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Failure Analysis of Semi-elliptical Master Leaf Spring of Passenger Car using Finite Element Method

By Yohannes Regassa, R. Srinivasa Moorthy & Ratnam Uppala

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Abstract - The design of leaf spring has been a constant challenge for automotive and manufacturing engineers and it has undergone multiple revisions [2, 3 and 4]. The aim of this paper is to investigate and analyze how failure occurs on the semi-elliptical master leaf spring of a commercial car by analytical approach and using FEM simulation to ascertain the failure condition and to provide a cost-effective design modification for the same. The currently used 10 mm thick master leaf fails repeatedly at a particular zone close to the spring hanger end. After multiple trials for different thickness values and materials, recommendations were given for a better and modified design of the master leaf spring.

Keywords : *failure analysis, leaf spring, von mises stress, spring steel.*

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I. INTRODUCTION

Leaf springs are crucial suspension elements used on light passenger vehicles to give a riding comfort. The leaf spring should absorb the vertical vibrations and impacts due to road irregularities by means of variations in the spring deflection so that the potential energy is stored in spring as strain energy and then released slowly, ensuring a more compliant suspension system. Leaf springs can serve both damping as well as springing functions. The leaf spring can either be attached directly to the frame at both ends or attached at one end, usually the front, with the other end attached through a shackle, a short swinging arm. The shackle takes up the tendency of the leaf spring to elongate when compressed and thus makes for softer springiness.

Failure prediction in large-scaled structures that are subjected to extreme loading conditions has been of utmost interest in the scientific and engineering community over the past century [4]. Failure of mechanical assembly component is a common phenomenon due to fracture that occurs almost everywhere in mechanical structures. The main cause of failure of leaf spring is due to large bending behavior [5-6].

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Figure 1 : Leaf Spring with Suspension Mechanism [5]

II. LITERATURE REVIEW

The shape of leaf springs has undergone multiple changes and revisions over time from 'flat' to 'elliptical' to the present-day shape of being parabolic. The parabolic spring is light-weighted, has superior capacity to store strain energy and offers better riding comfort and is widely used now-a-days in automotive applications. But it has manufacturing complications.

Different sub-assembly of vehicles, including leaf springs are made of steels with low strength and high ductility. Their failure modes are usually characterized by ductile tearing. Fatigue life prediction is based on knowledge of both the number of cycles the part will experience at any given stress level during that life cycle and environmental factors. The local strain-life method can be used pro-actively for a component during early design stage [7, 8]. For strain-based fatigue life prediction, Coffin-Manson relationship is normally applied [8], which is,

$$\varepsilon_a = \frac{\sigma'_f}{E} (2N_f)^b + \varepsilon'_f (2N_f)^c \quad (1)$$

Where, E is the material modulus of elasticity, ε_a is the true strain amplitude, $2N_f$ is the number of reversals to failure, σ'_f is the fatigue strength coefficient, b is the fatigue strength exponent, ε'_f is the fatigue ductility coefficient and c is the fatigue ductility exponent.

Meanwhile, two mean stress effect models commonly used are the Morrow [8] and Smith-Watson-Topper (SWT) [5] strain-life models.

Mathematically, the Morrow model is defined by,

$$\varepsilon_a = \frac{\sigma'_f}{E} \left(1 - \frac{\sigma^m}{\sigma'_f} \right) (2N_f)^b + \varepsilon'_f (2N_f)^c \quad (2)$$

The SWT model is defined by,

$$\sigma_{\max} \varepsilon_a = \frac{(\sigma'_f)^2}{E} (2N_f)^{2b} + \sigma'_f \varepsilon'_f (2N_f)^{b+c} \quad (3)$$

In 2008, Fuentes et al., studied leaf spring failure and concluded that the premature failure in the studied leaf springs which showed the fracture failure on a leaf was the result of mechanical fatigue and it was caused by a combination of design, metallurgical and manufacturing deficiencies [9].

III. FAILURE ANALYSIS

The existing design parameters are listed in Table 1.

Table 1 : Design Parameters

Parameter	Value
Material selected	20MoCr4 (ISO grade)
Total span length (eye to eye)	1200 mm
Camber height	137 mm
Width of master leaf leaves	60 mm
Normal static load	1500 N

The leaf spring considered is of simply supported beam type, where the central location of the

spring is fixed to the wheel axle. Therefore, the wheel exerts the force F on the spring and support reactions at the two ends of the spring come from the carriage. Maximum deflection, bending stress and Von-Mises stress distribution were estimated by considering the master leaf as a simply supported beam.

For uniform width of master leaf, the maximum stress and displacement were analytically calculated using,

$$\sigma_{\max} = \frac{3FL}{bh^2} \quad (4)$$

and

$$\delta_{\max} = \frac{3FL^3}{Ebh^3} \quad (5)$$

Where, E , F , L , b and h represent the Young's Modulus, normal load, span length, width and thickness of the master leaf.

IV. MODIFIED DESIGN

The spring steels commonly used for making leaf springs are low alloy steels like Carbon steel, Si steel, Mn steel, Si-Mn steel, Si-Cr steel, Mn-Cr steel, Cr-V steel, Si-Cr-V steel, Si-Ni-Cr steel, Ni-Cr-Mo steel and Cr-Mo steel. In this paper the material property selected for analysis is a Carbon steel of 56SiCr7, tempered in the temperature range of 400°C~550°C [10,11].

Table 1 : Spring steel standards - ISO683-14(1992-08-15) [6]

No	Specification		Composition %						
	Steel grade	grade	C	Si	Mn	P _{max}	S _{max}	Cr	Mo
1	59Si7	5	0.55-0.63	1.60-2.0	0.60-1.00	0.030	0.03	---	---
2	56SiCr7	3	0.52-0.59	1.6-2.0	0.70-1.00	0.030	0.03	0.2-0.4	----
3	61SiCr7	7	0.57-0.65	1.6-2.0	0.70-1.00	0.030	0.03	0.2-0.4	----
4	55SiCr63	2	0.51-0.59	1.6-2.0	0.50-0.80	0.030	0.03	0.55-0.85	----
5	55Cr3	8	0.52-0.59	0.15-0.4	0.70-1.00	0.030	0.03	0.70-1.0	----

V. FEM - BASED FAILURE ANALYSIS

The semi-elliptical master leaf was modeled using Solidworks 2012 software. Shackle and bushing were considered for boundary conditions only. Shot-peening and Nip stresses and the frictional effect were also omitted.

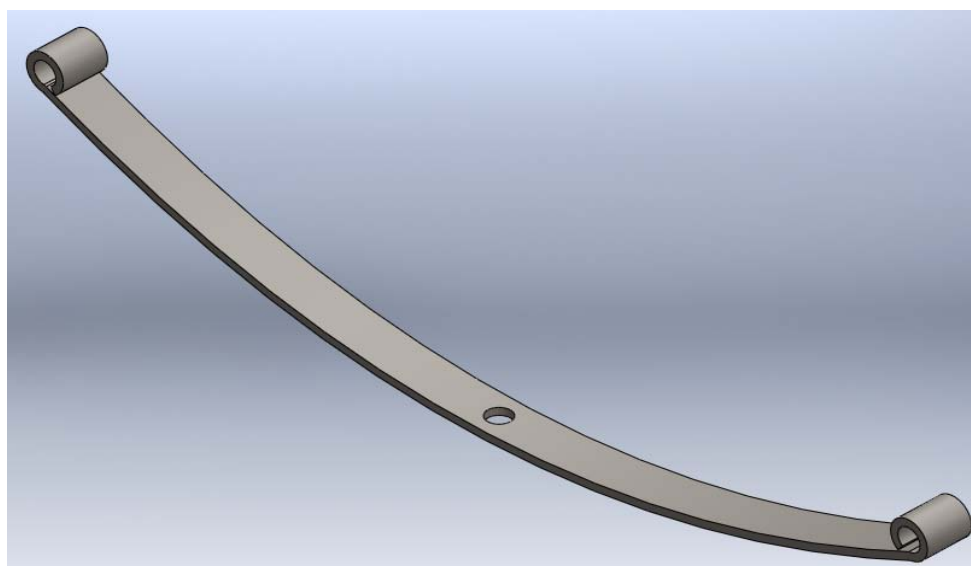


Figure 2 : Master Leaf (CAD Model by Solid Work 2012)

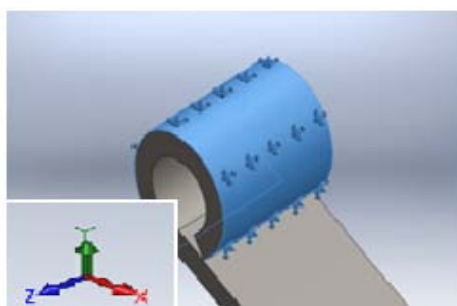


Figure 3 : Hanger end

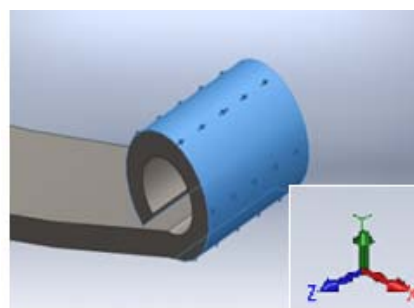


Figure 4 : Shackle end

Table 2 : Mesh Details

Jacobian points	4 Points
Element Size	9.49789 mm
Tolerance	0.474894 mm
Total Nodes	16022
Total Elements	8513

VI. RESULTS AND DISCUSSION

The post processing of the modeled master leaf (existing), gave the stress, strain and displacement plots as shown in Fig. 6.

It is evident that the Von-Mises stress at the hanger end is critical (604 MPa) and is close to the yield stress value (650 MPa), even in static loading conditions. Reversed fatigue loading affects the life of master leaf causing pre-mature failure in the same zone (near hanger end) as reported in the passenger car service station.

To overcome this failure, multiple trials have been made in terms of change of material and thickness of the semi-elliptical master leaf. Si steel substantially increases the elastic limit of the steel and improves the resistance to permanent set of springs.

Hence Si steel of ISO specification 56SiCr7 is chosen from the ISO spring steel standard shown in Table 1. Similarly, after repeated trials for varying thicknesses, 14 mm thickness is chosen for the uniform thickness of master leaf. The FEA results for the modified design were depicted in Fig. 7.

The fatigue test result (S-N curve) for dynamic loading of master leaf and the comparison of the results obtained were shown in Fig. 8 and Table 3 respectively.

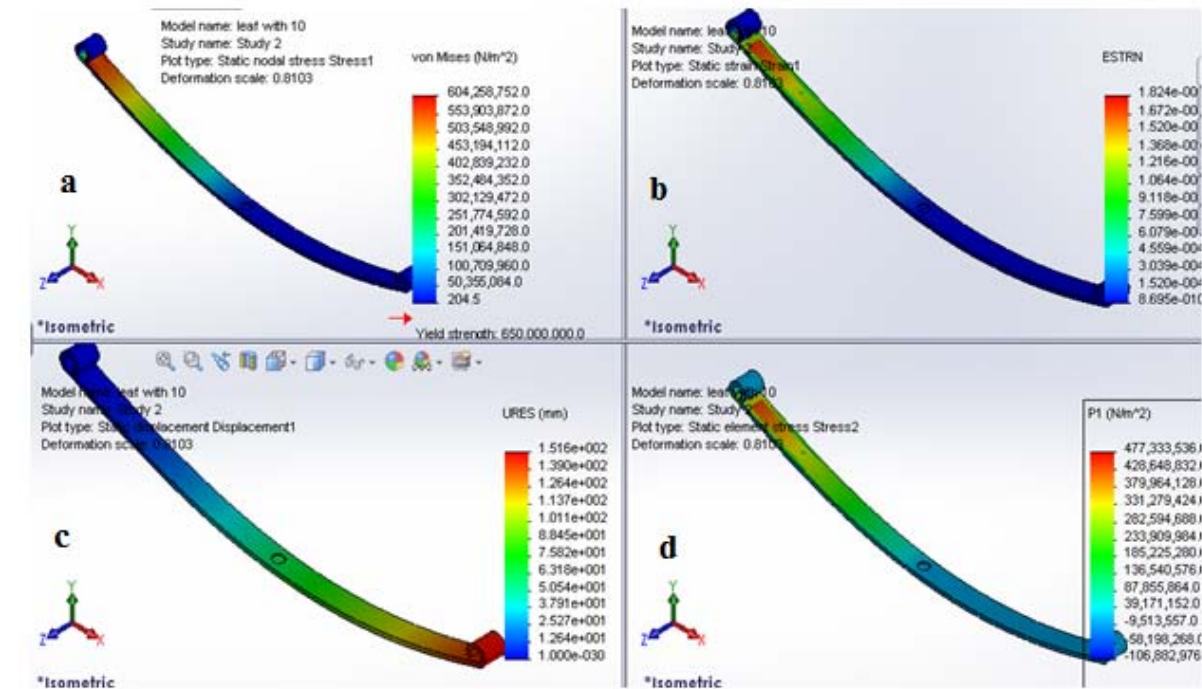


Figure 6 : For Existing Design
[(a) Stress plot (b) Strain plot (c) Displacement plot (d) First principal stress]

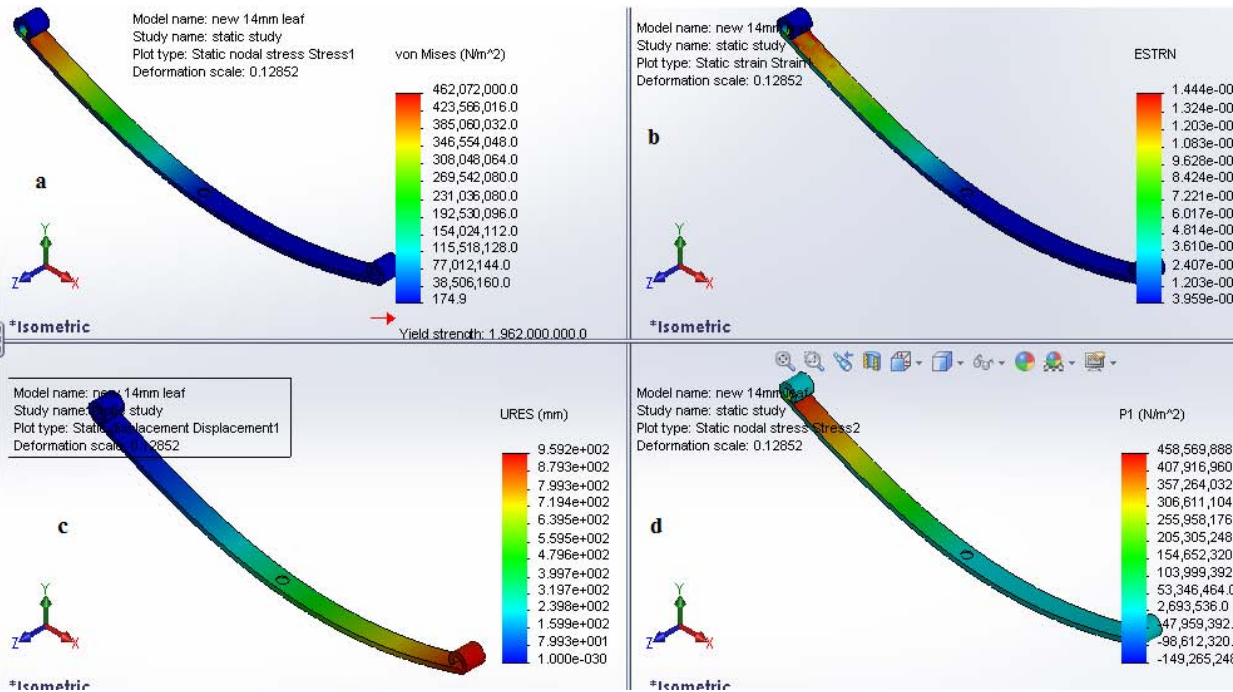


Figure 7 : For Modified Design
(a) Stress plot (b) Strain plot (c) Displacement plot (d) First principal stress

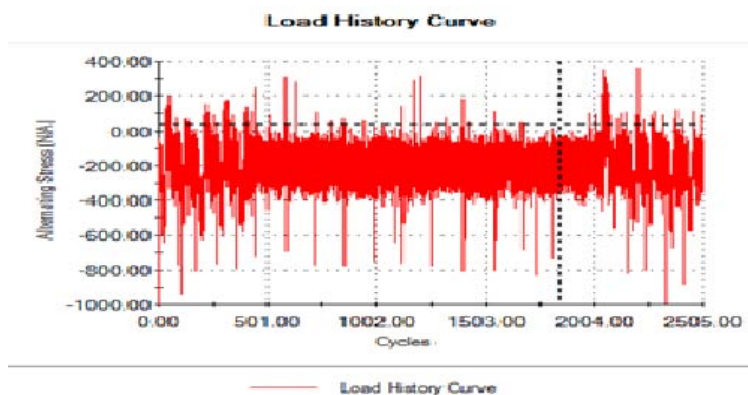


Figure 8: S-N Curve for Master leaf (Modified Design)

Table 3: Comparison of Results (Existing and Modified Designs)

S. No.	Parameter	Existing Design	Modified Design
<i>Analytical</i>			
1	Bending stress (MPa)	450	230
2	Max. deflection (mm)	7.7	28
<i>FEM-based</i>			
1	Von mises stress (MPa)	604	402
2	Resultant displacement (mm)	151.6	101
3	Elastic Strain	0.0304	0.0018

The following inferences can be taken from the above results:

- ✓ The revised design shows a marked reduction in Von Mises stress. The maximum Von Mises stress induced reduced by 33%. The yield strength of 56SiCr7 steel used in revised design is 1962 MPa, which is nearly 5 times that of maximum Von-Mises stress induced. This ensures high factor of safety and reliable operation even under dynamic conditions.
- ✓ The maximum bending stress induced (analytical) for static loading conditions reduced by 49%.
- ✓ FEM based resultant displacement registered 33% reduction.

Thus the modified design involving change of material with an increased thickness of 14 mm has substantial improvements in terms of reduction of Von Mises stress, higher yield strength, lessened resultant displacement and higher factor of safety. Hence the authors recommend this as a cost-effective solution, as desired by the customer.

The other alternatives like use of parabolic master leaf with varying thickness and use of composite materials are not advocated, since the objective was to give an economic and feasible design revision for the existing semi-elliptical master leaf, which is prone to frequent failure.

VII. ACKNOWLEDGMENT

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Investigation of Stresses in a Beam with Fixed Connection using Finite Difference Technique

By Muhammad Ziaur Rahman, Tousif Ahmed & Raihan Md. Imtiaz

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Abstract - In this study a simple rectangular beam has been analyzed. For simplicity two dimensional plane strain problem is considered. The beam is fixed at the left and right edges of the bottom surface. Uniformly distributed load is applied along the length of the beam. A new formulation, known as displacement potential function formulation has been used here. Stresses and displacements at different sections along the length and depth of the beam have been plotted and discussed. The curves are found to be in complete conformity with the available literature and physical condition of the beam. The present literature reveals that the fixed connections of the beam are extremely critical regions and have the highest probability to fail.

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Keywords : stresses, beam with fixed connection, finite-difference, displacement potential function.

I. NOTATIONS

Symbol	Meaning
x, y	rectangular co-ordinates
u, v	displacement component along x and y directions respectively
σ_x	normal stress component along x direction
σ_y	normal stress component along y direction
σ_{xy}	shear stress
ψ	displacement potential function
E	modulus of elasticity
μ	poisson's ratio
a	depth of the beam
b	length of the beam
h	mesh length in the x-direction
k	mesh length in the y-direction
R	ratio of the mesh length k/h
m	number of mesh points in x-direction
n	number of mesh points in y-direction

II. INTRODUCTION

Fixed connections are very common. Steel structures of many sizes are composed of elements which are welded together. A cast-in-place concrete structure is automatically monolithic and it becomes a series of rigid connections with the proper placement of the reinforcing steel. Fixed connections demand greater attention during construction and are often the source of building failures.

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This paper is an attempt to find out what actually happens in a beam with fixed connection at the bottom when it is loaded with uniformly distributed force. In this regard two dimensional elastic problem has been used. It enabled to manage the mixed mode of the boundary conditions as well as the zone of their transition.

In modern age, stress related problems are classical topics. But still there are some shortcomings in analyzing the problems. Specially at the boundary, where stress and deformation co-exist, the problem turns into a complex situation.

The formulation of two dimensional elastic problems used here was first introduced by Uddin[1], later Idris *et. al.* [5-7] used it for obtaining analytical solutions of a number of mixed boundary value elastic problems and Ahmed [2,8-9] extended its use where he obtained finite difference solution of a number of mixed boundary value problems of simple rectangular bodies. Later, Akanda developed a numerical scheme [10-12] by which he solved irregular shaped elastic bodies under mixed mode of loading. This study focuses on the solution of the problem of rectangular beam with fixed connection.

In this paper stresses and deformations at different longitudinal and transverse sections of the beam have been plotted and discussed elaborately. For the convenience of analysis the body is divided into 30 meshes in y direction and 10 meshes in x direction.

III. FORMULATION OF THE PROBLEM

Analysis of stresses in an elastic body is generally a three-dimensional problem. But in the cases of plane stress or plane strain, the stress analysis of three-dimensional body can easily be resolved into two-dimensional problem. The problem studied here has been considered to be a plane strain problem. In the case of the absence of any body forces, the equations governing the three stress components σ_x , σ_y , σ_{xy} under the state of plane stress or plain strain are as follows:

$$\frac{\partial \sigma_x}{\partial x} + \frac{\partial \sigma_{xy}}{\partial y} = 0 \quad (1)$$

$$\frac{\partial \sigma_y}{\partial y} + \frac{\partial \sigma_{xy}}{\partial x} = 0 \quad (2)$$

$$\left(\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} \right) (\sigma_x + \sigma_y) = 0 \quad (3)$$

Replacement of the stress components in Eqs. (1-2) by their relations with the displacement components u and v makes Eq.(3) surplus and Eq.(1) and (2) becomes :

$$\frac{\partial^2 u}{\partial x^2} + \left(\frac{1-\mu}{2}\right) \frac{\partial^2 u}{\partial y^2} + \left(\frac{1+\mu}{2}\right) \left(\frac{\partial^2 v}{\partial x \partial y}\right) = 0 \quad (4)$$

$$\frac{\partial^2 v}{\partial y^2} + \left(\frac{1-\mu}{2}\right) \frac{\partial^2 v}{\partial x^2} + \left(\frac{1+\mu}{2}\right) \left(\frac{\partial^2 u}{\partial x \partial y}\right) = 0 \quad (5)$$

This two equations can be used for the solution. But it is still difficult to solve for two functions simultaneously. To overcome the difficulty, the two equations are transformed into a single equation with a single function. So a new function called displacement potential function (ψ) is defined as a function of displacement components to reduce the no. of governing differential equations into a single equation like following :

$$u = \frac{\partial^2 \psi}{\partial x \partial y}$$

$$v = -\frac{1}{1+\mu} \left[(1-\mu) \frac{\partial^2 \psi}{\partial y^2} + 2 \frac{\partial^2 \psi}{\partial x^2} \right] \quad (6)$$

When the displacement components in Eqs.(4) and (5) are replaced by ψ (x,y) Eq.(4) is automatically satisfied and the only condition that ψ (x,y) has to satisfy becomes [1]:

$$\frac{\partial^4 \psi}{\partial x^4} + 2 \frac{\partial^4 \psi}{\partial x^2 \partial y^2} + \frac{\partial^4 \psi}{\partial y^4} = 0 \quad (7)$$

So the problem is now reduced in such a way that a single function ψ (x,y) has to be evaluated from the bi-harmonic Eq. (7), satisfying the boundary conditions that are specified at the boundary.

IV. BOUNDARY CONDITIONS

Fig. 1 shows the simple rectangular beam subjected to uniformly distributed load. Surface AB is under compression and points C, D are fixed. On the other hand AC and BD surfaces are free from any external load.

The normal and tangential stress components for the top surface AB are given by

$$\sigma_{xy}(x,y) = 0,$$

$$\sigma_x/E(x,y) = -1.10E-04 \text{ for } 0 \leq y/b \leq 1, x/a=0$$

The normal and tangential stress components for the surfaces AC and BD are given by

$$\left. \begin{array}{l} \sigma_{xy}(x,y) = 0 \\ \sigma_y(x,y) = 0 \end{array} \right\} \text{ for } 0 \leq x/a \leq 1 \text{ and } y/b=0, y/b=1$$

The normal and tangential stress components of the surface CD except the points C, D are given by

$$\sigma_{xy}(x,y) = 0,$$

$$\sigma_x(x,y) = 0 \text{ for } x/a=1, 1/15 \leq y/b \leq 14/15$$

The normal and tangential displacement components for the points C and D are given by

$$u(x,y)=0$$

$$v(x,y)=0 \text{ for } x/a=1 \text{ and } y/b= 0, 1/30, 29/30, 1$$

In order to solve the problem using the bi-harmonic equation (7) it is necessary to express the known boundary functions interms of the single potential function ψ like following

$$u = \frac{\partial^2 \psi}{\partial x \partial y} \quad (8)$$

$$v = -\frac{1}{1+\mu} \left[(1-\mu) \frac{\partial^2 \psi}{\partial y^2} + 2 \frac{\partial^2 \psi}{\partial x^2} \right] \quad (9)$$

$$\sigma_x(x,y) = \frac{E}{(1+\mu)^2} \left[\frac{\partial^3 \psi}{\partial x^2 \partial y} - \mu \frac{\partial^3 \psi}{\partial y^3} \right] \quad (10)$$

$$\sigma_y(x,y) = -\frac{E}{(1+\mu)^2} \left[\frac{\partial^3 \psi}{\partial y^3} + (2+\mu) \frac{\partial^3 \psi}{\partial x^2 \partial y} \right] \quad (11)$$

$$\sigma_{xy}(x,y) = \frac{E}{(1+\mu)^2} \left[\mu \frac{\partial^3 \psi}{\partial x \partial y^2} - \frac{\partial^3 \psi}{\partial x^3} \right] \quad (12)$$

So it is clear that all the boundary conditions can be discretized in terms of displacement potential function ψ with the help of finite difference technique.

V. SOLUTION PROCEDURE

a) Method of Solution

The finite-difference technique, frequently used to transform differential equations into corresponding algebraic equations has been used here to generate the algebraic equations of the bi harmonic Eq.(7) and partial differential Eq. (8-12) which are associated with boundary conditions. The discrete values of the potential function ψ (x,y) at the mesh points of the domain concerned (Fig. 2) are obtained from a system of linear algebraic equations resulting from the discretization of the governing equation and the prescribed boundary conditions.

The problem is discretized in a desired number of mesh points and the value of ψ (x,y) is evaluated only at these points. A false boundary, exterior to the physical boundary is introduced to keep the order of error of the difference equations to a minimum. The discretization of the domain concerned is shown in Fig. 2. The division into mesh points can be done in any regular and irregular shape. But due to the shape of the problem rectangular meshing is performed. The governing bi-harmonic equation which is used to evaluate the function only at the internal mesh points is expressed in its corresponding difference equation using central difference operators. Replacing the derivatives of Eq. (7) with central difference formulae, the finite difference equation stands like following

$$R^4\{\psi(i-2,j)+\psi(i+2,j)\}-4R^2(1+R^2)\{\psi(i-1,j)+\psi(i+1,j)\}-4(1+R^2)\{\psi(i,j+1)+\psi(i,j-1)\}+(6R^4+8R^2+6)\psi(i,j)+2R^2\{\psi(i-1,j-1)+\psi(i-1,j+1)+\psi(i+1,j-1)+\psi(i+1,j+1)\}+\psi(i,j-2)+\psi(i,j+2)=0 \quad (13)$$

Where $R=k/h$

Let $O(i,j)$ is an internal mesh point as shown in Fig.(2). From Eq.(13) it is evident that O has 13 mesh points including itself.

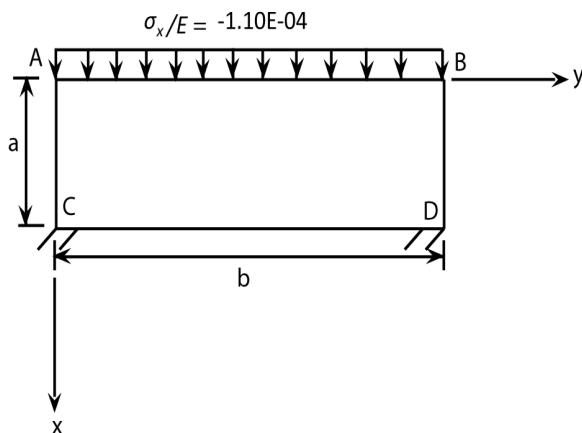


Figure 1 : Rectangular beam with fixed connection and subjected to uniformly distributed load

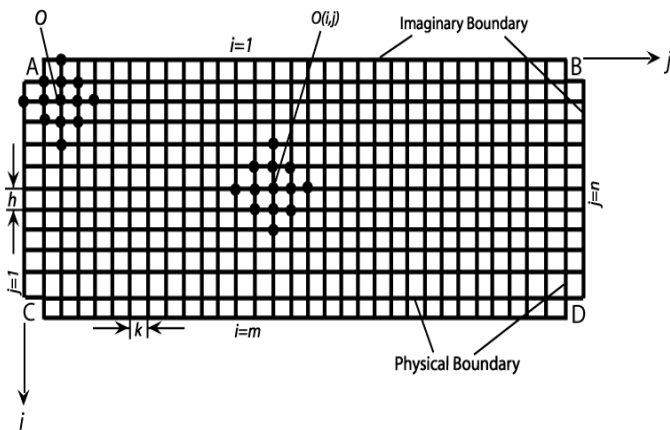


Figure 2 : Rectangular mesh generation of the domain in relation to the coordinate system and the finite-difference discretization of the bi-harmonic equation at an arbitrary internal mesh point

When O becomes an immediate neighbor of the physical boundary, this equation will contain mesh points both interior and exterior to the physical boundary. But it will not create any difficulty provided that a false boundary is introduced external to the physical boundary.

b) Management of Boundary Conditions

Boundary conditions are specified either in deformation parameter or stress parameter of combination of two. There are four possible combination of boundary condition. They are (1) normal stress and shear stress (2) normal stress and tangential displacement (3) shear stress and normal displacement (4) normal displacement and tangential displacement. Each mesh point on the physical boundary entertains two conditions at a time out of four. The computer program is organized in such a fashion that out of these two conditions one is used for evaluation of $\psi(x,y)$ at the concerned boundary point and the other point for the corresponding point on the exterior false boundary. Thus when the boundary conditions are expressed by their appropriate difference equations, every mesh point of the domain will have a single linear algebraic equation. Table 1 is the demonstration of boundary conditions for all the surfaces namely AB, CD, AC and BD.

Table 1 : Specification of the boundary conditions in relation to corresponding mesh points on boundary

Boundary	Given boundary conditions	Correspondence between mesh points and given boundary conditions	
		condition/mesh point	condition/mesh point
Top AB	σ_x, σ_{xy}	$\sigma_x/(2,j)$	$\sigma_{xy}/(1,j)$
Bottom CD (except point C and D)	σ_x, σ_{xy}	$\sigma_x/(m,j)$	$\sigma_{xy}/(m-1,j)$
Left AC	σ_y, σ_{xy}	$\sigma_y/(i,1)$	$\sigma_{xy}/(i,2)$
Right BD	σ_y, σ_{xy}	$\sigma_y/(i,n)$	$\sigma_{xy}/(i,n-1)$
Point C,D	u, v	$u/(m-1,2), (m-1,3), (m-1,n-1), (m-1,n-2)$	$v/(m,2), (m,3), (m,n-1), (m,n-2)$

Three point backward or forward difference formulae has been used here for the replacement of the derivatives of the boundary expressions. Because, the differential equations contain second and third order derivatives of the function ψ for which the application of the central difference formulae will lead to the inclusion of points exterior to the false boundary. As an example, the finite-difference expressions for the normal and tangential components of stress on top boundary AB closer to B are given by

$$\sigma_x(2,j) = \frac{E\mu}{(1+\mu)^2 R^3 h^3} \left[\left(\frac{3R^2}{\mu} - 5 \right) \psi(2,j) + 1.5\psi(2,j+1) + \left(6 - \frac{4R^2}{\mu} \right) \psi(2,j-1) + \left(\frac{R^2}{\mu} - 3 \right) \psi(2,j-2) + 0.5\psi(2,j-3) - \frac{3R^2}{2\mu} \{ \psi(1,j) + \psi(3,j) \} + \frac{2R^2}{\mu} \{ \psi(1,j-1) + \psi(3,j-1) \} - \frac{R^2}{2\mu} \{ \psi(1,j-2) + \psi(3,j-2) \} \right] \quad (14)$$

$$\sigma_{xy}(1,j) = \frac{E\mu}{(1+\mu)^2 R^2 h^3} \left[\frac{3R^2}{2\mu} \psi(1,j) + \left(3 - \frac{5R^2}{\mu} \right) \psi(2,j) + \left(\frac{6R^2}{\mu} - 4 \right) \psi(3,j) + \left(1 - \frac{3R^2}{\mu} \right) \psi(4,j) + \frac{R^2}{2\mu} \psi(5,j) - 1.5\{ \psi(2,j-1) + \psi(2,j+1) \} + 2\{ \psi(3,j-1) + \psi(3,j+1) \} - 0.5\{ \psi(4,j-1) + \psi(4,j+1) \} \right] \quad (15)$$

In Fig. 3(a) and (b) the discretization process used for the two equations above has been demonstrated.

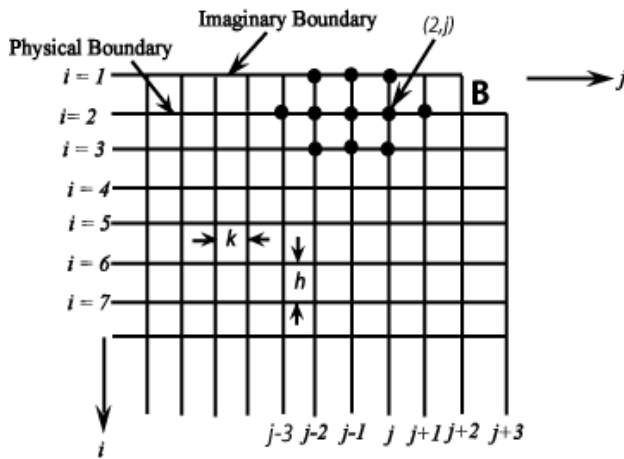


Fig. 3(a)

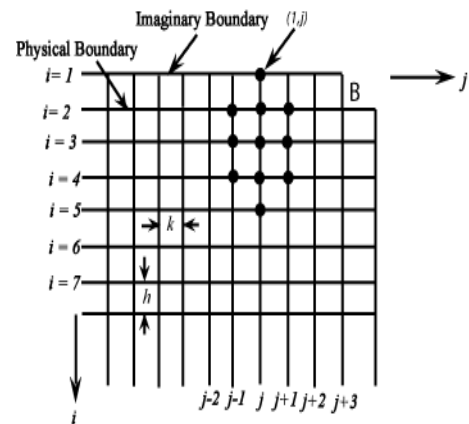


Fig. 3(b)

Figure 3 : Grid-points for expressing the boundary conditions on the top edge at points closer to B (a) for normal stress component σ_x (b) for tangential stress component σ_{xy}

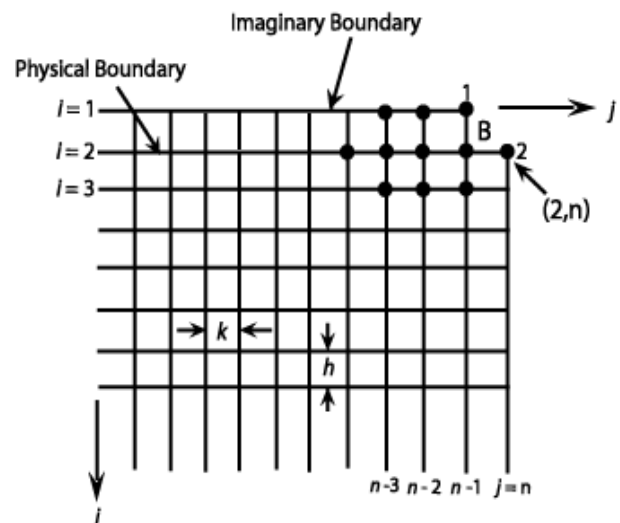


Figure 4 : Treating the transaction Point B

Table 2 : Management of the boundary conditions at corner mesh points

Corner point	Possible boundary conditions	Conditions used	Corresponding mesh points for evaluation of ψ
Point A	$[\sigma_x, \sigma_{xy}]$ on AB $[\sigma_y, \sigma_{xy}]$ on AC	$[\sigma_x, \sigma_{xy}, \sigma_y]$	(2,2),(1,2),(2,1)
Point B	$[\sigma_x, \sigma_{xy}]$ on AB $[\sigma_y, \sigma_{xy}]$ on AC	$[\sigma_x, \sigma_{xy}, \sigma_y]$	(2,n-1),(1,n-1),(2,n)
Point C	$[\sigma_y, \sigma_{xy}]$ on AC $[u, v]$ on CD	$[u, v, \sigma_y]$	(m-1,2),(m,2),(m-1,1)
Point D	$[\sigma_y, \sigma_{xy}]$ on AC $[u, v]$ on CD	$[u, v, \sigma_y]$	(m-1,n-1),(m,n-1),(m-1,n)

The finite difference equation corresponding to corner mesh point B of Fig. 4 is like following

a) *Solution of the system of algebraic equations*

A system of equations having a numerous number of unknowns needs to be solved in the present problem. An iterative method could be chosen in solving the system. But it has some short comings like it has

very slow convergence rate and it fails to produce any solution for other complex boundary conditions. Considering all these, a triangular decomposition method has been used here ensuring better reliability and better accuracy of solution in a shorter period of time. The matrix decomposition method used here solves the present system of equations directly.

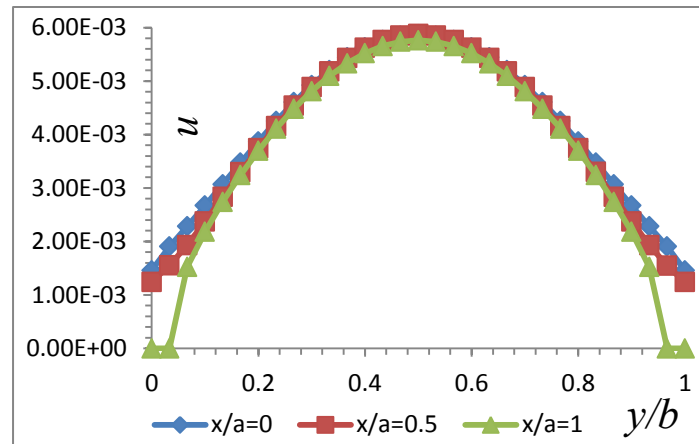


Figure 5 : Distribution of the displacement component u along the neutral axis of beam with fixed connection

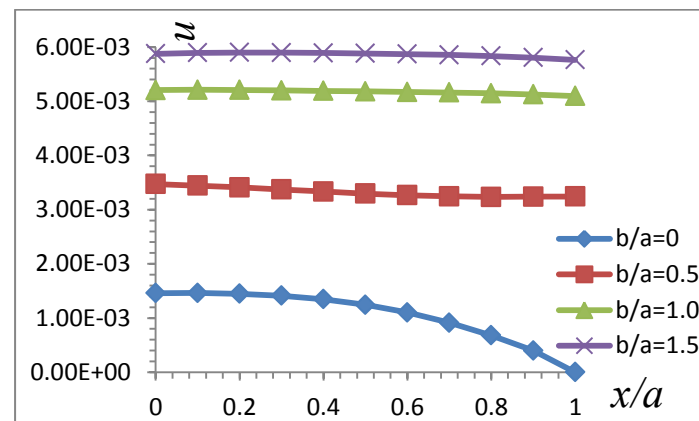


Figure 6 : Distribution of the displacement component u along the depth of beam with fixed connection

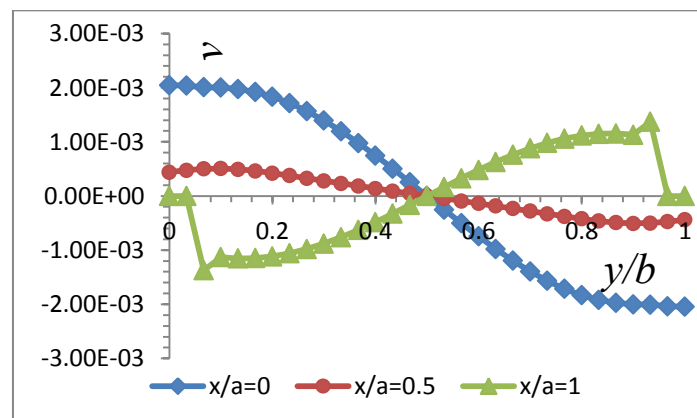


Figure 7 : Distribution of the displacement component v along the neutral axis of beam with fixed connection

VI. RESULTS AND DISCUSSION

Distribution of the deformation parameters and stress parameters has been plotted along the length and depth of the beam. All the solutions of interest obtained through the ψ formulation conform to the symmetric and anti-symmetric characteristics of the problem. Again the famous Saint-Venant's principle that the effects of sharp variation of parameter on the boundary die down and become uniform with the increase of distance of points in the body from the boundary is also supported by all the solutions obtained.

The elastic problem that has been studied here is considered to be made of ordinary steel having poisson ratio, $\mu=0.3$ and modulus of elasticity $E=200$ GPa. Uniformly distributed load is applied at the top surface of the beam. The stresses and deformation are symmetrically distributed about the transverse mid-section of the beam.

a) Distribution of u

Fig.5 shows the distribution of u , displacement along x axis, along the length of the beam. The distribution is symmetric about the mid-section of the beam. At the mid-section, value of u is maximum. Again u is zero at points C and D. All are in complete conformity with the physical condition of the beam. Change of the distribution of u with the increase of distance from the top fiber is also evident in the figure.

With the increase of distance, change in the distribution of u is minor.

Fig. 6 is a demonstration of the distribution of u along the depth of the beam. u remains almost constant along the depth. One exception is at the left most transverse section. Here at the point C, u is zero. It is in conformity with the physical condition because point C is fixed. The effect of b/a ratio along the neutral axis is also shown in the figure. With the increase of the ratio b/a , u increases. That means at the transverse sections nearer to mid transverse section, u is higher and it is highest at the mid section.

b) Distribution of v

Distribution of displacement along y axis, v is shown in the Fig.7. The distribution is counter symmetric about the transverse mid section of the beam. At the mid-section, v is zero. All these is in complete conformity with the physical condition and loading of the problem.

Fig. 8 shows the distribution of v along the depth of the beam. It is zero at the left most transverse section where value of x/a is 1. The effect of b/a ratio is also evident in the Fig. 8. With increasing b/a ratio, v decreases. That means at the inner sections the displacement along y direction dies down and at the mid-section there is no v value.

c) Distribution of σ_x

Fig. 9 shows the distribution of normal stress component σ_x with respect to x at various longitudinal sections of the beam.

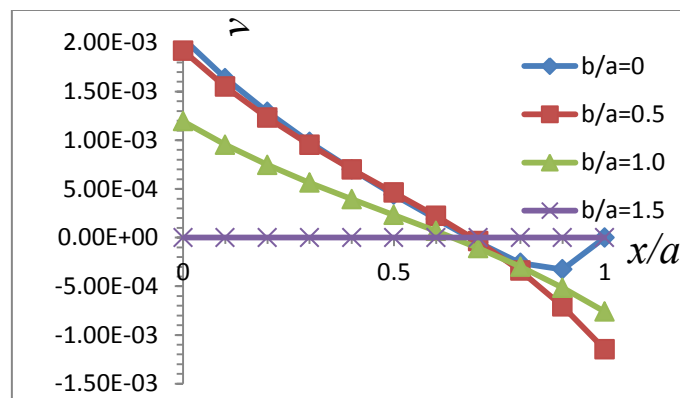


Figure 8 : Distribution of the displacement component v along the depth of beam with fixed connection

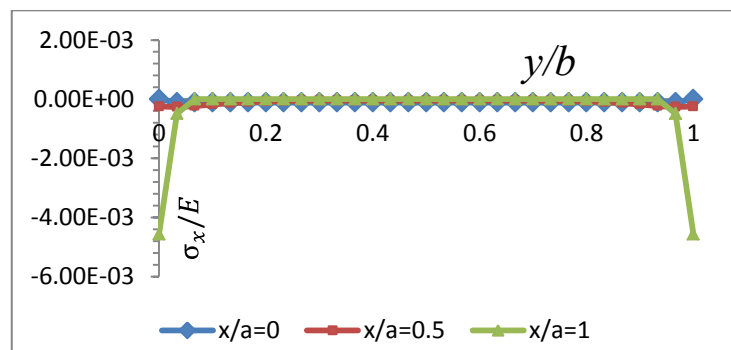


Figure 9 : Distribution of normal stress component σ_x at various longitudinal sections of a beam with fixed connection

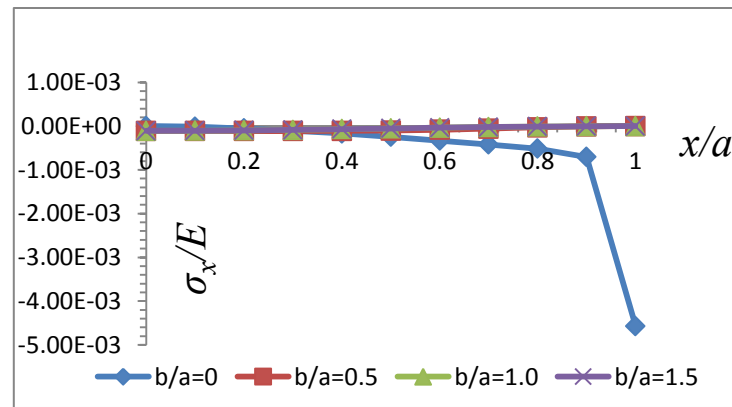


Figure 10 : Distribution of normal stress component σ_x at various transverse sections of a beam with fixed connection

From the figure it is observed that the distribution is symmetric about the transverse mid-section of the beam. It is also evident that the fixed region is the most critical region as far as the normal stress is concerned. It is supported by the available literature[9].

The effect of b/a ratio on the distribution of σ_x is shown in Fig.10. The value of σ_x decreases with an increasing b/a ratio. The most critical points are found at the region of fixed connection which is conforming to the available literature[9]. So it can be concluded from the distributions shown in Fig.10 and Fig.11 that the most critical points with respect to σ_x are around $x/a=1$, $y/b=0$ and $x/a=1$, $y/b=1$.

d) Distribution of σ_y

Fig. 11 is a demonstration of the distribution of normal stress component σ_y along neutral axis of the beam. The distribution is completely symmetric about the transverse mid-section of the beam. The distribution is negative at the top fiber and positive at the bottom fiber. At the mid longitudinal section, the value of σ_y is negligible. All these are in complete conformity with the physical condition of the beam. The most critical points as far as σ_y is concerned, are found at the fixed region and they are at around $y/b=0.03$ and 0.97 respectively. Again it conforms to the available literature[9].

From Fig.12 the effect of b/a ratio on σ_y can be studied. It is zero at almost all transverse sections except at $b/a=0.1$. At the section $b/a=0.1$, there are at least two critical points between

$$0.8 \leq x/a \leq 1.0$$

e) Distribution of σ_{xy}

Distribution of shear stress σ_{xy} at various longitudinal sections (Fig.13) of a beam with fixed connection reveals that shearing stress is almost zero at all the longitudinal sections. It conforms to the physical and loading condition of the beam under consideration. But again there are at least two critical points found at the fixed connection region which again reveals that

fixed connection is an extremely critical region where the probability of failure is maximum.

Finally Fig.14 is a demonstration of shear stress at different transverse sections of the beam. Shear stress is almost zero at different sections except at the fixed region, which again conforms to the physical condition of the beam.

VII. CONCLUSIONS

In the elementary formulas of strength of materials, the boundary conditions are satisfied in an approximate way. As a result it often fails to give the exact distribution of stresses at the restrained boundaries. The present displacement potential function approach is free from this type of limitations. So it is capable of representing the actual distribution of normal and shear stresses at critical regions. For example, from the elementary solution it is observed that the magnitude of the shearing stress is maximum at the mid-section of the beam. It is not supported by the numerical solution presented here,

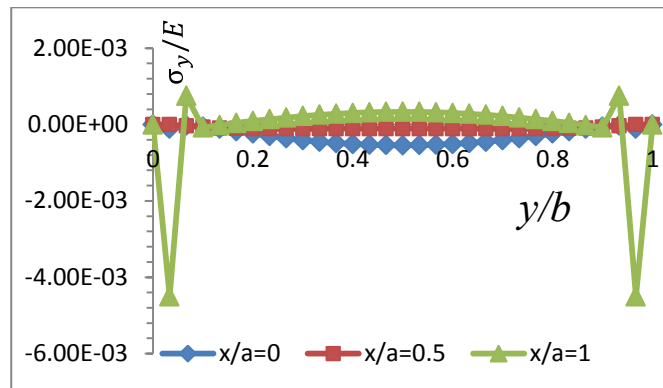


Figure 11 : Distribution of normal stress component σ_y at various longitudinal sections of a beam with fixed connection

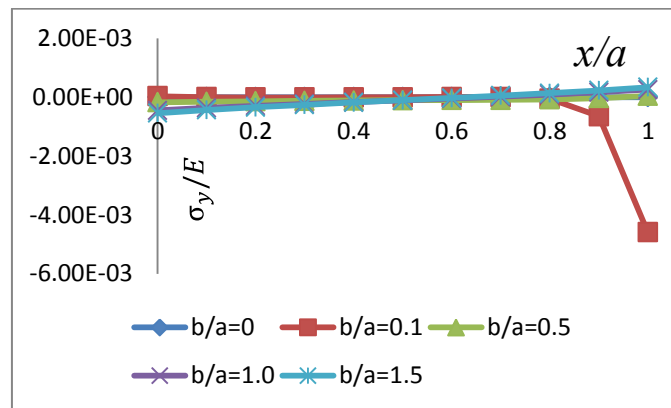


Figure 12 : Distribution of normal stress component σ_y at various transverse sections of a beam with fixed connection

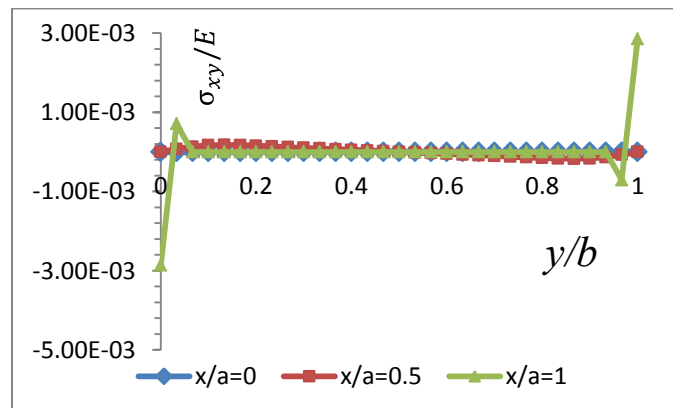


Figure 13 : Distribution of shear stress σ_{xy} at various longitudinal sections of a beam with fixed connection

because the approach used here reveals that shearing stress as well as normal stresses are maximum at the restrained boundaries. So, a more reliable and accurate study on the distribution of normal and shear stresses has been provided in the present literature. Besides, all the results obtained here are compared with the available literature and are in complete conformity with it and with the boundary and loading conditions. Such both the qualitative and quantitative results obtained through ψ -formulation of a beam with fixed connection

at the bottom surface and subjected under uniformly distributed load establish the soundness and acceptance of the approach adopted here.

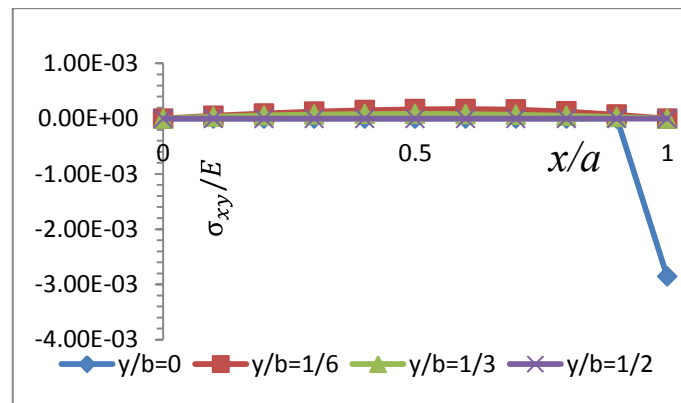


Figure 14 : Distribution of shear stress σ_{xy} at various transverse sections of a beam with fixed connection

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Pole Placement Approach for Controlling Double Inverted Pendulum

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Abstract - In this paper, we present in-depth analysis of the classical double inverted pendulum (DIP) system using the DIP modeling and the pole placement approach to control it. The double inverted pendulum system has the characteristics of multiple variables, non-linear, absolute instability; it can reflect many key issues in the progress of control, such as stabilization, non-linear and robust problems etc. DIP model is a simplified model of the anterior-posterior motion of a standing human. DIP has four equilibrium points (Down-Down, Down-Up, Up-Down, Up-Up). The objective of this paper is to keep the double pendulum in an Up-Up unstable equilibrium point. Modeling is based on the Euler-Lagrange equations, and the resulted non-linear model is linearized around Up-Up position. The built of mathematical model of double inverted pendulum plays a guiding role on the stability of the system. The eigen-values of the system which are the poles of the system have enormous influenced on stability and system response. Pole placement is the control method which places the poles at the desired position to control the system by calculating gain matrix of the system.

Keywords : *double inverted pendulum; linear time-invariant system; pole placement method.*

GJRE-A Classification : *FOR Code: 010207, 0913990*



Strictly as per the compliance and regulations of :



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Nita H. Shah ^α & Mahesh Yeolekar ^σ

Abstract - In this paper, we present in-depth analysis of the classical double inverted pendulum (DIP) system using the DIP modeling and the pole placement approach to control it. The double inverted pendulum system has the characteristics of multiple variables, non-linear, absolute instability; it can reflect many key issues in the progress of control, such as stabilization, non-linear and robust problems etc. DIP model is a simplified model of the anterior-posterior motion of a standing human. DIP has four equilibrium points (Down-Down, Down-Up, Up-Down, Up-Up). The objective of this paper is to keep the double pendulum in an Up-Up unstable equilibrium point. Modeling is based on the Euler-Lagrange equations, and the resulted non-linear model is linearized around Up-Up position. The built of mathematical model of double inverted pendulum plays a guiding role on the stability of the system. The eigen-values of the system which are the poles of the system have enormous influenced on stability and system response. Pole placement is the control method which places the poles at the desired position to control the system by calculating gain matrix of the system. In this paper, the performance of the pole placement method is analyzed by MATLAB to control the double inverted pendulum.

Keywords : double inverted pendulum; linear time-invariant system; pole placement method.

I. INTRODUCTION

The study of humanoid robot is currently one of the most exciting research projects. Even if some of those works have already demonstrated very reliable dynamic biped walking (Yamaguchi, Soga, Inoue & Takanishi, 1999; Hirai, Hirose, Haikawa & Takenaka, 1998; Nishiwaki, Sugihara, Kagami, Kanehiro, Inaba & Inoue, 2000), we believe it is still important to understand the mathematical theoretical background of biped locomotion. In standing, it has become common to consider the body as an (single\double\triple) inverted pendulum pivoted at the ankles. Moreover, up ride of a human shoulder is also considered as a motion of an (single\double\ triple) inverted pendulum (Jadlovská, 2011; Jadlovská & Jadlovská, 2010).

An inverted pendulum system is a typically non-linear, redundancy, uncertainty, strong coupling and natural characteristics of instabilities. All these features make it the ideal model of advanced control theory and typical experiment platform of test control results. There

are a number of different kinds of the inverted pendulum systems presenting a variety of control challenges. The most common types are the single inverted pendulum on a cart (Ohsumi & Izumikawa, 1995; Åström & Furuta, 2000), the double inverted pendulum on a cart (Zhong & Rock, 2001), the double inverted pendulum with an actuator at the first joint only (Pendubot) (Graichen & Zeitz, 2005), the double inverted pendulum with an actuator at the second joint only (Acrobot) (Hauser & Murray, 1990), the light weight rotary pendulum (Brockett & Hongyi, 2003).

In this paper, we have addressed the problem of stability of double inverted pendulum in the upright position using the pole placement method. For this, we have assumed that the double inverted pendulum is pivoted at the lower end of inner arm (see figure 1). The first step to achieve the objective is to understand the dynamics of the system of double inverted pendulum by developing the mathematical modelling of the system. In modelling, we have used Euler-Lagrange formulation to find equation of motion. In the second step, we linearized this non-linear system of double inverted pendulum in the up-up position and build up its linear state space model. The linearization is one of the most important issues for control of non-linear systems. There are lots of studies in the literature regarding linearization (Jordan, 2006; Wang, Chen & Zhou, 2000; Conga, Wanga & Hill, 2005). In the next step, the stability and controllability criteria showed that the system is unstable but it is controllable.

To control this unstable system, we have employed the pole placement method. In this method, the poles are the eigen-values of linear state space model and the calculated gain matrix places the poles at desired position to stabilize a system. In the simulation part of this paper, numerical and graphical simulations for control task are given to show the effectiveness of the proposed pole placement scheme.

II. MATHEMATICAL MODELING OF DIP

In this section, we will describe mathematical model necessary for stability and controllability analysis. The mechanism of the double inverted pendulum is shown in Figure 1 schematically. The mathematical model of DIP can be derived using the Euler-Lagrange equation. The form of the Euler-Lagrangian equation used here is:

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$$\frac{d}{dt} \frac{\partial L}{\partial \dot{\theta}} - \frac{\partial L}{\partial \theta} = \tau \quad (1)$$

Where $L = T - V$ is a Lagrangian, T is kinetic energy, V is potential energy, $\tau = [\tau_1 \ \tau_2]^T$ is

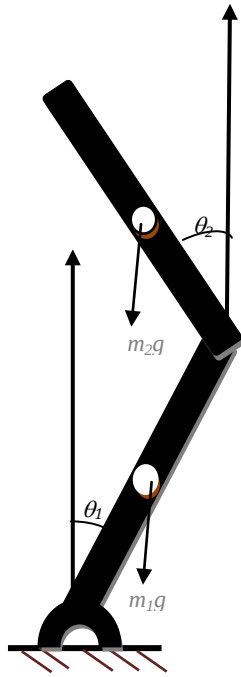


Figure 1 : Double Inverted Pendulum

the input generalized force vector produced by two actuators at the lower joint (ankle) and second at joint between to arm (knee), $\theta = [\theta_1 \ \theta_2]^T$ is generalized coordinate vector where θ_1 and θ_2 are angular positions of first arm, and second arm of the double pendulum. The kinetic and potential energies in terms of generalized coordinates can be determined as:

$$T = \begin{bmatrix} \frac{1}{2} (m_1 l_1^2 + I_1) \dot{\theta}_1^2 \\ \frac{1}{2} [m_2 (4l_1^2 \dot{\theta}_1^2 + 4l_1 l_2 \cos \theta_2 \dot{\theta}_1 \dot{\theta}_2 + l_2^2 \dot{\theta}_2^2) + I_2 \dot{\theta}_2^2] \end{bmatrix} \quad (2)$$

$$V = \begin{bmatrix} m_1 g l_1 \cos \theta_1 \\ m_2 g (2l_1 \cos \theta_1 + l_2 \cos \theta_2) \end{bmatrix} \quad (3)$$

Differentiating the Lagrangian $L = T - V$ by generalized coordinate's vector yields Euler-Lagrange equation (1) as:

$$\begin{aligned} & [m_1 l_1^2 + I_1 + 4m_2 l_1^2 + 4m_2 l_1 l_2 \cos \theta_2 + m_2 l_2^2] \ddot{\theta}_1 - [2m_2 l_1 l_2 \cos \theta_2 + m_2 l_2^2] \ddot{\theta}_2 + 2m_2 l_1 l_2 \sin \theta_2 \dot{\theta}_2^2 - 4m_2 l_1 l_2 \sin \theta_2 \dot{\theta}_1 \dot{\theta}_2 \\ & - (m_1 + 2m_2) g l_1 \sin \theta_1 - m_2 g l_2 \sin(\theta_1 - \theta_2) = \tau_1 \end{aligned}$$

$$[-2m_2 l_1 l_2 \cos \theta_2 - m_2 l_2^2] \ddot{\theta}_1 + [m_2 l_2^2 + I_2] \ddot{\theta}_2 + 2m_2 l_1 l_2 \sin \theta_2 \dot{\theta}_1^2 + m_2 g l_2 \sin(\theta_1 - \theta_2) = \tau_2$$

The matrix form of the system is

$$\begin{bmatrix} m_1 l_1^2 + I_1 + 4m_2 l_1^2 + 4m_2 l_1 l_2 \cos \theta_2 + m_2 l_2^2 & -2m_2 l_1 l_2 \cos \theta_2 - m_2 l_2^2 \\ -2m_2 l_1 l_2 \cos \theta_2 - m_2 l_2^2 & m_2 l_2^2 + I_2 \end{bmatrix} \begin{bmatrix} \ddot{\theta}_1 \\ \ddot{\theta}_2 \end{bmatrix} + \begin{bmatrix} 0 & 2m_2 l_1 l_2 \sin \theta_2 \\ 2m_2 l_1 l_2 \sin \theta_2 & 0 \end{bmatrix} \begin{bmatrix} \dot{\theta}_1^2 \\ \dot{\theta}_2^2 \end{bmatrix} + \begin{bmatrix} -4m_2 l_1 l_2 \sin \theta_2 \dot{\theta}_1 \dot{\theta}_2 - (m_1 + 2m_2) g l_1 \sin \theta_1 - m_2 g l_2 \sin(\theta_1 - \theta_2) \\ m_2 g l_2 \sin(\theta_1 - \theta_2) \end{bmatrix} = \begin{bmatrix} \tau_1 \\ \tau_2 \end{bmatrix}$$

a) Linearisation of the System

The tracking controller of DIP is designed using the Gain Scheduling method based on the linearisation of the system equations around certain equilibrium points. In this paper, we linearize the system at the vertical unstable equilibrium by taking.

$$\theta_1 \approx \theta_2 \approx 0,$$

$$\cos \theta_1 \approx \cos \theta_2 \approx 1$$

$$\sin \theta_1 \approx \theta_1; \sin \theta_2 \approx \theta_2;$$

$$\theta_1 - \theta_2 \approx 0; \cos(\theta_1 - \theta_2) \approx 1; \sin(\theta_1 - \theta_2) \approx \theta_1 - \theta_2$$

$$\dot{\theta}_1^2 \approx \dot{\theta}_2^2 \approx 0$$

So,

$$\begin{aligned} & \left[m_1 l_1^2 + I_1 + 4m_2 l_1^2 + 4m_2 l_1 l_2 + m_2 l_2^2 \right] \ddot{\theta}_1 + \left[-2m_2 l_1 l_2 - m_2 l_2^2 \right] \ddot{\theta}_2 \\ & - (m_1 + 2m_2) g l_1 \theta_1 - m_2 g l_2 (\theta_1 - \theta_2) = \tau_1 \\ & \left[-2m_2 l_1 l_2 - m_2 l_2^2 \right] \ddot{\theta}_1 + \left[m_2 l_2^2 + I_2 \right] \ddot{\theta}_2 + m_2 g l_2 (\theta_1 - \theta_2) = \tau_2 \end{aligned}$$

The matrix of the linear system

$$\begin{bmatrix} m_1 l_1^2 + I_1 + 4m_2 l_1^2 + 4m_2 l_1 l_2 + m_2 l_2^2 & -2m_2 l_1 l_2 - m_2 l_2^2 \\ -2m_2 l_1 l_2 - m_2 l_2^2 & m_2 l_2^2 + I_2 \end{bmatrix} \begin{bmatrix} \ddot{\theta}_1 \\ \ddot{\theta}_2 \end{bmatrix} + \begin{bmatrix} -(m_1 + 2m_2) g l_1 - m_2 g l_2 & m_2 g l_2 \\ m_2 g l_2 & -m_2 g l_2 \end{bmatrix} \begin{bmatrix} \theta_1 \\ \theta_2 \end{bmatrix} = \begin{bmatrix} \tau_1 \\ \tau_2 \end{bmatrix}$$

b) State space model for the above linear system

The state space model equation for the system is

$$\begin{aligned} \dot{x} &= Ax + Bu \\ y &= Cx + Du \end{aligned} \quad (4)$$

where

$$A = M^{-1}N \quad B = M^{-1}T \quad C = I_{4 \times 4} \quad D = 4 \times 4 \quad u = \begin{bmatrix} \tau_1 \\ \tau_2 \end{bmatrix}$$

$$M = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & m_1 l_1^2 + I_1 + 4m_2 l_1^2 + 4m_2 l_1 l_2 + m_2 l_2^2 & -2m_2 l_1 l_2 - m_2 l_2^2 \\ 0 & 0 & -2m_2 l_1 l_2 - m_2 l_2^2 & m_2 l_2^2 + I_2 \end{bmatrix}$$

$$N = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ (m_1 + 2m_2) g l_1 + m_2 g l_2 & -m_2 g l_2 & 0 & 0 \\ -m_2 g l_2 & m_2 g l_2 & 0 & 0 \end{bmatrix}$$

$$T = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 1 & 0 \\ 0 & 1 \end{bmatrix}; \quad x = \begin{bmatrix} \theta_1 \\ \theta_2 \\ \dot{\theta}_1 \\ \dot{\theta}_2 \end{bmatrix}$$

c) Values of Parameters

The values of parameters for the given double inverted pendulum are assumed as follows:

m_1 = mass of inner arm = 0.4 kg

m_2 = mass of outer arm = 0.5 kg

l_1 = length of inner arm = 5 m

l_2 = length of outer arm = 5 m

g = gravitational acceleration = 9.8 m/s²

So the corresponding values of state space matrices are as follows:

$$A = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -0.8276 & -1.4206 & 0 & 0 \\ -4.1012 & -2.1247 & 0 & 0 \end{bmatrix}$$

$$B = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ -0.0328 & -0.0908 \\ -0.0908 & -0.1775 \end{bmatrix}$$

III. STABILITY AND CONTROLLABILITY OF SYSTEM

a) Stability Criterion

A system (state space representation) is stable iff all the eigenvalues of the matrix A are inside the unit circle.

The eigen value of A of our system are: 0.0000 + 1.9939i, 0.0000 - 1.9939i, -1.0115, 1.0115 which are outside the unit circle because the modules of eigen values are greater than 1. So the system is unstable in absence of input force ($\tau_1 = 0$, $\tau_2 = 0$).

b) Controllability criterion

A system (state space representation) is controllable iff the controllable matrix $C = [B \ AB \ A^2B \ \dots \ A^{n-1}B]$ has rank n where n is the number of degrees of freedom of the system.

In our system, the controllable matrix $C = [B \ AB \ A^2B \ A^3B]$ has rank 4 which the degree of freedom of the system. So, the system is controllable.

If the system is controllable, then all set of distinct closed loop poles are assigned arbitrarily by output feedback gain (Kimura, 1975; Kimura, 1977; Wonham, 1967).

IV. POLE PLACEMENT

Block diagram of pole placement is given in Fig. 2. If the linearized system considered is completely state controllable, then poles of the closed-loop system may be placed at any desired locations by means of state feedback through an appropriate state feedback gain matrix K .

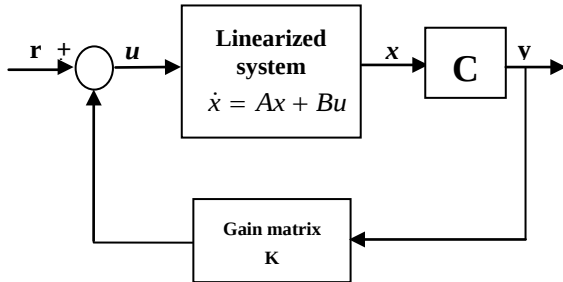


Figure 2: Block Diagram of Pole Placement

In this paper, we have used the following method to calculate state feedback gain matrix:

Pole placement with output feedback is displayed in Figure 2. In this paper, the reference signal, r , is taken zero. If an output feedback control

$$u = -Kx$$

is applied to (4), the closed-loop system becomes

$$\dot{x} = (A - KB)x$$

The poles assigned with output feedback are $\lambda = \{\lambda_1, \lambda_2, \lambda_3, \dots, \lambda_n\}$. The problem considered in this paper is finding gain matrix K for transferring the poles.

a) Gain Matrix Scheduling

Consider the controllability matrix $C = [B \ AB \ A^2B \ \dots \ A^{n-1}B]$ which is an $n \times pn$ order matrix. If the system is controllable, then the $\text{rank}(C) = n$. That means, it has only n linearly independent columns among the pn columns. Therefore, there will be many ways to construct an $n \times n$ similarity matrix which will give a multi-input controllable canonical form. In this paper, we use the following way:

Consider controllable matrix in n block as follows:

$$C = [b_1 \dots b_p : Ab_1 \dots Ab_p : \dots : A^{n-1}b_1 \dots A^{n-1}b_p]$$

Block 0 Block 1

Block $n-1$

Starting from the left in, this matrix, check each column, keeping count of the number of linearly independent columns we encounter. We may stop counting when n linearly independent columns are obtained.

Let the block in which we find the *last* (i.e., the n th) linearly independent column be denoted by the $(\mu-1)$ th block. Then the first block in which there are *no more* independent columns will be the μ th block. This μ is controllability index.

Rearranging these selected n linearly independent columns $b_1, b_2, \dots, b_p, Ab_1, Ab_2, \dots, A^{\mu-1}b_1, \dots$ we will get the invertible matrix M as:

$$M = [b_1 \dots A^{\mu_1-1}b_1 : b_2 \dots A^{\mu_2-1}b_2 : \dots : b_p \dots A^{\mu_p-1}b_p]$$

Where μ_i ($1 \leq i \leq p$) are the controllability indices of (A, B) .

The inverse of M is

$$M^{-1} = \begin{bmatrix} m_{11} \\ \vdots \\ m_{1\mu_1} \\ m_{21} \\ \vdots \\ m_{2\mu_2} \\ \vdots \\ m_{p1} \\ \vdots \\ m_{p\mu_p} \end{bmatrix}$$

Using this inverse matrix of M , calculate transformation matrix T as follows:

$$T = \begin{bmatrix} m_{1\mu_1} \\ \vdots \\ m_{1\mu_1} A^{\mu_1-1} \\ m_{2\mu_2} \\ \vdots \\ m_{2\mu_2} A^{\mu_2-1} \\ \vdots \\ m_{p\mu_p} \\ \vdots \\ m_{p\mu_p} A^{\mu_p-1} \end{bmatrix}$$

Using transfer matrix T , transferring of the matrix A and B are,

$$\bar{A} = T^{-1}AT, \bar{B} = TB$$

Using desired poles $\{\lambda_1, \lambda_2, \lambda_3, \dots, \lambda_n\}$, the transferred canonical form of the system is

$$\bar{A} - \bar{B}K = \begin{bmatrix} 0 & 1 & 0 & \cdots & 0 & 0 & 0 & 0 & \cdots & 0 & 0 & 0 & 0 & \cdots & 0 \\ 0 & 0 & 1 & \cdots & 0 & 0 & 0 & 0 & \cdots & 0 & 0 & 0 & 0 & \cdots & 0 \\ \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots \\ 0 & 0 & 0 & \cdots & 1 & 0 & 0 & 0 & \cdots & 0 & 0 & 0 & 0 & \cdots & 0 \\ -\alpha_1 & -\alpha_2 & -\alpha_3 & \cdots & -\alpha_{\mu_1} & 0 & 0 & 0 & \cdots & 0 & 0 & 0 & 0 & \cdots & 0 \\ 0 & 0 & 0 & \cdots & 0 & 0 & 1 & 0 & \cdots & 0 & 0 & 0 & 0 & \cdots & 0 \\ 0 & 0 & 0 & \cdots & 0 & 0 & 0 & 1 & \cdots & 0 & 0 & 0 & 0 & \cdots & 0 \\ \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots \\ 0 & 0 & 0 & \cdots & 0 & 0 & 0 & 0 & \cdots & 1 & 0 & 0 & 0 & \cdots & 0 \\ 0 & 0 & 0 & \cdots & 0 & -\beta_1 & -\beta_2 & -\beta_3 & \cdots & -\beta_{\mu_2} & 0 & 0 & 0 & \cdots & 0 \\ & & \vdots & & & & \vdots & & & & & \vdots & & & \\ 0 & 0 & 0 & \cdots & 0 & 0 & 0 & 0 & \cdots & 0 & 0 & 1 & 0 & \cdots & 0 \\ 0 & 0 & 0 & \cdots & 0 & 0 & 0 & 0 & \cdots & 0 & 0 & 0 & 1 & \cdots & 0 \\ \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots & \cdots \\ 0 & 0 & 0 & \cdots & 0 & 0 & 0 & 0 & \cdots & 0 & 0 & 0 & 0 & \cdots & 1 \\ 0 & 0 & 0 & \cdots & 0 & 0 & 0 & 0 & \cdots & 0 & -\gamma_1 & -\gamma_2 & -\gamma_3 & \cdots & -\gamma_{\mu_p} \end{bmatrix}$$

or

$$\bar{A} - \bar{B}K = \begin{bmatrix} 0 & 1 & 0 & \cdots & 0 \\ 0 & 0 & 1 & \cdots & 0 \\ \vdots & \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & 0 & \cdots & 1 \\ -\alpha_0 & -\alpha_1 & -\alpha_2 & \cdots & -\alpha_{n-1} \end{bmatrix}$$

Solving above matrix equation we will get gain matrix K.

b) Calculation of gain matrix

Applying above method, for the desired pole 0.1, -0.1, 0.1i, -0.1i, the gain matrix is

$$K = \begin{bmatrix} 93.8377 & -24.8771 & 0 & 0 \\ -24.1188 & 24.3614 & 0 & 0 \end{bmatrix}$$

V. SIMULATION

In absence of the input forces, the angles and their velocities increase rapidly which make the system unstable (see figure 3).

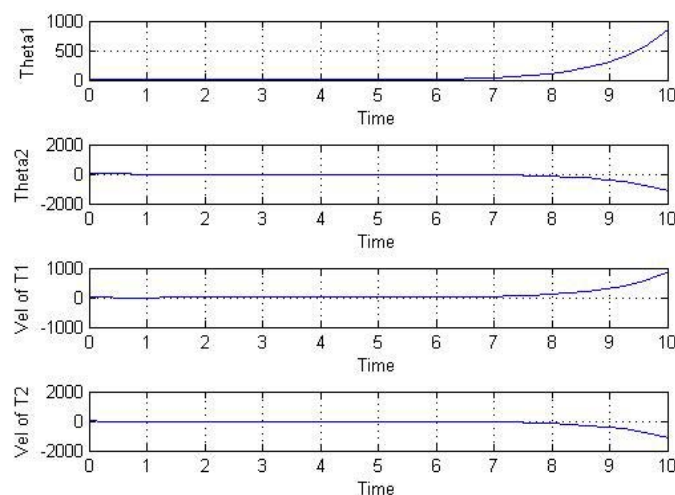


Figure 3: Uncontrolled System

By giving input force with measurement of gain matrix, the angles and their velocities will be slow down which make the system stable at the desired equilibrium place (see figure 4).

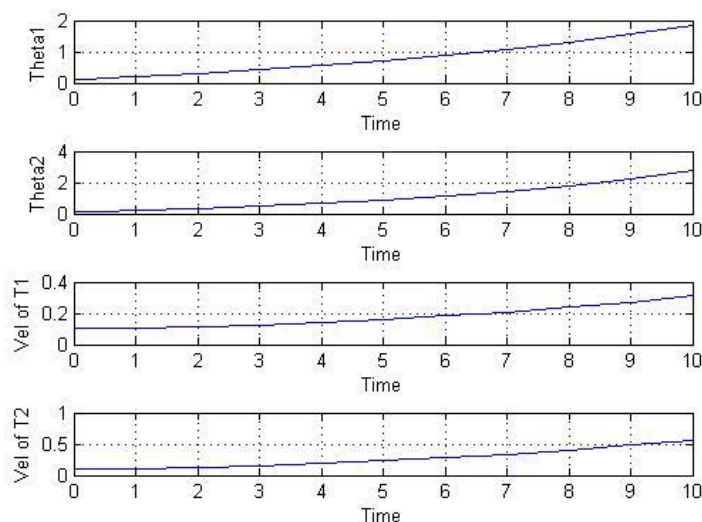


Figure 4 : Control System

VI. CONCLUSION

This study aims to understand what causes humanoids to fall, and what can be done to avoid it. Disturbances and modelling error are possible contributors to falling. For small disturbances in the walk of humanoid robot, it is simply behaving like a double inverted pendulum. So the results of this paper will be used in the development of the humanoid robot.

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- Very for a short time explain the tentative propose and how it skilled the declared objectives.

Approach:

- Use past tense except for when referring to recognized facts. After all, the manuscript will be submitted after the entire job is done.
- Sort out your thoughts; manufacture one key point with every section. If you make the four points listed above, you will need a least of four paragraphs.



- Present surroundings information only as desirable in order hold up a situation. The reviewer does not desire to read the whole thing you know about a topic.
- Shape the theory/purpose specifically - do not take a broad view.
- As always, give awareness to spelling, simplicity and correctness of sentences and phrases.

Procedures (Methods and Materials):

This part is supposed to be the easiest to carve if you have good skills. A sound written Procedures segment allows a capable scientist to replacement your results. Present precise information about your supplies. The suppliers and clarity of reagents can be helpful bits of information. Present methods in sequential order but linked methodologies can be grouped as a segment. Be concise when relating the protocols. Attempt for the least amount of information that would permit another capable scientist to spare your outcome but be cautious that vital information is integrated. The use of subheadings is suggested and ought to be synchronized with the results section. When a technique is used that has been well described in another object, mention the specific item describing a way but draw the basic principle while stating the situation. The purpose is to text all particular resources and broad procedures, so that another person may use some or all of the methods in one more study or referee the scientific value of your work. It is not to be a step by step report of the whole thing you did, nor is a methods section a set of orders.

Materials:

- Explain materials individually only if the study is so complex that it saves liberty this way.
- Embrace particular materials, and any tools or provisions that are not frequently found in laboratories.
- Do not take in frequently found.
- If use of a definite type of tools.
- Materials may be reported in a part section or else they may be recognized along with your measures.

Methods:

- Report the method (not particulars of each process that engaged the same methodology)
- Describe the method entirely
- To be succinct, present methods under headings dedicated to specific dealings or groups of measures
- Simplify - details how procedures were completed not how they were exclusively performed on a particular day.
- If well known procedures were used, account the procedure by name, possibly with reference, and that's all.

Approach:

- It is embarrassed or not possible to use vigorous voice when documenting methods with no using first person, which would focus the reviewer's interest on the researcher rather than the job. As a result when script up the methods most authors use third person passive voice.
- Use standard style in this and in every other part of the paper - avoid familiar lists, and use full sentences.

What to keep away from

- Resources and methods are not a set of information.
- Skip all descriptive information and surroundings - save it for the argument.
- Leave out information that is immaterial to a third party.

Results:

The principle of a results segment is to present and demonstrate your conclusion. Create this part a entirely objective details of the outcome, and save all understanding for the discussion.

The page length of this segment is set by the sum and types of data to be reported. Carry on to be to the point, by means of statistics and tables, if suitable, to present consequences most efficiently. You must obviously differentiate material that would usually be incorporated in a study editorial from any unprocessed data or additional appendix matter that would not be available. In fact, such matter should not be submitted at all except requested by the instructor.



Content

- Sum up your conclusion in text and demonstrate them, if suitable, with figures and tables.
- In manuscript, explain each of your consequences, point the reader to remarks that are most appropriate.
- Present a background, such as by describing the question that was addressed by creation an exacting study.
- Explain results of control experiments and comprise remarks that are not accessible in a prescribed figure or table, if appropriate.
- Examine your data, then prepare the analyzed (transformed) data in the form of a figure (graph), table, or in manuscript form.

What to stay away from

- Do not discuss or infer your outcome, report surroundings information, or try to explain anything.
- Not at all, take in raw data or intermediate calculations in a research manuscript.
- Do not present the similar data more than once.
- Manuscript should complement any figures or tables, not duplicate the identical information.
- Never confuse figures with tables - there is a difference.

Approach

- As forever, use past tense when you submit to your results, and put the whole thing in a reasonable order.
- Put figures and tables, appropriately numbered, in order at the end of the report
- If you desire, you may place your figures and tables properly within the text of your results part.

Figures and tables

- If you put figures and tables at the end of the details, make certain that they are visibly distinguished from any attach appendix materials, such as raw facts
- Despite of position, each figure must be numbered one after the other and complete with subtitle
- In spite of position, each table must be titled, numbered one after the other and complete with heading
- All figure and table must be adequately complete that it could situate on its own, divide from text

Discussion:

The Discussion is expected the trickiest segment to write and describe. A lot of papers submitted for journal are discarded based on problems with the Discussion. There is no head of state for how long a argument should be. Position your understanding of the outcome visibly to lead the reviewer through your conclusions, and then finish the paper with a summing up of the implication of the study. The purpose here is to offer an understanding of your results and hold up for all of your conclusions, using facts from your research and generally accepted information, if suitable. The implication of result should be visibly described. Infer your data in the conversation in suitable depth. This means that when you clarify an observable fact you must explain mechanisms that may account for the observation. If your results vary from your prospect, make clear why that may have happened. If your results agree, then explain the theory that the proof supported. It is never suitable to just state that the data approved with prospect, and let it drop at that.

- Make a decision if each premise is supported, discarded, or if you cannot make a conclusion with assurance. Do not just dismiss a study or part of a study as "uncertain."
- Research papers are not acknowledged if the work is imperfect. Draw what conclusions you can based upon the results that you have, and take care of the study as a finished work
- You may propose future guidelines, such as how the experiment might be personalized to accomplish a new idea.
- Give details all of your remarks as much as possible, focus on mechanisms.
- Make a decision if the tentative design sufficiently addressed the theory, and whether or not it was correctly restricted.
- Try to present substitute explanations if sensible alternatives be present.
- One research will not counter an overall question, so maintain the large picture in mind, where do you go next? The best studies unlock new avenues of study. What questions remain?
- Recommendations for detailed papers will offer supplementary suggestions.

Approach:

- When you refer to information, differentiate data generated by your own studies from available information
- Submit to work done by specific persons (including you) in past tense.
- Submit to generally acknowledged facts and main beliefs in present tense.



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References	Complete and correct format, well organized	Beside the point, Incomplete	Wrong format and structuring

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