DESIGN THE MAIN SHAFT OF WIDE WEAVING MACHINE
FOR EQUALLY SIDED REED ADVANCE

تصمــيم الـعمـــــود الرئيـــــى لماكينات النـــــــــيـج

العريضــة للتقدم المتــــاوى لجانـــى مشط ضم اللحمــه

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ملخـص البحـث :

نتيجة لإدارة ماكينات النسيج العريضة « مزدوجة مطواة السداء » من جانب واحد يحدث

إلتواء فى عمود الأدارة الرئيسى الذى يؤثر بدوره على التقدم غير المنتظم للمشط أثناء عملية ضم خيط

اللحمة – ويؤدى هذا بدوره إلى زيادة معدل الشد لخيط السـدى من جهة الأستقبال عنه لجهة الأرسال، فى

البحث الحالى تم حل مشكلة عدم إنتظام تقدم المشط وذلك بتقليل زاوية الألتواء لعمود الإدارة الرئيسى من

طريق التصميم الأمثل للعمود وبطريق التركيـات الميكانيكية المختلفة . كذلك تم إختيار أنسب مادة

وحساب أقل تكاليف لعمود الإدارة .

SUMMARY - As a result of driving wide weaving machines double warp beam from one side, torsion occurs in the main driving shaft, which in turn affects on the non-uniform advance of the reed. during the beat up of the weft yarn. This in turn leads to the increase in the rate of warp yarn tension from the checking side than the picking side., hence the warp yarn in the checking side will be consumed than that in the picking side.

In the present work the problem of non-uniform advance of the reed has been solved by reducing the twisting angle of the main driving shaft, through the optimum design of the shaft or by different mechanical devices, Also the selection of the most suitable material and the minimum cost of the driving shaft

1- INTRODUCTION

In weaving machines the drive of the main shaft is from one side of the machine. On this shaft cams are fixed which are responsible of moving the reed which in turn is responsible for weft beating-up.

In wide weaving machines, such as Sulzer weaving machine torsion occurs in the main shaft because of it's relative length. This leads to early advance of the reed to beat the weft from the driving side (checking side) than from the picking side.

It was found from the measurement of the variation in feeding rate on warp beams, over 100 wide weaving machines of the Sulzer type, that the feeding rate is higher for the warp beam in the checking side than that of the warp beam in the picking side. The reason of this defect is the advance of the reed from the picking side than the checking side, this leads to the increase in the rate of stress of warp threads in the checking side, which leads to the increase in the rate of let-off in this side (for double warp beam) than the other side, consequently the warp threads from the warp beam in the checking side are fed earlier. This will effect the percentage waste of warp threads.

The increase in the rate of stress of warp threads in the checking side ranges between 30% and 50% than the other side, this will result in variation in

the physical and mechanical properties of the produced fabric (1,2,3,4,5,6).

This necessitates the use of differential gears to compensate between the feeding rate and warp tension, because of the irregular advance of the reed during beating-up.

Therefore it is advised to solve the main cause of reed advance from one side earlier than the other side. The present work is an attempt to design a driving shaft that would have a minimum possible torsion angle (or equally torsion angle).

This is achieved by different ways, which are involve theoretical analysis or mechanical devices as follows:-

I   - Theoretical analysis which is applied to find optimum choice for dimensions and materials.

II.1 - The driving is supplied from the middle of the shaft. This mechanical device is achieved by two driving sahafts.

II.ii - The driving is supplied from the middle between every two cams. This device is achieved by three driving shafts.

II.iii- The existing main shaft can be changed to stepped-taper shaft.

NOMENCLATURE

C_f	: Fixed cost.
C_1	: Material cost.
C_2	: Machining cost.
d	: Nominal shaft diameter (mm).
d_1	: Big diameter.
d_2	: Small diameter
G	: Modulus of rigidity or shear modulus.
J	: Polar moment of inertia of an area.
K	: Torsional stiffness.
K_i	: Stress concentration faetor.
L_1	: Length at any point of the shaft.
L	: Total shaft length.
m	: Metre.
n	: Revolution per minute.
N	: Factor of safety. $= 1.3$
N_y	: Yielding faetor of safety.
P.E.	: Energy stored or absorbed or strain Energy in torsion.
R	: Shaft radius.
S_i	: Circumferential shearing strain distance.
S_e	: Fatigue strength of material.
$(St)_y$	: Yield strength of material.

t   : Lagging time.

t_1, t_2, t_3 : Lagging time at cam number one, two and three respectively.

T   : Twisting moment.

T_1, T_2, T_3 : Twisting moment at cams 1,2 and 3 respectively.

w   : Weight density of material.

θ^o : Angle of twist in radiant or degree

θ^o_{min} : Minimum twisting angle

$\theta^o_1, \theta^o_2, \theta^o_3$: Twisting angles at cams 1,2 and 3 respectively.

τ_{may} : Maximum shear stress

2- THEORETICAL

The maximum shearing stress oecurs at outer most fibers of the solid main shaft Fig.(1-a) and its magnitude is determined from the equation:

$$\tau_{max} = (16 \, T \, K_i) / (\pi \, d^3) \qquad \ldots (1)$$

The angle of twist θ for the main shaft (7,8,9) is detcrmined from the equation:-

$$\theta = (2.(P.E)) / T = (L.T) / (G.J) \qquad \ldots (2)$$

Eliminate "T" from equation 2, this introduces the geometrical parameters L,d and material parameter G. Thus equation 2 beeomes as follows with L minimum and d maximum giving minimum twisting angle:-

$$\theta_{min} = \sqrt{[((P_1 E) \times 64 \times L_{min} / (\pi \times G \times d^4))]} \qquad \ldots (3)$$

The next step wlll be to determine the variation of θ^o/m versus d (Figs. 2and3) for all feaslble existing designs for different materials (e.g. steeis and Titanium alloys).

Third step is the optimum design procedure and eliminating the parameter d using the stress equations 1and2 as follows:-

$$\theta_{min} = [\, 0.91 \, (\frac{L_{min}}{(P.E)^{0.33}}) \, (\frac{\tau_{max}^{\frac{4}{3}}}{G}) \frac{1}{K_i^{\frac{4}{3}}} \,]^{1.5} \qquad \ldots (4)$$

From equation (4) it is obvious that the value of combined shock and fatigue factor for load suddenly applied (heavy shock) K_i should be as big as possible for minimizing θ say K_i is equal 1.8. Also from equation 4 it is evident that τ_{max} should be at its lower limit which is given by the following equation:-

$$\tau_{max} \leqq S_e / [\, (1 + (\frac{S_e}{(S_t)_y}) \, N \,] \qquad \ldots (5)$$

The final step will be to look at the variation of $\theta°/m$ versus Υ'_{max} (Figs.4 and 5) typified by specific design range with the limit values on degree/m carried over from Figs.2 and 3. This is to define the optimum choice for θ_{min}/m with d and shear stress for real example in Figs.2-5 the optimum material for minimum twisting angle will be 1020, 1095, 2340, 4130 AISI steels and the titanium alloy would be the second choice. For the AISI steels, the minimum twisting angle obtained by the main shaft for wide range of energy would be 0.16 degree/m as governed by the restriction on value of energy equal to 1000 N.cm.

On the other hand, if the titanium alloy, should be chosen as next best alternative, the design would result in a minimum twisting angle of 0.234 degree/m obtained by main shaft of sulzer machine beat up system.

The previous choice of optimum condition, to find minimum twisting angle will solve partly the lagging time of cams (Fig.1), where in this figure the input torque is supplied to the main shaft at cam 3 from one side only, this device provides unequal torque distribution over the cams as shown in Fig.(1-c), which creates lagging time. This real existing problem can be solved by minimizing lagging time to approaches zero. This can be found by changing this device of Fig.1 to other, new devices.

2.1 LAGGING TIME

If the shaft is homogenous and has a uniform cross section the circumferential shearing strain distance S_i (Figure 1-b) is equal to:-

$$S_1 = \frac{L_i}{L} R \; \theta = (L_i \cdot \Upsilon_{max}) / G \qquad \qquad \ldots (6.1)$$

Where Lagging Distance $= S - S_i$ $\qquad \qquad \ldots (6.2)$

and

Lagging Time $= [(S - S_i) \; 60] / (\Pi.d.n)$ $\qquad \qquad \ldots (6.3)$

For present problem, (Fig.1) let us calculate the lagging time at three cams to find the differences in time, this is for the following example P.E= 12000 N.cm, d=120 mm, n=600 R.P.M., L = 440 cm.

For previous data find the twisting angle θ from Fig.2, which gives the following lagging time at three cams $t_1 = 78\times10^{-5}$ sec. , $t_2 = 41\times10^{-5}$ sec, $t_3 = 3.810^{-5}$ sec

The lagging time differences between first cam and third cam is equal to 74.2×10^{-5} sec. Due to lagging time differences, the yarns in the checking side of yarn beat-up will be strained more than the picking side. As aresult of this the strained side will be fed by a rate more than that of the other side. Hence the warp yarn will finish early from the warp beam in the strained side before the other side. Once one of the warp beams finishes they are replaced completely, and the yarn left is considered waste. Which is uneconomical. Therefore there must be another solution which gives $\theta_1 = \theta_2 = \theta_3$. This gives lagging time equal to zero. In the same time the main shaft effective length of power supply must be reduced. The previous two factors are achieved by mechanical devices.

3- MECHANICAL DEVICES

By establishing firstly a new design which provides equal power supply for all the three cams;this is can be don, by providing the power at mid-span for every two cams and manufacturing the shaft as stepped shaft or taper shaft, where the taper shaft twisting angle is equal to (10) :-

$$\theta = \frac{32\,T\,L}{3\,\pi\,G}\,\left(\frac{d_1^2 + d_1\,d_2 + d_2^2}{d_1^3 \cdot d_2^3}\right) \qquad \ldots (7)$$

This new design is shown in Fig.6. For this case and applying the last example where the energy P.E = 12000 N.cm is divided equallY to 6000 N.cm at the midspan of the cams.

Hence the medium cam will take 6000 N.cm while the right and left cams will take 3000 N.cm and 3000 N.cm respectively. Therefore from Fig.2 the stepped diameters can be fifound.

Big diameter   = 102 mm
Small diameter =   83 mm

The main function of this device is $\theta_1^o = \theta_2^o = \theta_3^o$ which provides lagging time approaches zero. In the mean time, the shaft diameter has been reduced from 120 mm to 102 mm and 83 mm. Inturn provids minimum material cost. Also the main shaft length has been divided to four effective lengths, this is very important factor to achieve optimum design eq. 3.4. Also this factor provides the chance for increasing the effective length (machine width) without fear of beating up lagging. This device has never been proposed before.

Secondly, if there is space restriction, the three driving shafts of Fig.6 could be reduced to two shafts where the drive is supplied at both sides of medium cam Fig.7. In the mean time the shaft is stepped. In this case if one applied the last example, the energy which is 12000 N.cm will be divided equally at both right and left of medium cam. In this case, the loss of power supply of the right spur gear will be very small compared with the case of one shaft system (Fig.1). The big and small diameters can be defined from Fig.2 and equal to 120 mm and 102 mm respectively. The function of this device gives small place restriction and equal twisting angles i..e $\theta_1^o = \theta_2^o = \theta_3^o$. This will lead to lagging time approaches zero and gives good mechanical and physical properties of the produced fabric. Also the diameter has been reduced from 120 mm to 102 mm. Inturn provides minimum cost of material. In addition the shaft length has been divided into two effective lengthes. This factor is very important to achieve optimum design from Equs.3 and 4.

Thirdly, If there is no space the existing design of one cylindrical driving shaft could be modifies to stepped taper shaft. Fig. 8. Assume areal example (2) where the percentage of power distribution over the three cams are 0.25 : 0.5 : 0.75 of supply power, these are the torques ratio of T_1 to T_2 to T_3 respectively. Also assume big diameter = 14 cm. The power supply is 6kw and the speed 500 R.P.M. From Equ. 2 assume $0^o = \theta_1^o = \theta_2^o = \theta_3^o$ hence the taper diameters are, 1.9 cm, 4.1 cm and 5.3 cm respectively. The important function of this device gives that the existing sulzer loom frame will not change, hence the modification should be easy in application, while the machining cost of this shaft is very high. Also in this device the reduction of shaft is very high. Also in this device the reduction of shaft diameter gives equal twisting angle $\theta_1^o = \theta_2^o = \theta_3^o$.

But the big difference between the diameters sizes influences fatigue strength (11) inturn endurance limit. Due to this, failure will occur in one of the diameters before the other diameter of the main shaft

4- OPTIMUM DESIGN FOR MINIMUM COST

Assume that the shape of the main shaft has been decided upon from basic functional considerations of the sulzer weaving machine that shape is illustrated in Fig. 1-a.

The total cost equation is as follows [9]:-

$$C_T = C_F + C_1 \ W \ (\frac{\pi d^4}{4}) \ L \ + C_2 \ L \ d \qquad \ldots (8)$$

The next step in the method of optimum design will be to express the following equations:

Torque gradient $= K = \pi d^4 \ G/32 \ L$ $\ldots (9)$

and

Maximum shear stress $= \tau_{max} = K_i \ (16T \ / \ \pi d^3)$ $\ldots (1)$

where

$$\tau_{max} \leq \frac{(S_t)_y}{2 \ N_y} \qquad \ldots (10)$$

Using the last three equations in Equ.8

Primary design equation is obtained:-

$$C_T = C_F + 8 \ (\frac{T^2 \ N_y^2}{K}) \ [\frac{C_1 \ W G}{(S_t)^2_y}] \ K_i^2$$
$$+ \ 4.6 \ (\frac{T^{5/3} \ N_y^{5/3}}{K}) \ (\frac{C_2 \cdot G}{(St)_y^{5/3}}) \ K_i^{5/3} \qquad \ldots (11)$$

From Equ.11 it would be wise to consider materials having the lowest values for the two terms:-

$$[\ G \ W \ C_1 \ / \ (\ St \ )^2_y \ ]$$

and

$$[\ C_2 \ G \ / \ (St)_y^{5/3}]$$

Finally, the lowest possible total cost C_T would be calculated from either Equs. 8 or 11.

5- CONCLUSION

For the AISI steels there is optimum minimum twisting angle obtained by main shaft for wide range of kinetic energy would be 0.16 degree/m as governed

by the restriction on value of energy 1000 N.cm. On the other hand, if titanium alloy should be chosen as the next best alternative, the design would result in a minimom twisting angle of 0.23 degree/m felt by main shaft of sulzer weaving machine beat up system.

The optimum choice of twisting angle, diameter and material can be found from Figs.2,3,4 and 5.

The last choice of optimum condition will be minimizing the differences of lagging time only for the existing main shaft. The complete optimum solotion uf making lagging time equal to zero is the three driving shaft device (Fig.6).

Three driving shaft device gives zero lagging time, if there is place restriction, the other two devices should be used (Figs.7and8).

All three devices give equal twisting angle at the three cams of the main shaft. Inturn the warp yarn will equally finish from the warp beams at both sides at the same time without waste of warp threads, which is economical.

Three driving shaft device provides the chance for increasing loom width without fear of beating up lagging. This device has never been proposed before.

The final optimum design of minimum twisting angle, minimum length, best diameter, good material can be found from Equs. 3 and 4 parallel with device of three driving shaft.

The minimum lowest possible total production cost of the main shaft has been illustrated from either Equs.8 or 11.

6- ACKNOWLE OGMENT

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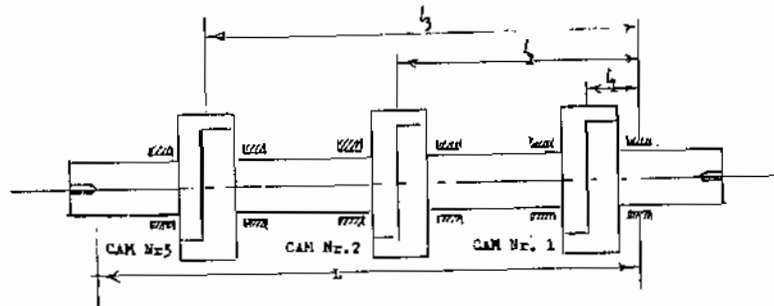
(a)One Driving Shaft
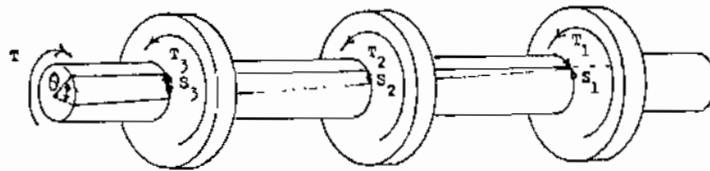
(b) Input – Output Torques(T –T ,T₂, T₃)
and circumferential Shearing Strain
Distances (S₁, S₂ ,S₃)
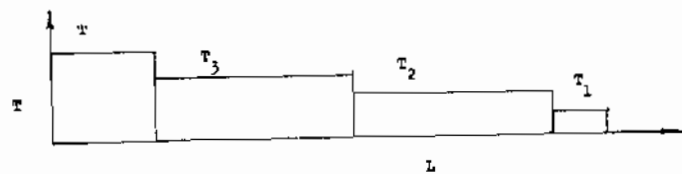
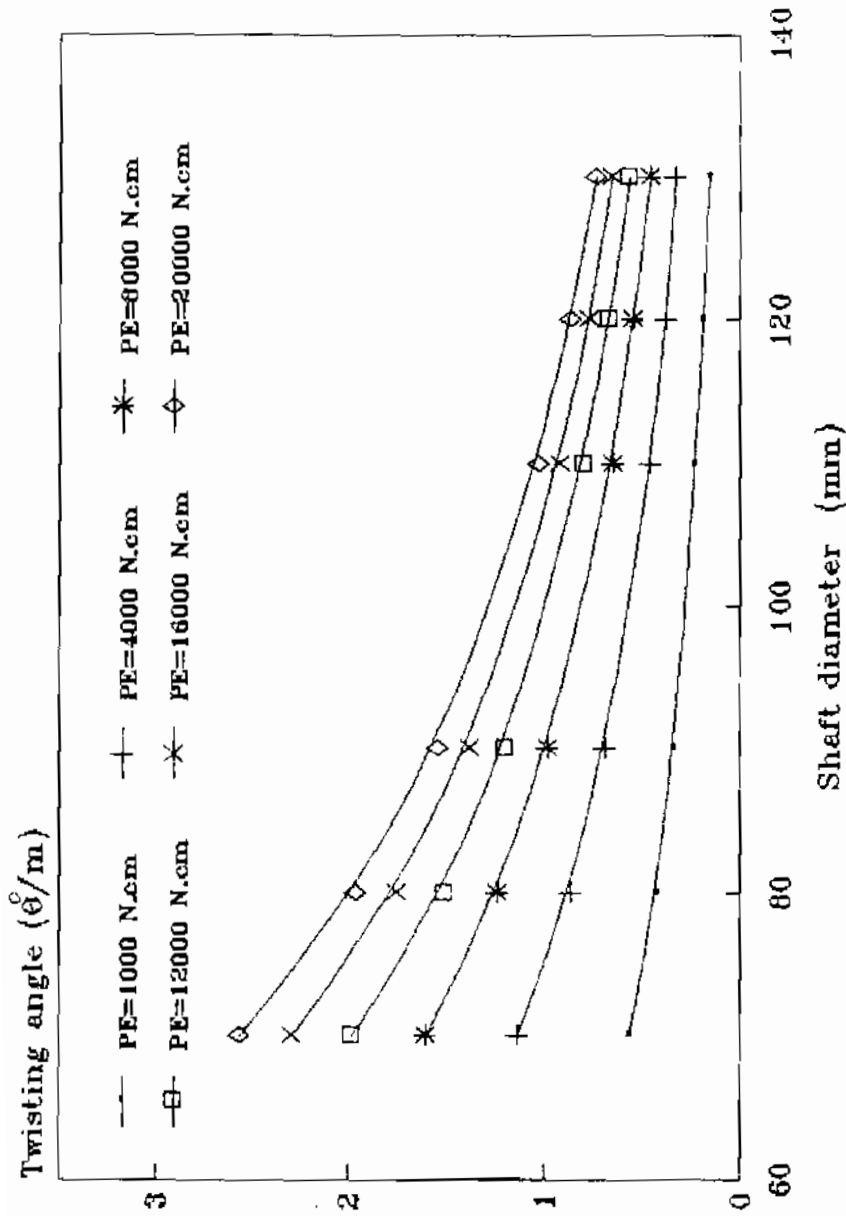(c) Torque Distribution On Cams

Figure (1) Main Shaft of Sulzer Loom

Fig.(2):Twisting angle $(\overset{o}{\theta}/m)$ versus
. shaft diameter (mm) for steel
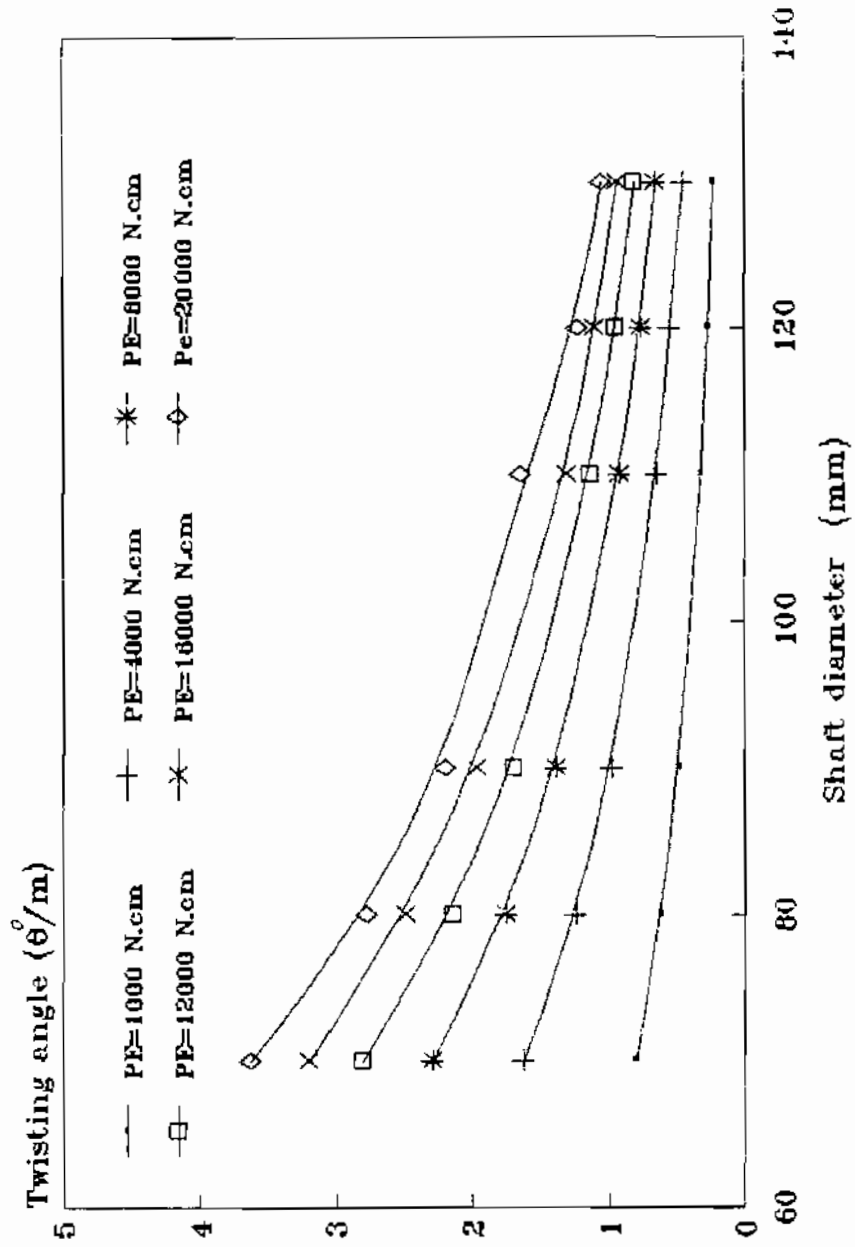
Fig.(3):Twisting angle (θ^o/m) versus , shaft diameter (mm) for titanium alloy

Twisting angle ($\overset{o}{\theta}/m$)

PE=1000 N.cm   PE=4000 N.cm   PE=8000 N.cm

PE=12000 N.cm   PE=16000 N.cm   Pe=20000 N.cm

Max. shear stress x 10~3 (N/cm~2)
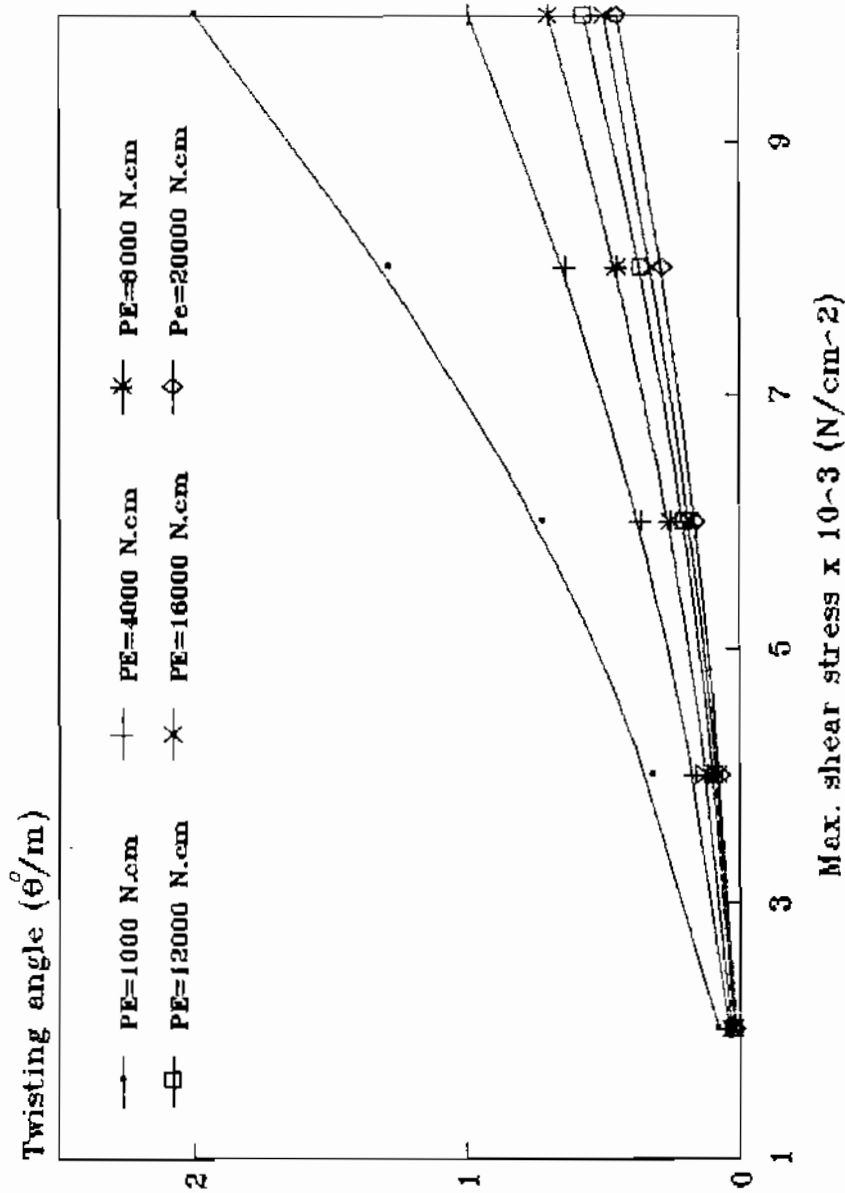
Fig.(4):Twisting angle ($\overset{o}{\theta}/m$) versus
    maximum shear stress for steel
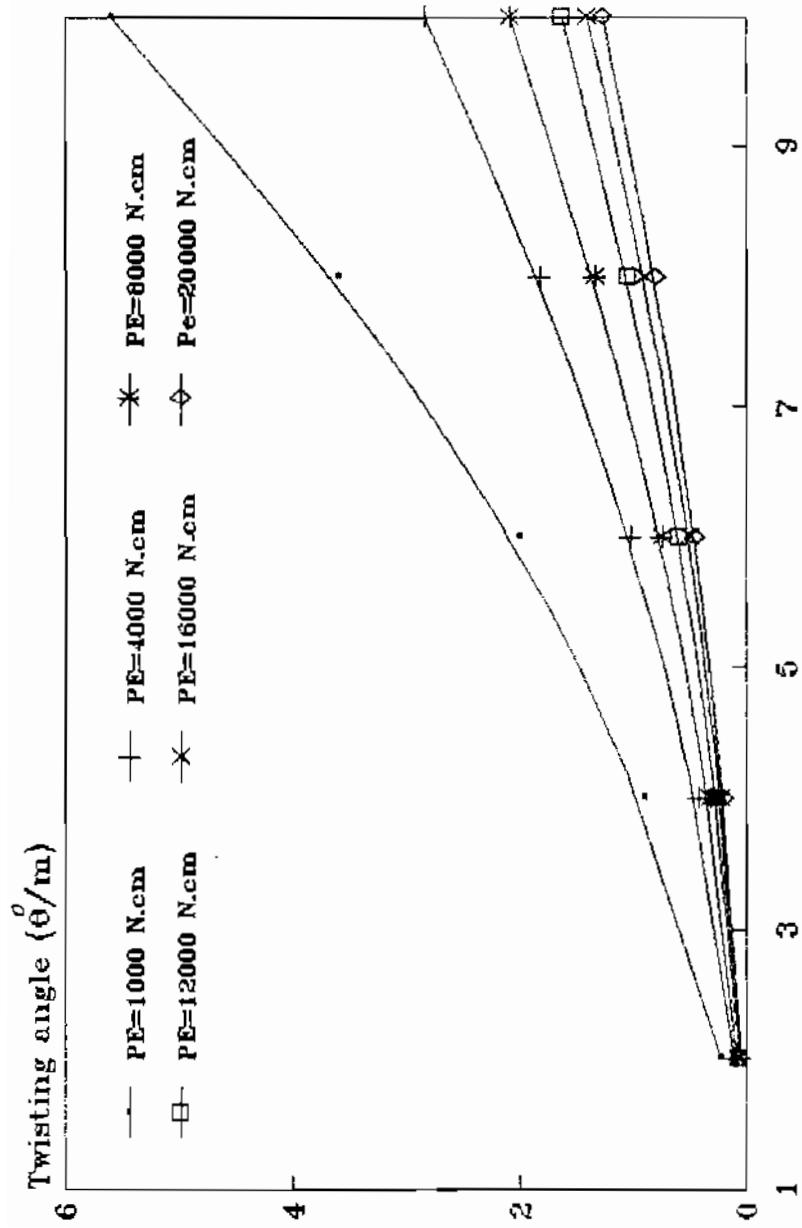
Fig.(5):Twisting angle (θ^o/m) versus
. max.shear stress for titanium alloy
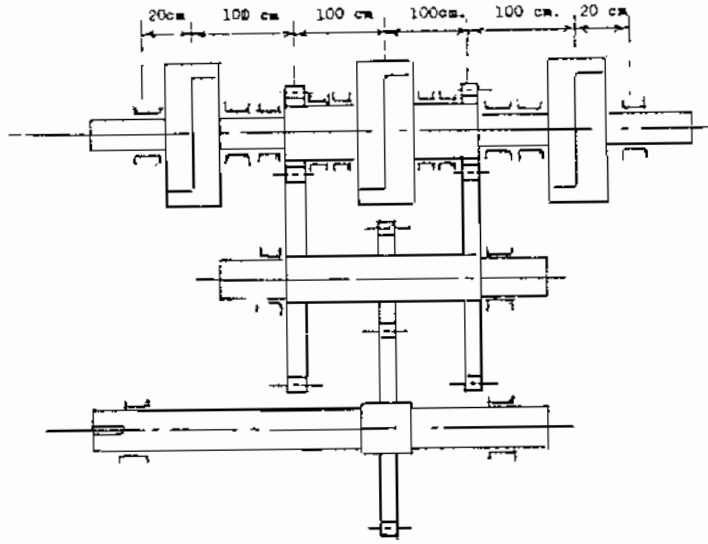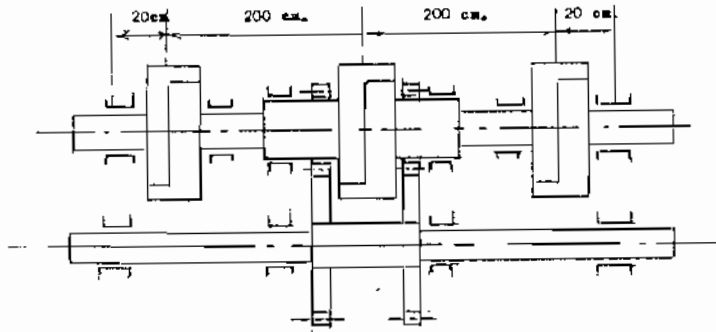
Figure (6) Device of Three Driving Shafts.
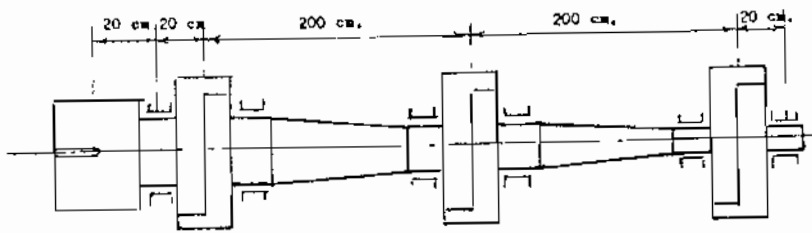

Figure (7) Device of Two Driving Shafts

Figure(8) Device of Stepped-Taper One Driving Shaft.