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Design, Analysis and Evaluation of the Sun Simulator Mechanism

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Abstract: In the satellite laboratories, sun simulator is one of the most important equipment in order to test satellite sun sensors. The behaviour of sun sensors is strongly influenced by the angle of sunshine. Therefore, increasing the accuracy in the testing mechanism of these elements is essential for their overall monitoring of their performance before launching to space. Sun simulator includes electronics and control units, structures, motion mechanisms, heat control, optics and light sources. In the design of the motion mechanism and the structure of this simulator, the considerations of the instruments such as the material of the structure, the type of joints, the structural components, motion mechanism of the projector, the static and dynamic loads applied to the structure and the thermal considerations caused by the projector heaters are very important. The purpose of this study is to design and analyse the loading, dynamical and thermal properties of the solar system simulator with precision positioning. Therefore, at the first, plans are proposed and the proper design is selected and preliminary design and partial design are done. Finally, after selecting the components of the system, various analyses and the correct operation of the device will be approved in order to evaluate the system's resistance to static and dynamic loads as well as vibrational and thermal loads.

Keywords: Sun simulator design; Attitude determination and control sub-system; Satellite; Vibrational and thermal analysis;

1. INTRODUCTION

In recent decade, the aerospace industry and its discoveries are the fields which have been very dynamic activities. Activities in this area require the growth of science and technology in the control and guidance field of aerospace systems. Space environment, due to unpredictable behavior of things and obstacles and also lack of precise knowledge of equipment workspace specifications are serious problem in this area. In order to exploratory and research travel to the space, spacecrafts and satellites are used. Therefore, in order to have accurate navigation for spacecraft or satellite in the environment of outside the atmosphere, one of the suitable guide is the sun and its light.

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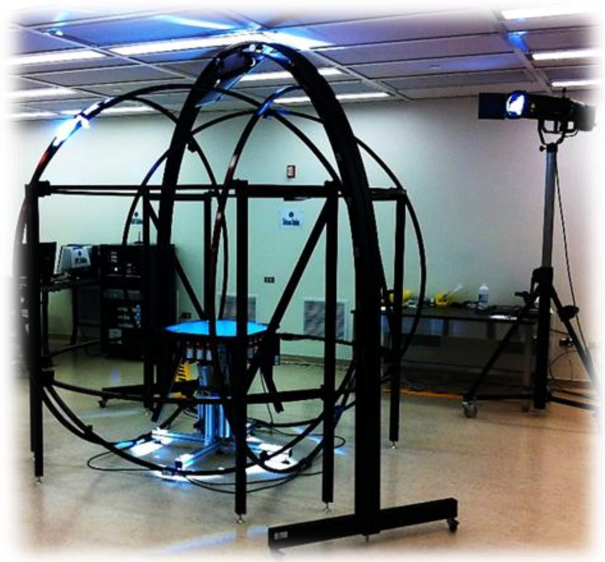


b

Fig. 1 Two samples of the device made for the sun simulator: a) a stimulus sample with a manual winch of Astofein; and b) a sun simulator with manual positioning of ADCS TB.

For this purpose, most of space crafts and satellites are equipped for measuring the angle of sunshine to make a detailed adjustment and reported information about the position of the spacecraft. The main challenge of space systems, the lack of access near and low cost to the real working space which are made for that space. For example, if needed check and verify one-stage of sensor angle which is mounted on the spacecraft from a functional point of view, It is necessary to test in the outside the atmosphere which is not feasible and has much costs. Therefore, designing and manufacturing equipment to simulate the work space of spacecrafts and satellites is necessary in the laboratory environment.

Satellite attitude and determination control laboratory system(ADCS) is designed to test satellites in order to guide and navigate equipment and their verification. Very precise equipment which two of them are shown in figure 1, have many different components, like test table system which has very sensitive degrees of freedom, sun simulator and etc.

In existing sample of this test device, adjustment system for projector positioning of sun simulator which simulates inversely the behavior of the sun. driven manually and there is no possibility to accurately adjust the position of the projector to prove correct operation of angle sun sensors.

The aims of this work are dynamic and thermal loading of sun simulator and positioning satellite test device with precise positioning. Therefore, at first proposed proposals are introduced and suitable proposal will be selected. In the following details of design are called and fully designed will be completed. Finally, render various analysis to evaluate the systems resistance to static and dynamic loads and also vibrating and thermal loads, and the correct operation of the device is confirmed.

2. CONCEPTUAL DESIGNS FOR THE MOVABLE PART OF SUN SIMULATOR

2.1 Introduction

In order to design a sun simulator, it is necessary to consider possible schemes for drive system. In this section, first, a few possible plans to fitting suitable system location and system actuation mechanism are investigated. The main challenges of the system and optimal solutions will be mentioned and the final design will be presented.

Existing challenges and design goals are as follows:

1.the structure must be non-magnetic

2. reduce the cost of construction as much as possible

3. ability to position the device

4. possibility to accurately position of the projector and mirror assembly

5. the presence of curvature of the path and the motion

6. high occupancy space

7. high thermal temperature of projector lamp

8. reduction of dynamic forces on the structure

9. possibility of changing the angle of reflection of light on the satellite

10. resistance of the structure for possible transverse load

11. resistance 0f the structure to dynamic load from the actuating system.

12. resistance to earthquakes and vibrational motion of the environment

13. low maintenance costs and optimal safety.

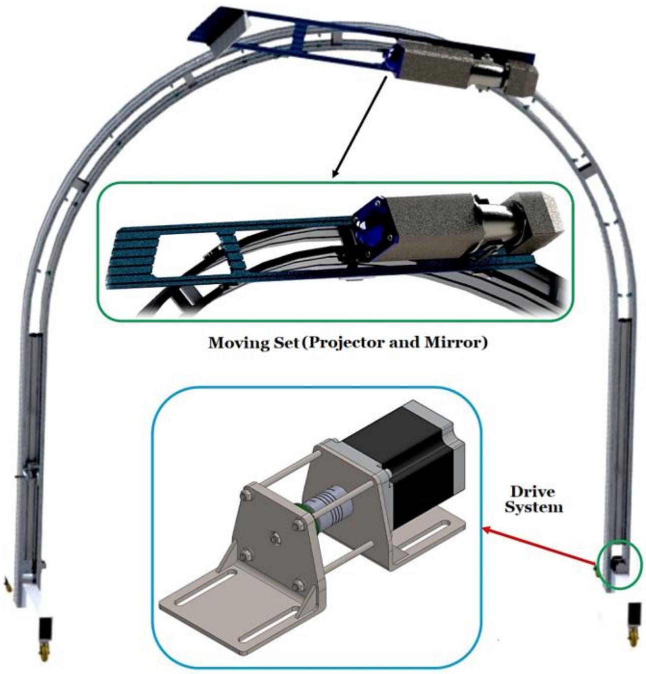


Fig. 2 Outline and main components of the sun simulator.

By considering all of these, the proposed scheme is presented in figure 2 which in following will be focused on design details and analysis.

2.2 First concept design: put the motor on the drive system

One of the plans is to put a flat-small-motor on aluminum base of lamp and pass a path of cable over the Pulley. In this plan only needs a cable path and by passing a few cables through pulley of motor and providing sufficient friction, the moving of base on the constant cable is guaranteed. In addition, the range of this system for adjustment sufficient preload on the cable is so important.

Using only one motor and a cable path is an advantage, because the cost of segment will be reduced and easier control. In return, the obligation to use a mechanism for passing the motor and accessories of motor over the rollers, increases the complexity of the system. In addition, by increasing the weight of moving set in the system and consequently the dynamic forces are important factors. As another disadvantages, the reliability of the system due to the possibility of damage to the moving set and also mention maintenance problems which all of these, together offer a partly weak design. The idea of this kind of moving set is taken from telecabin which is shown an example in figure 3.

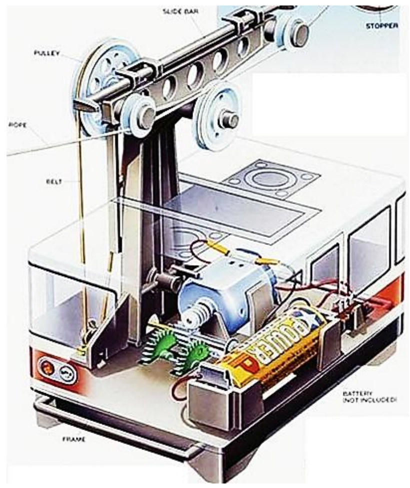


Fig. 3 Cable car (tele cabin) self-contained drive system [3].

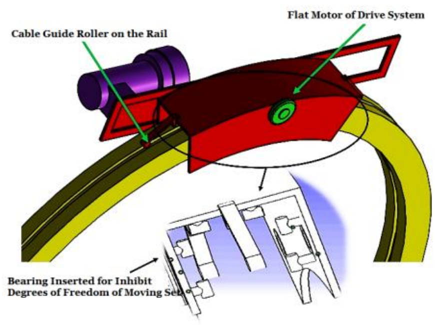


Fig. 4 First conceptual design for locating the motor of the driving system, bearings, rollers, and how to insert and direct the cable.

An overview of the conceptual plan 1 is presented in figure 4. In this figure, the green disk is a schematic of motor, the bulb is shown in purple, and as shown in the figure there is a mechanism to keep the cable in the middle to cross the roller over the motor and Pulley should be located which has a little space for guide cable on to the roller. the cable movement on the rail is also inside the groove on the path which this idea although dose not use the conductor roller, but versus power drop due to slipping cable connection and path will happen which is the disadvantage of cabling of this project.

2.3 Second conceptual design: Fitting the fixed motor next to the base

One of the **most** reasonable plans is to use a fixed motor on the base. The mechanism of this design is quite similar to the plans which is used in the telecabin figure4. There is a going round path and the base of the lamp is fixed and connected to the cable.

Fig. 5 Cable car rail drive unit in terminal [4].

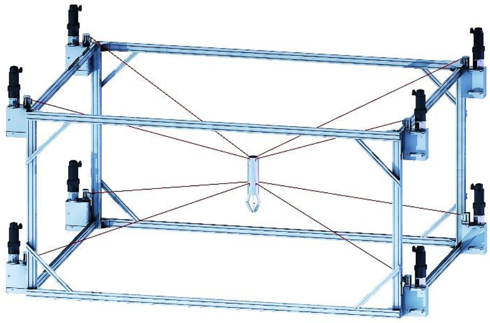


Fig. 6 Cable-Driven Parallel Robot [6].

In going round path for guide the rollers of cable must be inserted, additionally to adjust the preload inside the cable can tighten the strap which a Pulley is used outside the cable line.

The clear advantage of this design is, its simplicity, additionally the mass of movable has been reduced and does not require range.

One of the disadvantages of this project is cable stretching. In this type of mechanism, the cable is rotated over pulley many rounds and then crossed, according to direction of pulley rotation, always, one side of the cable is under more force, this is possible to effect the positioning of the movable and do not provide enough precision. This error is found when movable set is coming down from the opposite site and in this case if the movable set does not have stretch, there is a possibility of falling to the front.

2.4 use two fixed motors on both sides

Another plan which is exist, put two motors on both sides of the base, in this plan, the problem of freedom of cable on one side will not exist, because only one cable path is required. Like that rotation side of the two motors must be opposite of each other until one motor collect cable and opposite motor open the cable. this design is the same as the cable robots which controlled by the cable. In this system , to move the clamp in different direction, are used separate operators.

The important advantages of this project than previous project is resolving the problem of positioning accuracy because in this project always two movable side of cable are under stretch. Also, in this project is used one cable path too.

Perhaps paying for another motor can be the defect of this plan. Of course it is important to mentioning which in this plan, on the way back there were a number of rollers which were eliminated. And for this reason the cost of cable and roller is also saved, although in comparison with movable system it is not so important.

2.5 Conclusion

According to the description and complexity . slow acceleration, possibility of construction and maintenance costs, the second plan can be the optimum . in the following details of plan are selected and will be introduced and designed.

In the final section, dynamic analysis , tension and thermal analysis due to the operating condition of the machine happened and plan will be evaluated

3. REVIEW DETAILS OF MOVABLE MOTOR

3.1 Text Structure

As shown in figure 7 the middle part of the main structure is made of two semi-circular profiles which connected together by several small spacers, bending is the reason why two semi-circular profiles are used; if the thickness of the profile be high the possibility of bending will be discarded even there is a chance for bending, using a single piece of profile with high thickness will have a high mass which is not suitable for use.

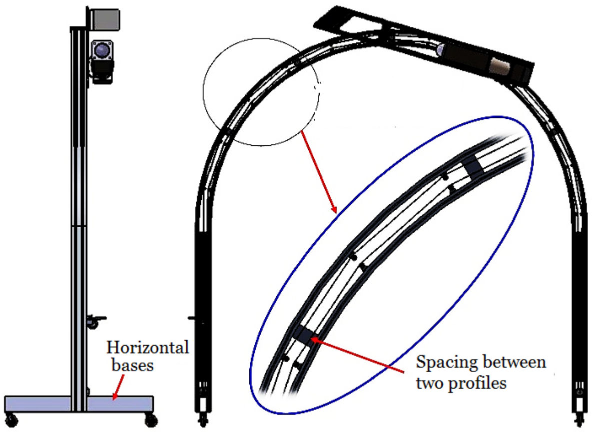


Fig. 7 Structure of the Sun Simulator.

In figure 7 is clear that in the lower part of structure for more stability is used an aluminum profile on both sides as a horizontal base.

3.2 Motor driven system

3.2.1 Motor

To select the right motor, the power required for the system must be calculated. Mass of the simulator system is considered 20 KG. Because this speed wills not ne high, therefore the stair motors can be used. The advantages of these motors are, produce more torque at low speed than high speed. Due to the mass of the simulator system amount of power which is required to be provided by the motor is calculated as follows. It is important to mention which maximum required force is where the simulator system is located vertically.

There for the motor without considering dynamic force must be able to apply force equal to 200N.m, the dynamic force is checked to ensure which is negligible.

The speed of slider movement of the lamp is approximating of distance traveled and time calculation , 0.2m/s will be obtained as a result , the acceleration should be about $0.15m/s^2$ so: it was ensured which the emount of this force is negligible against the weight force and it will be discarded.

To find motor power the Pulley of motor must be selected so the torque of arm can be obtained, similarly, to choose the right size of Pulley, at first the appropriate cable must be selected. There for at first must be discussed about cable. For now, for pass this stage radius of Pulley must be assumed 5cm. as result the motor torque is calculated as follows :

$$\tau = F * R = 200 * 0.05 = 10\ N.M \tag{1}$$

3.2.2 Cable

The cable should be able to withstand 200N.m , by selecting the confidence coefficient 7, can choose a suitable cable. Referring to the cables which are used in the industry, 19*1 cable is suitable cable for using , diameter of selected cable is 2mm.

3.2.3. Pulley

There are several choice for the Pulley system:

A) The first plan is passing a few rounds on Pulley and then the cable is going back, in this plan, friction between Pulley and cable is the reason for moving the movable and the slider (on the rails) figure 8 shows schematic of the first plan.

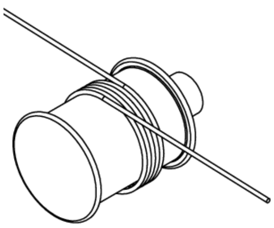


Fig. 8 Schematic of the first solution of cable twisting on drum.

One of the problem of this plan is the possibility of slip cable on the Pulley which , in spite of this, positioning will be in trouble. To avoid this, another plan will be presented

B) There is a solution to ensure non-slip which the cable are connected to the drum , so that, at first one side of cable is tightened to the drum and passed from rollers, then it is passed on the drum for one cable pass, and the other side of the cable will be tightened to the drum, figure 9 shows, cable closure to the drum, wrap cable on the drum for one path after passing cable from rollers causes to((results in)), when the cable is opening over drum from one side. From other side the cable will be wrapped over the drum , in this way movement of slipper is guramteed on the rails

Fig. 9 Tightening of the cable end to the drum.

C) Another design is using of timing belt and timing Pulley. In this design it is not necessary to wrapping belt around the Pulley and timing belt will ensure non-sliping.

Radius of the suns simulator is 2.5 meters so , the length of the path which is wraped over Pulley is calculated as follows:

$$L = r * \pi = 2.5\pi = 7.85\ m \tag{2}$$

Now if a drum with a radius of 5cm is assumed and its length is 5cm. for maximum torque of motor, the number of rounds which the cable, is calculated as follows:

$$\theta = \frac{L}{r} = \frac{7.85}{0.05} = 157\ rad = 25\ turns \tag{3}$$

The length of drum is 5cm with 2cm thickness for 25 rounds of cable is suitable. Even its length can be decreases to half(and two rounds around it). In this case , maximum power will be equal to

$$\tau = 200\ N * \big((50 + 4) * 0.01\big)m = 10.8\ N.m \tag{4}$$

The reason of motor torque correction is illustrated by figure 10.

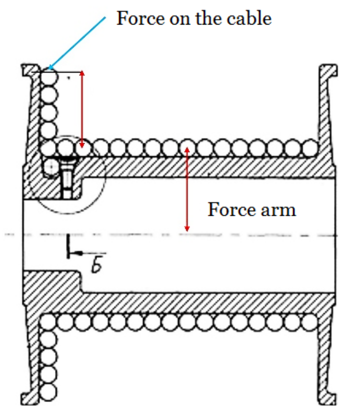


Fig. 10 Increase the power arm due to increasing the number of cable runs.

Now, by considering the torque and confidence factor of 1.5 , a motor with 15N.m torque should be selected but because the speed in the simulator of the sun is very slow, for reduce the cost an motor with half of this torque with a 1 to 2 gear box looks more suitable. Stepping motor(Nema 34) with dimension and appearance of the QSH8616 is used(figure11). Due to the suitable dimension of motor it can be placed on horizontal legs which are between the section of elevation structure(figure12), and the Pulley and coupling sets will be connected to the motor, select a coupling for stepper motor is not so hard. Due to the diameter of the motor shaft and transmitting power and by considering the error topics, the suitable coupling will be selected.

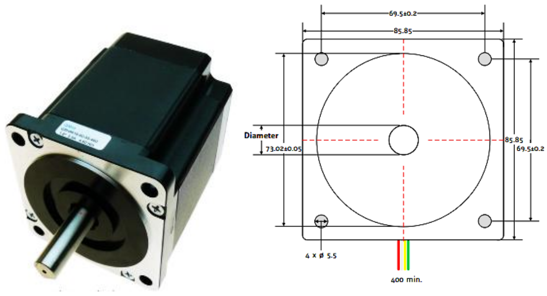


Fig. 11 Nema34 stepper motor

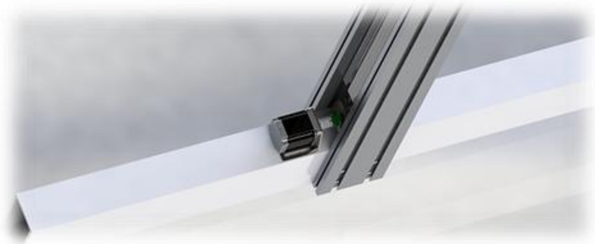


Fig. 12 Locating of a drive system.

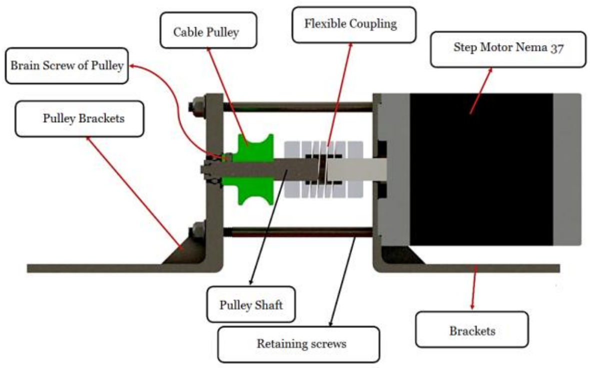


Fig. 13 Different parts of final power train design.

Movable part of motor and details of the bearing is shown in figure 13. Components and parts which are necessary to connect the horizontal base; by changing the position, to create bean grooves will be located.

3.3 Adjustment tension of cable

different designs can be used to adjust the preload inside the cable, in the following, different designs will be explained and eventually one design will be selected.

Use the weight of motor; a plan is to use the weight of motor set for preloading inside the cable, in this design the motor set is located on a plate and the plate will be hinged to the rest of the set until the weight of the motor stretches the cable or belt. figure 13 illustrates the explanation.

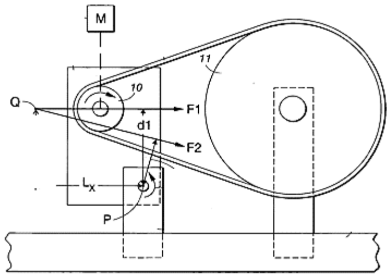


Fig. 14 Using motor weight to adjust the preload of the cable.

Using of this design is invalid, because by using of this design, repositioning and use another profile to pin in to the structure to place the motor on it, will be the problems.

Idler Pulley with additional weight: by using idler Pulley which is pinned on the rod which is mounted on the structure, can be adjusted the preload inside the cable, although looks simple and effeicient but has disadvantages of the previous design. This design is shown in figure 15.

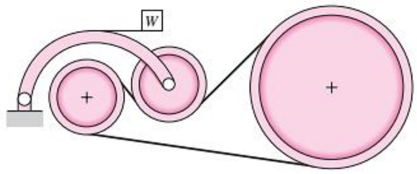


Fig. 15 Idler pulley with additional weight [7]

Idler Pulley with movable axis of rotation: in this design there should be a mechanism to take the axis of Pulley out of cable to be able to make enough load in cable. This design looks very simple and appropriate and there are no disadvantages of previous design. This design is shown in figure 16.

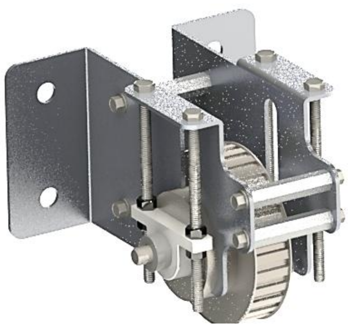


Fig. 16 Idler pulley with movable axis of rotation.

This mechanism is connected to the structure, then, by tightening the screws which move the axis of Pulley into the guide groove, can adjust the inside load of cable or trap. In order to three design which were introduced and advantages and disadvantages, the third design will be used as a tightening cable system in the intended device.

3.4 Cable guide rollers

It is necessary to use a conductive roller to guide the cable on the going round path. These rollers are located in the upper and lower rows (which these rows are between the two half-circle profile). The cable must be passed through the bottom of the lower roller and return from the top of the upper roller. Figure 17 is shown a piece designed to hold the cable guide roller in its place. This piece is threaded by 2 bolts and it is designed in such away which it can be moved in to the groove by slightly opening the bolts so there is no necessary to drilling in a half-circle profile.

Fig. 17 Cable guide rollers.

Actually by tightening the bolts, T-section friction with groove will be increased and the piece is kept inside it.

Viewing the cross-section of this piece in figure 18 helps to understand better the problem.

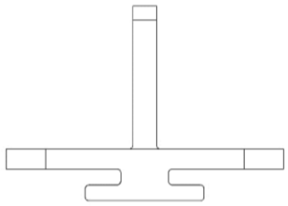


Fig. 18 Cross section of cable guide rollers holder.

How to connect the cable to the slider of lamp is a important point(figure19) which can be considered some plans. The simplest mode is use the hook which is attached to the slider of lamp.

Fig. 19 Cable connector hooks to moving bed.

3.5 Limiting system of movable on the path

One of the most important part of sun simulator system is designing constraints for slider inside the path. For this purpose there are different plans.

3.5.1 Circular rails

First plan is installing rails on an aluminum profile and using special sliders. in this design circular rails should be used. 'Rollon' company produce these rails(figure20). these rails which are produced by Rollon company are released in different radius. This should be ordered to the company for preparation. Using of such rails due to the limitation of its manufacturing companies and also unavailability seems very difficult but, instead, it does not have design problem and will have high confidence in the result. By the way, this design will not be used and suitable rollers will be design to constraint the slider on the path.

Fig. 20 ROLLON circular rails [7].

3.5.2 Designing a suitable roller

To design the roller to constraint the slider on the path, as it is clear, rollers in different parts to constraint and reduce the degree of freedom on the rails has been designed. On the four sides of the rails are such rollers which are as follows:

a) Front rollers: the images of these rollers are shown in figure 21. Ball bearings are also used to provide rotational movement; which two ball bearings are placed in this plan. The reason is, these rollers are expected to endurance fairly large load. Contact between side surface of the semi-circle profile and movable set, by these rollers to routing mode is provided.

In figure 21, there is a thorn in the front roller to deprivation axial movement of pin, compared to the piece which is located on pin. At the end it has been wormed threaded to prevent roller and bearing exits from their place.

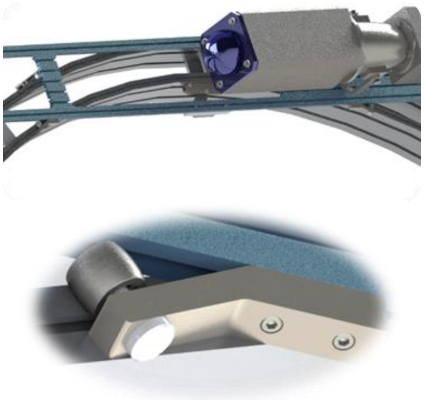


Fig. 21 Roller in the front and last of the moving bed.

b) Rollers on the top side: rollers on the top side are shown in figure 22. In this design there are two rollers from above and there is one roller from the side. Some pins which are used on the top rollers, are shown in the below picture. To prevent collapse of the set and increase the structural reliability, is also used a roller on the opposite side. Two upper rollers just tolerate the vertical load. Rails of rollers are the groove of profiles. L-shaped piece which is roller inhibitor, by a pin which has the ability to rotate toward movable set is connected to it. To allow rotation toward structure during moving.

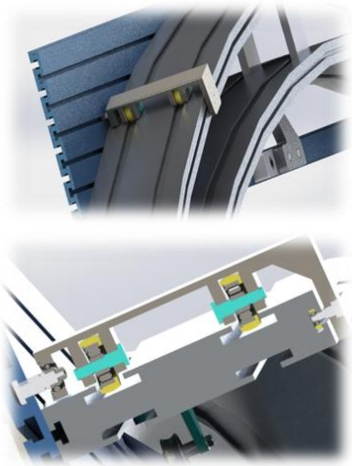


Fig. 22 Rollers on the top side of structure.

As mentioned, in this design there are two rollers on the top and a roller on the side; this design has been done conservatively and probably after design calculation only one roller will be required.

D) Lower roller: as shown in figure 23, the lower roller made from some pieces, the rollers are located on the piece which is shown in figure 23, task of this part is allow the set to separate to upwards and increase accuracy of positioning.

Fig. 23 Inhibitor rollers from below.

according to this design and previous designs there are two thorn spring in figure 24(is shown cross section of roller) which is the reason of using thorn spring. As shown in two pictures, there are a step which is located in the middle of the roller, for separating ball bearing and the place of thorn spring, and there are grooves inside roller (as it is shown in figure 24) to tightens the axial movement.

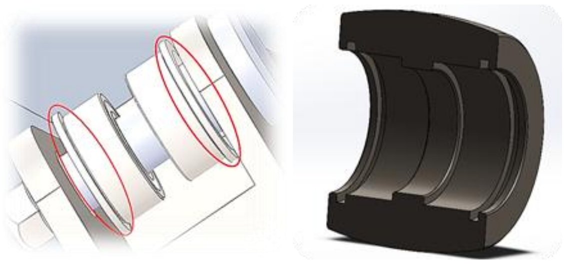


Fig. 24 Idler pulley with movable axis of rotation.

3-6- Mirror design system

The base of mirror is designed to change its angle and be fitted in the suitable place.

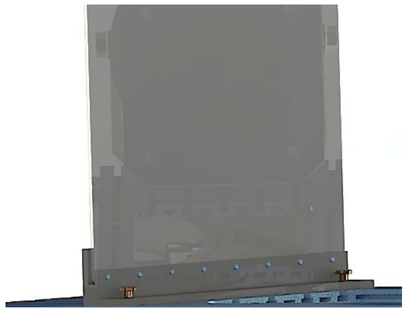


Fig. 25 Mirror design and its connections.

According to figure 25, screws which are used to connect the base of mirror to the slider profile, there are kind of screws which can be opened and closed by fingers easily. There is also a possibility to change the position on movable system.

3.7 Energy guide

To pass the wire of lamp, should be a suitable place. The gap between two grooves of profile seems to be a good place for guide energy. If the design of the upper rollers be corrected and one of the rollers be removed, inside the groove can also replace the guide energy (figure26 and figure27).

Fig. 26 Energy Guide.

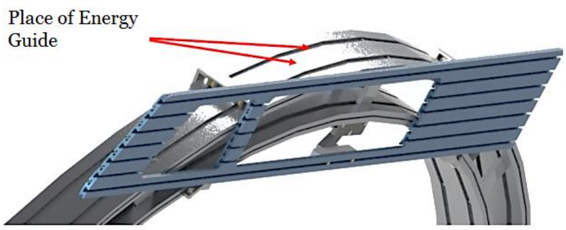


Fig. 27 Energy guide moving path.

3.8 Connect the cable to the movable set

In order to move the set by cable, it is necessary to connect the cable to the set by one or more points. On common cable connections, mostly there are hinges so leaping will be appearing which is not very suitable for precision of positioning.

From these designs, according to size of the cable ???? has been selected, which the location is also has been shown in the picture which there is no problem.]

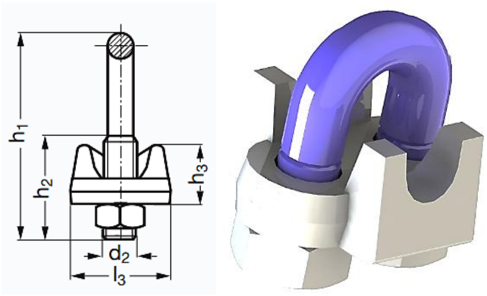


Fig. 28 Cable clamps used with different views.

As it is seen, by connecting the cable to the clamp, there is no touch when it is passing the lower roller and just there is a little movement of cable and there is no problem as view of astro company any more. So function of return pass will be continued easily (figure 29)

3.9 Summary and Conclusion

In this design, all possible problems are considered, surely, this is not the only plan and in the future, according to economical estimation and available facilities, there is possibility of changing some of details. Overall, this design can be introduced as the simplest, cheapest and the most accurate design. In the following, the details of this design will be designed and dynamic and tension analysis will be investigated.

Fig. 28 Locating and connecting the cable to the moving set.

4. DESIGNING AND FINAL FIXES FOR SENSITIVE COMPONENTS

In section 3, the initial design was presented for solar simulator. In this section final design will be specified.

4.1 Preliminary calculation of motor selection

Base on what was presented in conceptual design section, the final design of solar simulator system will be similar to figure 2. Generally, what was considered as movable system is a cable-driven-motor system. Different parts of this system are visible in the sectional view.

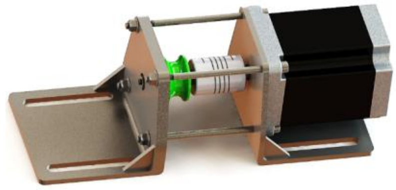


Fig.30 Overview of the actuator with the cable system.

Due to the mentioned problems, such as cable slip on the pulley (figure 30), the belt system is more appropriate than cable system. Actually instead of cable, timing belt drive will be used (figure31).

Now, analysis of the overall force on all components of system is necessary, in order to finish the initial design and then by software analysis, dimensions and geometry will be achieved.

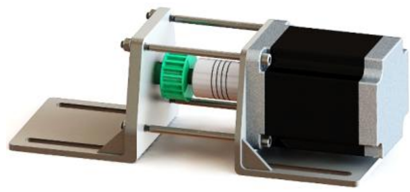


Fig. 31 Actuator system with time belt pulley

In the initial design, total mass of movable system is assumed 20Kg. load factor is assumed 2.5 for strike. Displacement of 50 Kg load will be the standard of designing; this force should be provided in all situation even in vertical direction by strain tension (figure 32).

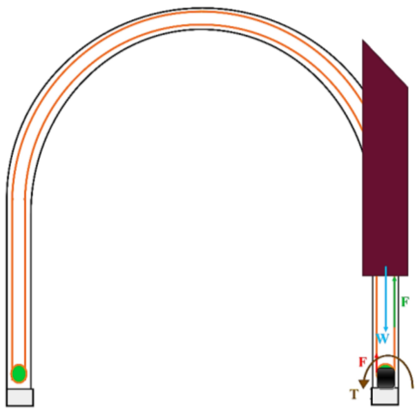


Fig. 32 The schematic of loading the actuator system in the worst possible mode (max necessary propulsion force).

According to the picture static equilibrium equations will be established. The motor torque will be obtained by specifying diameter of pulley (furmula1).

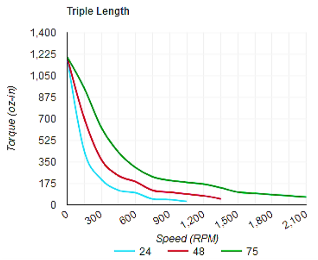
$$F = W = mg = 500\ N \ \& T = F.r \tag{5}$$

In this equation, radius of pulley is 'Y' and motor torque is 'T' in static situation. It should be noted, impact of dynamic forces due to acceleration and braking is so important which should be evaluated because of low speed of the solar simulator system, the effect of these forces will be very low. According to approximate calculations, velocity is assumed 0.2m/s and approximate acceleration time is 1 second which the acceleration is equal to $0.2m/s^2$, the dynamic force will be calculated with mass of 20Kg. based on these calculations, influence of inertial forces(F_i) compared to static forces which is related to the weight, are less than 1% and is negligible.(safety factor of these forces is included).(formula2)

$$F_I = ma = 20 \times 0.2 = 4\,N \;\; \& \, F_s = 500\,N \quad \Rightarrow \frac{F_I}{F_s} < 1\% \tag{6}$$

Now, if the rate of constant speed be 0.2 m/s and the force of belt be 500N , input power required with 85% efficiency will be 120W. if a timing belt is used with diameter of 40mm, required torque for the motor is calculated as follows: (formula3)

$$T = 500 \times 20 \times 10^{-3} = 10\,N.m = 1000\,N.cm \tag{7}$$



Spec	Notes	Triple length
Holding Torque -	oz-in (N-cm)	1203 (920)
Phase Current	Amps	6.3
Phase Resistance	ohms	0.56
Phase Inductance	mH	6.2
Rotor Inertia	oz-in-sec2 (kg-cm2)	0.03849 (2.70)
Weigh	oz (g)	141.1 (4000)

Fig. 33 Nema 34 with three Stacks features.

According to the load factor which is considered and manufactures catalogs, stepper motor 'NEMA34 triple stack' with maximum torque (920N.cm) is appropriate to choose. Curves of produced torque in different rounds and different voltages (12-24-48) are provided.

Alternative design of this type of motor is stepper motor with less length and step and has flexible gearbox. Since it does not require high speed, ability to reduce the speed and required torque will be possible.

According to the manufactures, sun gear box (4:1) can be used with dimensional and functional characteristics. In this situation, the torque of motor will be 250N.cm, so 'NEMA34 single length motor' can be used.

For compare the existing solutions, economic analysis will be useful.

Cost of motor which has torque of 250N.cm and lower is 40$ and cost of 4:1 sun gear box is 53$, therefore total cost will be 90$. While, cost of motor which has torque of 13N.m is about 80$ which is cheaper than the motor with gear box and in terms of construction, buy, assembly and maintenance are cheaper and easier.

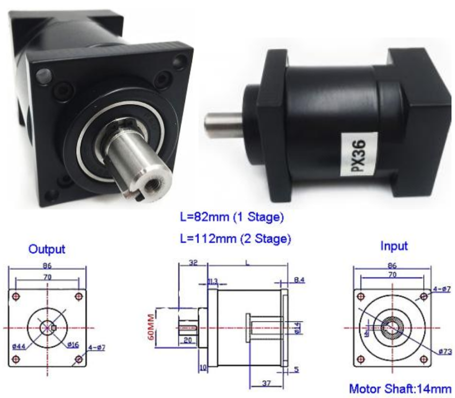
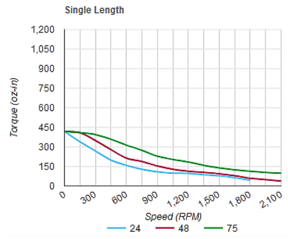


Fig. 34 Different views of 4:1 sun gearbox



Spec	Notes	Single length
Holding Torque -	oz-in (N-cm)	419 (296)
Phase Current	Amps	6.3
Phase Resistance	ohms	0.30
Phase Inductance	mH	1.90
Rotor Inertia	oz-in-sec2 (kg-cm2)	0.0275 (0.90)
Weigh	oz (g)	60.0 (1700)

Fig.35 Nema 34 with three Stacks features

4.2 design and select timing Pulley and belt

In order to diameter which was 40mm in the previous section, by referring to different standards and catalogs, depending on the transmission power of 130W and the speed ration of 1:1 the Pulley was selected from BBman company which is shown in figure 36. As it is shown, figure 36 has two adjustment screws for curb the axial movement. The width of the belt is about 15mm, and the diameter of pitch is about 40mm. shaft hole diameter is 6.5mm. the overall length of Pulley is also about 26mm. (figure 36)

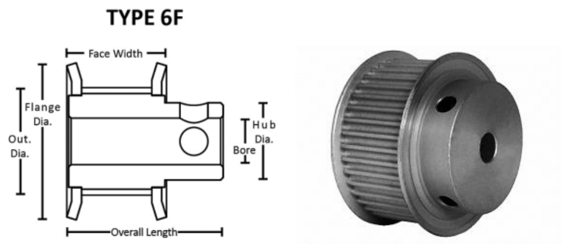


Fig. 36 The schematic of chosen pulley of BBman.

And the length of required belt is about 20m.

Because the rest of pulley set the path for movable set, they are similar to idler. So all of them will be chosen from same kind. In the route section which the belt passes above Pulley, exterior idler which has no gear will be used.

4.3 Designing and selection of flexible couplings

Flexible coupling is existed in different types with different applications. These kinds of coupling can tolerate the redial and angular axial asymmetry between the driving axis and movable axis.

Flexible couplings have four main tasks:

. Transfer the torque and speed of the driving to the movable.

. Neutralize the vibration

. Compensation for unbalancing

. Affection on the normal frequency of the system.

Generally, normal axial unevenness in small coupling is 0.055 inches and in large coupling is 0.03. Maximum angular misalignment is usually considered to be about $\pm 3^{\circ}$. Due to wide range of flexible coupling, there are no comprehensive categories for flexible coupling. So category of these coupling is not useful.

In stepper motor systems and servo systems, typically, there are two flexible coupling (rotex coupling and encoder coupling) (figure37).

Coupling is consist of two flexible pieces. Rotex coupling is made of high quality cast iron and all parts are machined to stop the vibration in high speed.

Spider of rotex coupling is made of high quality polyurethane materials(TPUS) rotex coupling is small and light but can tolerate high torque and is suitable for absorbing the vibrations and shocks of the system, operation temperature of rotex coupling is between 40°c- 90°c in normal mode and between 50°c-120°c in short periods. The material of rotex coupling are divided in to 3 categories: aluminum, cast iron and steel.

Fig. 37 Two common types of flexible couplings for control systems.

Encoder coupling are flexible, these couplings are made of aluminum, these kinds of coupling are suitable to control the situation. Encoder coupling can be used in CNC machines and servo motors.

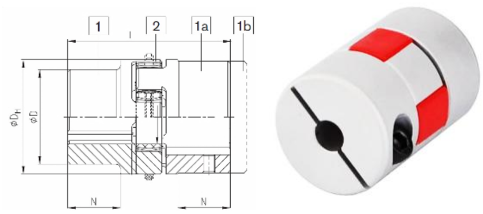


Fig. 38 Views of used Rotex coupling

According to the diameter of motor 'NEMA 34' which is 12.7mm so it is necessary the diameter of coupling hole be at least equal or smaller than this value, because it can reach this value by machining, in this condition, the outer diameter should be as large as possible to machining if rotex coupling is used(aluminum type), diameter of the shaft hole, 7.12mm are accessible for inlet and outlet. This coupling is a good choice but leaping is the main problem due to high load, so it is better to use encoder coupling.

By referring to the catalog, aluminum type with an outer diameter of 31.8 mm and inner diameter range of 5-15mm which maximum torque tolerable is 15N.m, is the suitable choice for this design.

So, both designs are partially usable, and base on market status and economic issues and positioning accurately best design can be chosen. in this paper encoder coupling will be used.

4.4 Designing and analyzing the main timing Pulley shaft

Figure 39 shows the shaft which is designed for Pulley. If assumed 10N.m torque is supposed to be applied in the middle of the thorn, which comes from 500N shear force, by assuming maximum bending in the location of thorn, with coefficient of bending stress k_t=2.15, k_{ts}=3 with the following geometric dimension. Static analysis can be performed: (formula 4)

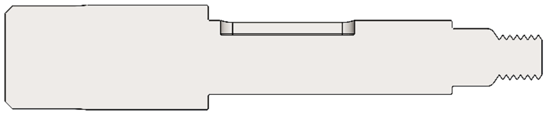


Fig. 39 Cross section of moving motor timing shaft.

$$\sigma_t = K_t \frac{32FL}{2\pi d^3} = 262\ Mpa \ \ \& \ \ \tau_s = K_{ts} \frac{16T}{\pi d^3} = 210\ Mpa \ \rightarrow \ \sigma' = \sqrt{\sigma_t^2 + 3\tau_s^2} = 450\ Mpa \qquad (8)$$

Assuming a 10145CD steel, which yield stress is equal to S_y=530Mpa, static safety factor on the set can be found(formula1)

$$n_s = \frac{S_y}{\sigma'} = \frac{530}{450} = 1.2 \qquad (9)$$

The load factor is assumed 2.5, real safety factor will be n=2.5*1.2=3. in fact, because all loads are very slow, safety factor with impact loading also models fatigue behavior.

Fig. 40 Final actuating system pulley shaft.

So the safety factor of fatigue can be assumed 1.2. so the diameter of 9mm for the hole of pulley is safe. The final shape of the shaft is visible in figure40

5. TENSION ANALYSIS OF SENSITIVE COMPONENTS

To ensure resistance of system against static and dynamic loads, it is necessary some critical components of the system are carefully analyzed, which comes in the following.

5.1 Static Analysis of pulley Timing

First, pulley timing will be analyzed, because it is under high load. This sample has been considered steel due to magnetic conductivity problems. Second suitable material is aluminum 7000, which most of components are considered aluminum.

Due to the variety of dent for pulley timing, for pulley timing which has 40 gears, the analysis will be repeated again. In this case, they have the same material and mesh, with static loading, by assuming only 10% of gears will withstand all load, tension and displacement are shown in figure 41 and 42.

As it is shown in figure 41 maximum tension occurs at the root of the dent and it is equal to 8.5Mpa which in comparison to crushing strength which is 172Mpa, is very low. According to the maximum displacement on surrounding of dents and it is equal 0.6mm and it will happen. As static analysis, this design is appropriate and the type of 40 dents will be selected.

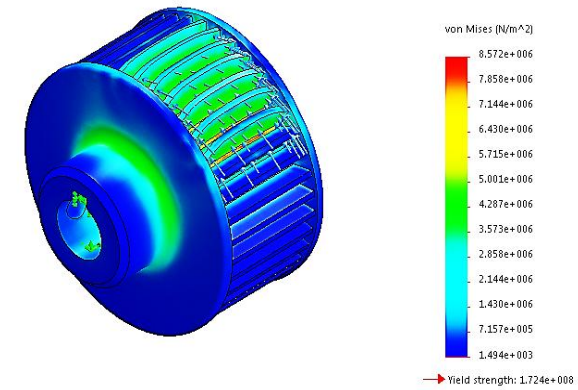


Fig. 41 Von Mises static analysis of a 40 teeth pulley.

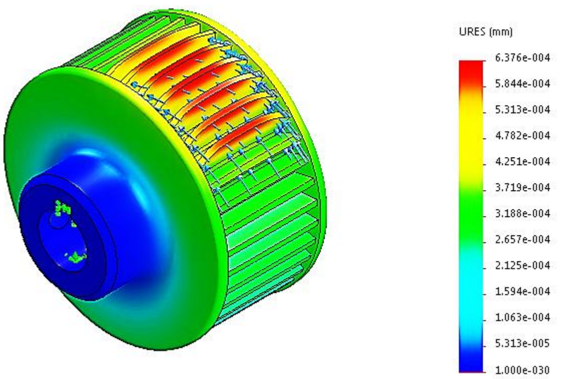


Fig. 42 Colorful curve of deflection of timing pulley with 40 teeth in static analysis.

5.2 Structural analysis with vertical and transverse force by SOLID WORKS

In this section, the main structure of the system 'railing of projector' will be analyzed. Due to aluminum and dimension of structure the total mass of the structure is calculated 500Kg.

The material is aluminum 7075-T6 which the yield stress is equal to 505Mpa and breaking strength is equal to 570Mpa. Density of the system to reduce the total weight of the system is less than steel and is equal to 2850Kg/m^2 .versus, modulus of elasticity is low and equal to 72Gpa. So deflection and displacements under the loading are so important.

First, it is necessary to determine the fixed positions and supports of structure, because the structure is located on the solid horizontal beams. connection between the rail and these horizontal beams which are connected by 8 screws, will be considered solid. So 16 internal surfaces of holes due to connection to the underlying structure will be assumed fixed.

Loading on the system has three separate parts:

1- Due to the size of the structure which is huge, and by considering gravity, the weight of structure which enters on the structure is considered
2- 500N vertical force which enters on the upper surface of the structure which comes from the weight of the system
3- 200N force as transverse unwanted forces and also bending force which enters to the structure.

After loading, next step is meshing. Meshing algorithm is four-point element with a size of 50mm.

Table 1 shows the reaction forces on the support, X direction and rail direction are the same which there is no force in this direction.

Table 1 Reaction forces on horizontal base.

Components	X	Y	Z	Resultant
Reaction force(N)	0.0367517	5430.81	199.973	5434.49
Reaction Moment(N.m)	0	0	0	0

Figure 43 shows equivalent static stresses on the structure. According to the graph, Maximum stress will be 12.3Mpa, which is very small compared to 505Mpa so the safety factor will be 41. Maximum and minimum structural stress values are visible. Minimum stress occurs in the side columns.

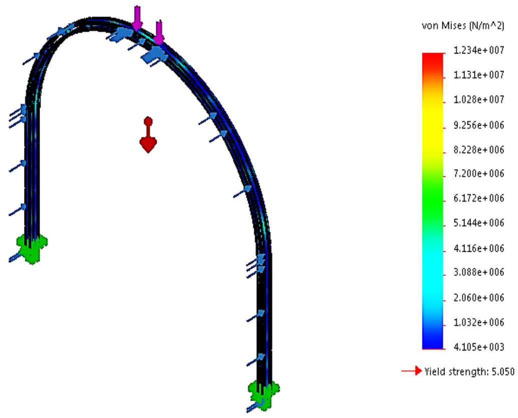
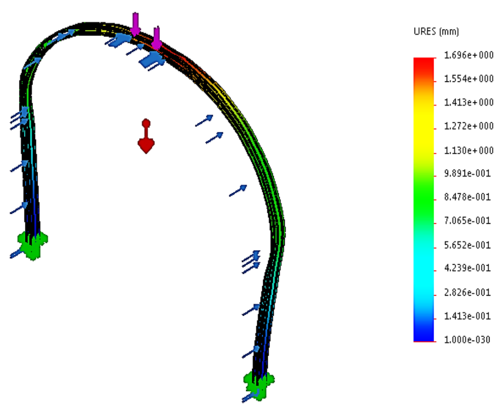


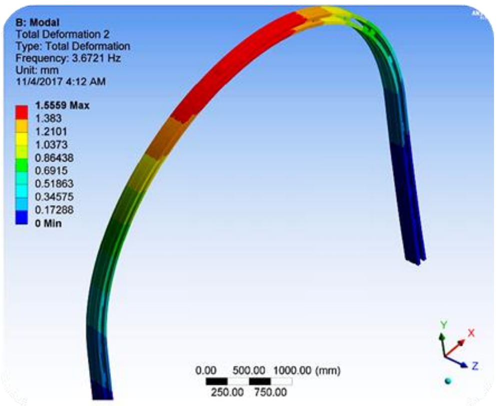
Fig. 43 Colorful curve of Von Mises stresses of the structure

Table 2 Modified static stress results

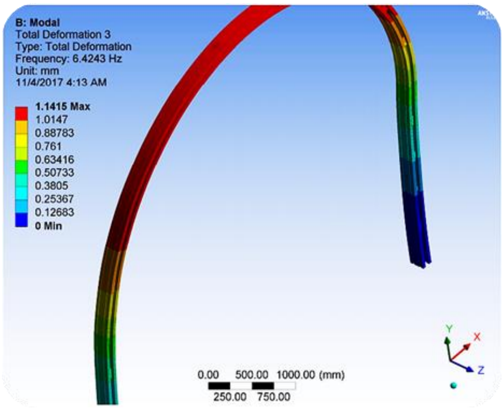
Name	Type	Min	Max
Stress1	VON: von Mises Stress	4.105e+003N/m^2 Node: 118719	1.234e+007N/m^2 Node: 19286

According to the figure 44, maximum displacement of the structure is approximately 1.7mm. according to the figure 44 this displacement is at the highest point of the structure and it is very small, so it does not occur an error in the system performance.





a) first natural modes 3.7Hz



b) first natural modes 6.4Hz

Fig. 44 Colorful curve of deflection of projector rail structure

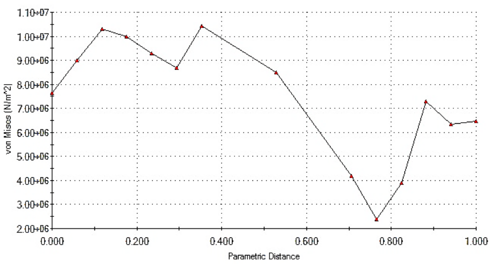
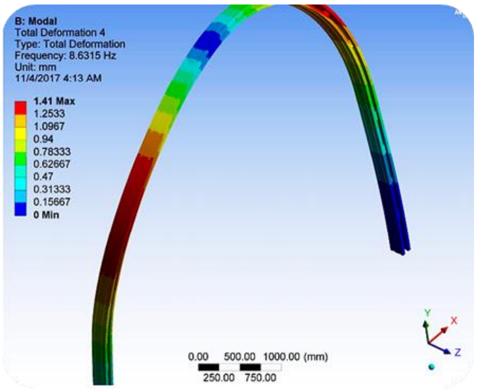
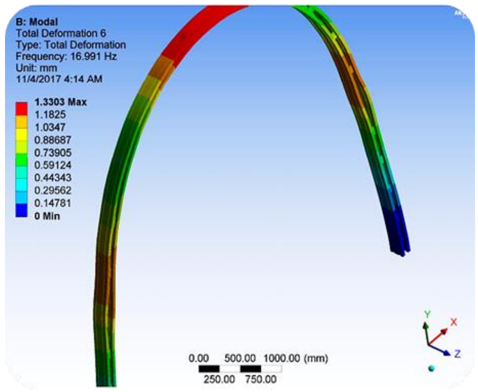


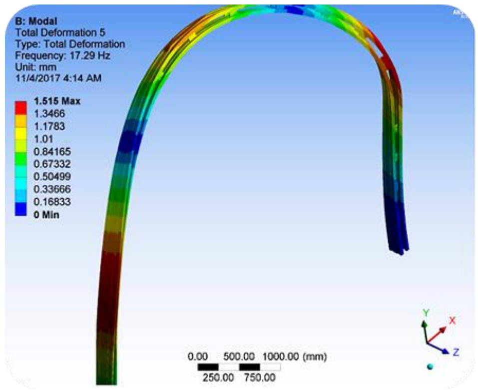
Fig. 45 Stress distribution around the hole of connection to the horizontal support structure



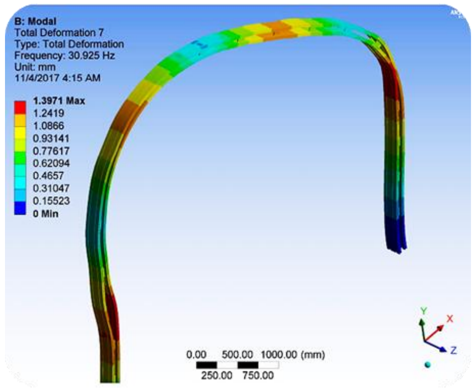
a) first natural modes 8.6Hz



b) first natural modes 17Hz



c) first natural modes 17.3Hz



d) first natural modes 31Hz

Fig. 46 Projector rail final design natural modes in Ansys

One of the sensitive areas of the structure is the holes of horizontal base. For this reason, the variation curve of the stress around one of the holes is shown in figure 45. As it is known the maximum stress in these areas is not more than 10Mpa.so the safety factor is more than 50. So the structure design is safe (from the static loading view)

5.3 Modal analysis with ANSYS workbench

The result of model analysis which is dynamic, will be analyzed with ANSYS workbench software to prove results. Figure 46 shows different modes. Natural frequencies are shown in table 3.

Table 3 Six first natural frequencies of the projector structure using the Ansys software

Mode	Frequency [Hz]
1.	3.6745
2.	6.4446
3.	8.6427
4.	17.046
5.	17.277
6.	31.026

The final results of the stress analysis are discussed in the following. Maximum displacement and maximum stresses in different vibrational directions are coming in the following table:

5.4 Frequency vibrations response of structural in vertical and lateral directions

Earthquake acceleration is $10mm/s^2$. The frequency response for the displacement of the highest point of the structure is plotted after the time analysis for the frequency range up to 50Hz. Details are shown in the following table.

To check in transverse direction, the input value is in the same value as before, and in Z direction.so, three natural frequencies are seen.

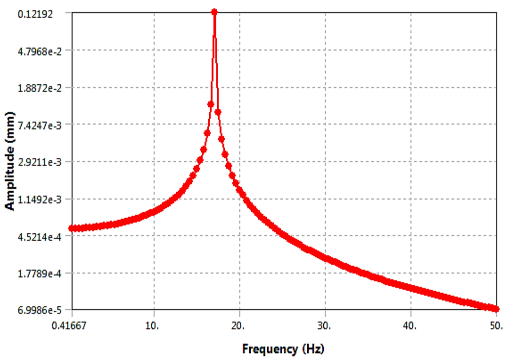


Fig. 47 Projector rail frequency response with vertical base excitation (Y direction)

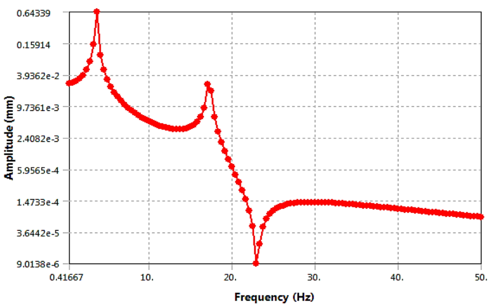


Fig.48 Projector rail frequency response with lateral base excitation

maximum displacement in this direction is 0.6mm which is very small and shows strength of the structure against dynamic and vibration loads.

5.5 Thermal analysis and effects of projector on ambient temperature

In order to modeling the thermal effects of the projector on the structure and environment, by assuming suitable air conditioning at the place and ambient temperature of 22 °C, thermal analysis has been done. In this analysis the upper surface is of the structure is the closest location to the projector and most sensitive location, temperature of this location is 100 °C. between the other surfaces and air the heat transfer is assumed convection type and a steady thermal analysis has been done in ANSYS software. The analysis characteristics are presented in table 4.

Table 4 Thermal loading of the structure

Object Name	*Temperature*	*Convection*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	1 Face	585 Faces
Definition		
Type	Temperature	Convection
Magnitude	100. °C (ramped)	
Suppressed	No	
Film Coefficient		Tabular Data
Coefficient Type		Average Film Temperature
Ambient Temperature		22. °C (ramped)
Convection Matrix		Program Controlled
Edit Data For		Film Coefficient
Tabular Data		
Independent Variable		Temperature
Graph Controls		
X-Axis		Temperature

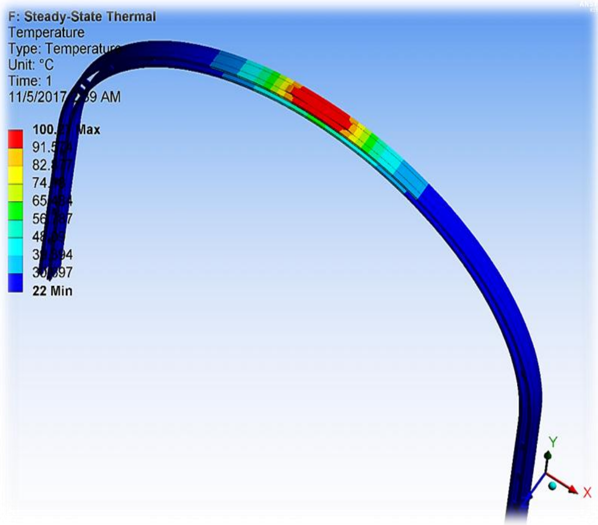


Fig. 49 Thermal distribution curve of the projector rail

After modeling and analyzing, temperature distribution of the structure will be determind.so the temperature reaches ambient temperature ($22°C$) at a distance of one meter above the surface.

6. TENSION ANALYSIS OF SENSITIVE COMPONENTS

By examining the system from dynamic, static, vibrational and thermal views, the structure and movable system have a suitable design, so it is possible to build this set up which has acceptable safety factor from fracture and deformation view. Dynamic analysis due to low acceleration in the system and excessive complexity on the rails seemed irrational, this is shown in the calculations in the first section and it does not analysis by software and just effects on the structure are investigated and kinematic is considered as a results of the calculation.

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