ASSIGNMENT

Module Code	MMD 509
Module Name	Kinematics &Dynamics of Machinery
Course	M.Sc in Machinery Design Engineering
Department	Mechanical & Manufacturing Engg.

...Igniting minds

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Batch	Full-Time 2011.
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Postgraduate Engineering and Management Programmes(PEMP)

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POSTGRADUATE ENGINEERING AND MANAGEMENT PROGRAMME – (PEMP)

Declaration Sheet

Student Name	P.Prabhakar		
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Course	Machinery design	Batch	Full-Time 2011.
Batch	FT-11		
Module Code	MMD509		
Module Title	Kinematics & Dynamics of Machinery		
Module Date	19-03-2012	to	14-04-2012
Module Leader	Asst Prof. Monish gowda		

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A copy of the extension approval must be attached to the assignment submitted.

Penalty for late submission

Unless you have submitted proof of mitigating circumstances or have been granted an extension, the penalties for a late submission of an assignment shall be as follows:

- Up to one week late: Penalty of 5 marks
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- More than Two weeks late: Fail - 0% recorded (F)

All late assignments: must be submitted to Academic Records Office (ARO). It is your responsibility to ensure that the receipt of a late assignment is recorded in the ARO. If an extension was agreed, the authorization should be submitted to ARO during the submission of assignment.

To ensure assignment reports are written concisely, the length should be restricted to a limit indicated in the assignment problem statement. Assignment reports greater than this length may incur a penalty of one grade (5 marks). Each delegate is required to retain a copy of the assignment report.

Declaration

The assignment submitted herewith is a result of my own investigations and that I have conformed to the guidelines against plagiarism as laid out in the PEMP Student Handbook. All sections of the text and results, which have been obtained from other sources, are fully referenced. I understand that cheating and plagiarism constitute a breach of University regulations and will be dealt with accordingly.

Signature of the student		Date	
Submission date stamp (by ARO)			

Signature of the Module Leader and date	Signature of Head of the Department and date

Abstract

In kinematics and dynamics of machinery, were kinematics involves in finding out the results of body in motion without considering the forces and dynamics involves in finding the result of the body in motion with considering the forces. These analyses are carried out by graphical, analytical and numerical method. The choice of approach to the solution is depend on the problem at hand and the available data.

In general mechanisms are used to obtain various types of motion, one of the motion is intermittent motion, to obtain the intermittent motion various type of mechanisms are used out of which cam mechanism provides the best intermittent motion is been explained in the part-A by comparing the advantage of cam over the other mechanism.

The wiper function is based on four bar mechanism were to obtain the crank rocker motion synthesis of link length is carried out and modeled in ADAMS 2010 motion & constrains are provided the result of displacement, velocity & acceleration is taken and the effect of change in result by changing the crank length and coupler length is funded and the results were compared and analysed.

The bearings are used to support the rotating shafts at the ends for the selection of the bearing the force on the bearing is required. In Part-C for the given mass, radius & rpm the shape is modeled in ADAMS and the force acting on the bearing was found and by changing the parameters of mass, radius and rpm the effect of change in the result was found and it is compared and analysed.

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List of Symbols

Symbol	Description	Units
ω	Angular acceleration	m/s^2
g	Acceleration due to gravity - 9.81	m/s^2
α	Angular velocity	m/s
ϴ	Angle	deg
s	Displacement	m
m	Mass	kg

PART-A

CHAPTER 1

1.0 Introduction & analysis of the statement:

Intermittent motion is the motion produced by mechanical device which starts and stops in a regular interval. The major considerations involved in adapting these mechanisms are a) The required indexing rate (number of steps per minute) b) Indexing accuracy required c) Dwell motion pattern required d) The load to be operated (heavy, medium or light duty) e) Cost consideration(an expensive mechanism is acceptable or to be medium or low cost).

The various mechanisms available for intermittent mechanism are:

Type of mechanism	Cycling rate (strokes/minute)	Dwell – motion ratio	Relative load capacity	Indexing precision	Relative cost
Cam	1000	Moderate	Very high	High	Moderate
Impulse ratchet	1500	High	low	Moderate	Low
Cycloid gearing	Few thousand	Very low	Low to moderate	Poor or fair	Moderate
Step motor	960000	Low to high	Low to high	Moderate	Moderate to high
Machine Geneva	Hundreds	Low	High	Moderate	moderate
Star wheel	Hundreds to thousand	Moderate	high	Moderate	Low to moderate

Table 1. 1 Various types of intermittent motion mechanism

From the above comparison table of the different mechanism shows *"Cam system is the best option as an intermittent motion mechanism"* in the various aspects like high load carrying capacity, high indexing precision and less relative cost of the mechanism etc.

1.2 Comparison of various mechanisms with cam:

Cam mechanism consist of usually two moving elements the cam and the follower. Cams can be designed to the required intermittent motion the cam accepts the input motion and provides the resultant motion to the follower which in turn will be as an intermittent motion.

1.2.1 Comparison between Cam & Step motor:

While comparing the cam mechanism with step motor the reliability is excellent for cam.

As the cam rotates the follower is guided, there will be the motion but in case of step motor the reliability depends on the power supply, and there will be more vibration in step motor as the rotor is held by the magnetic fields which behave like a spring. Cam mechanism also provides the mechanical advantage while function which is not possible by step motor. The motion control is excellent for cam than step motor. Step motor is not suitable for high rpm.

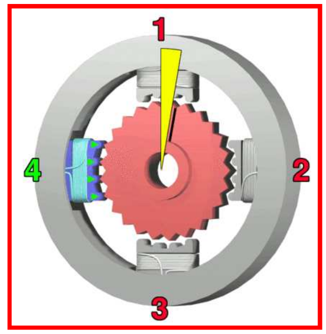


Fig 1. 1 Step motor[3]

1.2.2 Comparison between Cam & Ratchet mechanism:

The reliability of ratchet mechanism is good as it engages in tooth but not as good as cam mechanism and the ratchet mechanism don't offer any mechanical advantage in ratchet mechanism there is presence of impact while the ratchet engagement which is not there in cam. The indexing precision and the load carrying capacity are high for cam than ratchet mechanism.

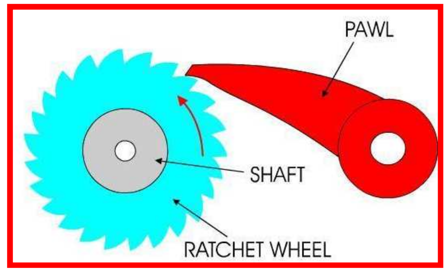


Fig 1. 2 Ratchet mechanism[2]

1.2.3Comparison of Cam & Cycloid gear mechanism

The cycloid gear has very less load carrying capacity than cam. The inputs to output the relative motion will be parallel in cycloid gear were as both parallel and perpendicular motion is achieved in cam. The cost of manufacturing of the cycloid gear is higher than cam. It has very less dwell motion ratio and less indexing precision than cam.

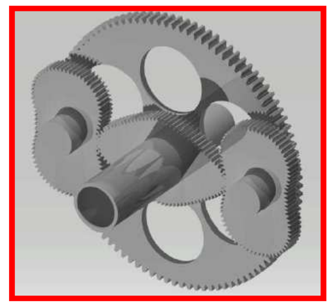


Fig 1. 3 Cycloid gear[4]

1.2.4 Comparison between Cam & Geneva mechanism:

The Geneva mechanism can be operated at relatively low SPM (strokes / minute) than cam mechanism. It can be used only for very slow motions. The load carrying capacity and the control over the load is less than cam mechanism.The indexing precision is less than cam mechanism.

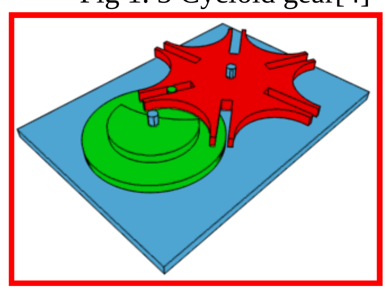


Fig 1. 4 Geneva mechanism[5]

1.2.5 Comparison between Cam& star wheel mechanism:

The star wheel mechanism has relatively low load carrying capacity than cam and has less indexing precision than cam. Fig 1.5 shows the 90^{o} indexing star wheel mechanism.

1.3 Case study:

To operate the diesel engine all the components must function very

Fig 1. 5 Star wheel

precisely and should have very precise relative motion to the piston movement. Cam shaft is used to accomplish this function. The cam shaft is the long cylinder shaft which has cam profiles in between called lobe; each lobe has a follower when the cam shaft rotates the follower moves up and down for each rotation by following the profile of the cam lobes. The followers are connected to the engine valves & fluid injectors by various linkages push rods & rocker arm. The valves are normally closed by springs. It opens & closes by push rod & rocker arm by rotation of cam lobe.

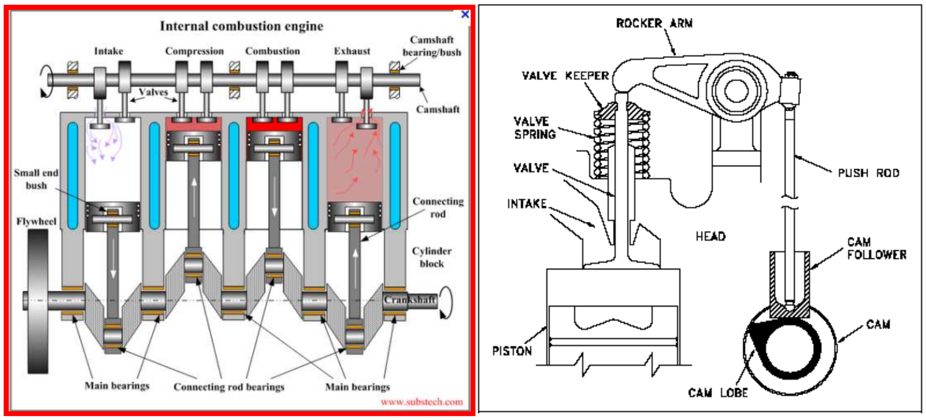


Fig 1. 6 Diesel engine valve mechanism[1]

1.3.1 Factors involved in selection of cam for intermittent motion:

In engine valve opening and closing is an intermittent motion, to operate this mechanism cam is used by which the process of oil intake, compression, combustion & exhaust takes place. The slight variation in valve operation will affect the function of engine as the cam has high indexing precision, good load carrying capacity and good control over the load than any other mechanism like cycloid gear, ratchet mechanism cam is used. As the engine is operated at the high temperature of *"Ambient to normal running temperature of about 190ºF"* [1] due to high temperature conduction takes place were the mechanism like stepper motor cannot be used but cam made of allowing those thermal expansion in result will not affect the intermittent motion. As the engine is operated at the high rpm, the mechanism like Geneva mechanism cannot be used which is suitable only for very less SPM (strokes/ minute)

1.4 Conclusion:

In the case study of diesel engine valve operation the cam is the only best option for intermittent motion and though in different mechanical assembly various mechanisms are used to obtain the intermittent motion other than cam but still cam has its own advantages and has wide application in mechanical parts and there is no substitute for cam in some mechanism.

PART-B

CHAPTER 2

Kinematic Synthesis and Analysis of Wiper Mechanism

2.0 Introduction

In cars wind shield are cleaned by wiper from the dust and water due to rain is swept away by the wiper which is operated using a motor. The wiper to and fro motion is achieved by the motor and worm gear. The wiper arm is of rubber which is always in contact with the wind shield and moves to and fro for cleaning.

Fig 2. 1 Car wiper[6]

2.1 Four bar mechanism:

Four bar mechanism generally consist of four links 1) Ground –The link of frame of reference 2) crank – The link connected to motor, 3) coupler – The link connected to crank and rocker 4) Rocker – The link from which the output motion is obtained. The four bar mechanism is used to change the type, speed and size of movement or motion. It can also convert vertical movement into horizontal movement as in the car wipers the coupler moves horizontally but the output rocker movement is in vertical.

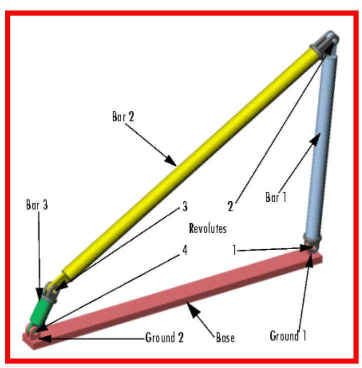


Fig 2. 2 Four bar mechanism[7]

2.2 Linkage synthesis:

Linkage synthesis is the process of calculating the dimensions for the four links in four bar mechanism based on which the desired output angle of rocker is achieved. The range of the wiper depends on the lengths of the links which is determined by linkage synthesis.

Given data:

i) Initial input = 0 to 180°

ii) Final output = 60 to 130°

iii) Number of steps = 4

iv) Number of precision points = 3

v) Function : $Y = X^2$

Chubchu spacing equation:

Chubchu's spacing equation provides the precision points of crank (X) and rocker(Y). These precision points are the points were there are no error and using which at the different angle of the crank the value of the angle of the rocker is obtained.

$$X_n = \frac{X_i + X_f}{2} - \frac{X_f + X_i}{2} \cos\left[\frac{2L-1}{2K} x\ 180\right]$$

$X_n =$ Chubchu spacing equation

$X_i =$ Initial position of crank link

$X_f =$ Final position of crank link

$L =$ Step number(1 to n)

$K =$ Total number of precision points

2.3 Calculation:

$$X_1 = \frac{4+1}{2} - \frac{4-1}{2} \cos\left[\frac{2(1)-1}{2(3)} x\ 180\right]$$

$$\boxed{X_1 = 1.2}$$

$$X_2 = \frac{4+1}{2} - \frac{4-1}{2} \cos\left[\frac{2(2)-1}{2(3)} x\ 180\right]$$

$$\boxed{X_2 = 2.5}$$

$$X_3 = \frac{4+1}{2} - \frac{4-1}{2} \cos\left[\frac{2(3)-1}{2(3)} x\ 180\right]$$

$$\boxed{X_3 = 3.8}$$

Function: $Y=X^2$

As $X_1 = 1.2$

$Y_1 = 1.2^2 = 1.44$

As $X_2 = 2.5$

$Y_2 = 2.5^2 = 6.25$

As $X_3 = 3.8$

$Y_3 = 14.44$

The values of precision points of crank(X) and rocker(Y) are:

$X_1 = 1.2$	$Y_1 = 1.44$
$X_2 = 2.5$	$Y_2 = 6.25$
$X_3 = 3.8$	$Y_3 = 14.44$

Finding the slope r_x with respect to input links.

$$\boxed{r_x = \frac{\theta_f - \theta_i}{x_f - x_i}}$$

$$r_x = \frac{180 - 0}{4 - 1}$$

$$\boxed{r_x = 60}$$

$$\theta_1 = (x_1 - x_i)r_x + \theta_i$$

$$\theta_1 = (1.2 - 1)60 + 0$$

$$\boxed{\theta_1 = 12^o}$$

$$\theta_2 = (x_2 - x_i)r_x + \theta_i$$

$$\theta_2 = (2.5 - 1)60 + 0$$

$$\boxed{\theta_2 = 90^o}$$

$$\theta_3 = (x_3 - x_i)r_x + \theta_i$$

$$\theta_3 = (3.8 - 1)60 + 0$$

$$\boxed{\theta_3 = 168^o}$$

$$r_y = \frac{\Phi_f - \Phi_i}{y_f - y_i}$$

$$r_y = \frac{130 - 60}{16 - 1}$$

$$\boxed{r_y = 4.66}$$

$\Phi_1 = (y_1 - y_i)r_y + \Phi_i$

$\Phi_1 = (1.44 - 1)4.66 + 60$

$\boxed{\Phi_1 = 62.05^o}$

$\Phi_2 = (y_2 - y_i)r_y + \Phi_i$

$\Phi_2 = (6.25 - 1)4.66 + 60$

$\boxed{\Phi_2 = 84.46^o}$

$\Phi_3 = (y_3 - y_i)r_y + \Phi_i$

$\Phi_3 = (14.44 - 1)4.66 + 60$

$\boxed{\Phi_3 = 122.63^o}$

The table shows at the precision points of crank(X) and rocker(Y) for the different angle of crank (ɵ) the angle of rocker(Φ) is:

$\Theta_1 = 12^o$	$\Phi_1 = 62.05^o$
$\Theta_2 = 90^o$	$\Phi_2 = 84.46^o$
$\Theta_3 = 168^o$	$\Phi_3 = 122.63^o$

For four bar mechanism – to obtain the crank rocker mechanism the fudenstain equation is used for finding the link lengths.

$\boxed{\cos(\theta - \Phi) = K_1\cos\Phi - K_2\cos\theta + K_3}$

$\cos(12 - 62.05) = K_1\cos 62.05 - K_2\cos 12 + K_3$

$\cos(90 - 84.46) = K_1\cos 84.46 - K_2\cos 90 + K_3$

$\cos(168 - 122.63) = K_1\cos 122.63 - K_2\cos 168 + K_3$

$\boxed{K_1=2.45;\ K_2=1.29;\ K_3=0.75}$

As given link length a=30mm

$\boxed{K_1 = \frac{d}{a}}$

$2.45 = \frac{d}{30}$

$d = 73.5\text{mm}$

$\boxed{K_2 = \frac{d}{c}}$

$1.29 = \frac{73.5}{c}$

$c = 56.97\text{mm}$

$$K_3 = \frac{a^2 - b^2 + c^2 + d^2}{2ac}$$

$$K_3 = \frac{30^2 - b^2 + 56.97^2 + 73.5^2}{2(30)(56.97)}$$

$b = 97.71\text{mm}$

Therefore the link lengths are:

a = 30mm

b = 97.71mm

c = 56.97mm

d = 73.5mm

2.4 Result interpretation from synthesis:

From the link lengths obtained from the calculation for the four bar mechanism satisfies the Grashof condition so it is possible to obtain the crank and rocker mechanism.

Grashof conditions:

i) The sum of smallest link + longest link length < The sum of length of the other two link length

= 30 + 97.71 < 56.97 + 73.5

This condition is satisfied as the value 127.71 < 130.47.

ii) The crank link should rotate 360° and the rest of the links should oscillate less than 180°.

As these conditions are satisfied it is possible to obtain crank and rocker mechanism. Therefore the result obtained by the synthesis can be used in ADAMS software to to find the results for kinamatic analysis.

2.5 Creation of wiper mechanism in ADAMS:

After synthesis of the link lengths from the calculation which is the result obtained are ground length is 73.5mm, crank length is 30mm, coupler length is 97.71mm and rocker length is 73.5mm. To create the same link length in ADAMS 2010 initially using CATIA V5 R16 software with the same link lengths mechanism is created and the mechanism is simulated and the crank and rocker motion is verified in CATIA then the X and Y axis co-ordinate for the marker is taken from CATIA which is the input to ADAMS as shown in the Fig 2.3. By keeping the co-ordinate as the input a marker is created in ADAMS as shown in the Fig 2.4.

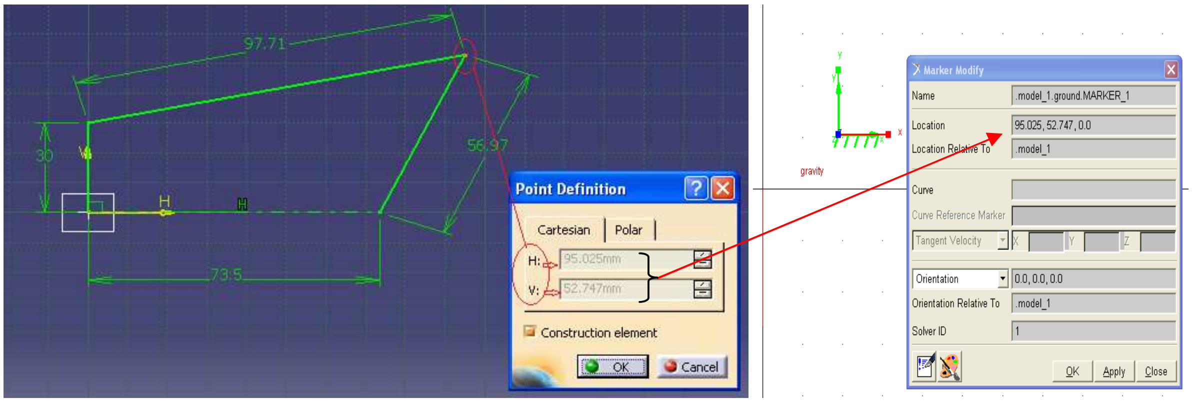
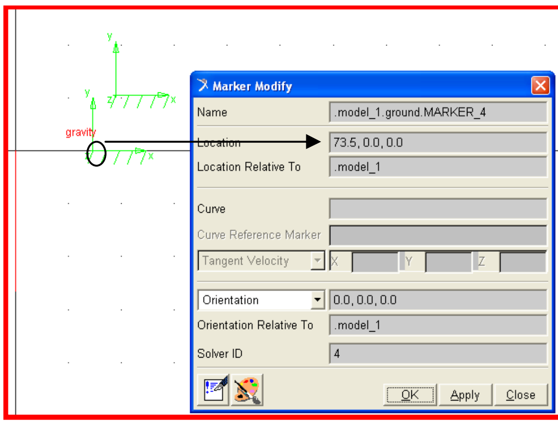


Fig 2. 3 Marker creation using CATIA input

By keeping the marker as the reference and as the ground length is known from the calculation as 73.5mm the other marker is created as shown in the Fig(). By using these two markers four bar mechanism is generated to the required link length as shown in the Fig().



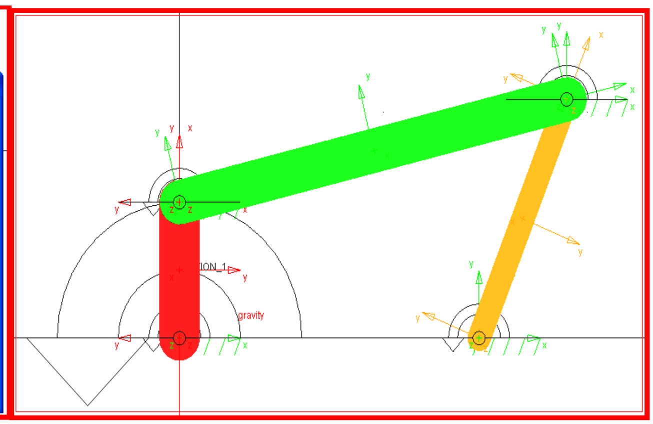


Fig 2. 4 Four bar link creation in ADAMS Fig 2. 5 Marker creation in ADAMS

2.6 Joints, Constrains &Motion for mechanism:

The four bar mechanism has four revolute joints as shown in the Fig 2.6

1) Between ground crank
2) Between crank and coupler
3) Between crank and rocker
4) Between rocker and ground

2.6.1 Revolute joint:

The revolute joint has one degree of freedom which can able only to rotate about its axis. The first joint between the ground and the crank & is provided as the crank is rotated without any to 360º fixed to the pivoted point on the ground. The second revolute joint is provided between the crank and the coupler as the crank rotates to the 360° the coupler end connected to the crank will also rotates to the 360° therefore it requires the revolute joint at this joint. The next revolute joint is between the coupler end and rocker end this is provided as for the input motion of the crank the rocker end will be oscillating in the path of radius. The last revolute joint between the rocker end and ground this is provided as this end has to swing about its axis without translation about the pivoted point for the input motion of the crank.

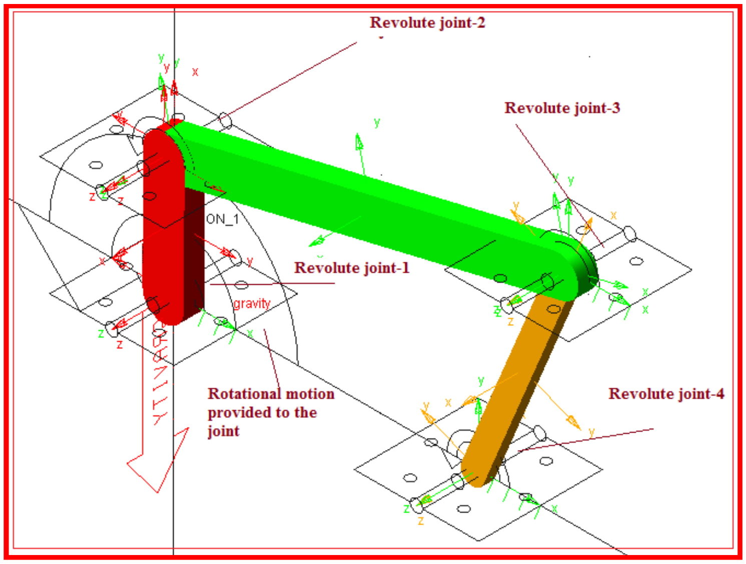


Fig 2. 6 Joints & motion for 4 bar mechanism

2.6.2 Rotational motion:

As the Grashof condition is satisfied by the link lengths and to obtain the crank and rocker mechanism the crank is to be rotated to the 360° the rotation motion is given to the revolute joint between the crank and the ground as shown in the Fig 2.6. As the motion is notion is not specified the default value is taken for the motion 30 degrees/second

2.6.3 Constrain for wiper:

The link for wiper is connected to rocker of 25mm length and the fix constrain is applied between the rocker and wiper so that the wiper oscillate to the same angle of the rocker as shown in the Fig 2.7.

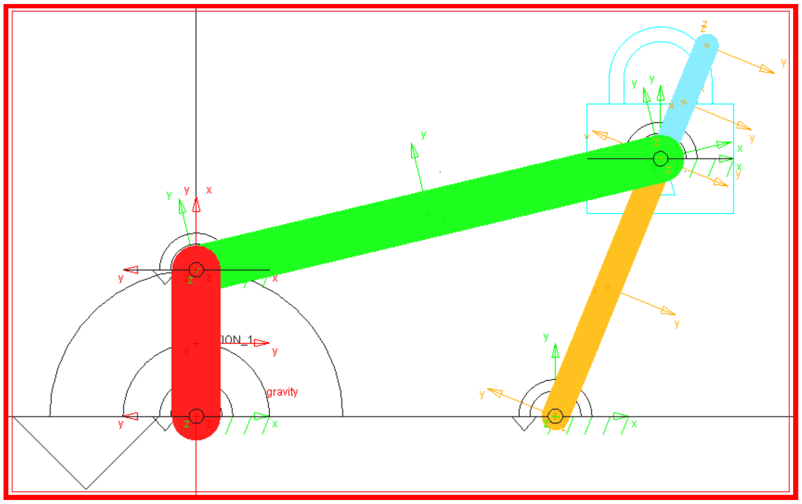


Fig 2. 7 Four bar mechanism with wiper

2.7 Kinematic analysis carried out in ADAMS:

For the lengths obtained in synthesis, the kinetic analysis is carried out for the mechanism in ADAMS by considering the link as mass less links and considering the rotation motion as 30 deg/sec the default value is taken the wiper length is taken as 25mm. The results of the 4 bar mechanism is plotted in graph are shown below.

2.7.1 Rocker's angular velocity graph:

In the Fig 2.9 the rocker link's angular velocity to time graph is shown. The angular velocity is the rate of angular displacement of the rocker. This shows the angular speed of the rocker and the axis about its pivot position. In the graph Y-axis shows the angular speed and the X-axis shows the time to complete one cycle.

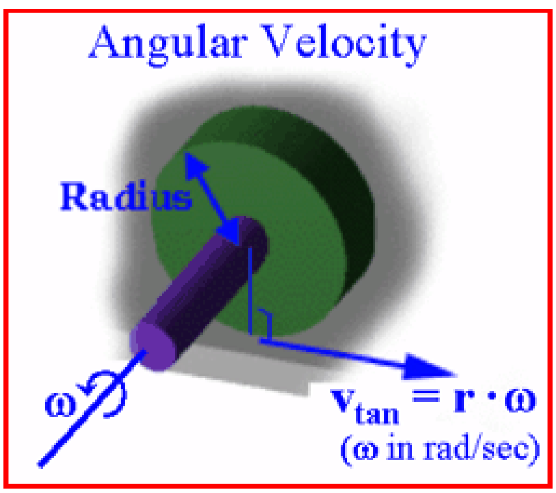


Fig 2. 8 Angular velocity [8]

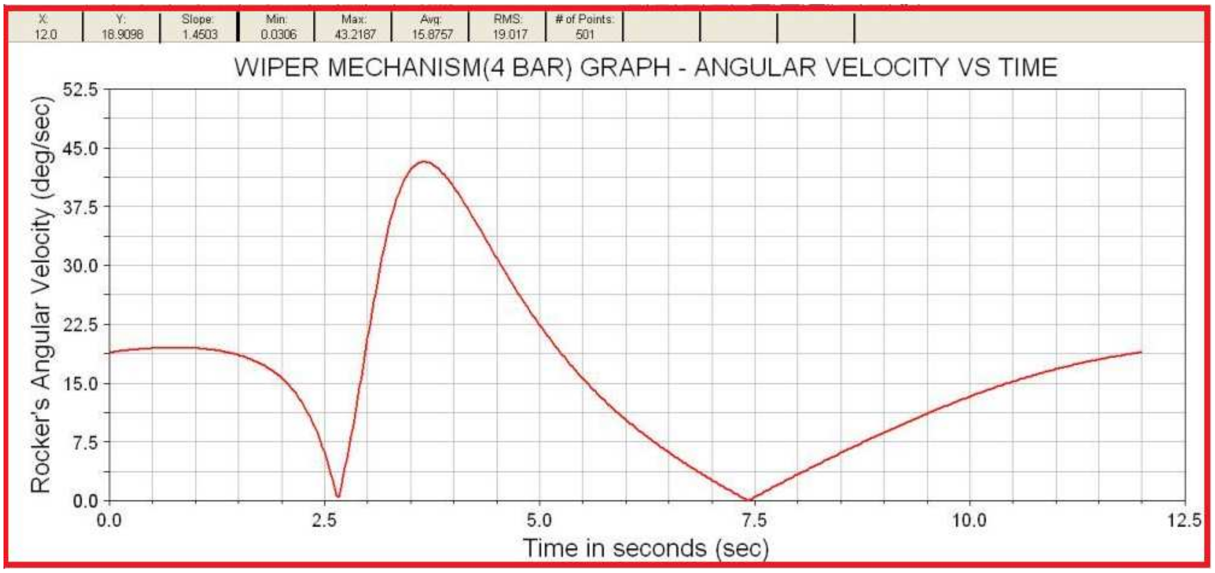
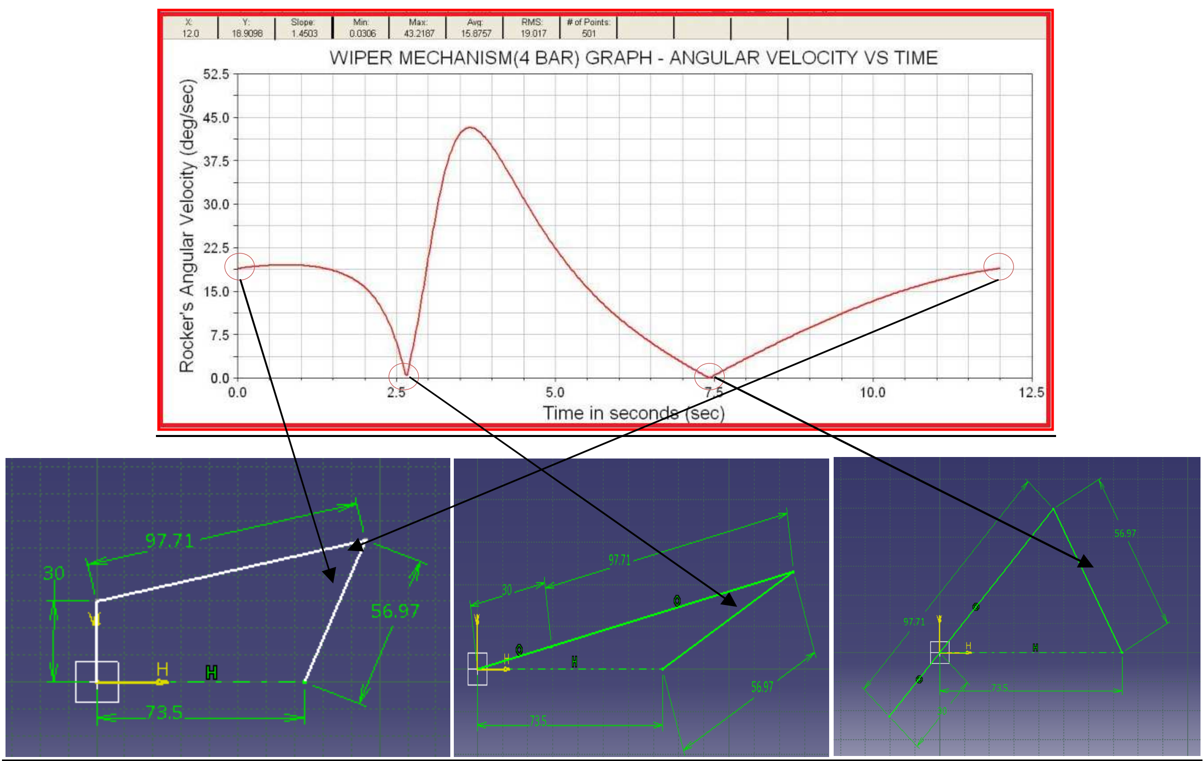


Fig 2. 9 Angular velocity to time graph

2.7.2 Comments on the result obtained:

- The crank initial position can be considered as 90º to the frame of reference at which the angular velocity of the rocker is 18.90 deg/sec as shown in Fig(a).



Fig(a) Fig(b) Fig(c)

- When the crank link and the coupler link becomes collinear while the crank angle is 28.80º the rocker reaches the maximum range as shown in Fig(b) were the angular velocity of the rocker 0.030 deg/sec.
- When the crank link and the coupler link becomes collinear again while the crank angle is -132.63º to the frame of reference the rocker reaches the other end of the maximum range were the angular velocity of the rocker comes to 0.030 deg/sec.
- When the crank again reaches the intial position of the 90º the cycle completes were the angular velocity is 18.90deg/sec.
- From the graph the time taken for the return stroke is 4.60 seconde as the cycle time is 12 seconds the time for the forward stroke is 7.40second there fore it comes under quick return mechanism and the quick return ratio is 0.62.

2.7.3 Wiper’s displacement graph:

In the Fig 2.11 the Wiper link’s displacement to crank angle graph is shown. The wiper displacement is the difference between the initial positions to the final position of the wiper with respect to the rotation of the crank of 360º in a cycle. In the graph Y-axis shows the Wiper displacement and the X-axis shows the crank angle to complete one cycle.

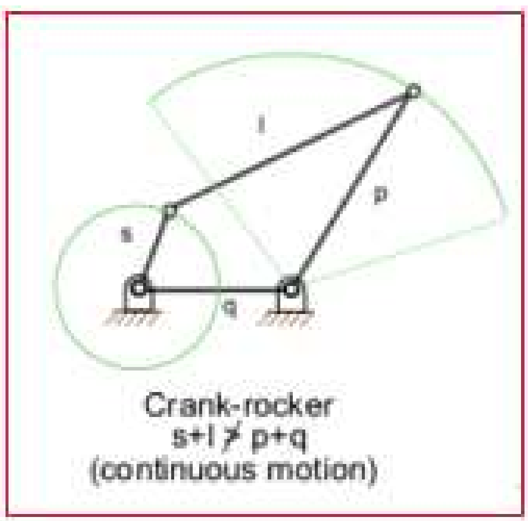


Fig 2. 10 Displacement of Wiper[9]

2.7.4 Rocker's displacement graph:

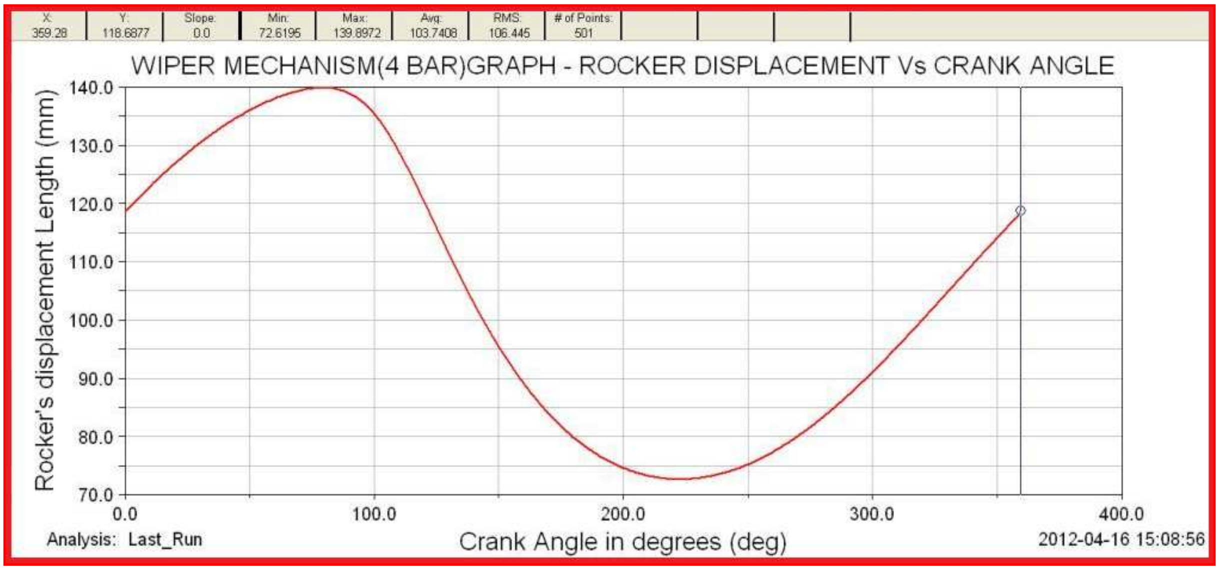


Fig 2. 11 Displacement to Crank angle graph

2.7.5 Comments on the result obtained:

The crank initial position can be considered as 90° to the frame of reference at which the position of the rocker 118.68mm when the crank reaches the angle of 28.80° the crank link and the coupler link becomes collinear and the rocker reaches the maximum range and completes the stroke were the maximum displacement of 118.68mm. When the crank reaches to the angle of 132.68° the rocker and the crank becomes collinear again the rocker reaches the maximum range of the other end were the rocker takes the minimum displacement of 72.61mm in graph and while the crank reaches the initial position of 90° the cycle completes and reaches to the initial value of 118.68mm. Therefore from the graph the total displacement taken by the rocker is (118.68 – 72.61) = 46.07mm. As the motion is to and fro oscillating motion we get sinusoidal curve per each cycle of the crank rotation as shown in the graph.

2.7.6 Rocker's angular acceleration graph:

In the Fig 2.13 the rocker link's angular acceleration to time graph is shown. The angular acceleration is the rate of change of angular velocity of rocker. In the graph Y-axis shows the angular acceleration and the X-axis shows the time to complete one cycle.

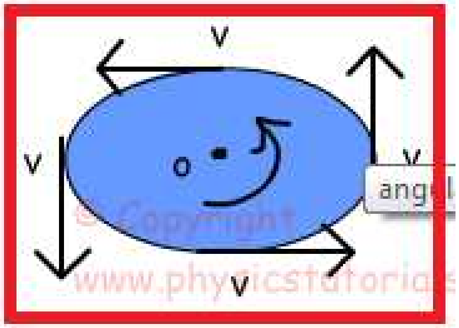


Fig 2. 12 Angular acceleration[10]

2.7.7 Rocker's angular acceleration graph:

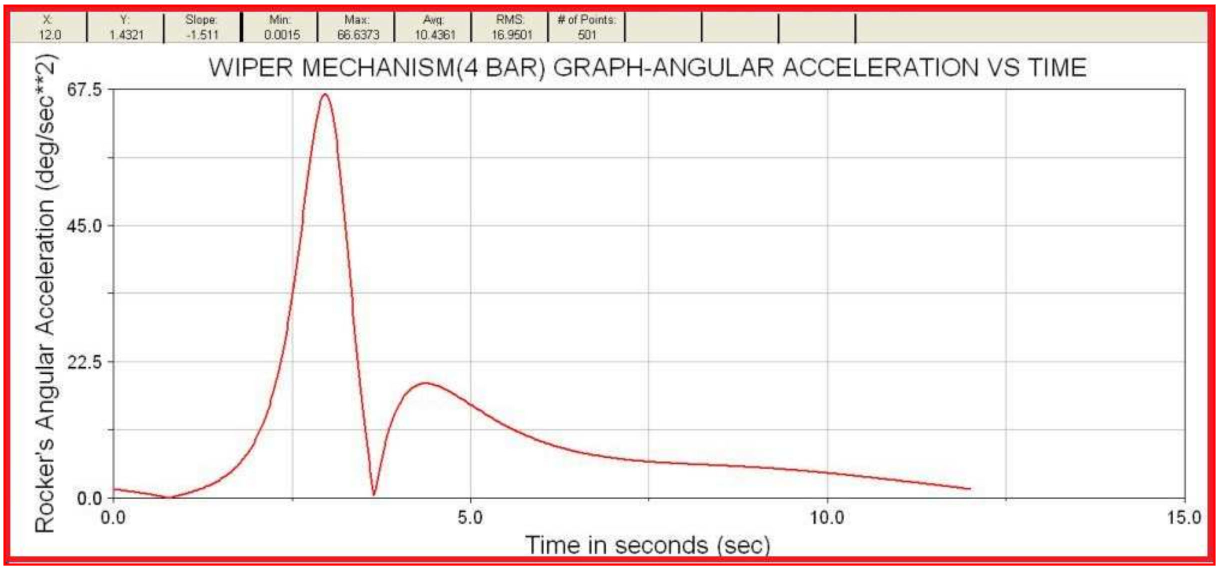


Fig 2. 13 Angular acceleration to time graph

2.7.8 Comments on the result obtained:

The crank initial position can be considered as 90º to the frame of reference at which the angular acceleration of the rocker is 1.43 deg/sec^2. At the crank angle of 28.80 º the crank link and the coupler link becomes collinear and the rocker reaches the maximum range and completes the stroke were the angular acceleration reaches to 0 deg/sec^2. And for the further rotation of the crank the rocker takes the return stroke during this When the crank reaches to the angle of 132.68º the rocker and the crank becomes collinear again the rocker reaches the maximum range of the other end were the return stroke completes in between the stroke the rocker angular acceleration reaches to the maximum of 66.63 deg/sec^2 and at the ending of the stroke that is at the crank angle of 132.68º again the acceleration of the rocker reaches to the 0 deg/sec^2. Then the forward stroke starts at the further rotation of the crank were the angular acceleration is very less and it is gradually reducing to 0 deg/sec^2 which takes more time than the time taken for forward stroke which satisfies the quick return mechanism.

2.8 Comparison of angular acceleration results:

The graph below Fig 2.14 shows the comparison of the result of the three conditions. Were Y-axis shows the angular acceleration and the X-axis shows the time to complete one cycle.

The result obtained for the lengths obtained by the synthesis.

A) The result obtained for the link length AB = 25mm.

B) The result obtained by changing the link length AB = 25mm.

C) The result obtained by changing the link length BC = 66.20mm.

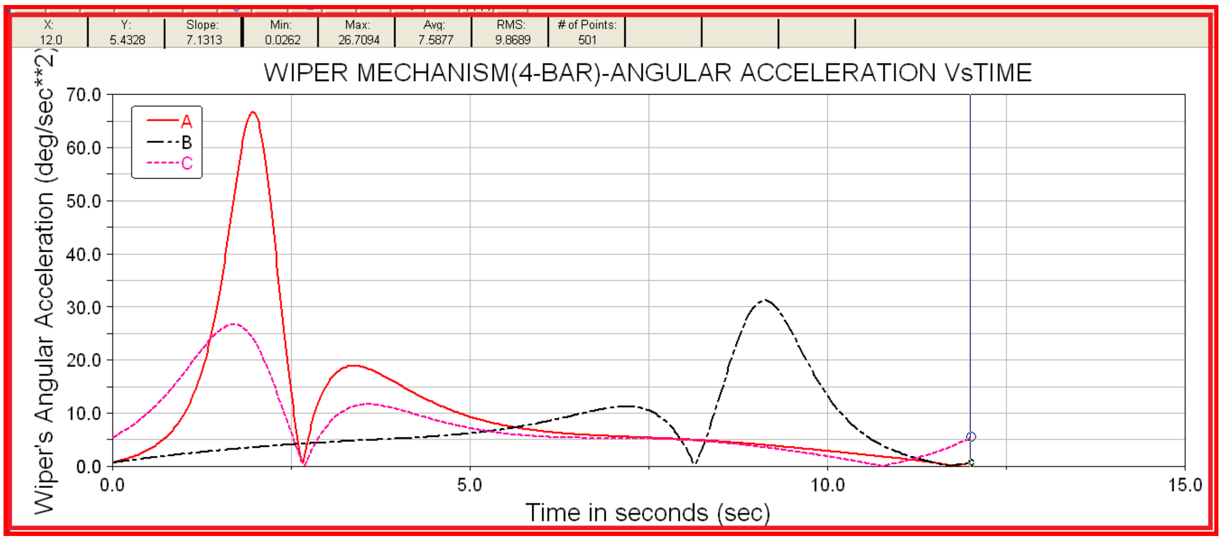


Fig 2. 14 Angular acceleration comparison graph

2.8.1 Comments on the result obtained:

In the graph the curve A takes the maximum acceleration than curve B & curve C as for all the 3cases the cycle time is 12 seconds for the curve A and curve B were the link lengths are 30mm and 25mm in the return stroke the acceleration is reaches the maximum and in return stroke the acceleration gradually reduces and reaches to zero and in forward stroke the acceleration is not so high as in return stroke but slightly increases and come back to zero with more time. But in curve C by changing the length BC the result obtained in the curve is in reverse to the previous cases of A and B in curve C in the return stroke the acceleration gradually increases from zero to reaches back to zero in more time of 8.5 seconds and for the return stroke though the graph reaches to maximum it reaches to zero and the forward stroke completes in short span of time.

2.8.2 Comparison of angular velocity results:

The graph below Fig 2.15 shows the comparison of the result of the three conditions. Were Y-axis shows the angular velocity and the X-axis shows the time to complete one cycle.

A) The result obtained for the lengths obtained by the synthesis.

B) The result obtained by changing the link length AB = 25mm.

C) The result obtained by changing the link length BC = 66.20mm.

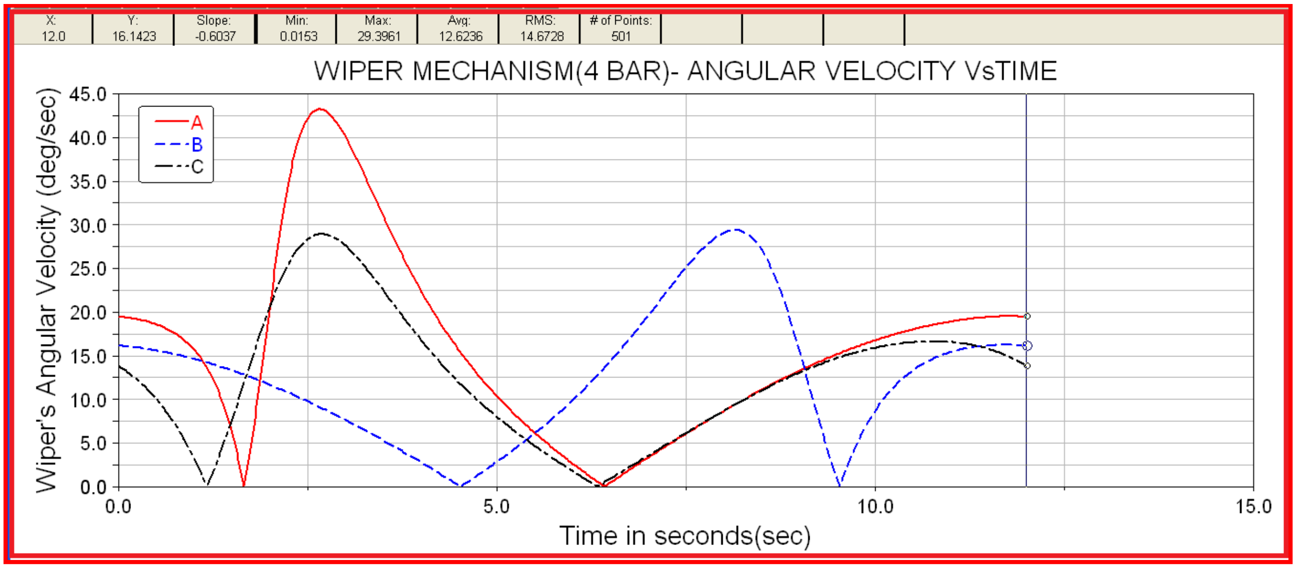


Fig 2. 15 Angular velocity comparison graph

2.8.3 Comments on the result obtained:

In the graph the curve A takes the maximum angular velocity than curve B &curve C as for all the 3cases the cycle time is 12 seconds for the curve A and curve B were the link lengths are 30mm and 25mm in the return stroke the angular velocity reaches the maximum and in return stroke the angular velocity gradually increases and again decreases reaches to zero the time taken for the velocity change in forward stroke is more and return stroke is less. But in curve C by changing the length BC the result obtained in the curve is in reverse to the previous cases of A and B in curve C in which the graph shows the time taken to complete the Angular velocity in forward stroke is equal to return stroke which doesn’t support the quick return mechanism.

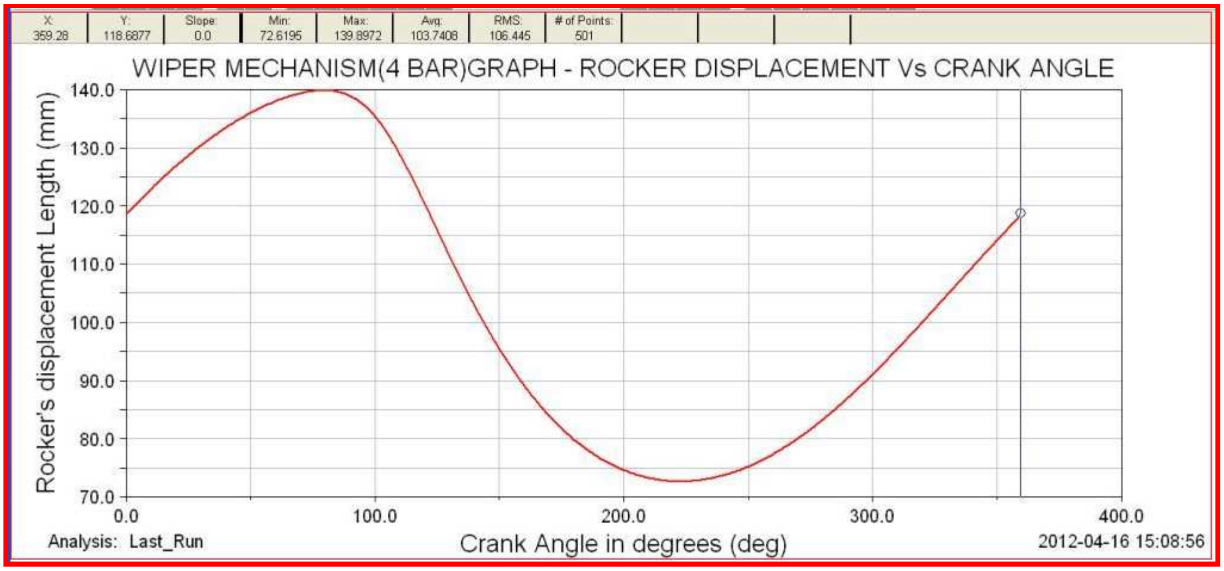
2.8.4 Comparison of wiper displacement results:

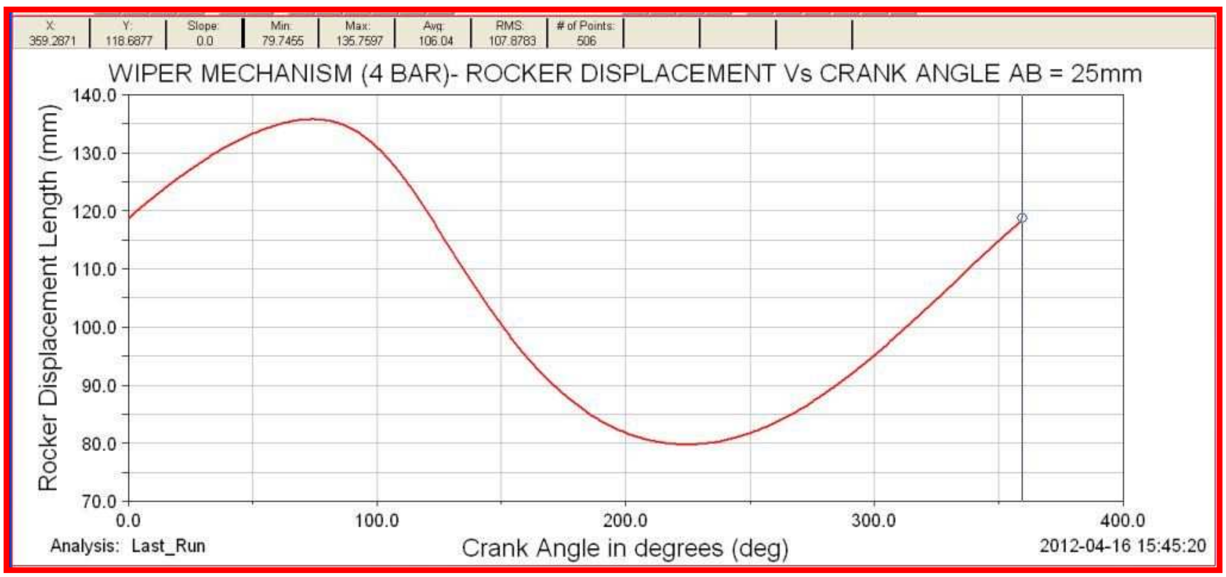
The graph below Fig 2.16 shows the comparison of the result of the three conditions. Were Y-axis shows the displacement and the X-axis shows the crank angle taken to complete one cycle.

A) The result obtained for the lengths obtained by the synthesis.

B) The result obtained by changing the link length AB = 25mm.

C) The result obtained by changing the link length BC = 66.20mm.





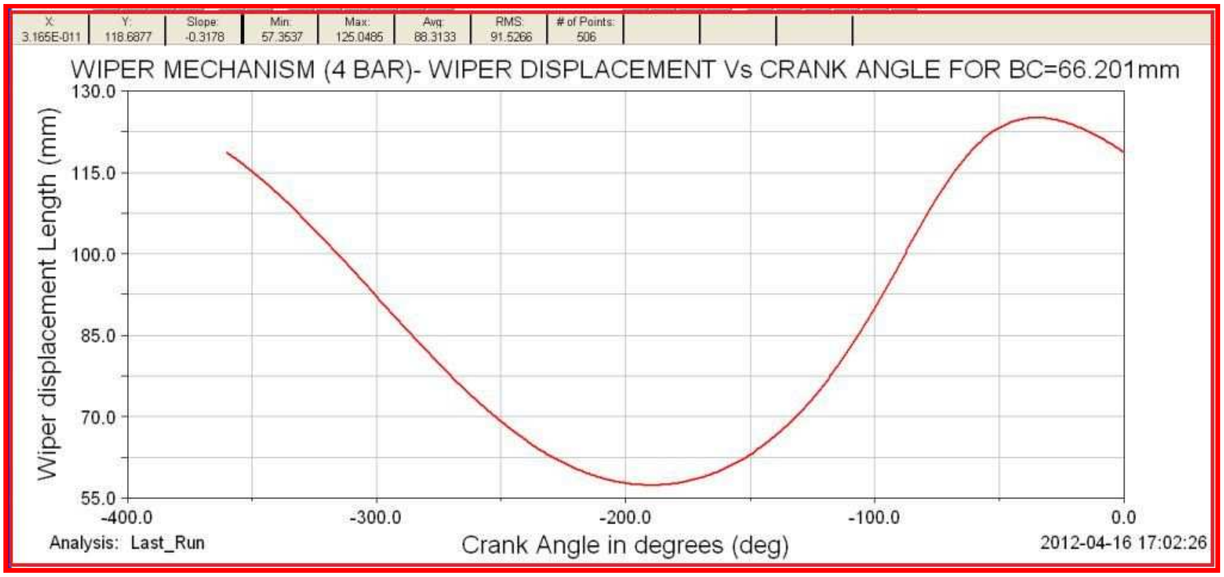


Fig 2. 16 Comparison of displacement result graph

2.8.5 Comments on the result obtained:

The wiper displacement graph is plotted against the crank rotation angle were in the A-graph for the crank length AB = 30mm the displacement of the wiper is (139.89 – 72.61) = 67.28mm. From the B-graph were the crank length is reduced to AB=25mm the displacement of the wiper is (118.68-79.74) = 38.94mm. As the result of reducing the crank length the mechanism is more of grashof condition and therefore the displacement is reduced to the greater extent. From the C-graph were the coupler length is changed were BC = 66.20mm the displacement of the wiper obtained is (125.04 – 57.36) =67.68mm from this result. In the C- graph the displacement is more than the other two graph as the coupler length is decreased the mechanism is closer to the non grashof condition and therefore the displacement is more than A & B graph.

2.8.6 Conclusion:

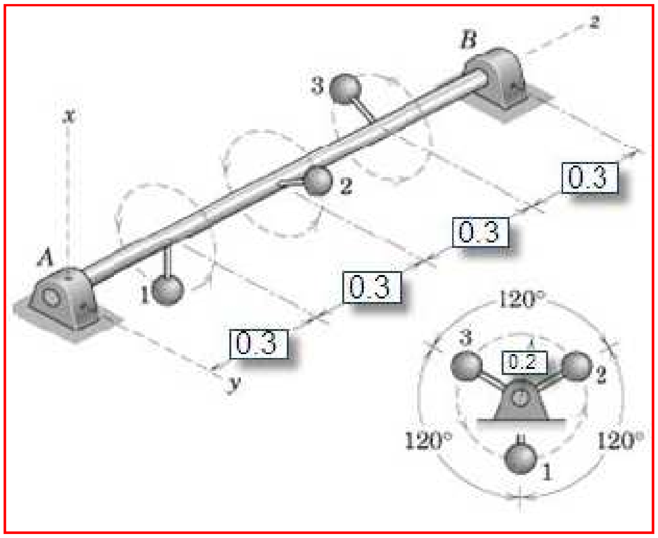
By comparing the graphs of displacement, angular velocity and angular acceleration of the three cases for the link length obtained in synthesis, by changing the crank length AB and by changing the coupler length BC the best result obtained among these is the link lengths constructed with the values obtained from the synthesis mechanism provides the best result as even the displacement value is slightly less than the value obtained by changing coupler length but since the synthesis length supports the quick return mechanism in better way than other results it is considered as the suitable lengths for 4bar wiper mechanism.

PART-C

Dynamic analysis of shaft

3.0 Problem statement:

A 1.2 m long shaft carries three small 2.0-kg masses attached by mass less rods. The shaft rotates at the constant speed of 1200 rev/min in 2 bearings. Compute the forces RA and RB acting on the bearings. Gravitational forces may be neglected. How do the forces RA and RB acting on the bearings change as the mass of spheres and the angular speed at which the mechanism is turned is varied? How do you think the forces RA and RB would change if the length of the three mass less rods were made longer than 0.2 m?



Consideration for calculation:

- The masses are rotating at the different planes.
- The central shaft is considered as mass less shaft.
- The masses are placed in equal distance and the weight of the masses is same therefore the masses are properly balanced.
- As the masses are properly balanced the forces acting on the in the both the sides will be same.

Given data:

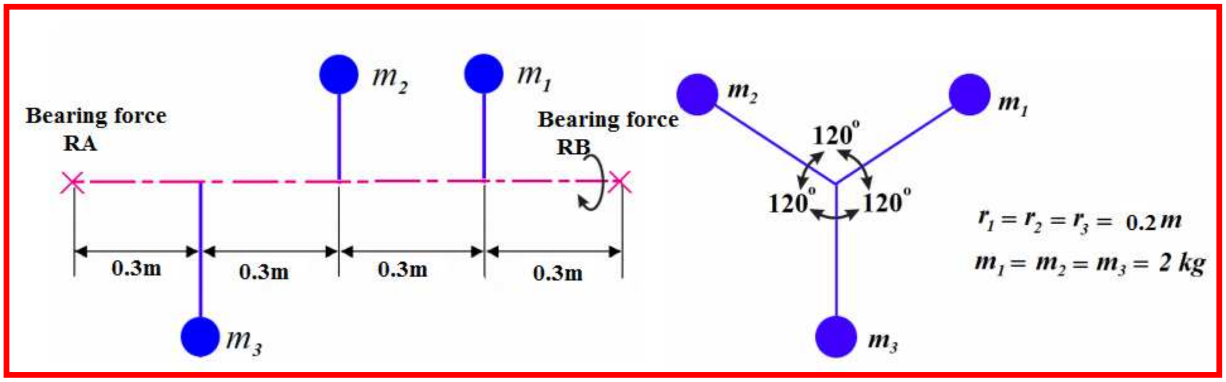
Weight of sphere = 2kg

Radius of the hanging mass = 0.2m

Total length of the shaft = 1.2m

Distance between the planes = 0.3m

3.1 Calculation:



PLANE	MASS (m) in kg	RADIUS (r) in m	mr	ɵ in degree	mr Cosɵ	mr Sinɵ	LENGTH (L)	mrL	mrL Cosɵ	mrL Sinɵ
A	A	R	RA	0	RA*Cosɵ	RA*Sinɵ	0	0	0.00	0.00
1	2	0.2	0.4	0	0.40	0.00	0.3	0.12	0.12	0.00
2	2	0.2	0.4	120	-0.20	0.34	0.6	0.24	-0.12	0.20
3	2	0.2	0.4	240	-0.20	-0.34	0.9	0.36	-0.18	-0.31
B	B	R	RB	0	RB*Cosɵ	RB*Sinɵ	1.2	1.2RB	1.2RB* Cosɵ	1.2RB* Sinɵ
									-0.18	-0.11

$1.2\text{RB} = \sqrt{(\text{mrlsin}ɵ)^2 + (\text{mrlcos}ɵ)^2}$

$1.2\text{RB} = \sqrt{(-0.18)^2 + (-0.11)^2}$

RB = 0.175

The force on bearing (F) = $mr\omega^2$

$\omega = \dfrac{2\pi N}{60}$

$\omega = \dfrac{2 \text{ x } \pi \text{ x} 1200}{60}$

ω = 125.663 rad/sec

$mr\omega^2 = 0.175 \text{ x } 125.663^2$

$mr\omega^2$ = 2763.45N

we know RB = 0.175 therefore RB cosɵ = 0.175 cos0 & RB sinɵ = 0.175sin0

Substituting o.175 for RB cosɵ in mr cosɵ & 0 for mr sinɵ

We get $RA = \sqrt{(mrsinɵ)^2 + (mrcosɵ)^2}$

$RA = \sqrt{(0.175)^2 + (0)^2}$

RA = 0.175

The force on bearing (F) = $mr\omega^2$

$$\omega = \frac{2\pi N}{60}$$

$$\omega = \frac{2 \text{ x } \pi \text{ x}1200}{60}$$

ω = 125.663 rad/sec

$mr\omega^2$ = 0.175 x 125.663^2

$mr\omega^2$ = 2763.45N

3.2 Modeling in ADAMS:

- The model is created by creating 4 ground marker 0.3m distances from the origin.
- Creation of the mass less rod of length 1.2m.
- Creation of 3 ground marker at the co-ordinates (0.3,-0.2), (0.6,-0.2) & (0.9,-0.2).
- Creation of the 3 mass less connecting the two ground marker created to the distance 0.2m apart.
- Rotating and positioning of the mass less rod to120º apart.
- Creation of spherical masses at the ends of the 3 mass less rod.

3.3 Constrains and joints for the model:

- Fix constrain is given between the 3 spherical masses and 3massless rods of 0.2m.
- Fix constrain is given between the central mass less shaft & 3 shaft of 0.2m connected to it.
- Revolute joint is given between the ground and the central shaft at the both ends.
- Rotation motion is given to the revolute joint at one end were the function is given as 7200d*time.

  As the rpm is 1200.

  $$Rad/\sec = \frac{2\pi N}{60}$$

$$\text{Rad/ sec} = \frac{2 \text{ x } \pi \text{ x } 1200}{60}$$

$$\text{Rad/ sec} = 125.6$$

deg/sec = 125.6 *(180/**π**)

deg/sec = 7200

- Changing the masses of the central shaft and the other 0.2m shaft as 0.05 in user input.

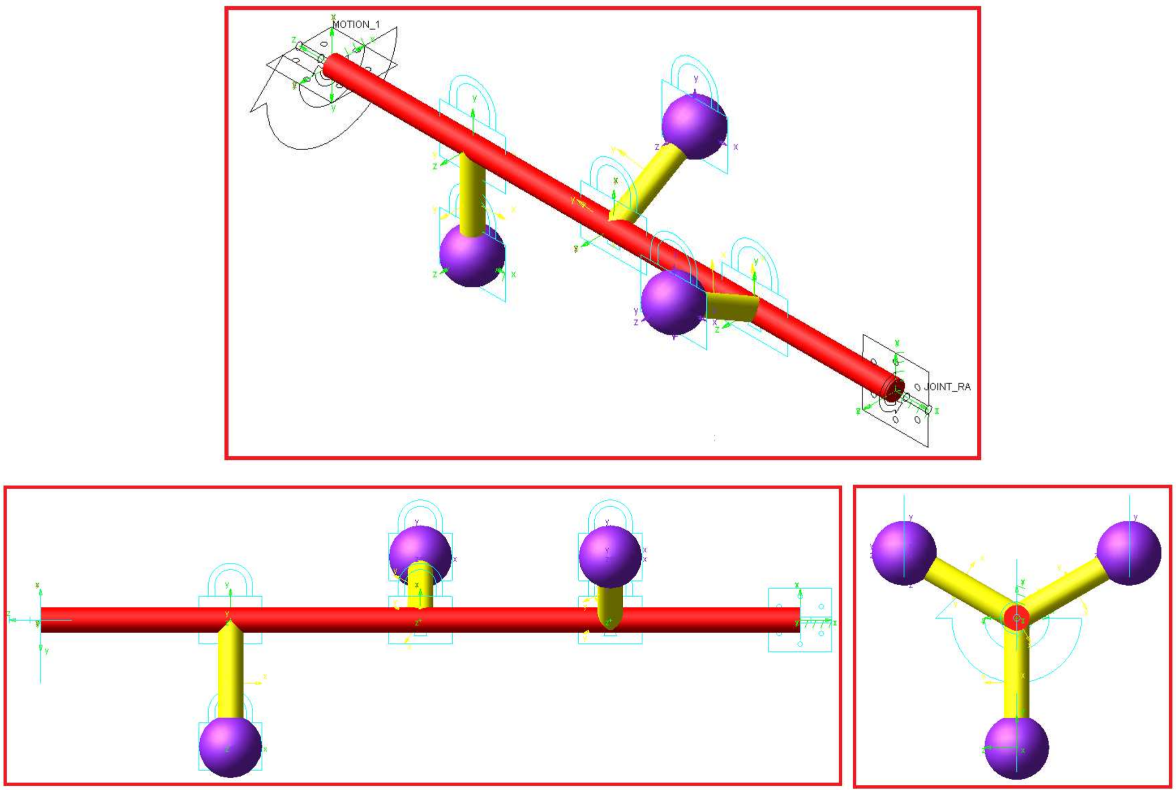


Fig 3. 1 Modeling in ADAMS

3.4 Results obtained from the ADAMS:

After running the simulation of model created in ADAMS the result is measured at the revolute joints which is provided between the central shaft and ground at the both ends were the result of the magnitude is plotted and shown in the graph below in Fig 3.2 as the load is equally distributed on the both the bearings as a result of distance from the masses to the bearing remains same the graph remains same for the both the joints.

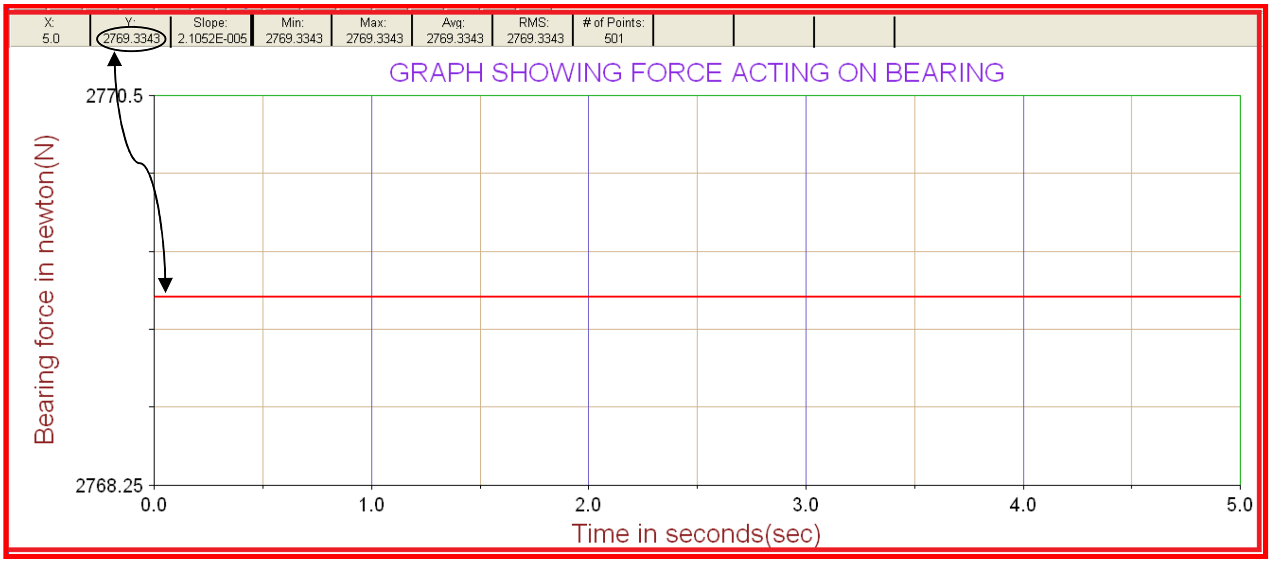


Fig 3. 2 Graph showing force on RA & RB bearing

3.4.1 Comments on the result obtained:

As the 3 masses are rotating at the same rpm there will be variation in forces along y-axis and along z-axis but there will be no forces will be acting along the x-axis which is the central axis of the shaft and the bearing. The resolved forces are shown in the graph which is the magnitude of the acting on bearing in the entire three axes. As the shaft is rotating at the 7200 rad/sec within the very short time the forces reaches to the maximum and remains constant in graph it is shown in straight line were the force reaches to the 2769.33N and remains constant which is the force acting on the bearing. As the masses are balanced properly as the masses are same and the radius is same for all three shaft and the angle between the all the 3 masses remains the same we get the constant force in the graph which indicates the masses are properly balanced.

3.4.2 Comparison of analytical result and result:

Analytical result value in Newton(N)	Result from ADAMS in Newton(N)	Units variation in Newton(N)	Percentage of variation in results.
2763.45	2769.33	5.87	0.22

Table 3. 1 Comparison of result of analytical & ADAMS

As the result obtained by the analytical has the variation of 0.22% which is very less and within the accepted level which can be overcome by the factor of safety. The result obtained from the ADAMS is the forces acting on the both of the bearings connected to the ends of the shaft.

3.5 Result obtained by changing the mass to 1kg:

For analyzing the result and to understand the effect of change in force in the bearing when the load is varied the mass value is reduced from 2kg to 1kg as shown in the Fig 3.3.

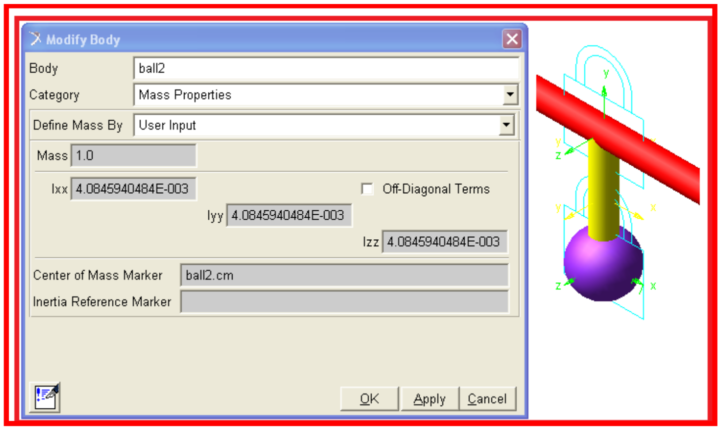


Fig 3. 3 Changing mass from 2kg to 1kg

3.5.1Results obtained from the ADAMS:

For the same lengths and the constrain of the model by changing the mass from 2kg to 1kg from the graph the magnitude of the force acting on the bearing fixed at the both the ends of the shaft 1401.76N shown in the Fig 3.4.

Fig 3. 4 Force on bearing RA & RB mass = 1kg

3.5.2 Comparison of result by changing masses:

The result obtained for the initial condition for the given radius and the mass is shown in the Fig 3.4 is 2769.33N and for the same radius and the same rpm by only changing the masses from the 2kg to 1kg the result obtained is 1401.76N.

ADAMS result value in Newton(N) – Mass = 2kg	ADAMS result value in Newton(N) – Mass = 1kg	Units variation in Newton(N)	Percentage of variation in results.
2763.45	1401.76	638.31	50.72

Table 3. 2 Comparision of result by changing mass = 1kg

3.5.3 Comments on the result obtained:

As the centrifugal force of the rotating shaft is $mr\omega^2$ were the "m" is the mass, "r" is the radius and "ω" is the angular velocity. As in this case the rod is taken as mass less rods the variables which affect the results are the mass of the balls, radius of the shafts and the rpm of the shaft. For the result shown in the graph in Fig() the only changes made among the "$mr\omega^2$" is "m" were the mass is reduced from 2kg to 1kg the variation in the mass is 50% and therefore the variation in the forces acting on the bearing RA and RB is 50.72% .

3.6 Result obtained by changing the radius to 0.25m:

For analyzing the result and to understand the effect of change in force in the bearing when the radius is varied the radius value is increased from 0.2m to 0.25 as shown in the Fig 3.5.

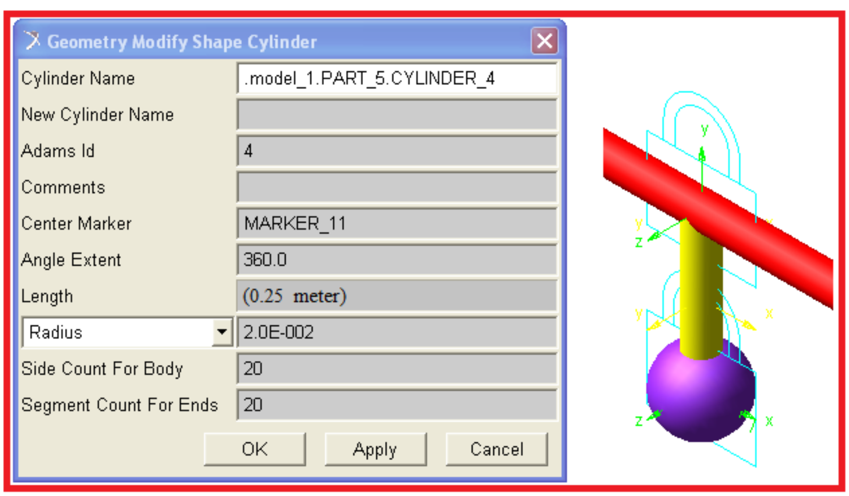


Fig 3. 5 Changing the radius to 0.25m in ADAMS

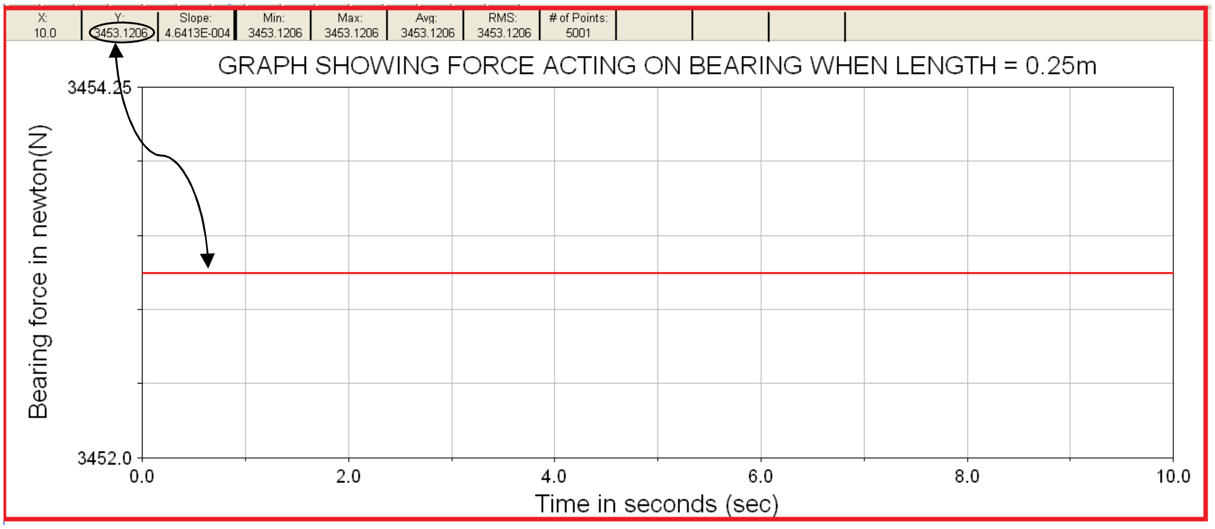


Fig 3. 6 Force on bearing RA & RB (radius = 0.25m)

3.6.1 Comparison of result by changing lengths:

The result obtained for the initial condition for the given radius and the mass is shown in the Fig() is 2769.33N and for the same masses and the same rpm by only changing the radius from the 0.2m to 0.25m the result obtained is 3453.12N.

ADAMS result value in Newton(N) – length = 0.2m	ADAMS result value in Newton(N) – length = 0.25m	Units variation in Newton(N)	Percentage of variation in results.
2763.45	3453.12	689.67	19.97

Table 3. 3 Comparison of result by changing length = 0.25m

3.6.2 Comments on The result obtained:

As the centrifugal force of the rotating shaft is $mr\omega^2$ were the “m” is the mass, “r” is the radius and “ω” is the angular velocity. As in this case the rod is taken as mass less rods the variables which affect the results are the mass of the balls, radius of the shafts and the rpm of the shaft. For the result shown in the graph in Fig3.6 the only changes made among the “$mr\omega^2$” is “r” were the radius is increased from 0.2m to 0.25m the variation in the length is 20% and therefore the variation in the forces acting on the bearing RA and RB is 19.97% .

3.7 Result obtained by changing the rpm to1000:

For analyzing the result and to understand the effect of change in force in the bearing when the rpm is varied the value of rpm is reduced from 1200rpm to 1000rpm as shown in the Fig 3.7. As in the ADAMS the value of rpm is given in deg/sec the 1000rpm is converted into the deg/sec by:

As the rpm is 1000.

$$\mathrm{Rad/\,sec} = \frac{2\pi \mathrm{N}}{60}$$

$$\mathrm{Rad/\,sec} = \frac{2 \text{ x } \pi \text{ x } 1000}{60}$$

$$\mathrm{Rad/\,sec} = 104.66$$

deg/sec = 104.66 *(180/**π**)

deg/sec = 6000

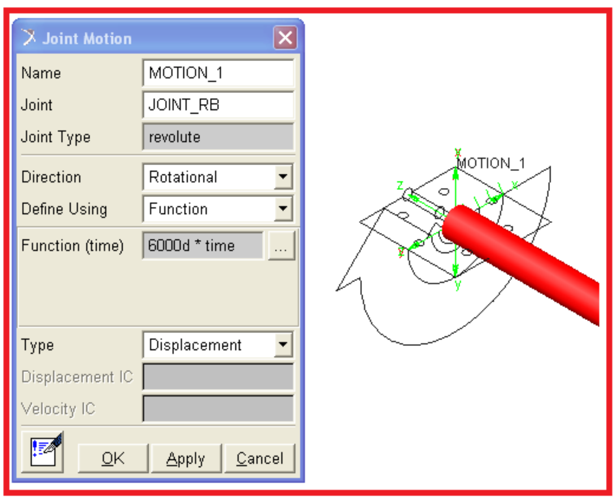


Fig 3. 7 Showing change in rpm in ADAMS

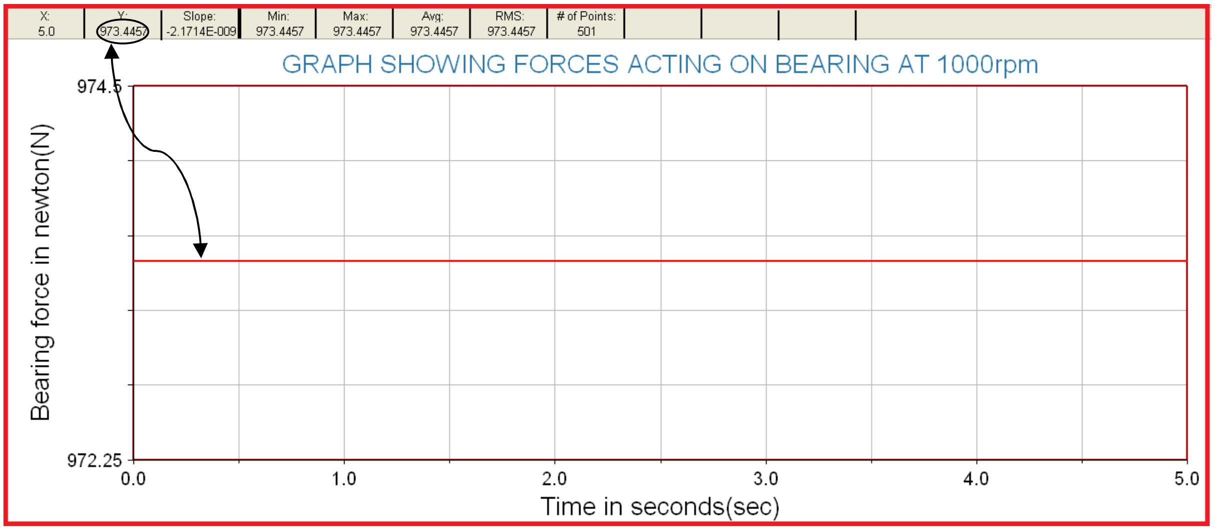


Fig 3. 8 Force on bearing (rpm = 1000)

3.7.1 Comparison of result by changing rpm:

The result obtained for the initial condition for the given radius, mass and the rpm is shown in the Fig() is 2769.33N and for the same masses and the same radius by only changing the rpm from the 1200rpm to 1000rpm the result obtained is 973.44N.

ADAMS result value in Newton(N) – rpm = 1200	ADAMS result value in Newton(N) – rpm = 1000	Units variation in Newton(N)	Percentage of variation in results.
2763.45	973.44	1790.01	64.77

Table 3. 4 Comparison of result by changing rpm =1000

3.7.2 Comments on The result obtained:

As the centrifugal force of the rotating shaft is $mr\omega^2$ were the "m" is the mass, "r" is the radius and "ω" is the angular velocity. As in this case the rod is taken as mass less rods the variables which affect the results are the mass of the balls, radius of the shafts and the rpm of the shaft. For the result shown in the graph in Fig() the only changes made among the "$mr\omega^2$" is "ω" were the angular velocity is increased from 7200rad/sec to 6000rad/sec the variation in the angular velocity is 16.66% and therefore the variation in the forces acting on the bearing RA and RB will be in the form of square as it is "ω^2" therefore the variation in the result is 64.77%.

3.8 Conclusion:

When the rotating masses are properly balanced the forces on bearing will be increasing with the increasing of masses, increase in radius and increase in rpm similarly the force will be reduced while decreasing any of the three parameter but out of these a slight variation in the rpm results in the tremendous change in the bearing forces.

4.0 Learning outcome:

The module introduces different types of mechanisms and their inversions and their relative motion of those mechanisms. The various types of joints and their degree of freedom and were to incorporate those joints in mechanism Gruebler’s equation to find the degrees of freedom of a mechanism, four bar mechanism Grashof, non-Grashof & transient conditions of four bar mechanism and finding the displacement, velocity and acceleration of four bar mechanism using graphical and analytical method. In linkage synthesis calculating the link lengths in four bar mechanism ware covered in kinematics part.

In the dynamics part, the basics of dynamics the various laws governing the dynamics, the static force analysis of the slider crank mechanism and dynamic force analysis of the slider crank using analytical and graphical method, comparison of both the results and interpretation of the results. Balancing of rotating masses in same plane and in different planes using analytical and graphical method. Balancing of reciprocating masses and the concept of partial balancing was covered in the module. Using the ADAMS 2010 software the various kinematics and dynamics related problem was solved in lab session.

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Specifications and Characteristics of Various Types of Intermittent Motion Mechanisms

Type of Mechanism	Reliability	Basic Mechanism Offers Mechanical Advantage	Related Components Required	Relative Size	Degree of Control Over:		Is Impact Present?	Is Jerk Present	Discussed in Chapter
					Motion Curves	Dwell-Motion Ratio			
Impulse ratchet	Good	No	Switch or pulse circuits	Small	Poor	High	Yes	Yes	7
Cam ratchet	Good to excellent	A little	None	Small to moderate	Fair	Poor	A little	Probably	7
Cam	Excellent	A little	None	Small to large	Excellent	Fair	No	Can be avoided	8
Instrument Genevas (external)	Good	No	None	Very small	None	None	Yes	Yes	9
Machine Geneva (external)	Excellent	No	None	Moderate	None	None	A little	Yes	9
Mutilated gearing	Fair to good	A little	None	Small	None	Fair	Yes	Yes	10
Cycloidal gearing	Fair to good	A little	None	Moderate	Poor	Poor	No	No	10
Differential gearing	Excellent	Yes	Clutches (sometimes)	Moderate to large	Poor	Poor	No	No	10
Clock and watch escapements (tuned)	Excellent	Very high	None	Very small to small	None	None	Yes	Yes	11
Machine escapement	Good	Yes, high	Control solenoid & drive circuits or control shaft	Small	None	High	Yes, high	Yes	12
Inverse escapement	Fair	No	Drive circuits	Small	None	High	Yes	Yes	12
Clutch-brake systems	Fair to good	No	Control circuits	Moderate to large	Fair	High	With some types	With most types, yes	13
Step motor	Fair to good	No	Drive circuits	Moderate	Fair	High	No	No	14
Star wheel	Excellent	No	None	Small to moderate	Fair	Poor	A little	Yes	15
Roll cam	Good	Yes	Solenoid and drive circuits	Small to moderate	Good	High	No	No	15

Type of Mechanism	Type of Input to Mechanism	Input Displacement		Output Displacement per Motion	Number of Output Dwells		Designed to Run at Fixed (Synchronous) or Variable (Asynchronous) Rate	Input-Output Parallel or Right Angle
		During Output Dwell	During Output Motion		Per Output Revolution	Per Input Revolution		
Impulse ratchet	Electrical pulses or cam loaded spring	5° to 30°	5° to 30°	1° to 45°	8 to 360	D.A.	Either	D.A.
Cam ratchet	Rotating cam	90° to 300°	60° to 270°	0.1° to 90°	4 to hundreds	1 to 2	Fixed	Parallel
Cam	Rotating shaft	90° to 300°	90° to 270°	90° to 270°	1 to 4	1 to 5	Fixed	Either
Instrument Genevas (external)	Rotating shaft	200° to 270°	160° to 90°	20° to 90°	4 to 8	1 to 2	Fixed	Parallel
Machine Geneva (external)	Rotating shaft	200° to 270°	160° to 90°	20° to 90°	4 to 8	1 to 2	Fixed	Parallel
Mutilated gearing	Rotating shaft	10° to 350°	10° to 350°	10° to many turns	1 to 36	1 to 5	Fixed	Parallel
Cycloidal gearing	Rotating shaft	Few degrees (theoretically 0°)	30° to 360°	30° to nearly 360°	1 to 12	1 to 12	Fixed	Parallel
Differential gearing	Two rotating shafts	0° to many turns	Fractions of a degree to many turns	Fraction of a degree to hundreds of degrees	One to hundreds	One to hundreds	Fixed	Either
Clock and watch escapements (tuned)	Stalled rotating shaft	0°	10° to 30°	10° to 30°	12 to 36	12 to 36	Fixed	Parallel
Machine escapement	Rotating shaft	Few degrees to many turns	Few degrees to many turns	Few degrees to many turns	1 to 100	Fraction of one to 100	Variable	Parallel
Inverse escapement	Rotating shaft or electrical pulses	5° to 300°	5° to 180°	4° to 60°	6 to 90	D.A.	Variable	Parallel
Clutch-brake systems	Rotating shaft	Partial revolution to many revolutions	Partial Revolution to many revolutions	A few degrees to many revolutions	One to hundreds	One to hundreds	Variable	Mostly parallel
Step motor	Electrical pulses	D.A.	D.A.	0.9° to 180°	2 to 400	D.A.	Either	D.A.
Star wheel	Rotating shaft	30° to 300°	60° to 320°	60° to 360°	1 to 8	1 to 3	Fixed	Parallel
Roll cam	Rotating shaft	180° to many turns	30° to many turns	30° to 360°	1 to 12	1 to 5	Variable	Parallel