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Design Improvement of Metallic Carpet Weaving Loom

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ABSTRACT

This paper presents a design improvement for the metallic looms used to weave hand knotted carpets. The existing looms face problems such as failure of worm and worm gear, and the inconvenience of hook covers in the upper and lower horizontal beams. In this paper improvements are proposed along these lines. First, the loading conditions of worm and worm gear are checked and force analysis was performed. This was followed by their design according to IS standards. To facilitate the design iterations a MATLAB programme was written which truly helps in checking any new worm gear set. Secondly, a structured horizontal beam with slotted disks is introduced with which requirements of hooks are avoided. For this, suitable number of disks is chosen to bear the tensions in carpet threads within allowable angular deflection of the beam.

Keywords: Carpet loom; Design; Worm and worm gear; Beam

1. INTRODUCTION

Traditionally, Indian or oriental carpets are knotted on wooden looms. These looms are not economically and functionally useful due to i) limited life (5-8 years); ii) Deforestation; iii) Laborious tensioning,; and iv) Non-uniform tension in the warps over the time [1,2]. The non-uniformity affects quality of the carpet. A systematic approach to improve the existing tools and processes used by artisans engaged in carpet sectors had been initiated in 2000 by IIT Delhi [3]. Since their developments, around 200 looms are in use in several places of the country namely, Shrinagar (J&K), Bhdohi (UP), Valsad (GUJ), and others. The metallic loom were designed considering all aspects of carpet weaving. The upper and lower beams are supported on side columns and locked in the plane normal to the plane of the warp threads using ratchet and pawl and worm gear, respectively. As carpet knotting to weave about six-inch carpet takes several days, the structure is subjected to steady loading. In order to verify analyticaliy, finite element (FE) analysis of the metallic loom was carried out to determine the critical stresses and deflection in its components so that optimum sizes and shapes of the structural members can be selected [4]. The FE model results are comparable with those from the analytical results. The weight optimisation of the metallic loom was carried out resulting in relatively lightweight and reduced cost [5]. The columns proposed in [5] are hollow square sections. In order to use fabricated structural sections, the columns are changed to fabricated structure of L-sections (Fig.1).

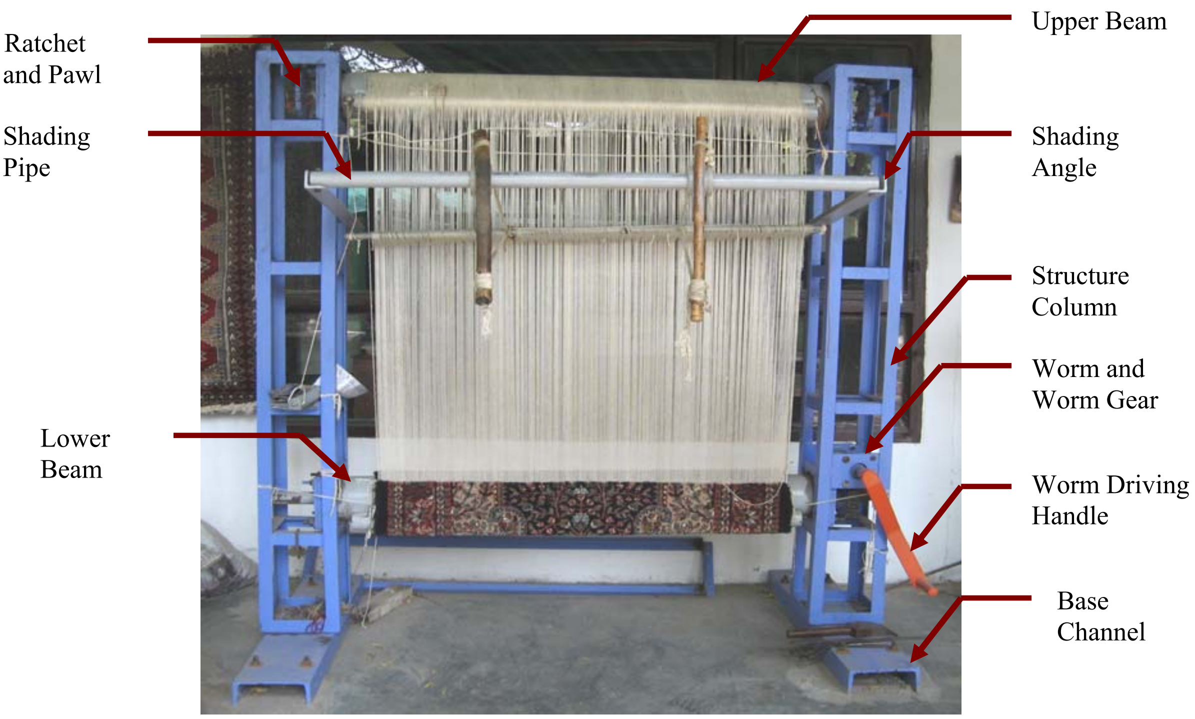

Fig. 1 New Design

2. WORM AND WORM GEAR

It was reported during the feedback collections that the worms and gears, shown in Fig. 2 were failing. Hence, their loading conditions were rechecked. The tension required to be produced by the gear set came to be 24 KN.(It should be 20 N/mm x1500mm=30KN) Then the worm set was redesigned according to the IS standard, and the material selection was done according to the standard data tables. The worm design has been finalized under static loading conditions. Material C40 was chosen for the worm wheel and worm gear.

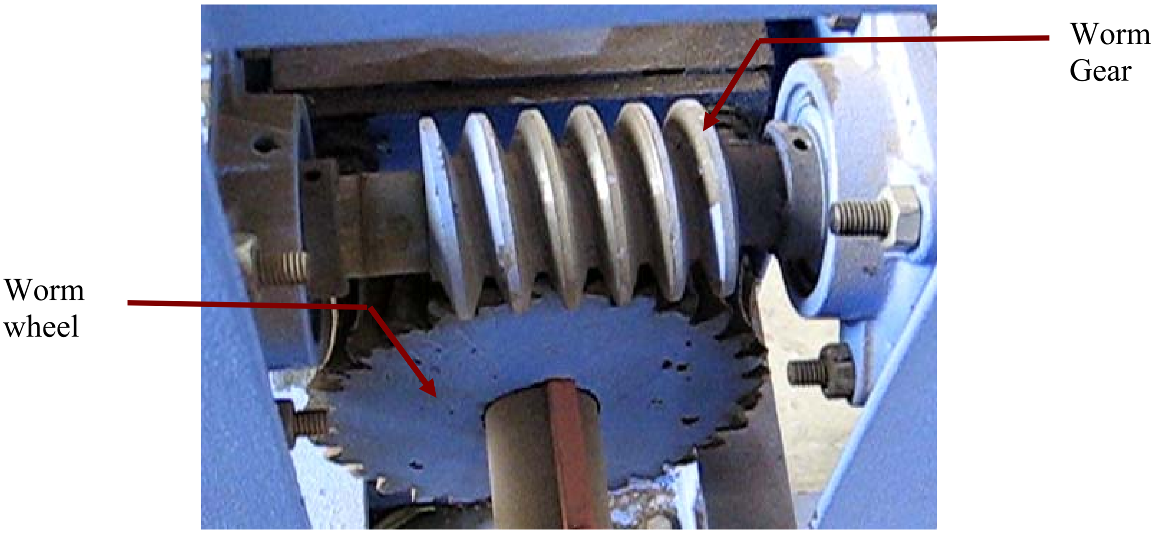

Fig. 2 Worm Wheel and Worm Gear

2.1 Load calculation

2.2.1 Input torque

The warp threads are tensioned by a weaver using worm and worm gear through a worm driving handle, as shown in Fig. 1. If the length of handle is l_h then the input torque a weaver can able is given by

$$T_{in} = F_w \times l_h \quad (1)$$

where F_w is force applied by the weaver. In the normal condition, a weaver can apply force of about 300 N at the handle and if the length of handle is 300 mm, maximum input torque applied by the weaver is 90 N-m.

2.2.2 Torque due to warp tension

Since all warp threads are wound uniformly over the upper beam, as shown in Fig. 3 (a), they are modeled as uniformly distributed load [4]. The same load is acting on lower beam, Fig.3(b). The torque due to warp load depends on total tension in warp threads and beam diameter, and is given by

$$T = F_t \times d/2 \ \ where\ F_t = w \times l \quad (2)$$

In which w is beam load per unit length due to the tension in the warps; d is diameter of beam and l is effective beam length over which threads are wrapped.

2.2.3 Gear tooth forces

It is required to select a standard worm and worm gear set which take input torque given by Eq. (1) and deliver the torque needed to tense the thread given by Eq. (2). Noting that the worm is the driving member, while the worm gear wheel is the driven member, the components of the gear tooth force between the worm and worm wheel are shown in Fig. 4.

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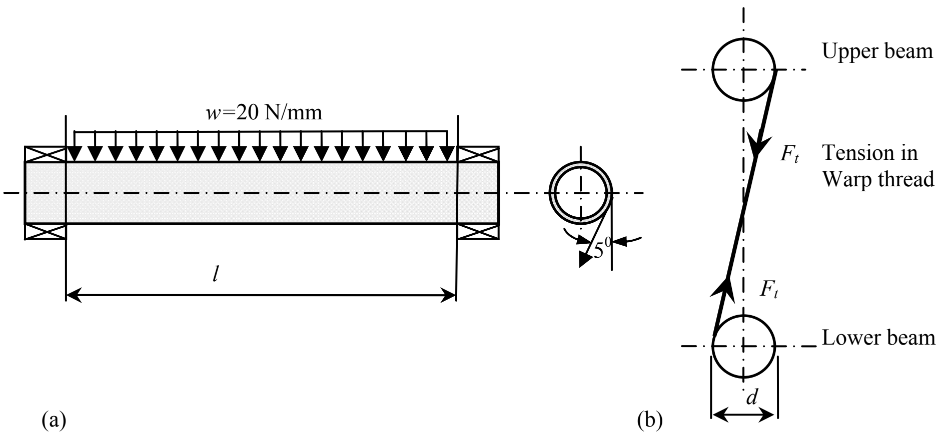

Fig. 3 (a) Warp thread load diagram on beam, (b) Warp thread tension

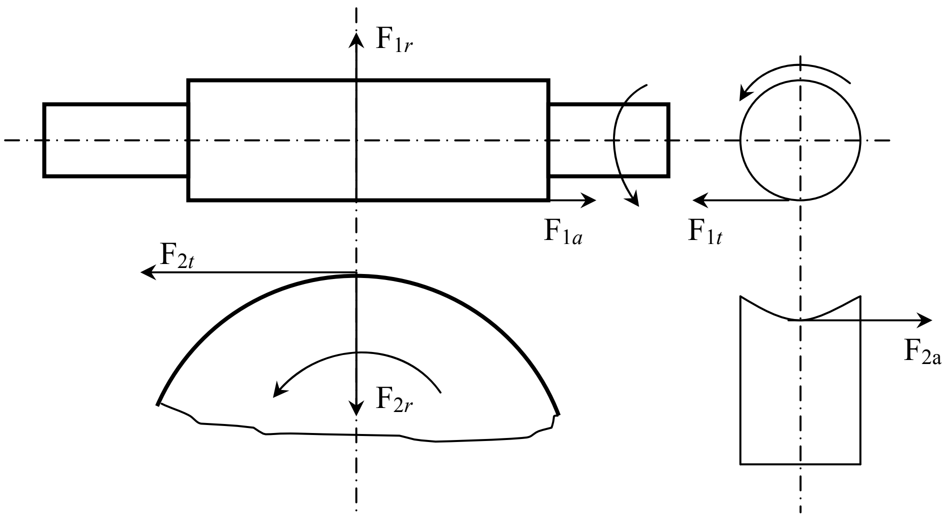

Fig. 4 Components of tooth forces

The tangential component on the worm is computed from the input torque as

$$F_{1t} = 2T_{in} / d_1 \tag{3}$$

where d_1 is diameter of pitch circle of worm. Knowing the tangential component, the axial and radial components on the worm are determined as follows [6]

$$F_{1a} = F_{1t} \times \frac{\cos\alpha\cos\gamma - \mu\sin\gamma}{\cos\alpha\sin\gamma + \mu\cos\gamma} \tag{4}$$

$$F_{1r} = F_{1t} \times \frac{\sin\alpha}{\cos\alpha\sin\gamma + \mu\cos\gamma} \tag{5}$$

where α is normal pressure angle, γ is the lead angle and μ is coefficient of friction between teeth of worm and worm wheel. Since the force acting on the worm wheel is equal and opposite reaction of the force acting on the worm wheel, hence,

$$F_{2t} = -F_{1a} \tag{5}$$

$$F_{2a} = -F_{1t} \tag{6}$$

$$F_{2r} = -F_{1r} \tag{7}$$

The torque delivered by the worm wheel is now obtained by

$$T_{out} = F_{2t} \times d_2 / 2 \tag{8}$$

where d_2 is diameter of pitch circle of worm wheel. To develop required tension in the warp thread the torque delivered by the worm wheel, T_{out}, should be greater than the torque needed to tense the threads, T, i.e., $T_{out} \geq T$

2.2 Load ratings for wear and strength

Indian Standard IS 7443:2002 is followed for load rating of worm gears. The permissible torque is limited by the lower of the following two values for wear.

$$T_{c1} = 0.00191 X_{c1} S_{c1} Y_z d_2^{1.8} m \tag{8}$$

$$T_{c2} = 0.00191 X_{c2} S_{c2} Y_z d_2^{1.8} m \tag{9}$$

The permissible torque values for strength are given by

$$T_{b1} = 0.0018 X_{b1} S_{b1} m l_r d_2 \cos\gamma \tag{10}$$

$$T_{b2} = 0.0018 X_{b2} S_{b2} m l_r d_2 \cos\gamma \tag{11}$$

Suffix 1 for worm and suffix 2 for worm wheel are used with the above notations wherever applicable. Various symbols used in Eqs. (8)-(11) are defined as per IS 7443:2002:

Symbols	Description	Units
D	Reference diameter	mm
l_r	Length of root of the worm wheel teeth	mm
M	Axial module	mm
N	Speed of rotation	min^{-1}
Q	Diameter factor	-
v_s	Rubbing speed	m/s
Z	Number of starts or teeth	-
S_b	Bending stress factor	-
S_c	Surface stress factor	
T_b	Permissible torque for strength	Nm
T_c	Permissible torque for wear	Nm
X_b	Speed factor for strength	-
X_c	Speed factor for wear	-
Y_z	Zone factor	-

2.3 Designation and Dimensions for worm gearing

IS: 3734-1998 is followed, for the dimensions and designation. According to this standard, a pair of worm gears shall be designated by the hand of the thread of the worm, number of starts of the worm (z_1), number of teeth of the wheel (z_2), diameteral quotient of the worm ($q=d/m$), module (m) and centre distance of the gear pair as Worm Gears R *z*1/*z*2/*q*/*m*-centre distance.

2.4 Computer MATLAB Codes

The process of selection of worm and worm gear is iterative type as shown above. Therefore, to ensure the correctness of the results and speed of calculating Computer MATLAB codes were generated as shown in Appendix A. The steps are demonstrated in next section.

3. NUMERICAL EXAMPLE: FOR LOOM SIZE 4' X 6'

Considering different yarn quality to be used in the loom design load of 20N for each warp thread is safe [4]. Therefore, for the loom of size 4' x 6' the dimensions of the loom are as follows [4]:

$$w = 20\,\text{N/mm}; \; l = 150\,\text{mm}; \; d = 117.8\,\text{mm}$$

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Using Eq. (2), the torque needed to tense warp threads is obtained as, $T = 1767$Nmm.

First trial for selection of worm and worm gear

Assuming that weaver can rotate the worm through handle with speed 30 rpm and with a 100 percent efficient gear set, the speed of worm wheel is obtained using $T_{in} \times n_1 = T \times n_2$ as $n_2 = 1.02\,rpm$. Hence, $n_2 / n_1 \cong 30$. Using IS 3734-1983 recommended transmission ratios, centre distances and corresponding z_1, z_2, q, m, we choose for first trial: 1/29/7.5/5.5-100. For this set dimensions of worm and worm wheel are chosen from the standard as follows:
Axial module, m=5.5 mm; No. of starts, z_1=1; Pitch circle diameter of worm, d_1=41.25 mm; Diameteral quotient, q=$7^1/_2$; Lead angle, γ=7°36'; No. of teeth on wheel, z_2=29;
Pitch circle diameter of wheel, d_2=159.5 mm. Accordingly loads are calculated as follows:

From Eq. (3), F_{1t}=4364 N. From IS7443:2002, Rubbing speed v_s=0.065 m/s and corresponding μ=0.08. Assuming pressure angle as α=20°, Eqs. (4) and (5) give F_{1a}=19906 N and F_{1r}=2820 N. From Eqs. (6)-(8), F_{2t}=-19906 N, F_{2a}=4364 N and F_{2r}=2820 N. From Eq. (9) torque delivered, T_{out}=1586 N. Therefore, $T \geq T_{out}$, i.e., the torque delivered is not sufficient for tensioning the threads. From wear and strength point of view, the permissible torque is limited by the lower of the four values for wear and strength given by Eqs. (8)-(11). A sample trial is are summarized in Table 1.

Table 1 Selection of worm and worm gear

Centre distance, mm	**Speed ratio**	**Worm set**	**Forces, N**				**Design check for load, Nm**				
	$i = n_1/n_2$	$z1/z2/q/m$	F_1	F_{1a}	F_{2t}	F_{2a}	T_{out}	T_{c1}	T_{c2}	T_{b1}	T_{b2}
100	30	1/29/7.5/5.5	4364	19906	19906	4364	1586	600	600	600	600

Check each iteration for

(1) $T_{out} \geq T$ for proper tensioning

(2) $T_{c1}, T_{c2}, T_{b1}, T_{b2} \geq T_{out}$ for safe in wear and strength

Using rest of the input data as given in Numerical example, i.e., α (pressure angle) = 20°, γ (lead angle) = 6.2°, σ_d = 140 for C-40 and 120 for C-20 material. Calculations are done. The results are tabulated below:
Force analysis:

Input				**Output**		
Worm set	Module (m)	Mean dia (d_2)	Mean dia (d_1)	Force analysis		
				T_r	T_s	Remarks
R1/31/9/4	4	124	36	1410	1566	Sufficient
R1/41/9/4	4	164	36	1410	1780	Sufficient
R1/51/11/3.25	3.25	165.75	35.75	1410	1832	Sufficient

Strength analysis:

Input					**Output**			
	Constants				C-20		C-40	
Worm set	X_{c1}	X_{c2}	X_{b1}	X_{b2}	T_{b1}	T_{b2}	T_{b1}	T_{b2}
R1/31/9/4	0.70	0.52	0.48	0.65	1370	1856	1576	2134
R1/41/9/4	0.72	0.53	0.48	0.65	1812	2455	2084	2823
R1/51/11/3.25	0.71	0.51	0.48	0.65	1316	1782	1513	2049

Comparing the results as $T_{c1}, T_{c2}, T_{b1}, T_{b2} \geq T_{out}$, since, the R1/31/9/4 worm set satisfying torque requirement specifically, it is chosen for fabrication and further testing in real looms.

4. REDISIGN OF BEAMS

In the development of seamless beams, the main objective was to keep the weight minimum. Since the weight of the beam directly affects the loom cost, the design of the beam based on disks was considered even though the manufacturing cost may be higher. This will also remove the requirement of the hooks.

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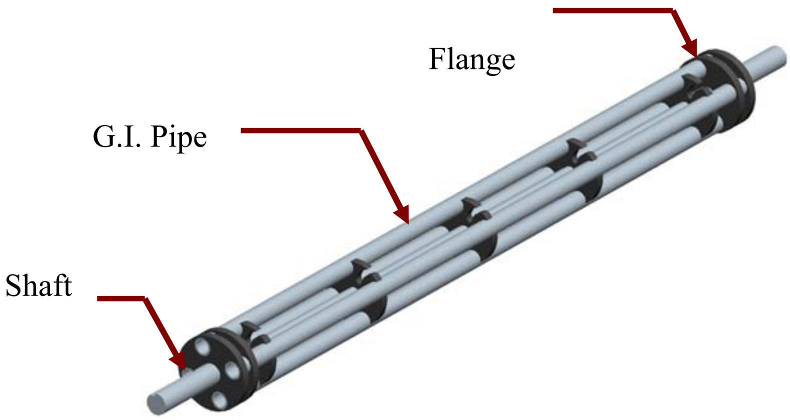

Fig. 5 Disk beam (for Carpet size: 4' X 6')

The following specifications are used for the disk beam:

Number of pipes at circumference as shown (N) = 4; Diameter of solid rod at middle (d) = 40mm; Outer diameter of pipe (d_0) = 33.4 mm; Inner diameter of pipe (d_i) = 26.64mm.

For checking the design of structure beam, equivalent diameter of beam (d_e) was calculated by balancing the inertia of an equivalent solid beam with structured beam, i.e.,

$$\frac{\pi}{64} * d_e^4 = N * \frac{\pi}{64} * (d_0^4 - d_i^4) + N * y^2 * \frac{\pi}{4} * (d_0^2 - d_i^2) + \frac{\pi}{64} * d^4$$

The strength of beam is then checked by Von-Mises stress theory [6]. Also, the maximum deflection was calculated using a MATLAB programme, as shown in Appendix A. The results with iterations are shown below:

Serial number	No. of pipes	No. of stiffeners	d_i/d_o mm	d mm	d_e mm	Maximum Von-Misses stress MPa	Maximum deflection mm
1	4	8	8/14	40	71.69	166.95	5
2	4	8	19/27	40	87	89	2.3
3	4	8	27/34	40	83.56	99.42	2.63
4	3	8	27/34	40	78.10	121.58	3.43

Since, the deflection in second row is lowest, it is chosen for fabrication and further testing in real looms.

5. STRUCTURED COLUMNS

The column of the metallic looms developed by IIT Delhi during 2000-2003 was having a cross section of 'C' type. It was facing the problem of twisting. The twist was generated because of the moments applied on the gear box. Also the motion in the worm shaft and the ratchet generated a reaction force in the columns. To overcome this problem, the structure of hollow trusses was proposed. The design consists of 4 angles standing vertically as shown in Fig. 1. The angles selected for the design are of the size of 40*40*4 mm. The sizes are taken from standard sizes available in the market. The column angles are having straight stiffeners in between the structures to maintain the required stiffness. This has avoided twisting.

6. CONCLUSIONS

In this paper improvements in the carpet loom are proposed to overcome the drawbacks in the existing metallic looms. The selection procedure of worm and worm gear is standardized according to IS standards. The iterative process is coded as a MATLAB programme. A structured horizontal slotted disks beam is also proposed which removed the requirement of hooks.

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Appendix A

MATLAB Codes

Force Analysis

```
function [obj1,obj2,obj3]=Forceanalysis(DV)
%===============================================
%DV=[omega,d1,d2,L,db,alpha,gamma,F_weaver,handle_length,mu]
omega=20;%thread force per unit lenth (N/mm)
d1=35.75; d2=165.75;
L=1200; % width of loom,i.e., carpet width in mm
db=117.5; %External diameter of beam,mm
alpha1=20;% pressure angle
gamma1=5.2;%lead angle
F_weaver=300; handle_length=300; mu=0.08;
%============================================
% input torque by weaver
Tin=(F_weaver*handle_length/1000);
%Force on the tooth of worm
alpha=3.14/180*alpha1; gamma=3.14/180*gamma1;
% tangentail force on the tooth of worm
Ft1=Tin*2*1000/d1;
%determination of axial and radial force
Fa1=Ft1*(cos(alpha)*cos(gamma)-
mu*sin(gamma))/(cos(alpha)*sin(gamma)+mu*cos(gamma));
Fr1=Ft1*sin(alpha)/(cos(alpha)*sin(gamma)+mu*cos(gamma));
%Forces on the tooth of wheel
Ft2=Fa1;%Opposite direction
%Torque supplied by the weaver at wheel
Ts=Ft2*(d2/1000)/2;
%Torque required for tensioning the warp threads, N-m
Tr=omega*L*db/(2*1000);
obj1=Ts;
obj2=Tr;
```

```
Ts;Tr
%============================================
Worm Design
close all;clear all;clc;
%R1/41/9/5-125
% Terminology
%z1/z2/q/m; z1=no of start on worm; z2=no. of teeth on worm wheel;
%q=diametral quotient=d1/m; m=module
%axial pitch of worm, px=pi*m
%Lead of worm, L=px*z1;
%Lead angle, gamma
%INPUT
%============================================================
z1=1;z2=51;q=11;m=3.25;
n1=51;%assumption rotation per mimute of worm
omega=20;%thread force per unit lenth (N/mm)
L=1200; % width of loom,i.e., carpet width in mm
db=117; %External diameter of beam,mm
alpha_degree=20;% pressure angle
alpha=alpha_degree*pi/180;
F_weaver=300;%assumption Load applied by weaver at handle
handle_length=300;%lenth of handle, mm
gamma=5.2;%Lead angle
%=================================================================
%rotaion of wheel
n2=n1*z1/z2;
% Dimensions of worm
d2=z2*m
%Centre distance,a
a=m*(q+z2)/2;
%speed ratio
i=z2/z1;
%Dimensions of worm
%pitch circle diameter
d1=m*q;
%addendum cirle diameter
da1=m*(q+2);
%Lead angle
gamma_degree=gamma*pi/180;
%dendendum cilce diameter
df1=m*(q+2-4.4*cos(gamma));
% Dimensions of worm wheel
da2=m*(z2+4*cos(gamma)-2);
df2=m*(z2-0.4*cos(gamma)-2);
%Rubbing velocity
vs=0.0000524*d1*(n1/cos(gamma_degree))
%============================================
mu=0.08;% read from the IS 7443:2002 Fig. 4
%===============================================
Xc1=0.71;Sc1=8.3;Yz=1.160; Xc2=0.51;Sc2=8.3; Xb1=0.48;Sb1=120;
Xb2=0.65;Sb2=120;
%Permissible Torque for Wear
Tc1=0.00191*Xc1*Sc1*Yz*d2^1.8*m
Tc2=0.00191*Xc2*Sc2*Yz*d2^1.8*m
c=0.2*m*cos(gamma_degree); I=da1+2*c; b=2*m*sqrt(q+1); H=b/I;
Lr=I*asin(H);
%Permissible Torque for strength
Tb1=0.0018*Xb1*Sb1*m*Lr*d2*cos(gamma_degree)
Tb2=0.0018*Xb2*Sb2*m*Lr*d2*cos(gamma_degree)
DV=[omega,d1,d2,L,db,alpha,gamma,F_weaver,handle_length,mu,];
```

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```
Forceanalysis(DV);
DV;
%===============================================
%=================================
%MATLAB programme for disked beam stress and deflection
%=================================
%N=no of pipes at circumference
%do=outer dia of pipe
%di=inner dia of pipe
%d=dia of solid rod at center of the disc
%de=equivalent dia of solid beam
%y=center to center distance of solid rod and pipe
%Ie=Equivalent Moment of Inertia for solid beam
%Is=Momonet of inertia of structural beam
%delta=deflection
%Dc=center to center distance between two beams
%tor=torque applicable at beam edge
%smx=bending stress
%taumax=shear stress maximum
%vonmax=max vonmises stress
%=================================
d=40;
pi=3.14;
do=33.7;
di=27.2;
N=4;
y=53.15;
w=20;
l=1500;
E=210000;
Is=N*pi/64*(do^4-di^4)+N*y^2*pi/4*(do^2-di^2)+pi/64*d^4;
de=(Is/pi*64)^(1/4);
Ie=pi/64*de^4;
delta=5/384*w*l^4/E/Is;
delta
de
Is
%=======================================================
%stress analysis
%=======================================================
tor=w*l*de/2;
Mbx=w*l*l/8;
taumax=tor*de/4/Ie;
smx=Mbx*de/2/Ie;
vonSMX=(smx/2)+sqrt((smx/2)^2+(taumax^2))
```