

ABSTRACT

This report discusses the background and current research done on the chosen topic, which is design of mechanisms for gantry crane experimental rig. The objective of the project is to come up with a design of mechanisms to drive the gantry cranes motions for an experimental crane rig. The scope of study focused on proposing a design of mechanism for the gantry crane designed for Mechanical Engineering Department of Universiti Teknologi PETRONAS based on the limitation of the gantry frame dimension. Gantry cranes are widely used in shipping terminals to handle freight containers. Advances in controlling the crane load sway has been achieved by conducting experiments in laboratory. However, the problem is the Mechanical Department of Universiti Teknologi PETRONAS does not have the gantry crane test rig for experimental purpose. Technique that is going to be used is proposing design of mechanism for the gantry crane by performing analysis on the crane operations. Justification on fundamentals that determine the gantry frame structural rigidity and flexibility due to the load is also being studied. This technique integrates the critical parameters that affect the performance of the driving mechanisms of the gantry crane. The parameters are being identified and manipulated in order to achieve the best driving mechanism. AutoCAD and ANSYS will be used in modeling and analyzing the gantry crane test rig. A gantry crane test rig will be constructed to perform an experiment to test the validity of the model. An enhanced mechanism of gantry cranes operations is essential for further research work, especially on the development of an automatic crane controller.

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1: INTRODUCTION

1.1: Background

Gantry cranes for experimental purpose are used to study and control the crane load sway while lifting and transporting item around a working area in a laboratory. Some of them are equipped with an enclosed track, while others use I-beams, or other extruded shapes, for the running surface for the trolley.

A gantry crane has a hoist in a fixed machinery house or on a trolley that runs horizontally along rails, usually fitted on a single beam (mono-girder) or two beams (twin-girder). Figure 1.1 below shows the simple structure of a gantry crane.

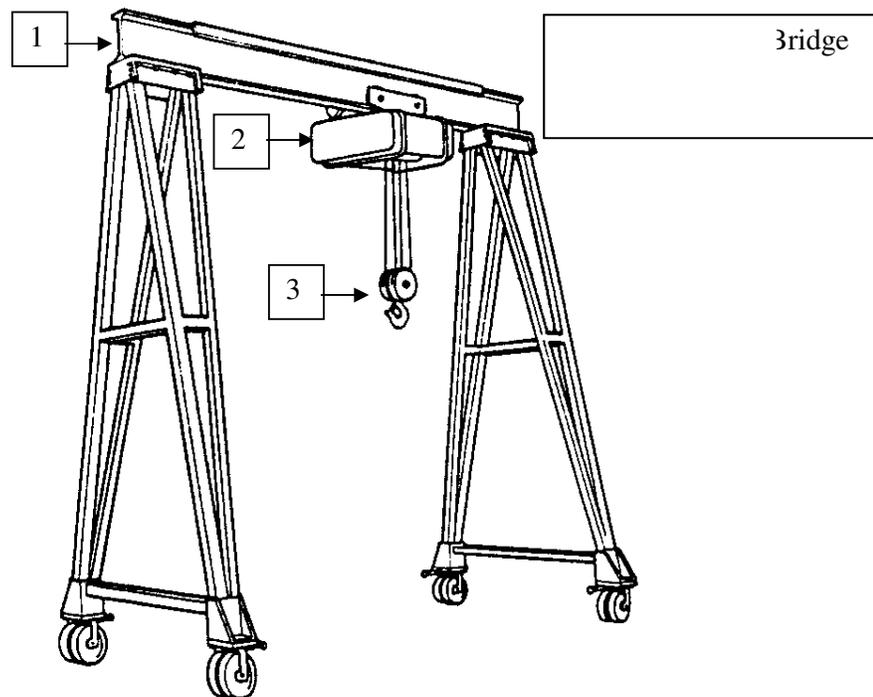


Figure 1.1: The Gantry Crane.

Many experiments and studies on the gantry cranes have been done but their emphasis is more on controlling the amount of sway of the spreader. One of the important components in designing the motion of the gantry cranes is its driving mechanism. Driving mechanism of gantry crane test rig comprises of horizontal motion of trolley along the rail and vertical motion of hoist.

1.2: Problem Statement

The main problem discovered is that there is no specific design of mechanism for the gantry crane designed for Mechanical Engineering Department of Universiti Teknologi PETRONAS. Besides, there will be maintenance problems arise if the department not internally constructs the gantry crane because the gantry crane is currently not available in Malaysia. The department needs to request the manufacturer of the gantry crane to come to the university in order to repair or do maintenance on the gantry crane. This will increase the cost to maintain the gantry crane and it is time consuming which can affect the research activities currently done by the researcher of the department on the gantry crane.

Therefore, there are many problem arise associated to the main problem such as the exact amount of power required to produce the motion of the moving trolley along the three axes which are x -axis, y -axis and z -axis which will determine the amount and type of actuator required to power up the motion of the trolley.

Other problem arise is the suitable cable that will be used to carry and withstand the load in terms of material's characteristics and features. Besides, the suitable material and method of constructing the path for the trolley to travel within the frame has to be specified as well as the parameter such as maximum acceleration of the trolley when travelling along the path of the frame.

These technical problems require various engineering solutions such as analysis on the trolley's profile motion of the crane as well as justification on material that will be used as the cable of the crane and the path for the trolley to travel within the frame.

1.3: Aim and Objective of Project

The aim of the project is to design the mechanism to drive the gantry cranes operations in three axes of motion for an experimental crane rig.

1.4: Scope of Project

The scopes of this project are:

1. To study the current mechanisms of experimental rig gantry crane.
2. To identify the criteria, parameters and requirements needed for the desired mechanisms for the gantry crane.
3. To design the mechanisms that meets the identified requirement.
4. To evaluate the design by making justification based on several alternatives to choose the desired mechanisms.
5. To analyze cost of investment and the ease of maintenance/assemble of the mechanisms.

The project will mostly emphasize on the design of mechanism for gantry crane as well as all the aspects that need to be specified in defining the mechanism such as power required, type and amount of actuator, method of moving the trolley and the hoist, criteria of cable to hold the load and design of track that the trolley will travel within the frame.

Also, this project will focus on the justification on various material and design of cable and the path of the trolley in term of ease of use and ease of maintenance because the crane will be used as an experimental purpose that need to be easily operated.

The project will not emphasize on the design of the gantry crane structure. The project is expected to come out with outputs in the form of a detail design of the mechanisms to drive the experimental rig gantry crane and specifications of actuators and other components involved in the project.

1.5: Significance of Project

Direct benefit that can be obtained from the project is the department will have the design of gantry crane for experimental rig in the laboratory as well as possible motion profiles of the gantry crane. Therefore, the researcher can construct the rig by using the information obtained from this study.

Others benefit obtained is that more research can be done regarding controlling the sway of the load attached to the trolley when move along the gantry frame. The gantry crane also can be used as a teaching tool for the mechanical engineering students at the laboratory.

Indirect benefit can be obtained from this project is a wide range of flexibility according to the requirements specified from the design of the mechanism for the gantry crane experimental rig. Advantage of this in-house construction is the department will not be depending on external expertise in constructing, operating and maintaining the gantry crane. This will reduce the cost of acquiring and maintaining the gantry crane and it is less time consuming which can speed up the research activities currently done by the researcher of the department on the gantry crane.

This project will go deep into the specification of the motion and in the end of this project is there will be a range of motion's profile that obtained and proposed to be the mechanism of the gantry crane for experimental purpose in the mechanical engineering laboratory at Universiti Teknologi PETRONAS.

2: LITERATURE REVIEW AND THEORY

One of the literature reviews done is by A. Zainuddin and P. Hussain. The paper focus on areas such as identify parameters in the motion of the trolley that affect the load's sway and develop motion profiles with the parameters that cause minimum load's sway [1]. The paper also provides details on size of the gantry crane experimental rig which used to test the validity of the models.

This paper gives a clear view on the type of driving mechanism and actuator that is suitable to be used in constructing a gantry crane experimental rig because the paper has successfully constructs a test rig which the specifications are listed as following:

1. The trolley and the crane bridge are driven by DC motors.
2. Utilize a pulley and cable system.
3. Operated with rotary encoders to feedback its velocity and position.
4. Incorporates four cables spread to each corner of the load (which is often called a spreader) suspended to the trolley.

The paper emphasizes more on the controlling the crane load sway instead of designing the mechanism of the gantry crane. The paper has concluded that the simplified linearized model is sufficient to predict a velocity profile of a trolley to generate minimum sway in the payload [1]. This will enhance this paper by incorporating the findings into the project.

The paper applies the technique of developing a mathematical model of the trolley along the gantry bridge with a suspended load. With a few assumptions, a linear model on the sway magnitude of the payload as a function of the trolley motion parameters can be used. By adopting a simple velocity profile of the trolley and manipulating the values of certain trolley motion parameters, minimum sway of the payload can be achieved [1].

Another study that was reviewed is by H. Butler, G. Honderd and J. V. Amerongen. The study presents a new method of “reference model decomposition” as an extension of model reference adaptive control [2]. The study focuses on area like illustrating the decomposition method by presenting an adaptive controller for a scale model of a gantry crane.

Besides, the study has manage to construct a scale model of gantry crane specifically on the cord length which can be varied between 0.63 m and 1.29 m, maximum trolley movements is 2.4m, position accuracy of 0.2 to 0.8 m, maximum load mass is 10 kg and minimum load mass of 5 kg required to keep the cord straightened.

On the other hand, the study has found the limitations of the laboratory scale model with respect to a real gantry crane which is all the physical dimensions are scaled down dramatically, to make the experimental setup fit within a laboratory environment and another limitation is a movements is only possible in two directions which are the trolley can move horizontally over the spanning rails and a motor in the crab can lift the load vertically [2].

The technique of illustrating the decomposition method by presenting an adaptive controller for a scale model of a gantry crane cannot be completely implemented in this study as the technique is only applicable to closed loop system instead of open loop system which practiced in this project.

Another study that reviewed is a PHD thesis by Kuan-chun Huang. The study investigates the dynamics and control of a Rubber Tyred Gantry (RTG) crane which is commonly used in container handling operations. Although simultaneous hoisting/trolley motion plus gantry level is limited for power-supply reasons in full-sized RTG cranes, the proposed experimental crane is purposely designed to be capable of actuating concurrently in those directions; that is, traveling motion of the trolley (x), transverse motion of the gantry (y), and hoisting (l). In addition to this capability, a fourth actuator is also provided to simulate rotational motion of the gantry, θ_v [3].

The experimental crane is to 1/8 scale, based on the dimensions to give the rig representative behavior of a full-sized RTG crane. The general specifications for each axis of the experimental crane rig are summarized in Table 2.1 below.

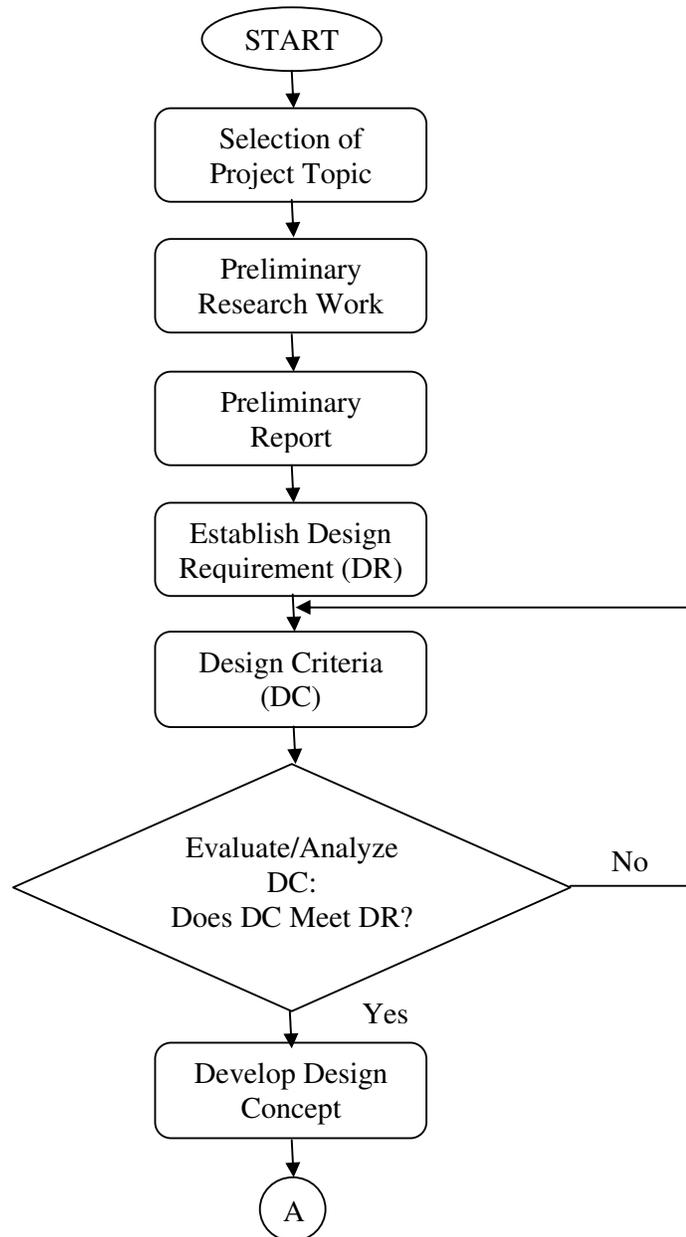
	x- axis (#1)	y- axis (#2)	Rotation (#3)	Hoist (#4)
Span	0 ~ 1500 mm	-500 ~ +500 mm	+/- 90°	700 ~ 1200 mm
Measured maximum speed	0.543 m/s	0.313 m/s	105.6 degree/s	0.255 m/s
Motors	DC permanent magnet	DC permanent magnet	DC permanent magnet	DC permanent magnet
Rails	A pair of linear sliding guide-rails	A pair of linear sliding guide-rails	NA	NA
Drive actuation	A steel cable connecting a drum mounted on the motor and a pulley at the far end of the rails	(cable/drum/pulley) Same as the x- axis	Experimental crane rotates the trolley about its vertical (central) axis	Rotate the hoist drum which drives the spreader underneath

Table 2.1: General Specifications of the Drives on the Experimental Rig.

In the study, the design and function of the experimental crane rig is illustrated. Mechanical and electrical aspects of the design for the experimental crane rig are discussed. This will enhance this paper by incorporating the findings into the project.

3: METHODOLOGY AND PROJECT WORK

In this project, there are tasks which have to be completed in order to achieve the aim and objectives of the project. The processes are briefly described in the process flow chart below:



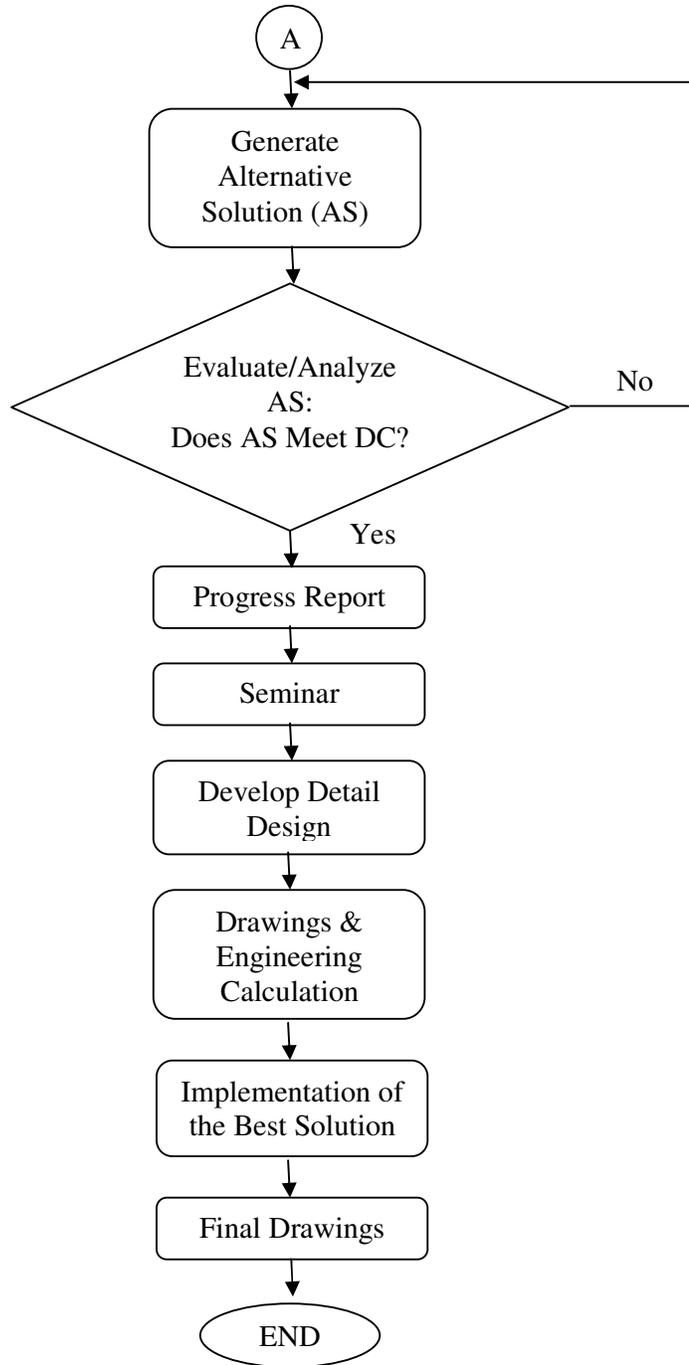


Figure 3.1: FYP Process Flowchart.

The timeline and brief description of each stage is shown in the Gantt chart attaches in Appendix A.

4: RESULT AND DISCUSSION

The detail about activities and parameters for each process which has been described in the process flowchart is elaborated as following:

4.1 Analyze problem & review literature

The main problem for this project, the main problem is there is no specific design of mechanisms for the gantry crane. Some related problems associated to the main problem are power required by the motor to move the trolley, type of material that will be used to construct the experimental rig gantry crane and range of resolution that will be posed by the encoder in order to give accurate readings on the distance travelled by the trolley and its speed.

Most published works, papers, journals, conferences and thesis to date are focusing more on crane dynamics instead of the mechanisms of the crane but there are some specifications of experimental rig gantry crane stated in the reviews. This will enhance this project by incorporating the findings into the model.

4.2 Establish Design Requirement

There are several motion requirement of the experimental gantry crane as specified by the researchers. These are shown in the Table 4.1.

Maximum traveling length (x)	1000 mm
Maximum transverse length (y)	1000 mm
Maximum hoisting length (ℓ)	1000 mm
Maximum speed (x-axis)	0.5 m/sec
Maximum speed (y-axis)	0.3 m/sec
Maximum speed (ℓ -axis)	0.5 m/sec
Type of encoder (all axis)	Rotary
Encoder's resolution (all axis)	± 0.5 mm

Table 4.1: Requirements for Experimental Rig Gantry Crane.

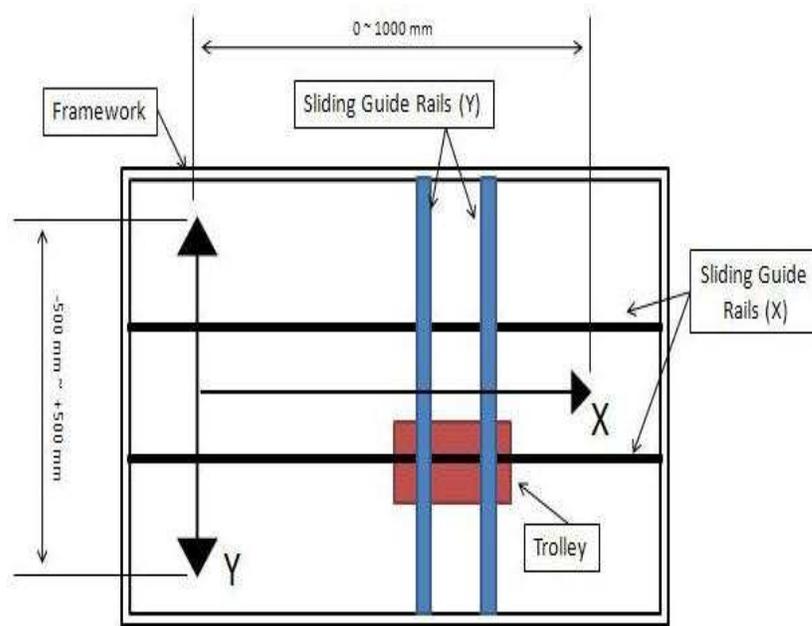


Figure 4.1: A Plan View of the Experimental Crane Rig.

The common physical parameters for the experimental rig which given by the researcher are as follows:

The mass of the gantry	8 Kg
The mass of the trolley (including the hoisting devices such as motor, drives and reeling drum etc.)	3 Kg
The mass of the spreader	2 Kg
The mass of the load	2 Kg

Table 4.2: Common Physical Parameters for Experimental Rig Gantry Crane.

Design criteria can be defined as criteria that designers should meet in designing some system or device. The gantry crane experimental rig in this project will consist of **three major components** which are as the following:

- i. Driving Mechanism
- ii. Actuator and Encoder
- iii. Guiding System

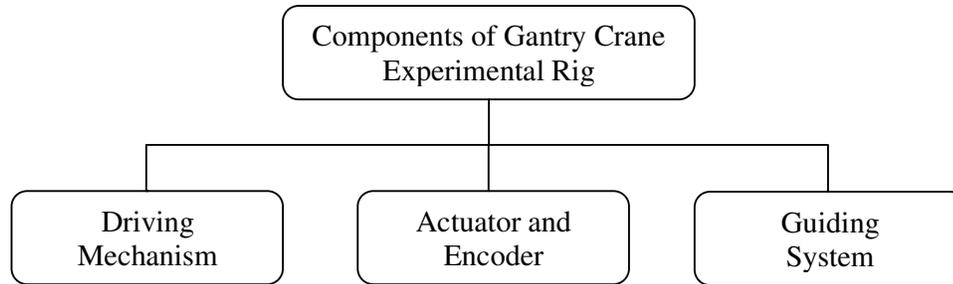


Figure 4.2: Components of Gantry Crane Experimental Rig

Each component will have its own design criteria but in this project, the **general design criteria** of the driving mechanism are as follows:

- i. Cost effective – design concept must be selected from the least capital and operational cost.
- ii. Ease of maintenance/assembly – design concept must be selected from the least complex assembling and maintaining way.
- iii. High accuracy and precision – encoders must meet the requirement which is ± 0.5 mm.
- iv. Reliability and durability – design concept must be selected from the alternative solutions of a system or component to perform its required functions under stated conditions for a specified period of time

Design criteria for each component in this project are determined to be as follows:

- i. Driving Mechanism
 - a. Smoothness and performance – driving mechanism must be selected from least friction produced from the motion of trolley in x-axis.
 - b. Repairable – driving mechanism must be selected from the hardware components of a gantry crane that are designated for repair.
 - c. Load capacity – driving mechanism must be selected from the alternative solutions which can withstand the 2 kg load.

- ii. Actuator and Encoder
 - a. Precision positioning – actuator and encoder must be selected from the most accurate and precise positioning
 - b. High holding torque – actuator must be selected from the alternative solutions which can encounter the 2 kg load.

- iii. Guiding System
 - a. Smoothness and performance – guiding system must be selected from the least friction produced from the motion of trolley in x-axis.
 - b. Travel life – guiding system must be selected from the alternative solutions which can withstand a long service time.

4.3 Develop Design Concept

Generate alternative solutions based on the design criteria. **Alternative solutions** are the solutions that allowing or necessitating a choice between two or more things. In this project, the alternative solutions for the gantry frame are:

- i. Driving Mechanism
 - a. Belt and pulley system
 - b. Chain and sprocket system
 - c. Cable and pulley system

- ii. Actuator and Encoder
 - a. Brushless DC motor (built-in encoder)
 - b. Stepper motor (built-in encoder)
 - c. AC motor (built-in encoder)

- iii. Guiding System
 - a. Shaft guiding
 - b. Profile rail guides
 - c. Precision rail guides

4.4 Evaluate / Analyze Design Concepts

The alternative solutions are being evaluated and analyzed based on the design criteria stated in the design requirements section. Justification on each alternative solution will be made to choose only one solution that is the most effective solution in term of costs, accuracy and ease of maintenance/assemble. Some engineering calculation and assumptions will be made to assist in making the decision on which is the best solution chosen at the end of the justification step.

For simplicity, the following **assumptions** are made:

1. The elasticity of the crane structure elements, dissipation effects like rolling resistance and losses in the drive mechanism, and effect such as wind forces, are neglected.
2. The load is assumed to be concentrated at a point and hanging at the end of a massless cable with negligible length changes due to load swing.
3. The rotation of the lumped mass m_s about its own mass centre is ignored.

4.4.1 Actuator & Encoder

As for the actuators, the justification was undertaken by calculating the relevant gains using the physical limit of the actuators (Table 4.3) and the maximum spans (Table 4.4) of the experimental rig given by the researchers. Justifications on the physical limit of the actuators can assist in setting up the gains correctly so as not to exceed the capacity of the actuators therefore useful in determining the best actuators to be used in the Experimental Rig Gantry Crane.

The maximum linear velocity is determined according to the following equation which is,

$$V_{max} = \omega \cdot r \quad \dots \dots \dots (4.1)$$

Where,

$$\omega = \frac{rpm \cdot 2\pi}{60} \quad \dots \dots \dots (4.2)$$

Therefore,

From Table 4.3, we substitute the value of the rpm and the drum diameter into the equation above to obtain the value of V_{\max} . The **maximum linear velocity** of the actuator is recorded into the Table 4.3 below:

Actuator	Max. rpm with 180 V DC input voltage setup (rpm)	Existing input voltage setup (volts)	Existing max. rpm (rpm)	Drum diameter (meter)	Max. linear velocity (m/sec)
X-axis	300	60	100	0.05	0.7854
Y-axis	120	90	60	0.05	0.3142
Hoist	120	90	60	0.05	0.3142

Table 4.3: The Physical Limit of Actuator Velocities.

After establishing the physical limit of actuator velocities, the next step is to select a motor type from the alternatives solution of actuator and encoder in the design concept development based on the required specification which listed below.

- i. Brushless DC motor (built-in encoder)
- ii. Stepper motor (built-in encoder)
- iii. AC motor (built-in encoder)

 BLDC motor	 Stepper motor	 AC motor
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Table 4.4: The Alternative Solution of Actuator.

Justification is made by comparing all the characteristic of alternative solution that comply with the required specification shown in the table below.

	Cost (initial & operational)	Holding torque	Ease of Maintenance	Lifespan	Total
Weight	4	3	2	1	
BLDC motor	8	9	9	6	83
Stepper motor	7	7	7	9	72
AC motor	9	8	8	6	82

Table 4.5: The Decision Matrix to Select the Best Actuator.

From the table shown above, the best actuator which calculated to be the highest score is **Brushless DC motor** based on the required characteristic of the actuator times the weight of each responsible characteristic.

After establishing the actuator in the previous section, the next step is to calculate the Frictional Torque in order to determine the most suitable capacity of gearbox/motor for the experimental rig.

From equation below,

$$\tau = \begin{bmatrix} \frac{F_x}{m_s} \\ \frac{F_y}{m_s} \\ \frac{F_\ell}{m_s} \\ 0 \\ 0 \end{bmatrix}$$

For example, if torque, T_x , is provided by the x – axis actuator with drum radius, r_1 , then,

$$T_x = F_x \cdot r_1 \quad \dots \dots \dots (4.5)$$

Therefore,

—

From the equation (4.6), it is understood that the torque exerted by a motor is the actual variable control input of the experimental rig at that point. However, as mentioned in previous section, the motors are actually speed (or voltage) controlled. Thus it is necessary to find a relationship between the input voltage to a motor drive module and the torque output from that relevant motor.

The representative free-body diagram for a motor-drive axis is as shown in Figure 4.3,

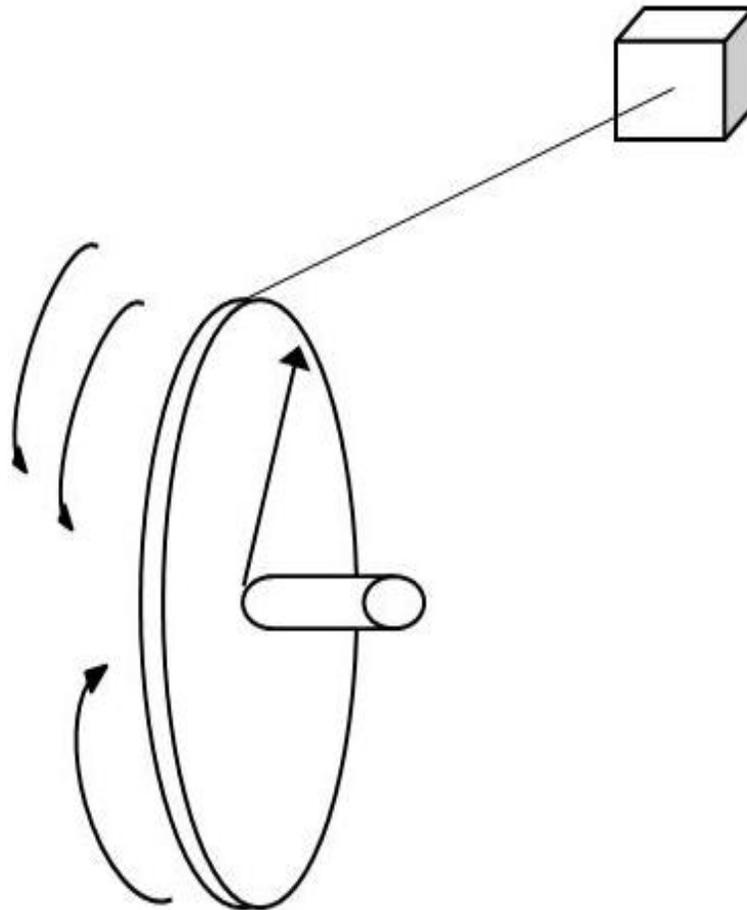


Figure 4.3: Free-body Diagram for a Motor-drive Axis.

The equation of motion is,

$$I_{eq,n}\ddot{\theta}_n + B_n = T_n \quad \dots \dots \dots (4.7)$$

Where;

$I_{m,n}$: The mass moment of inertia of the actuator including the motor, gear box, drum. n = 1, 2, 3, 4.

I_n : The mass moment of inertia of the rest of the moving components of the drive on that axis. n = 1, 2, 3, 4.

$I_{eq,n}$: The equivalent mass moment of inertia of the axis. $I_{eq,n} = I_n + I_{m,n}$

Because the mechanism in Figure 4.3 is no longer in steady state conditions (i.e.it is accelerating), then the equation will become,

$$I_{eq,n}\ddot{\theta}_n = T_{d,n} \quad \dots \dots \dots (4.8)$$

With the angular acceleration, $\ddot{\theta}_n$, can be assumed to be given by,

$$\ddot{\theta}_n = \frac{\dot{\theta}_{t1+\Delta t} - \dot{\theta}_{t1}}{\Delta t} \quad \dots \dots \dots (4.9)$$

Where Δt is the period of time needed for the actuator to reach another steady-state with angular velocity, $\dot{\theta}_{t1+\Delta t}$.

From equation (4.8),

$$\ddot{\theta}_n = \frac{T_{d,n}}{I_{eq,n}} \quad \dots \dots \dots (4.10)$$

On substituting equation (4.9) into equation (4.10),

$$\dot{\theta}_{t1+\Delta t} - \dot{\theta}_{t1} = \frac{T_{d,n}}{I_{eq,n}} \Delta t \quad \dots \dots \dots (4.11)$$

Also it can be seen that,

$$\dot{\theta}_{n,t1\Delta t} = K_{\theta,n} V_{in(n,t1+\Delta t)} + \dot{\theta}_{offset,n} \quad \dots \dots \dots (4.12)$$

$$\dot{\theta}_{n,t1} = K_{\theta,n} V_{in(n,t1)} + \dot{\theta}_{offset,n} \quad \dots \dots \dots (4.13)$$

So, substituting equations (4.12) and (4.13) into equation (4.11) gives the following,

$$V_{in(n,t1+\Delta t)} = V_{in(n,t1)} + \frac{T_{d,n}}{I_{eq,n} K_{\theta,n}} \Delta t \quad \dots \dots \dots (4.14)$$

Equation (4.14) shows a relationship between the desired torque, $T_{d,n}$, and the voltage, $V_{in(n,t1+\Delta t)}$, required to obtain it. In order to make use of the torque-to-voltage conversion, it is necessary to measure the mass moments of inertia of the x, y and ℓ axes of the experimental rig.

First the combined damping and frictional torque term, B_n , in equation (4.7) is assumed to be reasonably represented by,

$$B_n = C_n \dot{\theta}_n + T_{f,n} \quad \dots \dots \dots (4.15)$$

Where the damping coefficient, C_n and the frictional torque $T_{f,n}$ are both taken to be constants.

From Figure 4.3 a mass, m , is attached to the end of the cable wrapped around the pulley (no-slip conditions assumed throughout), the position as shown in Figure 4.4, then the equation of motion is,

Where,

So equation (4.16) becomes,

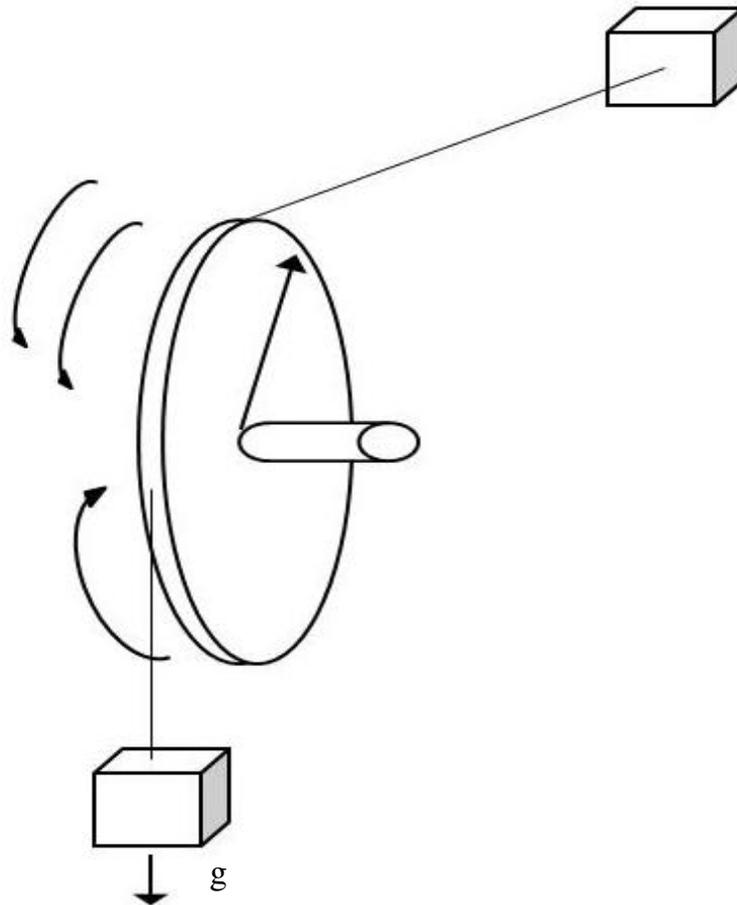


Figure 4.4: The mechanism for measuring the combined torque term B_n .

$T_{f,n}$ can be measured by actual measurement which is by increasing gradually the weight, mg , until the mechanism just starts to move at a very slow speed (< 0.1 radian/sec) so that the effect of $I_{eq,n}\ddot{\theta}_n$ and $C_n\dot{\theta}_n$ in equation (4.17) can largely be ignored. Because $T_{f,n}$ is a dynamic frictional torque, the mechanism in Figure 4.4 must be pushed slightly each time to overcome the static frictional torque threshold.

Therefore,

$$T_{f,n} = mgr_n \quad \dots \dots \dots (4.18)$$

Substitute the value of $m = 2$ Kg and $g = 9.81$ ms⁻² into the equation 4.18. Then, substitute the respective value of r_n of each axis into the equation to obtain the respective value of $T_{f,n}$ for each axis.

Factor of safety (FoS) known as the fraction of structural capability over that required, or a multiplier applied to the maximum expected load (frictional torque) to which the actuator of the gantry crane assembly will be subjected. In this case the FoS is 3 according to the design requirement set by the researcher. The actual frictional torque after multiplying the measured frictional torque, $T_{f,n}$ with the FoS are shown in Table 4.6.

	x - axis	y - axis	ℓ - axis
Frictional Torque, T_f	2.5 Nm x 3 = 7.4 Nm ($T_{f,1}$)	4.4 Nm x 3 = 13.2 Nm ($T_{f,2}$)	0.98 Nm x 3 = 3.0 Nm ($T_{f,4}$)

Table 4.6: The measured result for $T_{f,n}$.

By further increasing the weight, mg , beyond the frictional torque, the mechanism in Figure 4.3 obviously starts moving appreciably. In order to move the load, the actuator must be able to overcome the frictional torque so that the load can be moved to the desired location of the experimental rig.

Therefore, after making the justification aided with the engineering calculation and some assumptions, this project proposes the actuators of each axis which has the optimal requirements needed to function for the gantry crane which shown in the Table 4.7.

Axis	Frictional Torque M_o [Nm]	Standstill Current I_o [A]	Nominal Speed at Rated Supply Voltage			Rotor Inertia J [kg cm ²]
			230 V AC n [1/min]	400 V AC n [1/min]	480 V AC n [1/min]	
x	11.4	4.77	1000	2000	2400	9.1
y	16.5	4.5	-	1200	1500	24
ℓ	4.34	3	-	1500	2000	6.2

Table 4.7: The Proposed Capacity of Actuators for Each Axis of Gantry Crane.

This project also proposes the required dimensions of the actuators which based on the capacity of the respective actuators of each axis. The schematic diagram which shows the configuration of the dimensions is shown in the Figure 4.5 and the value of the dimension is shown in the Table 4.8.

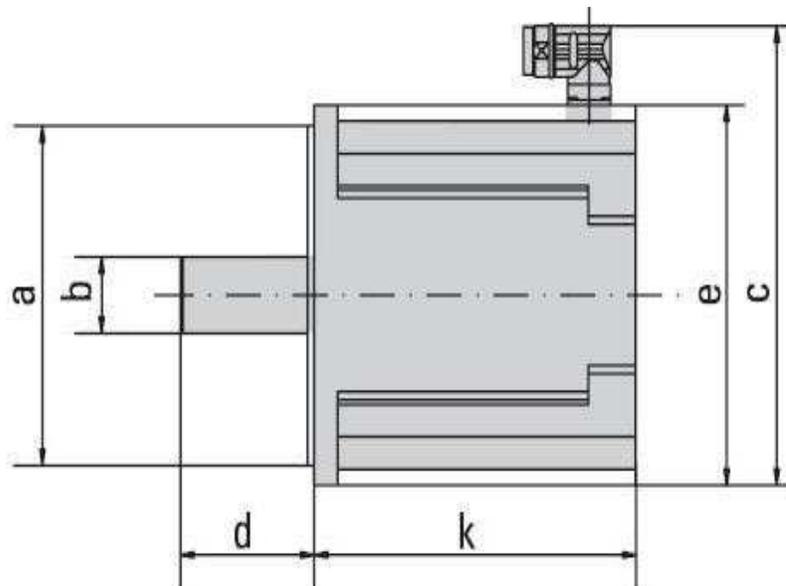


Figure 4.5: Dimensions Configuration of the Actuator.

Axis	a/mm	b/mm	c/mm	d/mm	e/mm	k/mm
x	110	24	147	50	108	189.5
y	130	32	177	58	138	178.7
ℓ	110	24	147	50	108	158.5

Table 4.8: The Proposed Dimension of Actuators of Gantry Crane.

4.4.2 Guiding System

As for the guiding system, there are several design requirements and physical limitations of the guiding system which given by the researcher shown in the table below.

Axis	Stroke, s (mm)	Load, W (N)	Travel Speed (m/s)	Rail's Profile	Number of Ball Circuits, n	Required Life, L_h (hours)	Frequency, f (cycle per minute)
x	1000	88.3	0.7854	Round	4	3500	100
y	1000	49.1	0.3142	Round	5	3500	100

Table 4.9: The Design Requirements & Physical Limitation of the Guiding System.

There are some additional features must be taken into consideration in selecting the suitable guiding system. As for guiding system for x-axis and y-axis, the features are listed below.

- i. Ball Bushing Twin Pillow Blocks (Open Type) for x-axis.
- ii. Ball Bushing Pillow Blocks (Closed & Adjustable Type) for y-axis.
- iii. Available with standard double acting integral seals.
- iv. Can be adjusted to take out diametrical clearance.
- v. Easily mounted and secured with four mounting bolts.
- vi. Available with standard lubrication access.

After establishing the physical limit of actuator velocities in the previous section, the next step is to select a guiding system from the alternatives solution of guiding system in the design concept development based on the required specification which listed below.

- i. Shaft guiding
- ii. Profile rail guides
- iii. Precision rail guides



Table 4.10: The Alternative Solution of Guiding System.

The selection was undertaken by calculating the relevant high score by using the decision matrix. Justification is made by comparing all the characteristic of alternative solution that comply with the required specification shown in the table below.

	Cost (initial & operational)	Sealing Performance	Simplicity	Stroke	Total
Weight	4	3	2	1	
Shaft Guiding	9	9	9	8	89
Precision Rail Guides	7	8	7	9	75
Profile Rail Guides	6	7	8	7	68

Table 4.11: The Decision Matrix to Select the Best Guiding System.

From the table shown above, the best guiding system which calculated to be the highest score is **Shaft Guiding** based on the required characteristic of the guiding system times the weight of each responsible characteristic.

After establishing the guiding system in the previous section, the next step is to calculate the Ball Bushing Bearing Life Expectancy and Load Capacity in order to determine the most suitable capacity of guiding system for the experimental rig.

From equation below,

$$W_R = \frac{P}{K_0 \cdot K_S \cdot K_L} \quad \dots \dots \dots (4.19)$$

Where;

W_R : Required dynamic load capacity (lb_f or N)

P : Resultant of externally applied loads (lb_f or N)

K_0 : Factor for direction or resultant load

K_S : Shaft hardness factor

K_L : Load correction factor

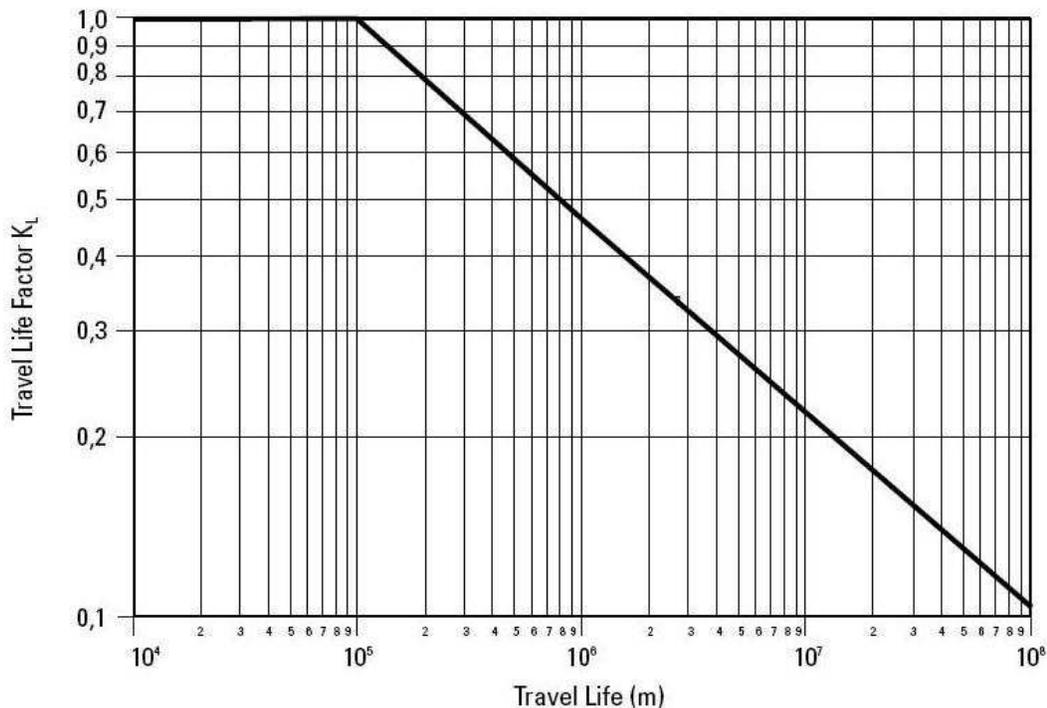


Figure 4.6: Travel Life Chart.

The load correction factor, K_L , can be found from Figure 4.6. To determine K_L , for the gantry crane required travel life, look for the value on the horizontal axis – Travel Life Factor – left side of the chart. Interpolate as necessary because it is a Log-Log curve. That is the value of the load correction factor.

This project currently is using shaft hardness (Rockwell HRC) with value of 60. For shafts that do not meet 60, shaft hardness factor K_s must be applied. To determine K_s , simply refer to Figure 4.7 with the shaft Rockwell hardness, find the value on the horizontal axis – Shaft Hardness – bottom of the chart. Move vertically up until it intersects the curve. Then move horizontally until it reaches the vertical axis – Shaft Correction Factor – left side of chart.

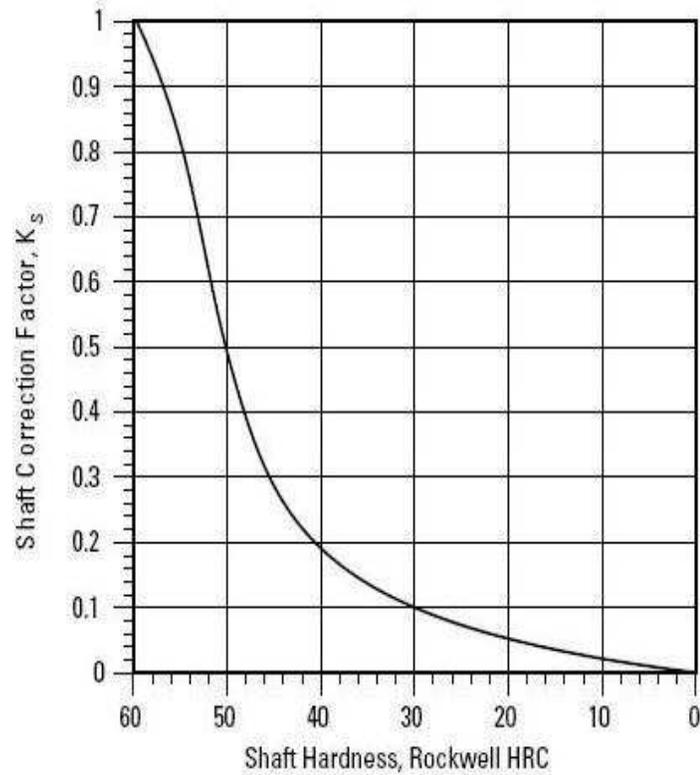


Figure 4.7: Shaft Hardness Chart.

4.4.2.1 The x – axis Guiding System

In order to determine the correct Ball Bushing Bearing size for this axis, some calculation will be made by using all the information extracted from the previous two charts. From the Design Requirements & Physical Limitation of the Guiding System section, the bearing/shaft system is subjected to a load of 88.3 N perpendicular to the direction of travel. The load is distributed equally among four closed type ball bushing bearing. The carriage reciprocates over a 1.0 m stroke at a frequency of 100 complete cycles per minute. The minimum service life required is 3500 hours.

The first step is to determine the average load on each ball bushing bearing.

$$P = \frac{W}{n} = \frac{88.3 \text{ N}}{4} = 22.1 \text{ N} \quad \dots \dots \dots (4.20)$$

Next, determine the equivalent travel life in meters:

$$L_m = 2 \cdot s \cdot f \cdot L_h \cdot 60 \quad \dots \dots \dots (4.21)$$

Where;

L_m : Required travel life (m)

s : Stroke (m)

f : Frequency in cycles per minute

L_h : Service life (hours)

On substituting all the value into equation (4.21),

$$\begin{aligned} L_m &= 2 \cdot s \cdot f \cdot L_h \cdot 60 \\ L_m &= 2 \cdot 1.0 \cdot 100 \cdot 3500 \cdot 60 \\ L_m &= 4.2 \cdot 10^7 \text{ m} \end{aligned}$$

From Figure 4.6 (Travel Life Chart), the travel life factor (K_L) is 0.15.

From Figure 4.7 (Shaft Hardness Chart), the shaft hardness factor (K_S) is 1.

For closed type ball bushing bearings, the minimum value of K_0 is 1, the assumed value for this calculation.

The required dynamic load capacity is obtained by using the following formula:

$$W_R = \frac{P}{K_L \cdot K_S \cdot K_0} \quad \dots \dots \dots (4.22)$$

On substituting all the value into equation (4.22),

$$W_R = \frac{22.1}{0.15 \cdot 1 \cdot 1} = 147.3 \text{ N} / 33.11 \text{ lb}$$

Therefore, after making the justification aided with the engineering calculation and some assumptions, this project proposes the guiding system of x - axis which has the optimal requirements needed to function for the gantry crane which shown in the Table 4.12 and the assembly diagram of the guiding system is shown in Figure 4.8.

Nominal Diameter (inches)	Minimum Depth of Hardness	Support Rail Mass (lb/in)	Pillow Block Mass (lb)	Dynamic Load Capacity (lb _f)
.5	.04	.06	.46	720

Table 4.12: The Proposed Capacity of Guiding System for x - axis of Gantry Crane.

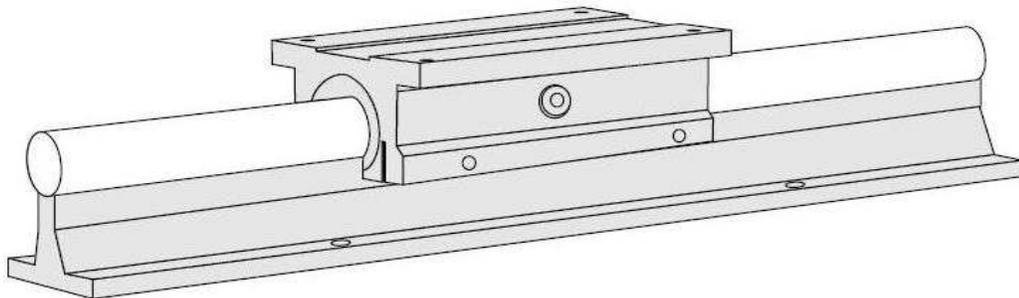


Figure 4.8: Proposed Guiding System for x-axis of Gantry Crane.

This project also proposes the required dimensions of the actuators which based on the capacity of the respective actuators of each axis. The schematic diagram which shows the configuration of the dimensions is shown in the Figure 4.9 and the value of the dimension is shown in the Table 4.13.

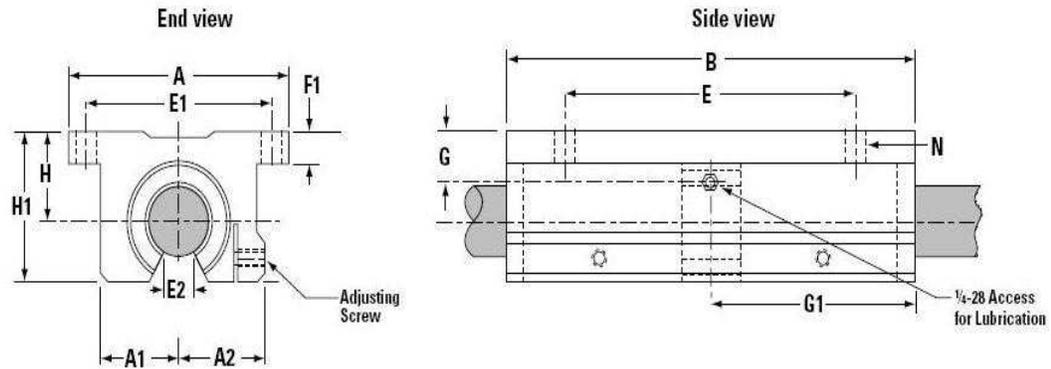


Figure 4.9: Dimensions Configuration of Guiding System (x-axis).

A	A1	A2	B	E	E1	E2	F1	G	G1	H	H1
				±.010	±.010	min.				±.003	
2	.69	.75	3.5	2.5	1.688	.31	.25	.56	1.75	.687	1.13

(Dimension in inches)

Table 4.13: The Proposed Dimension of Guiding System for x- axis of Gantry Crane.

4.4.2.2 The y – axis Guiding System

In order to determine the correct Ball Bushing Bearing size for this axis, some calculation will be made by using all the information extracted from the previous two charts. From the Design Requirements & Physical Limitation of the Guiding System section, the bearing/shaft system is subjected to a load of 49.1 N perpendicular to the direction of travel. The load is distributed equally among four closed type ball bushing bearing. The carriage reciprocates over a 1.0 m stroke at a frequency of 100 complete cycles per minute. The minimum service life required is 3500 hours.

The first step is to determine the average load on each ball bushing bearing.

$$P = \frac{W}{n} = \frac{49.1 N}{5} = 9.8 N \quad \dots \dots \dots (4.20)$$

Next, determine the equivalent travel life in meters:

$$L_m = 2 \cdot s \cdot f \cdot L_h \cdot 60 \quad \dots \dots \dots (4.21)$$

Where;

L_m : Required travel life (m)

s : Stroke (m)

f : Frequency in cycles per minute

L_h : Service life (hours)

On substituting all the value into equation (4.21),

$$\begin{aligned} L_m &= 2 \cdot s \cdot f \cdot L_h \cdot 60 \\ L_m &= 2 \cdot 1.0 \cdot 100 \cdot 3500 \cdot 60 \\ L_m &= 4.2 \cdot 10^7 \text{ m} \end{aligned}$$

From Figure 4.6 (Travel Life Chart), the travel life factor (K_L) is 0.15.

From Figure 4.7 (Shaft Hardness Chart), the shaft hardness factor (K_S) is 1.

For closed type ball bushing bearings, the minimum value of K_0 is 1, the assumed value for this calculation.

The required dynamic load capacity is obtained by using the following formula:

$$W_R = \frac{P}{K_L \cdot K_S \cdot K_0} \quad \dots \dots \dots (4.22)$$

On substituting all the value into equation (4.22),

$$W_R = \frac{9.8}{0.15 \cdot 1 \cdot 1} = 65.3 \text{ N} / 14.68 \text{ lb}$$

Therefore, after making the justification aided with the engineering calculation and some assumptions, this project proposes the guiding system of y - axis which has the optimal requirements needed to function for the gantry crane which shown in the Table 4.14 and the assembly diagram of the guiding system is shown in Figure 4.10.

Nominal Diameter (inches)	Minimum Depth of Hardness	Support Rail Mass (lb/in)	Pillow Block Mass (lb)	Dynamic Load Capacity (lb _f)
.25	.027	.01	.10	60

Table 4.14: The Proposed Capacity of Guiding System for y - axis of Gantry Crane.

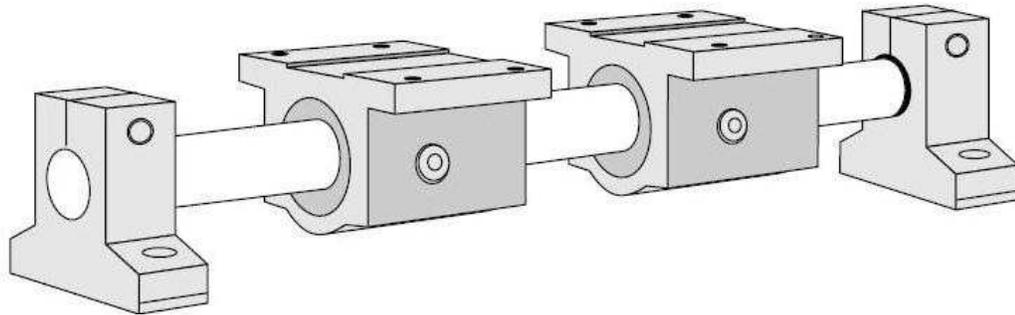


Figure 4.10: Proposed Guiding System for y-axis of Gantry Crane.

This project also proposes the required dimensions of the actuators which based on the capacity of the respective actuators of each axis. The schematic diagram which shows the configuration of the dimensions is shown in the Figure 4.11 and the value of the dimension is shown in the Table 4.15.

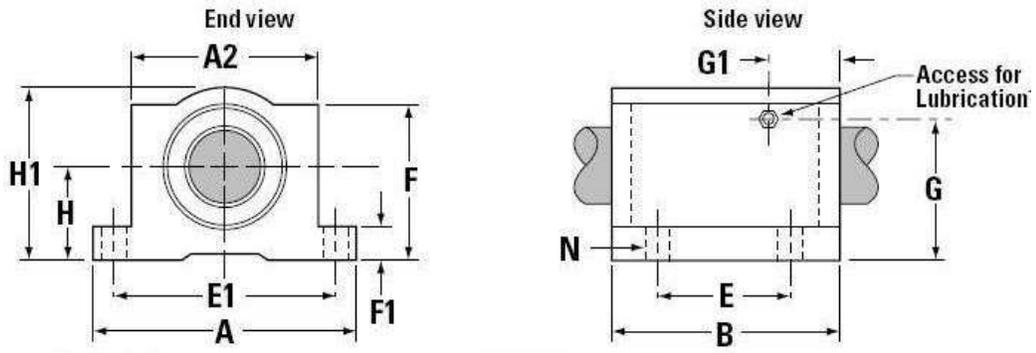


Figure 4.11: Dimensions Configuration of Guiding System (y-axis).

A	A2	B	E ±.010	E1 ±.010	F	F1	G	G1	H ±.003	H1
1.63	1	1.19	.75	1.313	.75	.19	.60	.41	.437	.81

(Dimension in inches)

Table 4.15: The Proposed Dimension of Guiding System for y- axis of Gantry Crane.

4.4.3 Driving Mechanism

As for the driving mechanism, there are several design requirements and physical limitations of the driving mechanism which given by the researcher shown in the table below.

Axis	Max System Capacity (kg)	Max System Weight (kg)	Coefficient of Friction
x	2	.5	0.14
y	2	.5	0.14

Table 4.16: The Design Requirements & Physical Limitation of the Driving Mechanism.

There are some additional features must be taken into consideration in selecting the suitable guiding system. As for driving mechanism for x-axis and y-axis, the features are listed below.

- i. **Low coefficient of friction** - A rating of only 0.14 reduces friction, wear, vibration, downtime and maintenance.
- ii. **Abrasion Resistance** – Selected driving mechanism outlasts other driving mechanisms and most metals under abrasive conditions.
- iii. **Toughness** - It has the highest impact resistance of all thermoplastics.

After establishing the physical limit of actuator velocities in the previous section, the next step is to select a guiding system from the alternatives solution of guiding system in the design concept development based on the required specification which listed below.

- i. Belt and pulley system
- ii. Chain and sprocket system
- iii. Cable and pulley system

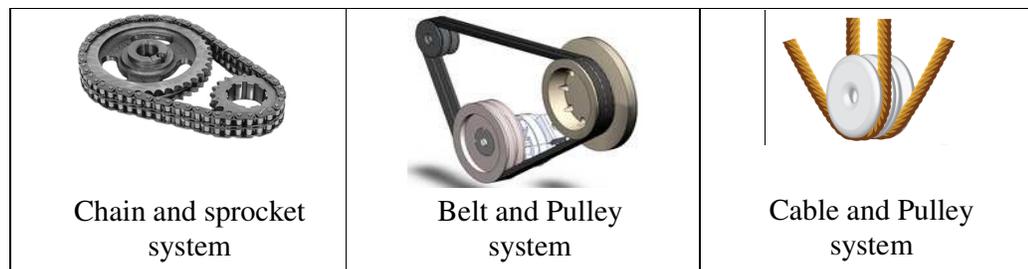


Table 4.17: The Alternative Solution of Driving Mechanism.

The selection was undertaken by calculating the relevant high score by using the decision matrix. Justification is made by comparing all the characteristic of alternative solution that comply with the required specification shown in the table below.

	Cost (initial & operational)	Medium Speed Capability	Interlocking Performance	Ease of Maintenance	Total
Weight	4	3	2	1	
Cable & Pulley system	9	8	7	9	83
Belt & Pulley system	7	8	8	8	76
Chain & sprocket system	5	7	9	7	66

Table 4.18: The Decision Matrix to Select the Best Driving Mechanism.

From the table shown above, the best guiding system which calculated to be the highest score is **Cable & Pulley System** based on the required characteristic of the guiding system times the weight of each responsible characteristic.

After establishing the guiding system in the previous section, the next step is to propose the driving mechanism of both axis which has the optimal requirements needed to function for the gantry crane which shown in the Table 4.19 and the assembly diagram of the driving mechanism is shown in Figure 4.12.

Materials	Electroplated	Bushings	Lubrication
Stainless Steel	Zinc Electroplated	Bronze bushings – designed for high load and medium speed	Permanent Self Lubrication, Oil- impregnation

Table 4.19: The Proposed Capacity of Driving Mechanism for Both axis of Gantry Crane.



Figure 4.12: Proposed Driving Mechanism for Both axis of Gantry Crane

This project also proposes the required dimensions of the driving mechanism which based on the capacity of the respective driving mechanism of each axis. The schematic diagram which shows the configuration of the dimensions is shown in the Figure 4.13 and the value of the dimension is shown in the Table 4.20.

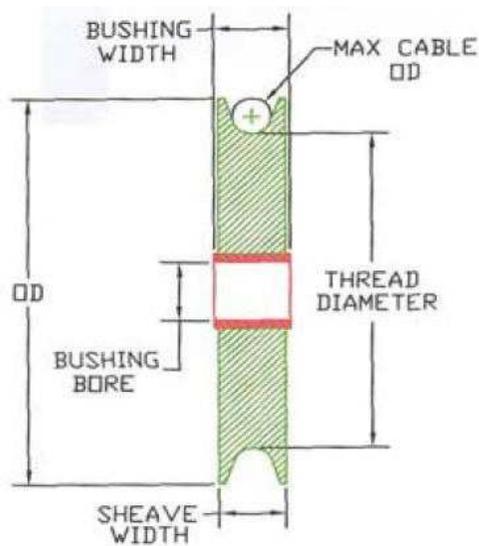


Figure 4.13: Dimensions Configuration of Driving Mechanism

Max Cable Size	OD	Bushing Bore	Sheave Width	Thread Diameter	Bushing Width	Groove Depth
.19	2	.5	.44	1.5	.47	.21

(Dimension in inches)

Table 4.20: The Proposed Dimension of Driving Mechanism for Both axis of Gantry Crane.

4.5 Develop Detail Design

The experimental crane is based on dimensions to give the rig representative behavior of a full-sized RTG crane. The rig is designed to be capable of operating either manually or fully automatically using the electrical drives and controls. Based on the physical limit of actuator velocities which determined from the calculation and the maximum spans of the experimental rig that is given by the researchers, we can describe the mechanical aspects of the experimental crane rig. The aspects can be divided into three main areas as following:

4.5.2 The x – axis

The x – axis is representative of the direction of trolley along the top-beam of a full-sized crane. One pair of linear sliding guide-rails provides smooth linear movement with a maximum span of 1000 mm (0 to 1000 mm). A DC permanent magnet gearbox/motor is used to drive the subassembly underneath the rails along the x – axis. This subassembly includes the y – axis sliding guide rails and the trolley as shown in Figure 4.4. A steel cable, connecting a drum mounted on the motor and a pulley at the far end of the rails, provides the drive actuation.

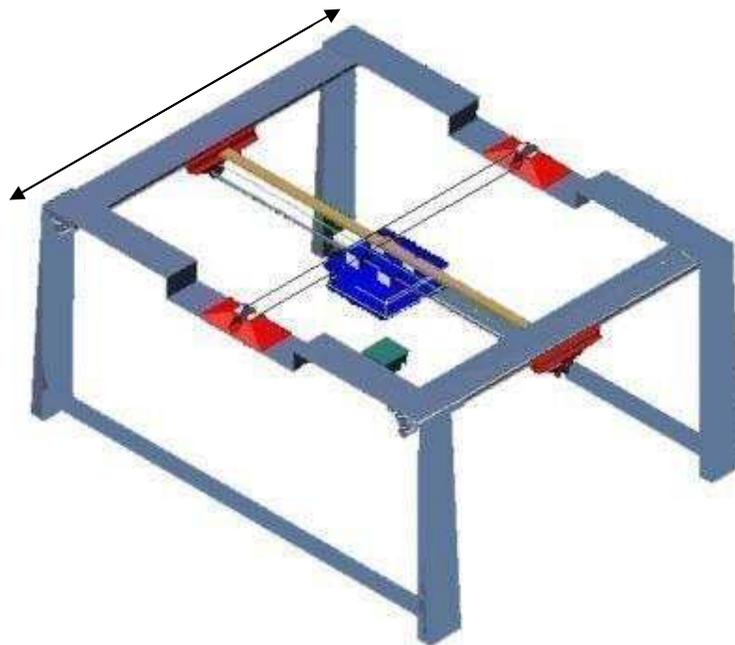


Figure 4.14: Direction of x – axis Movement of Gantry Crane

4.5.3 The y – axis

The y – axis represents the direction of gantry motion in a full-sized crane. Another pair of linear sliding guides provides smooth linear movement with a maximum span of 1000 mm (-500 mm to 500 mm). A DC permanent magnet gearbox/motor is used to drive the trolley underneath the rails along the y – axis as shown in Figure 4.4. The trolley is actuated by a similar drive system (cable/drum/pulley) as for the x – axis.

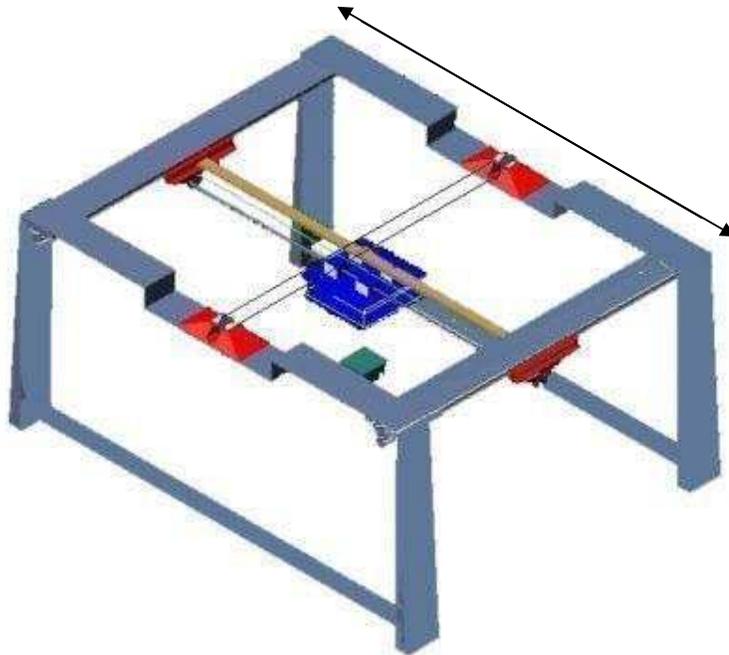


Figure 4.15: Direction of y – axis Movement of Gantry Crane

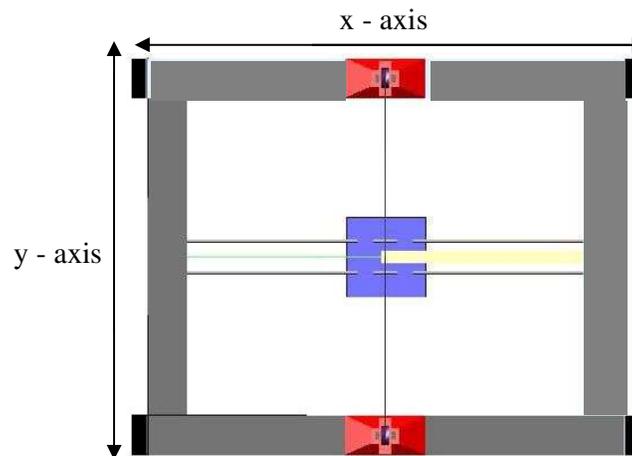


Figure 4.16: Direction of Both axis Movement of Gantry Crane

4.5.4 The - axis

The hoist subassembly provides the hoist/lower motion, as in a full-sized crane. A DC permanent magnet gearbox/motor is used to rotate the hoist drum which drives the spreader underneath. The maximum span along the - axis is 1000 mm (500 to 1500 mm).

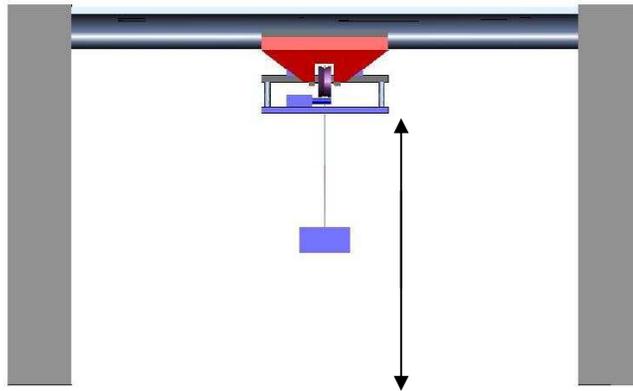


Figure 4.17: Direction of - axis Movement of Gantry Crane

From the engineering calculation and assumptions, the best design concept of the gantry crane experimental rig is listed as following:

- i. Actuator and Encoder – Brushless DC motor (built-in encoder)
 - Permanent magnets allow for a very good degree of efficiency, fast acceleration as well as rotational speeds.
 - Integrated hall sensor electronically provides feedback about the rotor position.
 - Significantly higher degree of efficiency and power density than induction motors (for the same power, about 35% volume and weight reduction).
 - The linear torque curve allows for a very big speed range at full motor power and thus better tuning to the required load ratios.
 - Mechanical exchangeability to standard stepper motors means lower construction costs and higher parts diversity.
 - Highest lifetime and running smoothness in brushless techniques with precision ball bearings.

- ii. Driving Mechanism - Cable and pulley system
 - Most simple and inexpensive system
 - Low initial and maintenance cost
 - Little or no maintenance

- iii. Guiding System – Shaft Guiding
 - Economical and simple.
 - Unlimited stroke.
 - High sealing performances.
 - Corrosion resistant.

Finally, the detail design of the driving mechanisms for the experimental rig gantry crane will be developed based on the best solution selected in the previous step. The design will combine all the mechanical and electrical components and systems which have been carefully justified in one final design. The detail design is shown in Appendix B.

5: CONCLUSION

Driving mechanism, guiding system, actuator and encoder are critical components of a gantry crane experimental rig and they play an important role in constructing the gantry crane. A careful study on the components can result the department to come out with the most efficient gantry crane experimental rig which can cause a huge benefit to the department in making further research on the gantry crane. In future, some tools including engineering software such as AUTOCAD might be used to generate the detail design and ANSYS will help in analyzing the structure of the experimental rig gantry crane when subjected to some amount of loads.

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[2] – H. Butler, G. Honderd and J. V. Amerongen (1991). *Model Reference Adaptive Control of a Gantry Crane Scale Model*. *IEEE Control Syst. Mag.*, Vol. 9. January, 1991.

[3] – Kuan-chun Huang (1997). *Integrated Sensing, Dynamics and Control of a Mobile Gantry Crane*. *Ph.D. Thesis, The University of Edinburgh. 1997.*

APPENDIX A

Project Milestone for FYP

Task Name	Feb-09				Mac-09				Apr-09				May-09				June-09				July-09				Aug-09				Sep-09				Oct-09			
	1	2	3	4	1	2	3	4	1	2	3	4	1	2	3	4	1	2	3	4	1	2	3	4	1	2	3	4	1	2	3	4	1	2	3	4
Analyze Problem & Literature Review	█	█	█	█																																
Selection of project topic & preliminary research	█	█	█	█																																
Preliminary report submission			█	█																																
Establish Design Requirement		█	█	█																																
Obtain & evaluate design criteria			█	█																																
Develop Design Concept					█	█	█	█	█	█	█	█																								
Generate alternative solutions & evaluate design concept					█	█	█	█	█	█	█	█																								
Progress report submission											█	█																								
Evaluate/Analyse Design Concept											█	█	█	█	█	█																				
Develop Detail Design													█	█	█	█																				
Develop few types of model													█	█	█	█																				
Analyzing and improving the best model																	█	█	█	█																
Interim report submission																																				
Prepare Drawings & Engineering Calculation																	█	█	█	█	█	█	█	█												
Compare & Implement the Best Solution																									█	█	█	█								
Prepare Final Drawing																													█	█	█	█				
Re-analyze, re-design and re-implement																													█	█	█	█	█	█	█	█

Table A: Task Gantt chart

APPENDIX B

1 - Assembly design

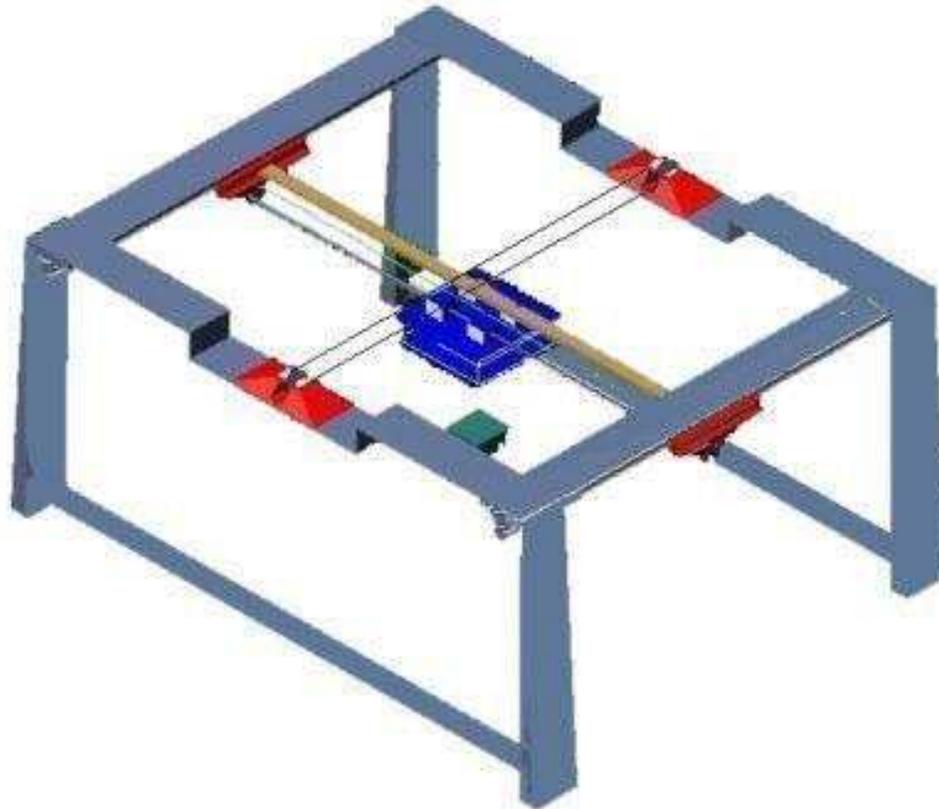
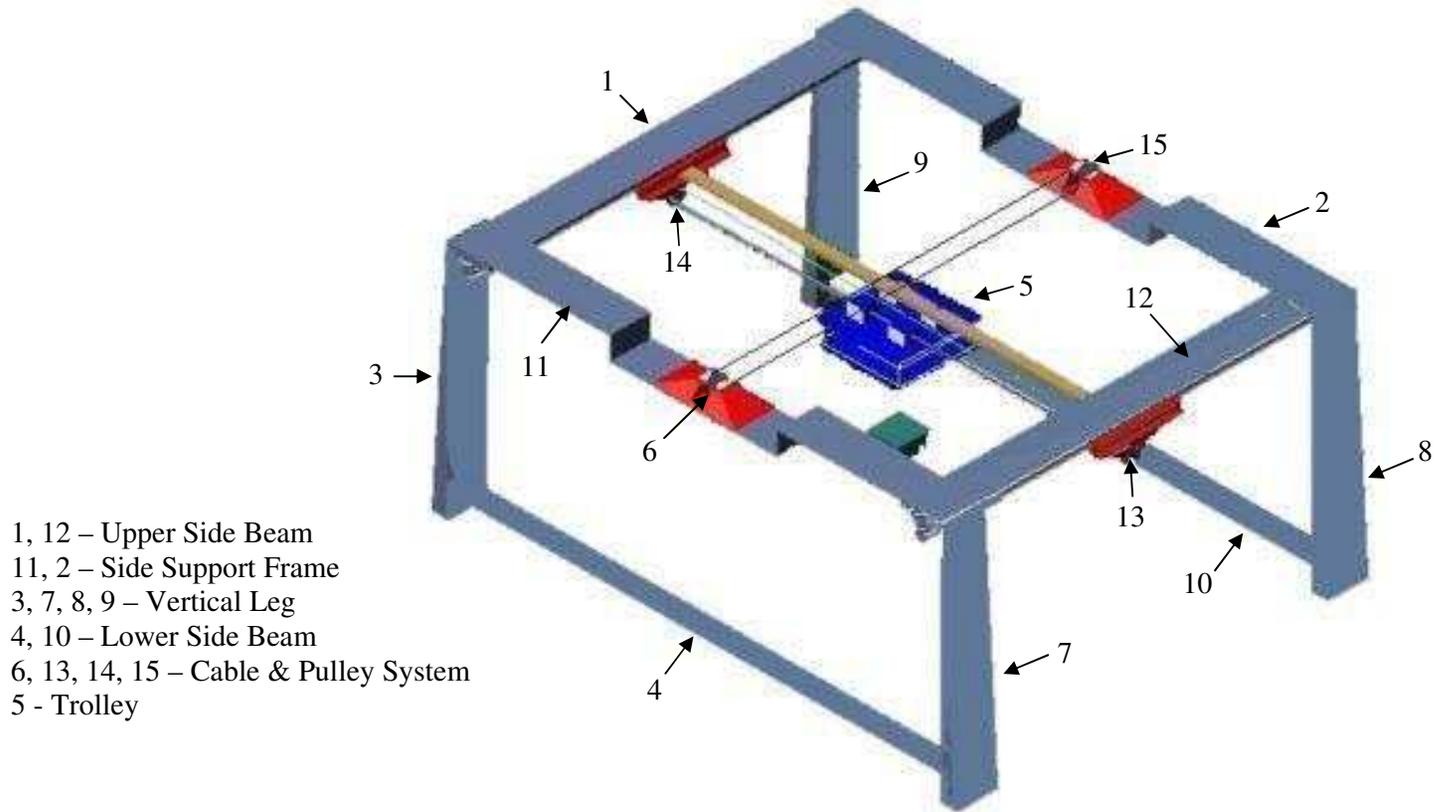


Figure B-1: Assembly Design

2 – List of Parts and Components of Gantry Crane



- 1, 12 – Upper Side Beam
- 11, 2 – Side Support Frame
- 3, 7, 8, 9 – Vertical Leg
- 4, 10 – Lower Side Beam
- 6, 13, 14, 15 – Cable & Pulley System
- 5 - Trolley

Figure B-2: List of Parts and Components of Gantry Crane

3 – Front View Dimension

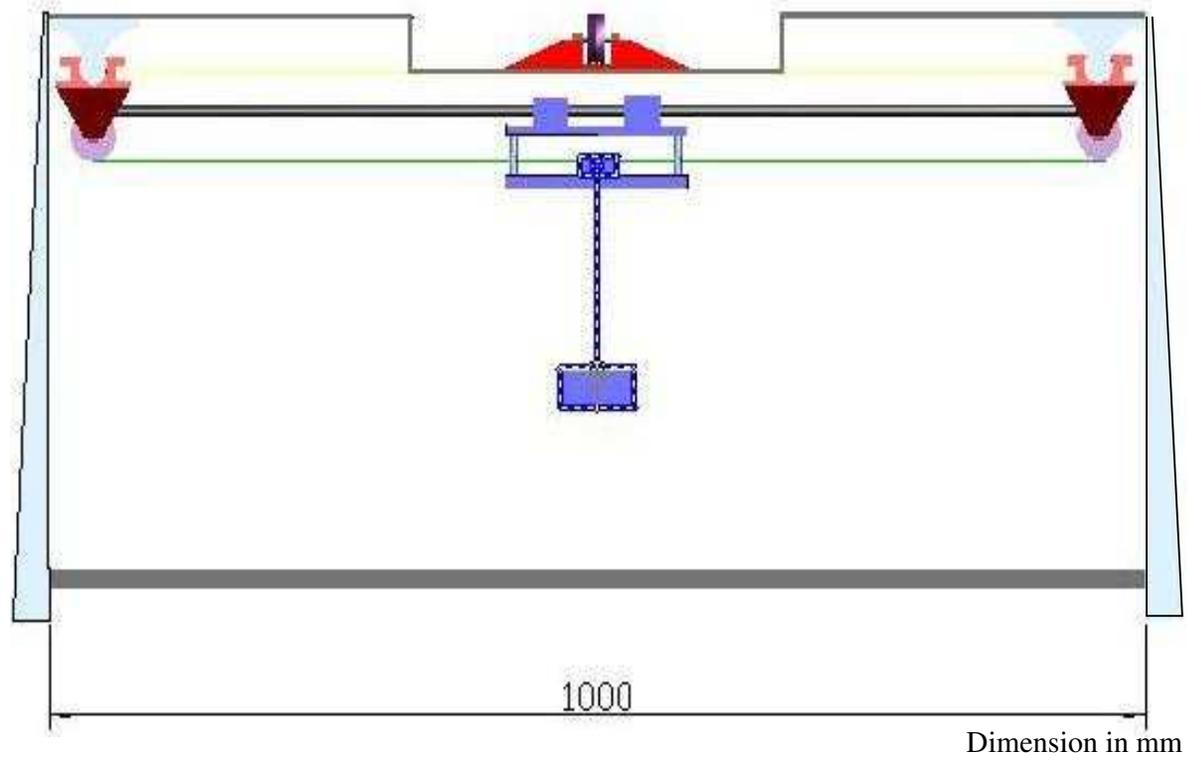


Figure B-3: Front View Dimension

4 – Side View Dimension

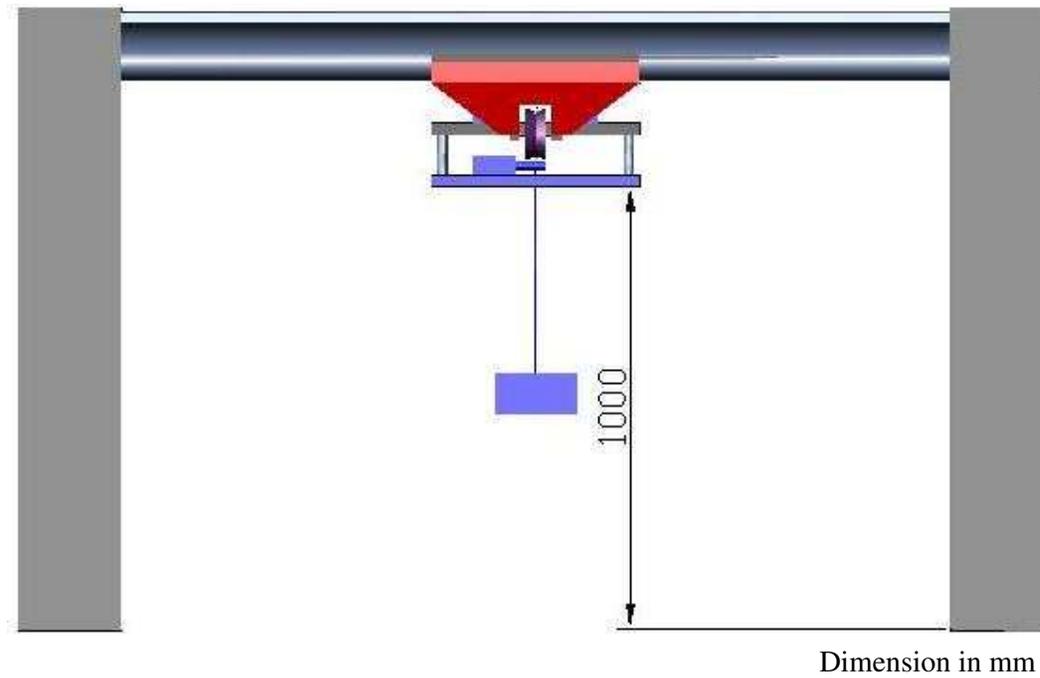


Figure B-4: Side View Dimension

5 – Top View Dimension

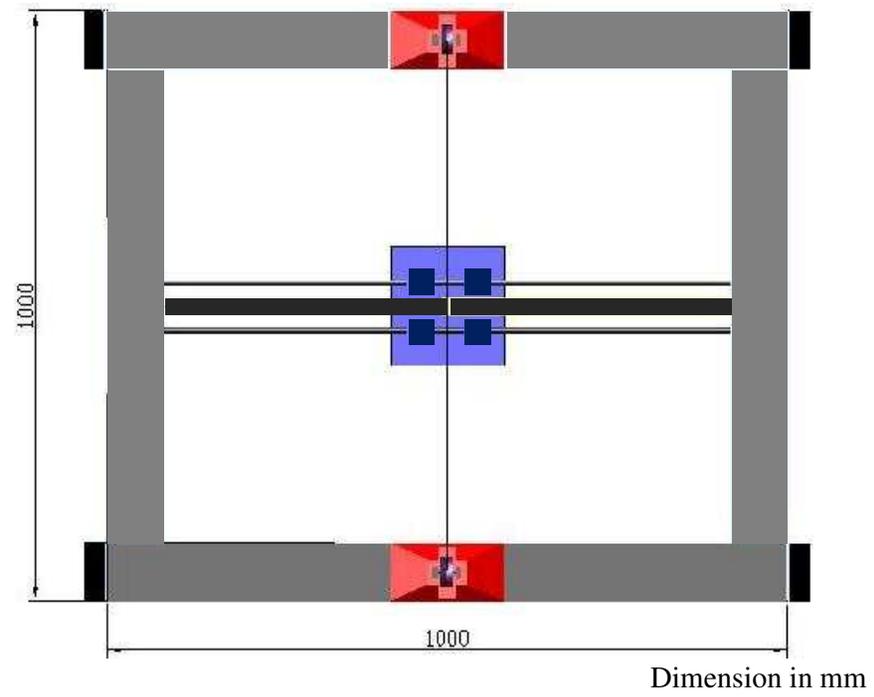


Figure B-5: Top View Dimension