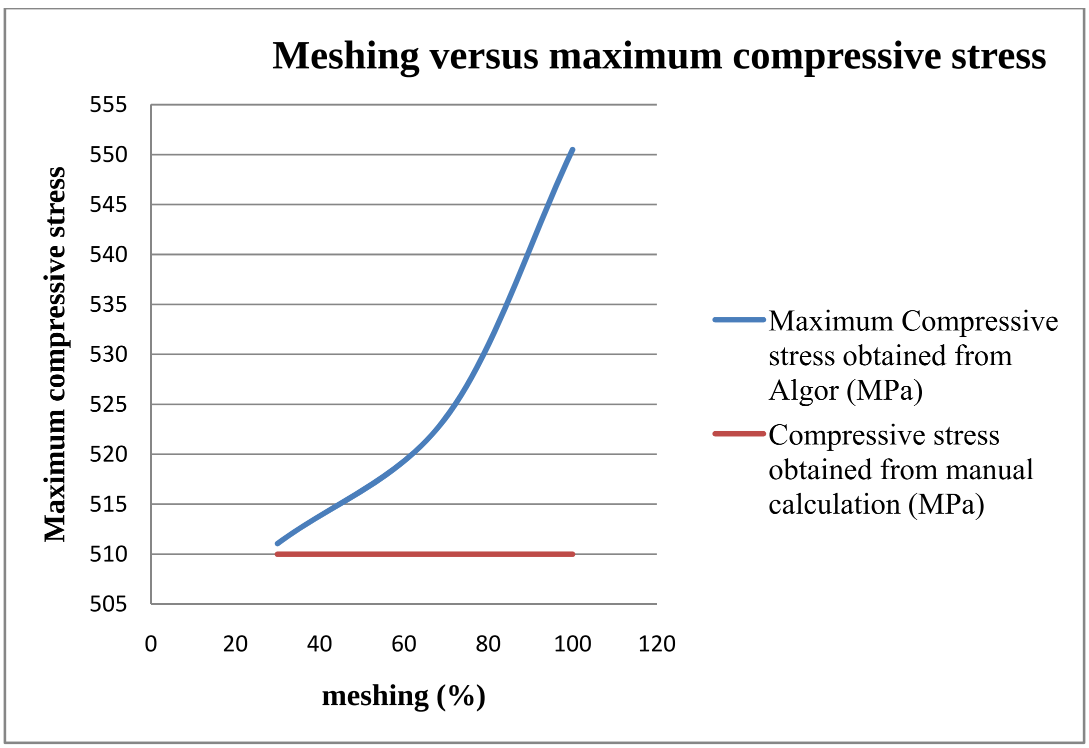


Figure 4.22: Finite element analysis using 30% meshing

Table 4.1: Percentage error between manual calculation and finite element analysis

Mesh density (%)	Maximum Compressive stress obtained from Algor (MPa)	Compressive stress obtained from manual calculation (MPa)	Percentage error (%)
100	550.5	510	$\frac{550.5 - 510}{510} \times 100\%$ $= \; 7.94\%$
70	523.71	510	$\frac{523.71 - 510}{510} \times 100\%$ $= \; 2.69\%$
30	511.06	510	$\frac{511.06 - 510}{510} \times 100\%$ $= \; 0.21\%$



Graph 4.1: Meshing versus compressive stress

Graph 4.1 shown the accuracy of result obtained using Algor is increasing by refined the mesh size of the element in analysis from four to one. Refined mesh includes significantly more element in analysis and with smaller size. This represents improvement in the geometry accuracy in the vicinity of the discontinuity and also solution for accuracy. Here, the most accurate compressive stress value obtained using Finite element analysis where 511.06Mpa is is compare to the ASTM no.20 allowable compressive stress value where is 572Mpa. ASTM no.20 Compressive strength, σc_{all} is 572 MPa (shigley's, 2008). It seems that the value obtained is much smaller compare to allowable compressive stress value so, material ASTM no.20 is safe to use for the gear and pinion design.

4.9 PISTON ARM STRESS ANALYSIS

The piston arm when operate in a retract condition, there will be a tensile force exert on the arm here, analysis is perform to measure whether the material that under maximum tensile force able to resist or withstand without breaking by using the selected material of stainless steel AISI 302. Stainless steel was selected because it has great corrosion resistance properties when exposed to environmental. The stresses over the arm need to be determined due to the gate mechanism operate over long period operating condition and to avoid failure occurred.

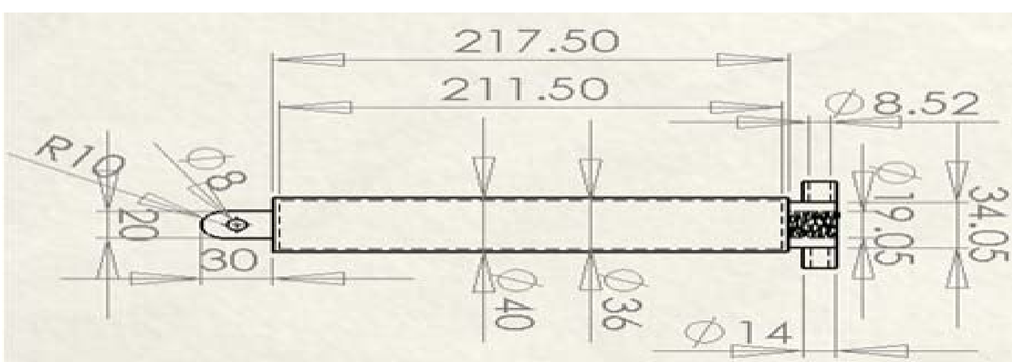


Figure 4.23: Piston arm with full dimension

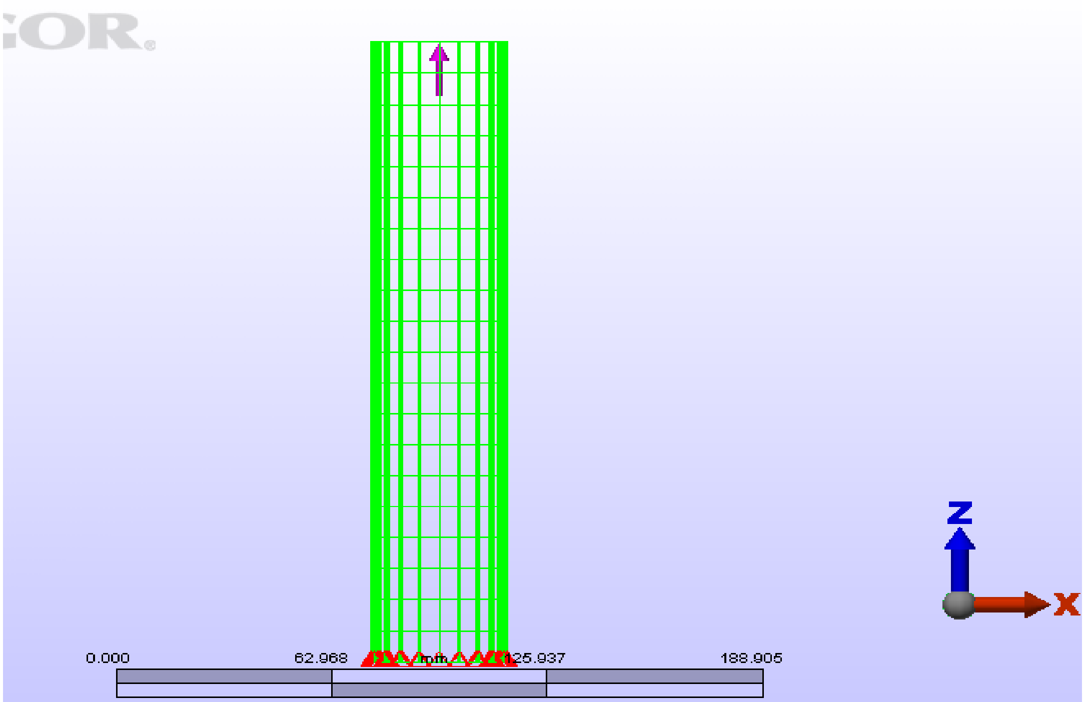


Figure 4.24: Piston arm finite element analysis under tensile load

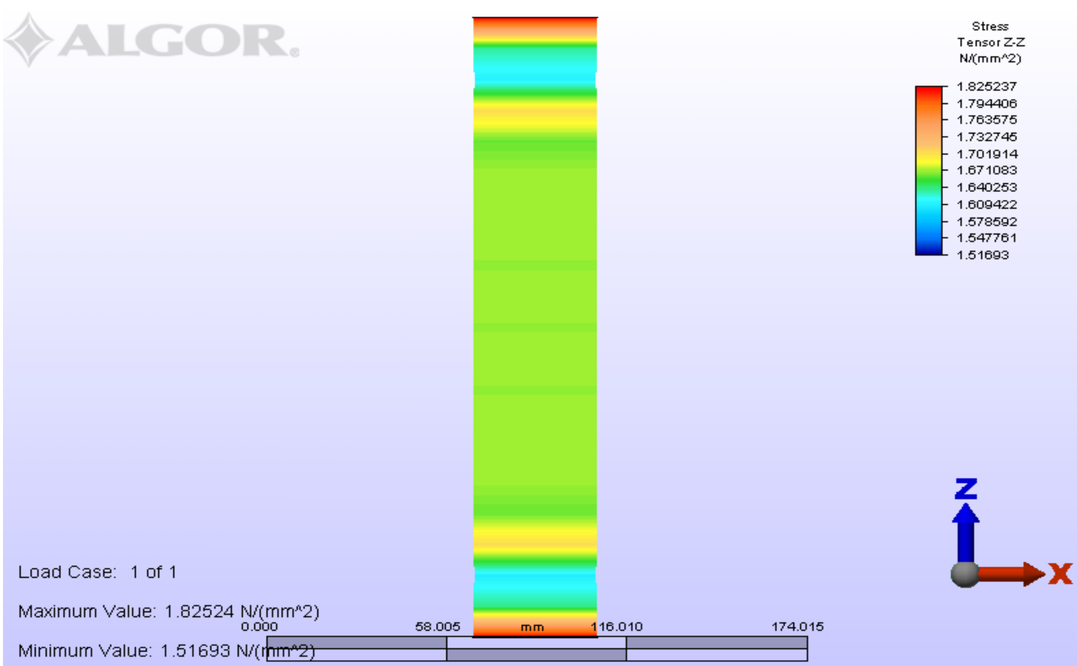


Figure 4.25: Tensile stress finite element analysis for piston arm

Stainless steel AISI 302 for annealed condition, the tensile allowable stress value is 260MPa (Beer, 2006). The tensile stress value obtained using Finite element analysis referring to Figure 4.25 is 1.82524MPa. Here, the value obtained in Finite element analysis seems much lower than the allowable stress value and the material in used is safe. The material selection also depend on the stainless steel AISI 302 material properties, where is the corrosion resistance factor.

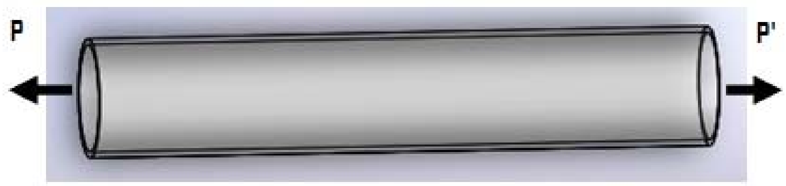


Figure 4.26: Piston arm hollow part (tensile force exert on body where pulling the gate)

Manual calculation for tensile stress over the piston arm can be obtained using Eq. (4.38)

$$\sigma = \frac{P}{A} \tag{4.38}$$

Force required moving the gate that obtained using Eq. (4.7),

P = 392.4N

Cross sectional area of piston arm hollow part can be obtained using Eq. (4.39)

$$A = \left(\frac{\pi(\text{outer diamter})^2}{4}\right) - \left(\frac{\pi(\text{inner diamter})^2}{4}\right) \tag{4.39}$$

By referring to Equation 4.38,

$$\sigma = \frac{P}{A} = \frac{392.4\text{N}}{(\frac{\pi(0.040)^2}{4}) - (\frac{\pi(0.036)^2}{4})} = 1.6435\text{MPa}$$

Tensile stress allowable for stainless steel AISI 302, σ_{all} is 260 MPa (Beer, 2006).$\sigma < \sigma_{all}$, The material under the tangential loading obtained a compressive stress value 510Mpa and the obtained stress still below the allowable compressive limit, so the material is safe to use.

Table 4.2: Percentage error between manual calculation and finite element analysis

Data	Maximum Tensile stress (MPa)	Percentage error (%)
Measured using software (Algor)	1.8254	$\% \text{ error} = \frac{1.8254 - 1.6435}{1.6435} \times 100$ $\% \text{ error} = 11.07\%$
Manual calculation	1.6435	$\% \text{ error} = \frac{1.8254 - 1.6435}{1.6435} \times 100$ $\% \text{ error} = 11.07\%$

From table 4.2, the result obtained using Finite element analysis is 1.8254MPa and result obtained using manual calculation is 1.6435MPa. The percentage error between manual calculation and Finite element analysis is 11.07%. The percentage accuracy can be reducing by increasing the number of element in analysis through refinement.

4.10 COST ANALYSIS

The cost considerations play an important rule in the design decision process. Here the design is based on simplicity so the component required is much lesser compare to market available product. Bill of material is prepared in this section and by comparing to market available product prices and make sure the product design for unit price is affordable by customer.

Figure 4.27: Fabricated automatic gate mechanism

Table 4.3: Bill of material

Part	Description	Quantity	Price per unit (RM)	Total unit price (RM)
Body top cover and bottom cover	Material in use Carbon steel grade ASTM- A36 (1200mmx 1500mm x 2mm)	1	70.00	70.00
ACME lead screw	Total Length: 315mm, Thread length: 175mm, Pitch: 3/4 inch, Diameter pitch 06424inch. Material stainless steel 302	1	85.00	85.00
Roller slider	High density polyethylene	2	4.20	8.40
Motor	Dc motor with following specification: torque 9.101N.m,speed 400rpm,and power 381.227 watt	1	175.00	175.00
Gear	Gear type spur gear, pitch diameter 60mm ,ratio pinion: gear 1:2, pinion to gear center 37.5mm, material ASTM no 20	1	50.00	50.00
Pinion	Gear type spur gear, pitch diameter 15mm, material ASTM no 20	1	40.00	40.00
single row deep groove ball bearing	Bore diameter 10mm,thickness 9mm, outer diameter 30mm and bore diameter 15mm,thickness 11mm and outer diameter 35mm	2	25.00	50.00
Screw (to hold roller slider)	Major diameter:4mm, Pitch:07mm Pitch diameter: 3.5453367mm	2	0.50	1.00
Piston arm	Material Carbon steel grade stainless steel AISI 302 (thickness 2mm)	1	30.00	30.00
Receiver and transmitter	/	1	100	100
TOTAL				609.4

Cost comparison in between market product and design product prices can be obtained using Eq. (4.40)

$$\text{Cost save} = \text{market product price} - \text{designed product price} \qquad (4.40)$$

$$\text{Cost save} = \text{RM}1400 - \text{RM}609.4$$

$$\text{Cost save} = \text{RM}790.6$$

From the survey through related automatic gate mechanism shop lot, the unit average price for automatic gate mechanism is RM1400. The design estimated or calculated price obtained is RM609.4, by comparing the market available product price with designed price, it seem that the price is much lower and customer could purchase the product with RM790.60 of cost saving.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

5.1 INTRODUCTION

In this chapter it will include with the overall summary regarding the design of automatic gate mechanism project. The results or outcomes of act or process will be concluded as the final judgment reached after consideration. Recommendations are also mentioned in this chapter for future improvement or development for this project.

5.1 CONCLUSION

As a conclusion, mechanical engineering design theories and finite element analysis is used in this project to solve several problems. There are two important factors had taken into consideration for the design process. Those factors are Failure prevention and design of mechanical element. Several analyses are performed to select suitable or appropriate material and standard sizing of component for this automatic gate mechanism. After the material and standard sizing of component is selected, bill of material is listed out and compared with prices of the product available in the market. This is done to ensure that the designed product has a reasonable price and within the customer affordable range. From the cost analysis that is done, the mechanism price is much cheaper than those available in the market with RM 790.60 of cost reduction and more people will afford to install one of this automatic gate mechanism. The product designed is also much safer and reliable because all of the critical parts for this mechanism is analyzed under maximum loaded or stresses condition. From the above argument, the overall objective for this project has been achieved successfully.

5.2 RECOMMENDATIONS

In order to improve the design of automatic gate mechanism, more features should be added to the design concept for example, voice remote, keypad, several type of sensor and mechanism auto unlock system. The vibration of the mechanism also needs to be analyzed to prevent some unpleasant noises to occur during operating condition. Normally, the vibration is caused by the gearing system of the mechanism.

REFERENCES

Automated Gates Ltd. 2009. Automatic Gate Design, US Patent 5984330

David H Myszka. 2005. MACHINES AND MECHANISMS Applied Kinematic Analysis third Edition. Prentice Hall.

Ferdinand P.Beer. 2006. MECHANICS OF MATERIALS Fourth Edition in SI Units. Mc Graw Hill.

MAG Malaysia Automatic Gate. 2005.Cubic Under Ground System, US Patent 505445

Moleykutty George. 2008. “Speed Control of Separately Excited DC Motor”. American Journal of Applied Sciences 5 (3): 227-233.

Parker Hannifin Corporation. 2008. Screw drive piston operator, US Patent 4904437

Prestige Door and Gates Ltd. 2008. Gate system, US Patent 5904336

R.C.Hibbeler. 2007. Engineering Mechanics Dynamics 11th edition in SI units. Prentice Hall.

Richard G Budynas shigley’s. 2008. Mechanical Engineering Design. Mc Graw Hill.

Robert L Mott. 2004. Machine Element in Mechanical Design Fourth Edition. Prentice Hall.

Shoewu, O. and O.T. Baruwa. 2006. “Design of a Microprocessor Based Automatic Gate”. Pacific Journal of Science and Technology. 7(1):31-44.

V2 Automatismos. 2007. Auto Gate System US Patent 5544008

APPENDIX A (Standard ACME Thread Dimensions Table)

Table 12.3 Standard ACME Thread Dimensions

Nominal Major Diameter (in.)	Threads per inch, n	Pitch (in.) $p = \frac{1}{n}$	Nominal Pitch Diameter (in.)
1/4	16	0.0625	0.2043
5/16	14	0.0714	0.2614
3/8	12	0.0833	0.3161
7/16	12	0.0833	0.3783
1/2	10	0.1000	0.4306
5/8	8	0.125	0.5408
3/4	6	0.1667	0.6424
7/8	6	0.1667	0.7663
1	5	0.2000	0.8726
1 1/8	5	0.2000	0.9967
1 1/4	5	0.2000	1.1210
1 3/8	4	0.2500	1.2188
1 1/2	4	0.2500	1.3429
1 3/4	4	0.2500	1.5916
2	4	0.2500	1.8402
2 1/4	3	0.3333	2.0450
2 1/2	3	0.3333	2.2939
2 3/4	3	0.3333	2.5427
3	2	0.5000	2.7044
3 1/2	2	0.5000	3.2026
4	2	0.5000	3.7008
4 1/2	2	0.5000	4.1991
5	2	0.5000	4.6973

APPENDIX B (Typical Properties of Selected Material Used in Engineering)

Appendix B. Typical Properties of Selected Materials Used in Engineering[1,5]

Material	Density kg/m³	Ultimate Strength: Tension, MPa	Ultimate Strength: Compression,[2] MPa	Ultimate Strength: Shear, MPa	Yield Strength[3]: Tension, MPa	Yield Strength[3]: Shear, MPa	E Modulus of Elasticity, GPa	G Modulus of Rigidity, GPa	Coefficient of Thermal Expansion, $10^{-6}/°C$	Ductility, Percent Elongation in 50 mm
Steel										
Structural (ASTM-A36)	7860	400			250	145	200	77.2	11.7	21
High-strength-low-alloy										
ASTM-A709 Grade 345	7860	450			345		200	77.2	11.7	21
ASTM-A913 Grade 450	7860	550			450		200	77.2	11.7	17
ASTM-A992 Grade 345	7860	450			345		200	77.2	11.7	21
Quenched & tempered										
ASTM-A709 Grade 690	7860	760			690		200	77.2	11.7	18
Stainless, AISI 302										
Cold-rolled	7920	860			520		190	75	17.3	12
Annealed	7920	655			260	150	190	75	17.3	50
Reinforcing Steel										
Medium strength	7860	480			275		200	77	11.7	
High strength	7860	620			415		200	77	11.7	
Cast Iron										
Gray Cast Iron										
4.5% C, ASTM A-48	7200	170	655	240			69	28	12.1	0.5
Malleable Cast Iron										
2% C, 1% Si, ASTM A-47	7300	345	620	330	230		165	65	12.1	10
Aluminum										
Alloy 1100-H14 (99% Al)	2710	110		70	95	55	70	26	23.6	9
Alloy 2014-T6	2800	455		275	400	230	75	27	23.0	13
Alloy-2024-T4	2800	470		280	325		73		23.2	19
Alloy-5456-H116	2630	315		185	230	130	72		23.9	16
Alloy 6061-T6	2710	260		165	240	140	70	26	23.6	17
Alloy 7075-T6	2800	570		330	500		72	28	23.6	11
Copper										
Oxygen-free copper (99.9% Cu)										
Annealed	8910	220		150	70		120	44	16.9	45
Hard-drawn	8910	390		200	265		120	44	16.9	4
Yellow-Brass (65% Cu, 35% Zn)										
Cold-rolled	8470	510		300	410	250	105	39	20.9	8
Annealed	8470	320		220	100	60	105	39	20.9	65
Red Brass (85% Cu, 15% Zn)										
Cold-rolled	8740	585		320	435		120	44	18.7	3
Annealed	8740	270		210	70		120	44	18.7	48
Tin bronze (88 Cu, 8Sn, 4Zn)	8800	310			145		95		18.0	30
Manganese bronze (63 Cu, 25 Zn, 6 Al, 3 Mn, 3 Fe)	8360	655			330		105		21.6	20
Aluminum bronze (81 Cu, 4 Ni, 4 Fe, 11 Al)	8330	620	900		275		110	42	16.2	6

(Table continued on page 747)

APPENDIX C (Results of Tensile Tests of Some Metals Table)

Table A–20

Results of Tensile Tests of Some Metals* *Source:* J. Datsko, "Solid Materials," chap. 32 in Joseph E. Shigley, Charles R. Mischke, and Thomas H. Brown, Jr. (eds.-in-chief), *Standard Handbook of Machine Design,* 3rd ed., McGraw-Hill, New York, 2004, pp. 32.49–32.52.

			Strength (Tensile)					
Number	**Material**	**Condition**	**Yield S_y, MPa (kpsi)**	**Ultimate S_u, MPa (kpsi)**	**Fracture, σ_f, MPa (kpsi)**	**Coefficient σ_0, MPa (kpsi)**	**Strain Strength, Exponent m**	**Fracture Strain ε_f**
1018	Steel	Annealed	220 (32.0)	341 (49.5)	628 (91.1)†	620 (90.0)	0.25	1.05
1144	Steel	Annealed	358 (52.0)	646 (93.7)	898 (130)†	992 (144)	0.14	0.49
1212	Steel	HR	193 (28.0)	424 (61.5)	729 (106)†	758 (110)	0.24	0.85
1045	Steel	Q&T 600°F	1520 (220)	1580 (230)	2380 (345)	1880 (273)†	0.041	0.81
4142	Steel	Q&T 600°F	1720 (250)	1930 (210)	2340 (340)	1760 (255)†	0.048	0.43
303	Stainless steel	Annealed	241 (35.0)	601 (87.3)	1520 (221)†	1410 (205)	0.51	1.16
304	Stainless steel	Annealed	276 (40.0)	568 (82.4)	1600 (233)†	1270 (185)	0.45	1.67
2011	Aluminum alloy	T6	169 (24.5)	324 (47.0)	325 (47.2)†	620 (90)	0.28	0.10
2024	Aluminum alloy	T4	296 (43.0)	446 (64.8)	533 (77.3)†	689 (100)	0.15	0.18
7075	Aluminum alloy	T6	542 (78.6)	593 (86.0)	706 (102)†	882 (128)	0.13	0.18

*Values from one or two heats and believed to be attainable using proper purchase specifications. The fracture strain may vary as much as 100 percent.

†Derived value.

APPENDIX D (Size Factor Table)

0.999 0.75

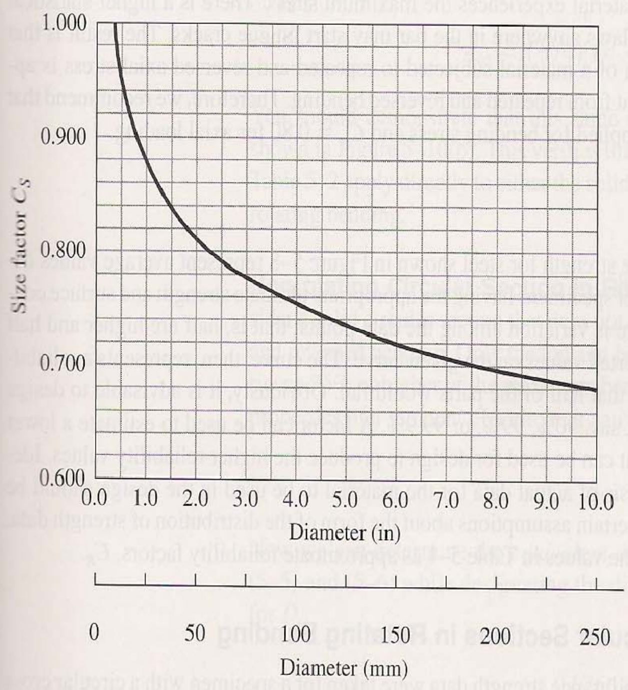


FIGURE 5–9 Size factor

TABLE 5–2 Size factors

Size Range	For D in inches
U.S. customary units	
$D \leq 0.30$	$C_S = 1.0$
$0.30 < D \leq 2.0$	$C_S = (D/0.3)^{-0.11}$
$2.0 < D < 10.0$	$C_S = 0.859\text{-}0.02125D$
SI units	
Size Range	For D in mm
$D \leq 7.62$	$C_S = 1.0$
$7.62 < D \leq 50$	$C_S = (D/7.62)^{-0.11}$
$50 < D < 250$	$C_S = 0.859\text{-}0.000837D$

APPENDIX E (Reliability Factors Table)

TABLE 5–1
Approximate reliability factors, C_R

Desired reliability	C_R
0.50	1.0
0.90	0.90
0.99	0.81
0.999	0.75

APPENDIX F (Comparison of Bearing Types Table)

TABLE 14–1 Comparison of bearing types

Bearing type	Radial load capacity	Thrust load capacity	Misalignment capability
Single-row, deep-groove ball	Good	Fair	Fair
Double-row, deep-groove ball	Excellent	Good	Fair
Angular contact	Good	Excellent	Poor
Cylindrical roller	Excellent	Poor	Fair
Needle	Excellent	Poor	Poor
Spherical roller	Excellent	Fair/good	Excellent
Tapered roller	Excellent	Excellent	Poor

APPENDIX G (Recommended Design Life for Bearings Table)

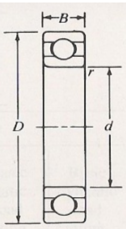
TABLE 14–4 Recommended design life for bearings

Application	Design life L_{10}, h
Domestic appliances	1000–2000
Aircraft engines	1000–4000
Automotive	1500–5000
Agricultural equipment	3000–6000
Elevators, industrial fans, multipurpose gearing	8000–15 000
Electric motors, industrial blowers, general industrial machines	20 000–30 000
Pumps and compressors	40 000–60 000
Critical equipment in continuous, 24-h operation	100 000–200 000

Source: Eugene A. Avallone and Theodore Baumeister III, eds., *Marks' Standard Handbook for Mechanical Engineers,* 9th ed. New York: McGraw-Hill, 1986.

APPENDIX H (Bearings Selection Data Table)

TABLE 14–3 Bearing selection data for single-row, deep-groove, Conrad-type ball bearings



A. *Series* 6200

Bearing number	Nominal bearing dimensions							Preferred shoulder diameter		Bearing weight	Basic static load rating, C_o	Basic dynamic load rating, C
	d		*D*		*B*		*r**	Shaft	Housing			
	mm	in	mm	in	mm	in	in	in	in	lb	lb	lb
6200	10	0.3937	30	1.1811	9	0.3543	0.024	0.500	0.984	0.07	520	885
6201	12	0.4724	32	1.2598	10	0.3937	0.024	0.578	1.063	0.08	675	1180
6202	15	0.5906	35	1.3780	11	0.4331	0.024	0.703	1.181	0.10	790	1320
6203	17	0.6693	40	1.5748	12	0.4724	0.024	0.787	1.380	0.14	1010	1660
6204	20	0.7874	47	1.8504	14	0.5512	0.039	0.969	1.614	0.23	1400	2210
6205	25	0.9843	52	2.0472	15	0.5906	0.039	1.172	1.811	0.29	1610	2430
6206	30	1.1811	62	2.4409	16	0.6299	0.039	1.406	2.205	0.44	2320	3350
6207	35	1.3780	72	2.8346	17	0.6693	0.039	1.614	2.559	0.64	3150	4450
6208	40	1.5748	80	3.1496	18	0.7087	0.039	1.811	2.874	0.82	3650	5050
6209	45	1.7717	85	3.3465	19	0.7480	0.039	2.008	3.071	0.89	4150	5650
6210	50	1.9685	90	3.5433	20	0.7874	0.039	2.205	3.268	1.02	4650	6050
6211	55	2.1654	100	3.9370	21	0.8268	0.059	2.441	3.602	1.36	5850	7500
6212	60	2.3622	110	4.3307	22	0.8661	0.059	2.717	3.996	1.73	7250	9050
6213	65	2.5591	120	4.7244	23	0.9055	0.059	2.913	4.390	2.18	8000	9900
6214	70	2.7559	125	4.9213	24	0.9449	0.059	3.110	4.587	2.31	8800	10 800
6215	75	2.9528	130	5.1181	25	0.9843	0.059	3.307	4.783	2.64	9700	11 400
6216	80	3.1496	140	5.5118	26	1.0236	0.079	3.504	5.118	3.09	10 500	12 600
6217	85	3.3465	150	5.9055	28	1.1024	0.079	3.740	5.512	3.97	12 300	14 600
6218	90	3.5433	160	6.2992	30	1.1811	0.079	3.937	5.906	4.74	14 200	16 600
6219	95	3.7402	170	6.6929	32	1.2598	0.079	4.213	6.220	5.73	16 300	18 800
6220	100	3.9370	180	7.0866	34	1.3386	0.079	4.409	6.614	6.94	18 600	21 100
6221	105	4.1339	190	7.4803	36	1.4173	0.079	4.606	7.008	8.15	20 900	23 000
6222	110	4.3307	200	7.8740	38	1.4961	0.079	4.803	7.402	9.59	23 400	24 900
6224	120	4.7244	215	8.4646	40	1.5748	0.079	5.197	7.992	11.4	26 200	26 900

APPENDIX I (Physical Constants of Materials Table)

†Often preferred.

Table A–5

Physical Constants of Materials

Material	Modulus of Elasticity E		Modulus of Rigidity G		Poisson's Ratio ν	Unit Weight w		
	Mpsi	GPa	Mpsi	GPa		lbf/in^3	lbf/ft^3	kN/m^3
Aluminum (all alloys)	10.4	71.7	3.9	26.9	0.333	0.098	169	26.6
Beryllium copper	18.0	124.0	7.0	48.3	0.285	0.297	513	80.6
Brass	15.4	106.0	5.82	40.1	0.324	0.309	534	83.8
Carbon steel	30.0	207.0	11.5	79.3	0.292	0.282	487	76.5
Cast iron (gray)	14.5	100.0	6.0	41.4	0.211	0.260	450	70.6
Copper	17.2	119.0	6.49	44.7	0.326	0.322	556	87.3
Douglas fir	1.6	11.0	0.6	4.1	0.33	0.016	28	4.3
Glass	6.7	46.2	2.7	18.6	0.245	0.094	162	25.4
Inconel	31.0	214.0	11.0	75.8	0.290	0.307	530	83.3
Lead	5.3	36.5	1.9	13.1	0.425	0.411	710	111.5
Magnesium	6.5	44.8	2.4	16.5	0.350	0.065	112	17.6
Molybdenum	48.0	331.0	17.0	117.0	0.307	0.368	636	100.0
Monel metal	26.0	179.0	9.5	65.5	0.320	0.319	551	86.6
Nickel silver	18.5	127.0	7.0	48.3	0.322	0.316	546	85.8
Nickel steel	30.0	207.0	11.5	79.3	0.291	0.280	484	76.0
Phosphor bronze	16.1	111.0	6.0	41.4	0.349	0.295	510	80.1
Stainless steel (18-8)	27.6	190.0	10.6	73.1	0.305	0.280	484	76.0
Titanium alloys	16.5	114.0	6.2	42.4	0.340	0.160	276	43.4

APPENDIX J (Mechanical properties of Three Non-Steel Metals Table)

Table A–22

Mechanical Properties of Three Non-Steel Metals

(a) Typical Properties of Gray Cast Iron

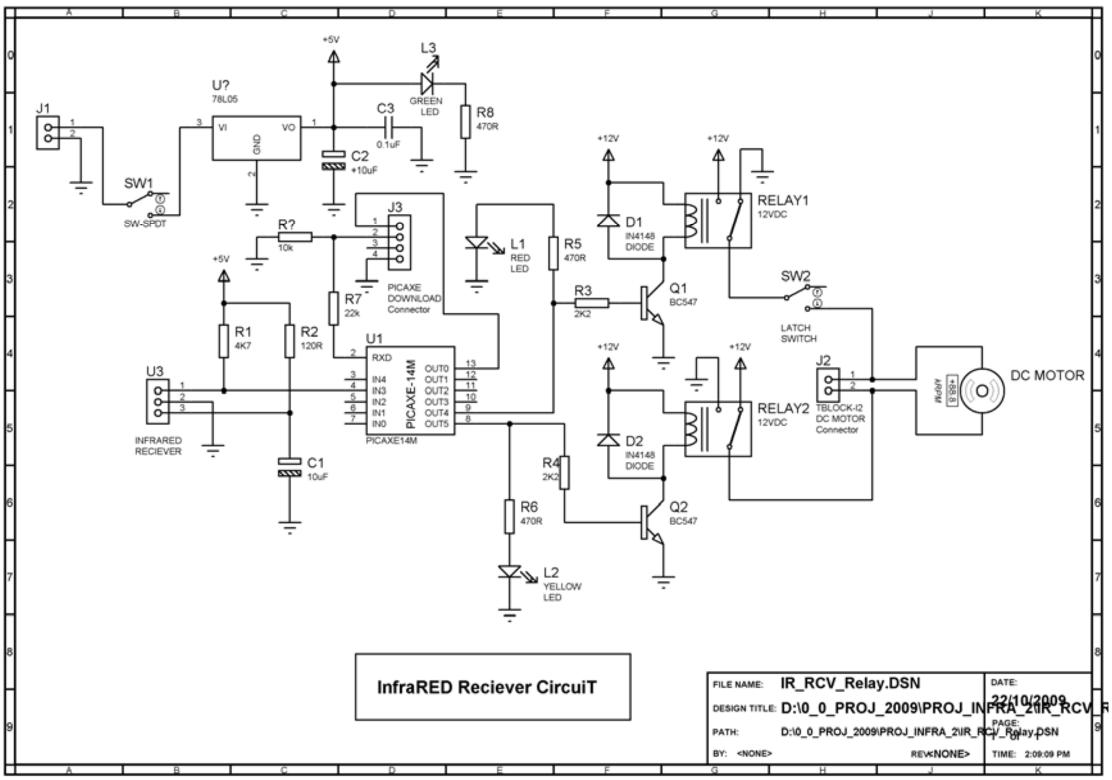
[The American Society for Testing and Materials (ASTM) numbering system for gray cast iron is such that the numbers correspond to the *minimum tensile strength* in kpsi. Thus an ASTM No. 20 cast iron has a minimum tensile strength of 138 MPa. Note particularly that the tabulations are *typical* of several heats.]

ASTM Number	Tensile Strength S_{ut}, MPa	Compressive Strength S_{uc}, MPa	Shear Modulus of Rupture S_{su}, MPa	Modulus of Elasticity, GPa Tension†	Torsion	Endurance Limit* S_e, MPa	Brinell Hardness H_B	Fatigue Stress-Concentration Factor K_f
20	152	572	179	9.6–14	3.9–5.6	69	156	1.00
25	179	669	220	11.5–14.8	4.6–6.0	79	174	1.05
30	214	752	276	13–16.4	5.2–6.6	97	201	1.10
35	252	855	334	14.5–17.2	5.8–6.9	110	212	1.15
40	293	970	393	16–20	6.4–7.8	128	235	1.25
50	362	1130	503	18.8–22.8	7.2–8.0	148	262	1.35
60	431	1293	610	20.4–23.5	7.8–8.5	169	302	1.50

*Polished or machined specimens.

†The modulus of elasticity of cast iron in compression corresponds closely to the upper value in the range given for tension and is a more constant value than that for tension.

APPENDIX K (Schematic Diagram of Infrared Receiver Circuit)



APPENDIX L (Schematic Diagram of Infrared Transmitter Circuit)