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Design and Development of Coupling for Unequal Torque System

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Abstract: *The problems occurs at the steel industries, as the overall subject of this paper is the frequent failure of output shaft of gear motor at the pay-off reel and design and development of new type of coupling which will operates according to the situation of the nature of the rotation of pay-off reel rotation and the nature of the rotation of the 4HI cold rolling mill and which will perform efficient working under maximum torque condition without fail. As the mechanism used in the coupling is a ratchet pawl mechanism which work is to provide only one directional rotation and restrict the other. Also the use of spring loaded inserts to, which when reverse motion required, mechanism to engage the coupling hence the newly designed coupling is best suitable as per the nature of the rotation of the pay-off and 4HI cold rolling mill.*
Keywords: *Pay-off reel, 4HI cold rolling mill, Gear motor, Coupling, Ratchet and pawl mechanism.*

I. Introduction

The gear motor is used in cold rolling mill to give motion to the Pay-off reel which is movable with automatic guide table also un-wrapping the coil and the beginning of the process. Pay-off is the standby unit to reduce feeding time of the coil to the cold rolling mill. A cold rolling mill, also known as a reduction mill or mill, has a common construction independent of the specific type of rolling being performed. Cold rolling occurs with the metal below its recrystallization temperature (usually at room temperature), which increases the strength via strain hardening up to 20%. It also improves the surface finish and holds tighter tolerances. Commonly cold-rolled products include sheets, strips, bars, and rods, these products are usually smaller than the same products that are hot rolled. Because of the smaller size of the work pieces and their greater strength, as compared to hot rolled stock, four-high or cluster mills are used. Cold rolling cannot reduce the thickness of a work piece as much as hot rolling in a single pass. Cold-rolled sheets and strips come in various conditions: full-hard, half-hard, quarter-hard, and skin-rolled. Full-hard rolling reduces the thickness by 50%, while the others involve less of a reduction. Skin-rolling, also known as a skin-pass, involves the least amount of reduction: 0.5-1%. It is used to produce a smooth surface, a uniform thickness, and reduce the yield point phenomenon. It is also used to break up the spangles in galvanized steel. Skin-rolled stock is usually used in subsequent cold-working processes where good ductility is required. Other shapes can be cold-rolled if the cross-section is relatively uniform and the transverse dimension is relatively small. Cold rolling shapes requires a series of shaping operations, usually along the lines of sizing, breakdown, roughing, semi-roughing, semi-finishing, and finishing. If processed by a blacksmith, the smoother, more consistent, and lower levels of carbon encapsulated in the steel makes it easier to process, but at the cost of being more expensive.

II. Force analysis of shaft of gear motor

The shaft of gear motor goes failure at the weakest section due to overloading, a tangential load act by the metal sheet i.e. torque of cold rolling mill that comes over the Pay-off reel up to 30000 Nm, self-torque 1281 Nm, and bending loading. Also it was found that casing of gear motor goes failure, i.e. it broke. The Gear motor is of 5.5KW and 41 rpm and that of Cold rolling mill driving motor is of 1000KW and initial rpm of motor is 200mpm i.e. meter per minute. The shaft of gear motor goes failure in duration of 1-2 months and each motor cost up to 60,000 Indian rupees. Due to periodic failure of gear motor, company changed the gear motor with separate gear box and motor. One of the reasons of failure is lack of concentration of worker while working.

In Continuous process industry many types of roll are used for various applications; however process rolls used are classified in to two as working & conveying. Working rolls are used for changing the product being process. This included cold rolling mill, hot rolling mill rolls used for reduction of sheet thickness & shape. Printing press rolls used for transferring pattern to product. Corrugation rolls used for changing the profile of sheet. Conveying rolls are mainly having application of transferring product from one place to other. This type rolls are smaller diameter and are generally non driven to large diameter used for transferring sheets, paper, and textile products in continuous process industry. The gear motor is used in pay-off reel to wound the metal coil, that motor is connected with Pinch roll of pay-off reel by claw coupling. One lever provided for engage and dis-engage of shaft of gear motor. Pinch rolls each of diameter 250mm. Shaft of gear motor is of SAE4140 i.e. 197 BHN hard. Power of gear motor is 5.5KW and 41rpm and that shaft is a step shaft largest diameter of 60mm and end diameter 40mm.

A. Force Acting On The Shaft Of Gear Motor

Stress Raiser in shaft due to various parameters that are acting on shaft of Gear motor is;

a. Shear stresses due to torque transmission by motor of 5.5KW

b. Bending stresses due to forces acting by,
   i) Weight of Pinch roll
   ii) Weight of sheet
   iii) Weight of shaft itself
   iv) Sheet tension on forward direction when main system starts with full

c. Stresses due to combined torsional and bending load

B. Stress Analysis Of Shaft Of Gear Motor

The stress analysis for the shaft of gear motor is carried out for static analysis in the following ways,
1) Analytical Analysis and
2) Finite Element Analysis.

For the analysis the technical specifications are shown in table 4.1 in which the total length and other parameters are mentioned and also seen in figure 4.1. For the static analysis the shaft is considered like a horizontal cantilever shaft fixed at its one end. The forces acting over the bridle roll has been designed and used for analysis of shaft. The strip tension on forward & backward is treated as belt tension. The static stress analysis for the shaft of gear motor is as follows.

Table I

Available Data		
Motor Specification	Gear motor	Motor Of Cold Rolling Mill
Motor Make	PBL Motor	Kirloskar
Output shaft Diameter in mm	60 mm	200mm
RPM	41 rpm	382rpm (Initial)
Power in KW	5.5 KW	1200KW
Torque in N-m	1281	30000

Table II

		Pay-Off 4 HI	Cold Rolling Mill
Diameter	Pinch Roll	250mm	
	Work Roll		250mm-410mm
	Back up Roll		1150mm-1250mm
Length	Pinch Roll	1550mm	
	Work Roll		1550mm
	Back up Roll		1550mm
Weight	Pinch Roll	500kg	
	Work Roll		1000-3500 kg
	Back up Roll		18000 -25000 kg
No. of	Pinch Roll	2	
	Work Roll		2
	Back up Roll		2
Load/ Pressure	Acted by each Cylinder	80kg	750000kg
	No. of Cylinder	2	2

C. Static Analysis Of Shaft Of Gear Motor For Sudden Failure

I. Design Torque of shaft of Gear motor

Power of Gear motor = 7.5 HP =5.5 KW
Rpm of gear motor = 41 rpm
Therefore Power of shaft, $P= (2\pi NT)/60$
T = 2751 N-m

Kl – Load Factor

II. Design Torque of motor of Cold rolling Mill

While the pay-off en-gauge with main rolling mill, as pay-off is used for providing the coil to the main rolling mill system , and at instance that pay-off get dis-en-gauged ,while during the dis-en-gauging the pay - off whole torque of mail rolling mill comes on the pay of and shaft of gear motor ,that drives the pay-off system goes failure. So, Power of motor that drives the main rolling mill P=1200KW

Initial rotation of rolls of main rolling mill is up to 300mpm

Rpm of motor = 300000/1288 = 232 rpm

Power of motor $P = (2\pi NT)/60Kl$

$\mathbf{T_2 = 108664}$ **N-m**

III. Torque Transmitted from main CRM system to the Roll of Pay-off System through metal Sheet in the form of tangential Force

$Ft=T/R1$ $\quad Ft = 530068$ N

As that much amount of force exerted by sheet on the Roll of Pay-off System

Hence Total Amount of Torque on the Roll of Pay-off is,

$Ft1 = T2/R2$ $\quad T2 = 15902$N-m

Material and their properties of shaft of gear motor

As considering Factor of Safety i.e. F.S = 2

Therefore Tensile /Bending stresses in shaft

$\sigma_b = \sigma_t = S_{yc}/FS = 340$Mpa

And Shear stresses in shaft

$\tau = S_{ys}/FS = 187.5$ Mpa

For Maximum shear force and bending moment

$R_a + R_b = 11438$

Taking moment @ A

$R_b *1.55 = 58.86*1.88 + 11438 *1.55/2$

l – length of span = 1.55m

$R_b = 5618$ N

$R_a = 11438 - 5618 = 5820$ N

Equivalent Twisting Moment

$T_e = \sqrt{(M^2+T^2)}$

$T_e = 16470$ N-m

Equivalent Bending Moment

$Me = ½ (M+\sqrt{(M^2+T^2)})$

$Me = 10380$ N-m

Stresses induced in the shaft due to Bending Moment

$\sigma_b = 489$ N/mm2 >Allowable bending stresses=340 N/mm2

Stresses induced in the shaft due to Twisting Moment

$\tau = 388.33$ N/mm^2 >Allowable Shear Stresses=187.5 N/mm^2

Stress	Allowable stress	Analytical	Software
Max. Shear stress	187.5 Mpa	388.33Mpa	472 Mpa
Max. Bending stress	340 Mpa	489 Mpa	512 Mpa

Table III Comparison between allowable stresses and analytical stresses

From the table III the maximum shear stress for impact loading is more than allowable shear stress calculated by taking factor of safety 2 as per design data book. This will lead to shaft failure because of impact loading.

III. Validation Using Finite Element Method

A. Axis Symmetric Loading

Problems involving three dimensional axis symmetric solids subjected to axis symmetric loading reduce to two dimensional problems. Because of total symmetry about z axis all deformations and stresses are independent of rotational angle. Thus, the problem needs to be looked at as two-dimensional problem.

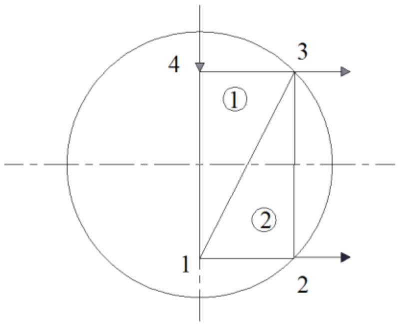

Fig 1. Represent Shaft loading

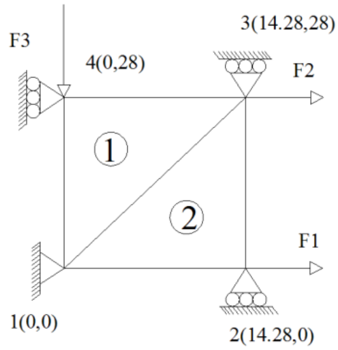

Fig.2. Elements of Shaft

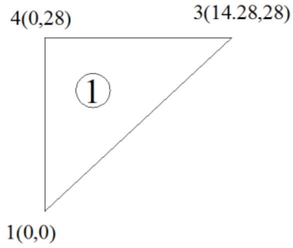

Fig.3. Element 1

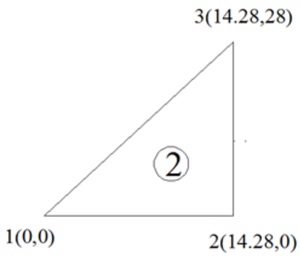

Fig.4. Element 2

From Properties of material, For SAE4140
E= 2*105 N/mm2 and
Coefficient of Friction for Steel to steel contact i.e. μ= 0.7

$$D = \frac{E(1-\mu)}{(1+\mu)(1-2\mu)}\begin{bmatrix} 1 & \frac{\mu}{1-\mu} & 0 & \frac{\mu}{1-\pi} \\ \frac{\mu}{1-\mu} & 1 & 0 & \frac{\mu}{1-\pi} \\ 0 & 0 & \frac{1-2\mu}{2(1-\mu)} & 0 \\ \frac{\mu}{1-\mu} & \frac{\mu}{1-\mu} & 0 & 1 \end{bmatrix}$$

For both Element
$J = z_{23}r_{13} - z_{13}r_{23} = 399.84$
$= 399.84$
$A = \frac{1}{2}|J| = 199.92\text{mm}^2$

$$B1 = \begin{bmatrix} \frac{z_{23}}{|J|} & 0 & \frac{z_{31}}{|J|} & 0 & \frac{z_{12}}{|J|} & 0 \\ 0 & \frac{r_{32}}{|J|} & 0 & \frac{r_{13}}{|J|} & 0 & \frac{r_{21}}{|J|} \\ \frac{r_{32}}{|J|} & \frac{z_{23}}{|J|} & \frac{r_{32}}{|J|} & \frac{z_{31}}{|J|} & \frac{r_{32}}{|J|} & \frac{z_{12}}{|J|} \\ \frac{N_1}{r} & 0 & \frac{N_2}{r} & 0 & \frac{N_3}{r} & 0 \end{bmatrix}$$

The Stiffness Matrix are obtained by Finding,
K =2πr*A*BT*D*B For each element
For Element 1, K1 = 2πr*A*B1T*D*B1
For Element 2, K2 = 2πr*A*B2T*D*B2
Now, Applying Boundary condition
F= Ku
Where F1= F2= 250248 N
F3 = 113796 N
$U_{2x} = 43.51$mm
$U_{3x} = 68.04$mm
$U_{4y} = 39.15$mm
To find the stresses in the Element Considering each element

$$\sigma = D\,B\,U$$

For Element 1

$$\sigma = D\,B_1\,U$$

For Element 2

σ =D B_2 U

Hence, it can be clearly seen that the maximum shear stresses are obtained above the given value and i.e. 402 .89 Mpa

IV. Design Of Coupling

The design of coupling is on the basis torque transmission, using ratchet and Pawl mechanism and also using the slider mechanism to restrict the relative motion between. A ratchet mechanism is based on a gear wheel and a pawl that follows as the wheel turns. When the gear is moving in one direction, the pawl slides up and over the gear teeth, sending the pawl into the notch before the next tooth. The pawl is then jammed against the depression between the gear teeth, preventing any backwards motion. Ratchet mechanisms are very useful devices for allowing linear or rotary motion in only one direction.

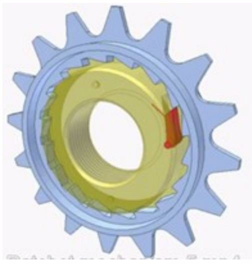

Fig. 8.1 Ratchet and Pawl used in Freewheel of Bicycle

Design of Ratchet and Pawl

Assuming the no. of teeth of ratchet (z)

Z = 15

The Pitch angle $\emptyset = \frac{360}{z} = \frac{360}{15}$

$\emptyset = 24°$

Circumference Force on the Coupling (Fc)

Fc = Tc / r1

Tc –Torque on coupling =1281 N-m

R1 –Radius of Ratchet =42.5mm

Fc =1281 / 0.0425

Fc =30141 N

Bending Stresses on Ratchet

$Fb = \sigma_b * b * e^2/(6 * b * SF)$

Where:

σ_b =Bending Stresses

b = Face Width (mm)

e = Root length (mm)

e =Depth of teeth (h) *tan (60-(360/z))

h =Depth of teeth (mm)

SF = Factor of Safety

r_f =Tooth root radius (m)

r_f =(Outside Dia.(D) –(2h)) /(2000)

From Catalog of SRT- SBT (Page No. 3)

a) Ratchet

No. of Teeth = 15

Pitch = 12.56 mm

Outside Data =200 mm

Face width = 25mm

Depth of Teeth = 7.4

b) Pawl (Size of Pawl is based on Ratchet)

Diameter of Ratchet rod = 8 mm

Face width of Pawl = 25 mm

Pitch of Ratchet =12.56 mm

Material Properties (Catalog SRT-SBT) AISI 1046

Taking Factor of Safety FS =2

Allowable Stresses are

T = 310 Mpa
σ_b = 257.5 Mpa

Calculation for Bending Stresses
a) e = Root length = depth of teeth (h)× tan (60 -360/z)
e = 6.24*(tan (60-(360/15))
e = 4.54 (depth of teeth = 6.24)
b) rf =teeth root radius =
= (Outside Dia. (D) – 2h)/2000
= (180-(2*6.24)/2000)
= 0.08376
Fb = 1772 N

Torque required rotating the Roll of Pay-off System
T = Fb × rf
T =148.42 N-m < Torque of Gear motor i.e. =1281 N-m
Bending stress calculation for Ratchet and Pawl
As Ratchet and Pawl is of same Material so,
For Ratchet
Bending Force (Fb) =σ b × A
A – Area of Bending =Area of Contact of Pawl and ratchet
A = Face width (b) × surface of contact (h)
A =75 mm2
Bending Stresses (σb) = Fb /A = 23.62 MPa < Allowable Shear Stress = 257MPa
Bending Stress on Powel
Bending Force (Fb) =σ b × A
A – Area of Bending =Area of Contact of Pawl and ratchet
A = Face width (b) × surface of contact (h)
Surface contact of Pawl with Ratchet (h) = 2mm (Circular surface has line contact)
A= 50 mm^2
Bending Stresses (σb) = Fb /A = 35.42 MPa < Allowable Shear Stress = 257MPa

Shear Stresses on the Inserts of Coupling
Shear Force on the shaft of coupling (Ft) = τ × A

$$F_t = \frac{T}{r} = \frac{1281}{0.068} = 21350 \text{ N}$$

r- radius of Insert = 15 mm
Shear Stresses τ = Ft /A = 21350/ ((π/4)*152)
τ = 120 Mpa < allowable Stresses of Material= 310 Mpa
All the Stresses in Insert of Coupling, Ratchet and Pawl are less than allowable Stresses Hence, Design of Coupling is Safe.

V. Result & Conclusion

The analytical and FE analysis of shaft of gear motor are compared for all cases. The analytical analysis for the static analysis indicates the stresses increases during the impact loads. The tabular representation of the analytical & Inventor compared with the allowable stresses. From the table for both the cases the maximum shear stress value obtained by Impact loading is greater than the allowable stress which results in sudden failure of shaft.

Table IV

Stresses	Analytical Results	Inventor Results	FEM Results	Allowable limit
Max Shear Stresses (MPa)	388 Mpa	472 Mpa	402.89 Mpa	187.5 Mpa

The FE analysis is also shows the region of high stresses at the step of 60 mm diameter near bearing region from which the failure is known to happen frequently. Hence analysis proves that the shaft is heavily loaded and on some instances like while processing higher gauge material responsible for the extra tractive effort developed on the shaft Further the actual loading condition would have the additional load or shocks due to improper tension, jerk, and vibration and overloading, misalignment of shaft of gear motor, traction variation at exit section etc. This entire factor shall contribute to increase in the load and hence the stress also increased. So, my suggestion related to newly Designed coupling mentioned in the thesis will definitely help to reduce on the stresses on the shaft of motor.

Design of coupling in on the basis of Freewheel of Bicycle whose working is to provide one direction rotation and restrict other, means it rotate freely when disengage and when engage it rotate with shaft of Roll and only Motor torque in action not the Main CRM system Torque. Hence use of newly designed coupling reduces the failure rate of shaft of Gear motor.

Bending Stresses on the teeth of Ratchet and Pawl are analytically found out and all are less than the allowable stresses i. e. Calculated bending Stresses (34.42 Mpa) and allowable stresses are 257 Mpa in both of the component i.e. ratchet and Pawl.

Ratchet gearing may be used to transmit intermittent motion, or its only function may prevent the ratchet wheel from rotating backward. Ratchet gearing of this latter form is commonly used in connection with hoisting mechanisms of various kinds, to prevent the hoisting drum or shaft from rotating in a reverse direction under the action of the load.

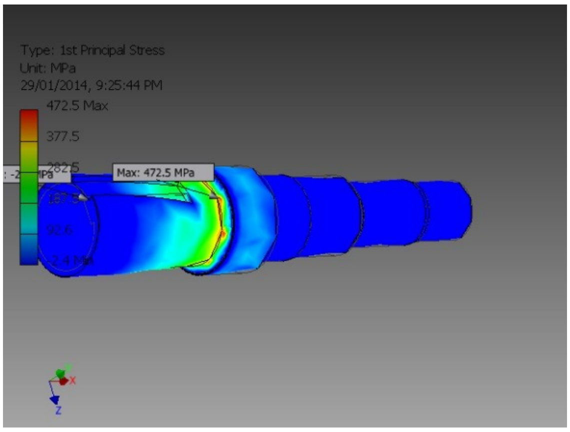

Fig.5. Shear Stresses on shaft of gear motor

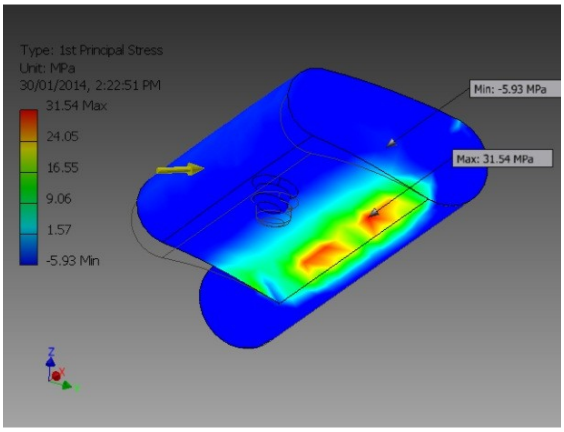

Fig .6. Maximum bending stresses on Pawl

VI. Conclusion

The studies executed in the scope of stress analyses of the shaft of gear motor have revealed the followings:

1. Stress analyses performed by both analytical and Inventor-12.0 software show that maximum stresses are generated at step of 40 mm diameter near bearing zone. This indicates that impact loading near that zone i.e. shaft at least end of step 40mm diameter fails suddenly due to over loading.
2. The component to be preventing failure, Max Shear Stresses should be less than yield strength of material. In this case it is greater than the allowable yield strength of material considering FOS 1 which is generally considered for the mechanical structural components.
3. The static loading for bending stress up to 512 N/mm2 but maximum shear stress is 472N/mm2, which is exceeding the allowable stress of 187.5 N/mm2. This result in sudden failure of drive end shaft.
4. May one of the reason for sudden failure is lack of concentration of worker during operation.
5. As Ratchet gearing may be used to transmit intermittent motion, or its only function may be to prevent the ratchet wheel from rotating backward. Ratchet gearing of this latter form is commonly used in connection with hoisting mechanisms of various kinds, to prevent the hoisting drum or shaft from rotating in a reverse direction under the action of the load.
6. Bending stresses in teeth of ratchet wheel and in pawl are calculated and all the stresses are less than the allowable stresses, ie .calculated stresses are 17.71Mpa and allowable stresses are 257Mpa, hence design of coupling is safe.

VII. Future scope

1. Selecting proper material for shaft having stresses more than 2504Mpa, so shaft may sustain in the maximum loading and impact loading condition.
2. Developed design of shaft according to the stresses induced in the shaft and modify diameter, machined it, use in the motor that in the scrap. So company gets its total value.
3. Select the Proper model of motor having the least diameter of shaft is more than 75mm.
4. Maintain the speed of both Gear motor and motor of cold rolling mill by installing new gear box for cold rolling mill .disengagement of Gear motor Shaft from pinch roll of Pay-OFF goes easy.

5. May include rotation policy for worker those working on cold rolling mill after four hour work.
6. Dynamic analysis of shaft of gear motor should be carried out.
7. Dynamic analysis of Coupling has to be carried which may helpful in vast of application in engineering field.

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