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Design and Analysis of Shaft Used in Self-Driven Step Mechanism by Human Weight

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***Abstract:** In this paper, shaft used in self-driven step mechanism is designed and analysed by analytical method and using CAD software like ANSYS-V14.0 under static loading**.** Design and analysis includes numerical formulation to find out various factors such as bending and twisting moment, Stresses, deflections parameters under static loading using analytical method and Ansys-v14.0 and validation of results. Results of finite element analysis software are within the limits as compared with analytical solution to get conclusion for optimization of shaft.*

1. Introduction

Fig. 1 shows Self-cleaning step which includes combination of sector gear, pinions and idler gears driven by sector gear, which is driven by potential energy of human weight, whenever human puts his foot over oscillating top of the step. Idler gears are mounted in between pinions. There are three shafts mounted on the pinions which carry bristles on its periphery to clean footwear.

FIGURE 1. Self-driven step.

Shaft is a common important machine element. It is a rotating member, in general, has a circular cross-section is used to transmit power [4]. The shaft may be hollow or solid. The shaft is supported on bearings it rotates a set of gears or pulleys for the purpose of power transmission. The shaft is generally acted upon by bending moment, torsion axial force. Design of shaft primarily involves in determining stresses at critical point in the shaft that is arising due to aforementioned loading, deflection at various points.

A shaft is a rotating member, usually of circular cross-section for transmitting power using gears. It is supported by bearings supports. In self-driven step mechanism elements like shaft, bearing, gears are critical members over which direct load acts. These members are subjected to bending and twisting moment. So It is desirable to design components of product to expected levels of reliability (or failure probability), with the target level of reliability depending on specific consequences of failure. This would allow the development of risk-informed design methods may result in potential savings. Reliability-based designs promote consistency, allow more efficient designs, are more flexible rational than working stress or safety factor methods because they provide consistent levels of safety over various types of structures [1]. Probabilistic design allows a quantification of risk that can help to avoid over- or under-design problems while ensuring that safety quality levels are economically achieved [2]. Over design requires more resources than necessary leads to costly products. Avoiding over-design helps to conserve product materials reduce manufacturing resources, machining accuracy, quality control, etc. during processing. Under-designed products are prone to failures, making the products unsafe unreliable. This increases the risks of product liability lawsuits, customer dissatisfaction, even accidents [3]. Much research has been done in recent years on quantifying uncertainties in engineering systems their effects on reliability

2. Design of shaft

2.1. Material for shaft

Most shafts are made from steel, either low- or medium-carbon. However, high quality alloy steel, usually heat treated, may be chosen for critical applications. Other metals, e.g. brass, stainless steel or aluminium, may be used where corrosion is a problem or lightness is required. Small, light-duty shafts, e.g. in household appliances, may be injection- moulded in a plastic material such as nylon or delrin[4]

2.2. Analytical calculation

Firstly we have to find failure stress for maximum loading to design the shaft for given set of condition.

- Consider 60kg load acting on shaft

Table 1 Design Parameters for 60 Kg Load.

Parameters of shaft	Symbol	Value	Unit
Diameter	D	5	mm
Length	L	609.6	mm
Point Load	P	588.6	N

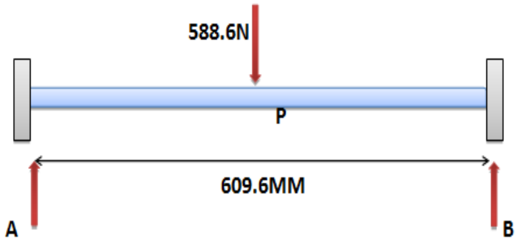


FIGURE 2. Simply supported shaft.

Conditions -Shaft is roller (bearing) supported
-Point load 'P' act on centre of shaft
-Material is stainless steel
-Modulus of elasticity E-193X10^3 Mpa

i) Reactions RA and RB
RA-588.6+RB=0
RA=588.6-RB
RAx0+588.6x304.8-RBx609.6=0
=179.40x10^3 =609.6RB
RB=294.29 N
RA=588.6-294.29
RA=294.29 N

ii) Shear force calculations (considering forces acting on left side of the section)
S.F. at A=0
S.F. just to right of A=294.29N
S.F. just to left of P=294.29N
S.F. just to right of P=294.29-588.6
= -294.29N
S.F. just to left of B=294.29-588.6
=-294.29N
S.F. at B=0

iii) Bending Moment Calculations
Since there is no moment acting on the beam, there shall not be sudden variation in BM.
B.M. at A=0
B.M. at P
P=294.29x304.8=86.69x10^3 Nmm
B.M.at B =0

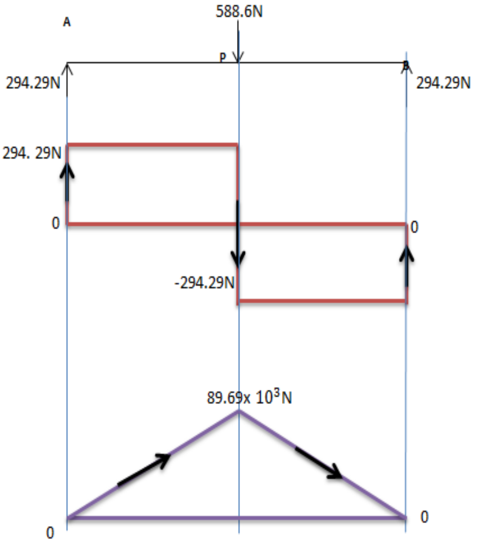


FIGURE 3. Force bending moment diagram.

iv) Deformation Calculations
Moment of inertia of the circular shaft about neutral axis= I

$$I=\frac{\pi d^4}{64}$$

$$I=\frac{\pi 5^4}{64}$$

$$I=30.67 mm^4$$

Using conjugate beam method,

$$\delta= \frac{WL^3}{48\ EI}$$

$$\delta= -469.64 \text{ mm}$$

v) Equivalent twisting moment calculation
Twisting moment can be found by,

$$\frac{T}{J}=\frac{I}{R}=\frac{G\theta}{L}$$

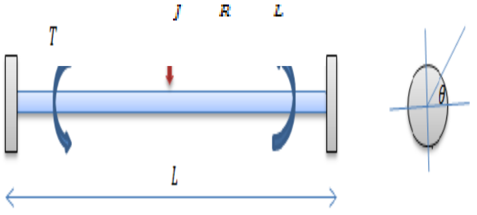


FIGURE 4.Twisting couple diagram.

Polar moment of inertia:

$$J=\frac{\pi d^4}{32}$$

=61.35mm4

Shear stress:

$$\tau=\frac{P}{A}$$

=29.97MPa

Angle of Twist θ :

$$\frac{\tau}{R}=\frac{G\theta}{L}$$

d

There for twisting ıd by,

$$\frac{T}{61.35}=\frac{193X\,10^3\,X0.037}{609.6}$$

Now find fa ıons (588.6 n force) using

- **Maximum shear stress theory:**

Step 1: calculation of maximum shear stress

$$\tau max=\frac{1}{2}\sqrt{(\sigma y-\sigma x)^2+4\tau^2}$$

Where, $\sigma y=\frac{P}{A}$ = 30 N/mm2

$$\sigma x=0$$

Ʈmax=33.54 Mpa

Calculation of factor of safety (FOS)

$$\tau max=\frac{0.5XSyt}{FOS}$$

Where, Syt= = 207 Mpa for stainless steel.

FOS=3.0 for 588.6 N

- Conceder 58.8 N force

Table 2. Design parameters for 58.8 N force.

Parameters of shaft	Symbol	value	Unit
Diameter	D	5	mm
Length of shaft	L	609.6	mm
Point load	P	588.6	N
Moment of inertia	I	30.67	mm4
Coefficient of friction	µ	0.1	-
Modulus of elasticity	E	193x 10^3	Mpa

i) Calculation of Normal force (N)

For designing of self-cleaning step we use polypropylene bristles. Hence coefficient of friction offered when there is leather to metal & metal to plastic contact is in the range of 0.1 -0.3. Which is depending on contact pressure and surface roughness.[5]

There for we take coefficient of friction (µ) as 0.1[6]

$$\mu=\frac{N}{F}$$

$$0.1=\frac{N}{588.6}$$

N=58.86 N

ii) Deflection of shaft (δ) at normal force

Using conjugate beam method,

$$\delta=\frac{WL^3}{48\ EI}$$

$$\delta=\frac{-58.86X609.8^3}{48X193X10^3\ X30.67}$$

δ=-46.97 mm

iii) Bending moment of shaft at normal force

RAx0+58.6x304.8-RBx609.6=0
RB=29.4 N
RA=29.4 N
B.M. at A=0
B.M. at P=8.96x10^3 Nmm
B.M.at B=0

- Conceder 177.7 N force

Table 3. Design parameters for 117.7 N force.

Parameters shaft	Symbol	value	Unit
Diameter	D	5	mm
Length of shaft	L	609.6	mm
Point load	P	1177.7	N
Moment of inertia	I	30.67	mm4
Coefficient of friction	µ	0.1	-
Modulus of elasticity	E	193X10^3	Mpa

i) Calculation of Normal force (N)

$$\mu=\frac{N}{F}$$

$$0.1=\frac{N}{1177.2}$$

N=117.72 N

ii) Deflection of shaft (δ) at normal force

Using conjugate beam method,

$$\delta=\frac{WL^3}{48\ EI}$$

$$\delta=\frac{-117.72X609.8^3}{48X193X10^3\ X30.67}$$

δ=-93.95 mm

iii) Bending moment of shaft at normal force

RAx0+117.72x304.8-RBx609.6=0
RB=58.86 N
RA=58.86 N
B.M. at A=0
B.M. at P=17.92x10^3 Nmm

B.M.at B=0

- Conceder 58.8 N force and shaft diameter 10 mm

Parameters of shaft	Symbol	value	Unit
Diameter	D	10	mm
Length of shaft	L	609.6	mm
Point load	P	588.6	N
Moment of inertia	I	490.87	mm4
Coefficient of friction	μ	0.1	-
Modulus of elasticity	E	$193X10^3$	Mpa

Table 4. Design Parameters for 58.8 N force.

i) Calculation of Normal force (N)

$$\mu=\frac{N}{F}$$

$$0.1=\frac{N}{588.6}$$

N=58.86 N

ii) Deflection of shaft (δ) at normal force

Using conjugate beam method,

$$\delta=\frac{WL^3}{48\ EI}$$

$$\delta=\frac{-58.86X609.8^3}{48X193X10^3X490.87}$$

δ=-2.93 mm

iii) Bending moment of shaft at normal force

RAx0+58.6x304.8-RBx609.6=0
RB=29.4 N
RA=29.4 N
B.M. at A=0
B.M. at P=8.96x10^3 Nmm
B.M.at B=0

- Conceder 177.7 N force and shaft diameter 10 mm

Parameters of shaft	Symbol	value	Unit
Diameter	D	10	mm
Length of shaft	L	609.6	mm
Point load	P	1177.2	N
Moment of inertia	I	490.87	mm4
Coefficient of friction	μ	0.1	-
Modulus of elasticity	E	$193X10^3$	Mpa

Table 5. Design Parameters for 177.7 N force.

i) Calculation of Normal force (N)

$$\mu=\frac{N}{F}$$

$$0.1=\frac{N}{1177.2}$$

N=117.72 N

ii) Deflection of shaft (δ) at normal force

Using conjugate beam method,

$$\delta=\frac{WL^3}{48\ EI}$$

$$\delta=\frac{-117.72X609.8^3}{48X193X10^3\ X490.87}$$

δ=-5.87 mm

iii) Bending moment of shaft at normal force

RAx0+117.72x304.8-RBx609.6=0
RB=58.86 N
RA=58.86 N
B.M. at A=0
B.M. at P=17.92x10^3 N.MM
B.M.at B=0

Analysis using FEA software

With finite element modelling a three-dimensional (3D) finite element model is developed to simulate the behaviour of shaft. Analysis is done using finite element software package ANSYS 14.

Fig.5 shows properties of stainless steel material used for shaft.

Modelling- Model created from work page.

Meshing: Meshing of model includes selection of element size, type and number of elements use [7]

- Relevance centre-coarse
- Element type- hexahedron
- Nodes-45
- Number of element-22

A. Analysis for 588.6 N force

Table 6 Parameter 60 Kg Load of Person.

Parameters of shaft	Symbol	value	Unit
Diameter	D	5	mm
Length of shaft	L	609.6	mm
Point load	P	588.6	N
Moment of inertia	I	30.67	mm4
Modulus of elasticity	E	193X 10^3	Mpa

i) Bending moment at point 'p'

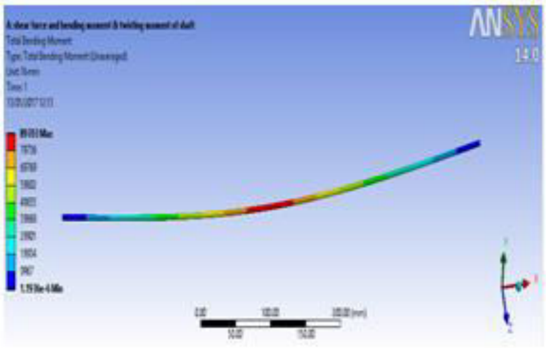


FIGURE 8. Bending moment for 588.6N force.

ii) Deformation at point 'P'

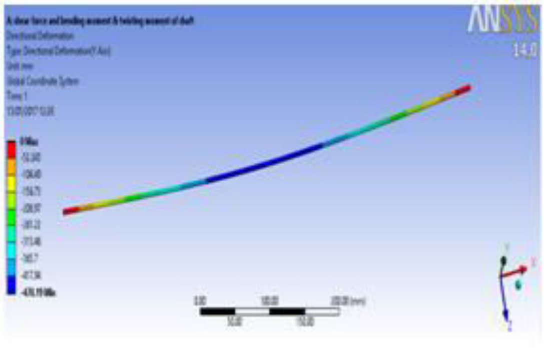


FIGURE 9. Deformation for 588.6 N force.

B. Analysis for 58.86 N normal force

Table 7 Parameter for 60 Kg Weights of People.

Parameters of shaft	Symbol	value	Unit
Diameter	D	5	mm
Length of shaft	L	609.6	mm
Normal force	N	58.86	N
Moment of inertia	I	30.67	mm4
Coefficient of friction	M	0.1	-
Modulus of elasticity	E	193X 10^3	Mpa

i) Directional deformation

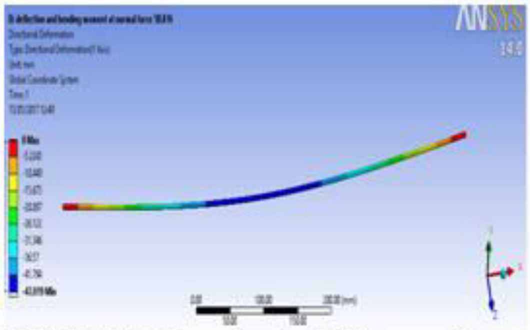


FIGURE 10. Deformation for 58.86 normal forces.

ii) Total bending moment

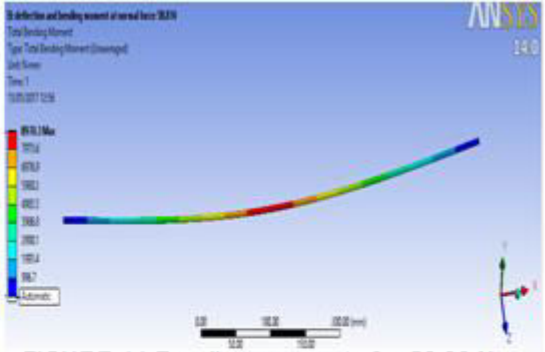


FIGURE 11 Bending moments for 58.86 N forces.

C. Analysis for 117.7 N normal force

Table 8 Parameter for 120 Kg Weights of People.

Parameters of shaft	Symbol	value	Unit
Diameter	D	5	mm
Length of shaft	L	609.6	mm
Normal force	N	117.7	N
Moment of inertia	I	30.6	mm4
Coefficient of friction	µ	0.1	-
Modulus of elasticity	E	193X 10^3	Mpa

i) Directional deformation

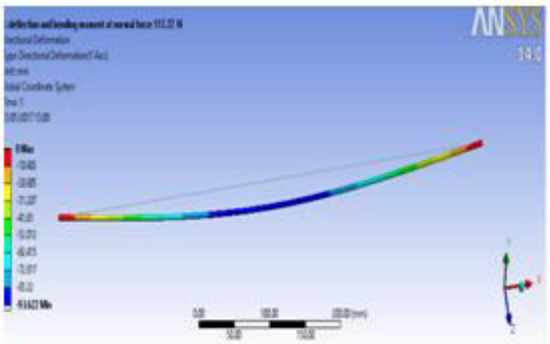


FIGURE 12. Deformation for 117.7 N force.

ii) Total bending moment

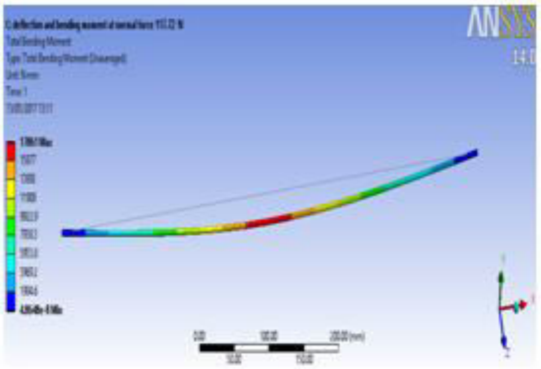


FIGURE 13. Bending moment for 117.7 N forces.

D. Analysis for shaft diameter 10 mm and 58.8 N normal force

Table 9 Parameter 60 Kg Weight of People and Shaft Diameter 10 mm.

Parameters of shaft	Symbol	value	Unit
Diameter	D	10	mm
Length of shaft	L	609.6	mm
Normal force	N	58.86	N
Moment of inertia	I	490.8	mm4
Coefficient of friction	µ	0.1	-
Modulus of elasticity	E	193X	Mpa

		10^3	

i) Directional deformation

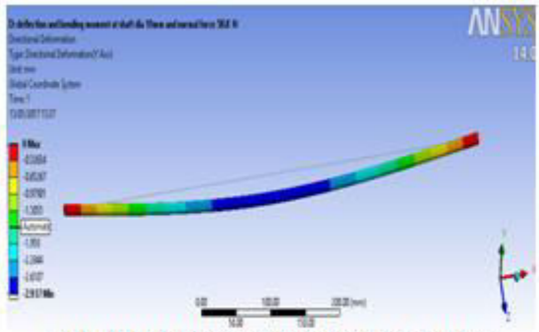


FIGURE 15. Deformation for 58.8 N force.

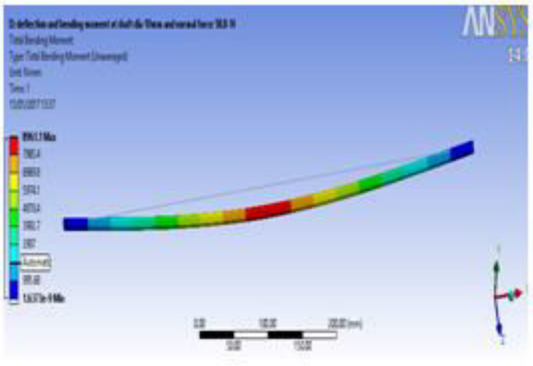


FIGURE. 16 Bending moment for 58.8 N force.

ii) Total bending moment

E. Analysis for shaft diameter 10 mm and

Parameters of shaft	Symbol	value	Unit
Diameter	D	10	mm
Length of shaft	L	609.6	mm
Normal force	N	177.7	N
Moment of inertia	I	490.8	mm4
Coefficient of friction	M	0.1	-
Modulus of elasticity	E	193X10^3	Mpa

i) Directional deformation

ii) Total bending moment

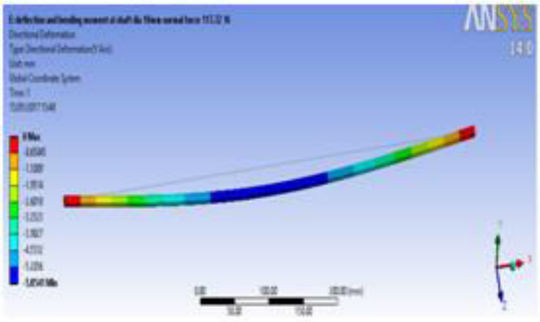


FIGURE 17. Deformation for 177.7 N force

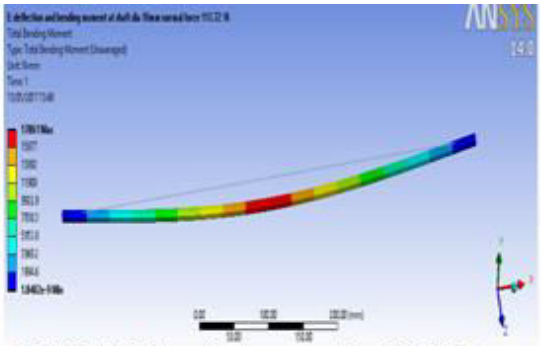
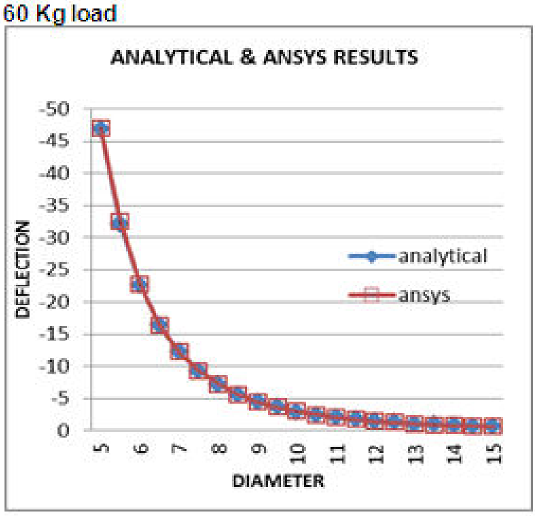


FIGURE 18. Bending moment for 58.8 N forces.

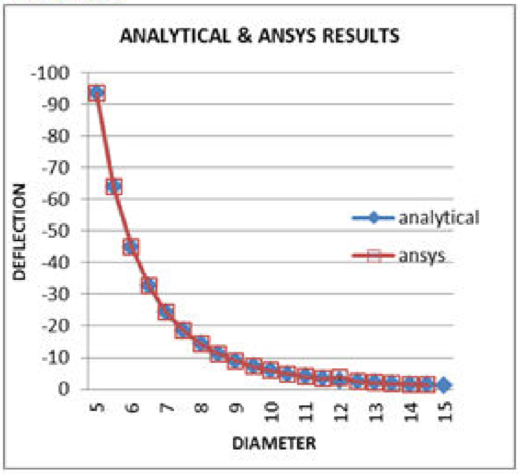
4. Validation

Validation is comparison of analytical results with CAD results. Analytical results obtained above are compared with Ansys results.

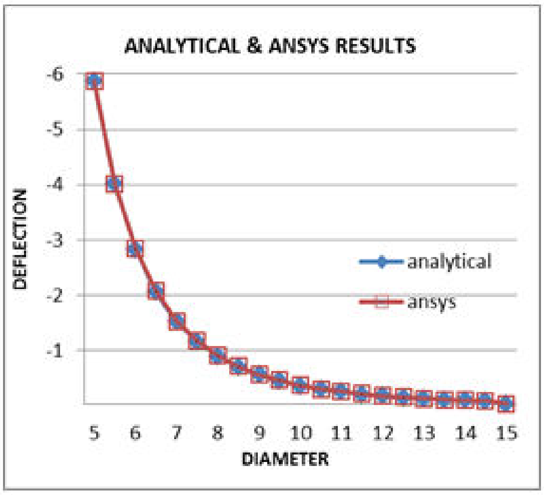
Graph 1 Deflection Verses Diameter of Shaft for 60 Kg load



Graph 2 Deflection Verses Diameter of Shaft for 120 Kg load



Graph 3 Deflection Verses Diameter of Shaft for 60 Kg load when shaft is centrally supported.



Graph 4 Deflection Verses Diameter of Shaft for 120 Kg load When Shaft is Centrally Supported.

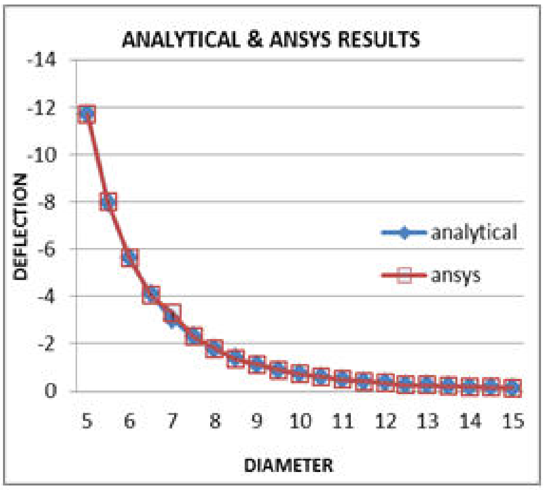


Table 11 Results Obtained For Various Loading Conditions and Dimensions in Section-II

	Parameter	Force (N)	D	Analytic results	Ansys results
A	Bending moment (Nmm)	588.6	5	$89.69 X 10^3$	$89.69 X 10^3$
	Twisting moment (Nmm)			718.66	735
	Max Deflection (mm)			-469.64	-470
B	Max Deflection	58.8	5	-47	-46.97
	Bending moment (Nmm)			$8.970 X 10^3$	$8.96 X 10^3$
C	Max Deflection	117.2	5	-93.62	-93.95
	Bending moment (Nmm)			$17.81 X 10^3$	$17.92 X 10^3$
D	Max Deflection (mm)	58.8	10	-2.93	-2.93
	Bending moment (Nmm)			$8.61 X 10^3$	$8.96 X 10^3$
E	Max Deflection (mm)	117.2	10	-5.58	-5.87
	Bending moment (Nmm)			$17.6 X 10^3$	$17.92 X 10^3$

5. Conclusion

From graph-1 and graph-2 it is conclude that when 60 and 120kg of load acts on shaft, deflection is -46.97 and -93.95mm respectively when diameter of shaft is 5mm, Which is too much from design view and not acceptable.

From graph-1 and graph-2 it is conclude that, when diameter of shaft increases strength of shaft also increases. i.e. when D=10 mm, deflection is -2.93 and -5.87 mm for 60 &120 kg load respectively, Which is not considerable.

From graph-3 and graph-4 it is conclude that, when shaft is supported centrally by bearing, load on shaft is equalised. Hence deflection of shaft is minimum. Hence shaft can also be optimise by selecting the diameter 6 mm with central bearing support, which gives deflections -2.84 and -5.63 mm for 60kg & 120 kg respectively cut down excessive material cost with optimum performance.

If 10 mm diameter shaft is used, then deflection is -2.93 & -5.87mm for 60kg and 120kg load, hence clearance between sole of shoe and bristles is -2.93 or -5.87, Which is not considerable . So we can select bristles with medium stiffness (i.e. not so hard and not so soft) which effectively removes cleats or dust particles from shoe.

For optimisation purpose i.e. minimizing excessive material, optimum performance and cost cutting, shaft diameter should not be too larger or too small. Small diameter may cause more deformation and larger diameter causes excessive material costing.

6. References

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