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HERRINGBONE GEAR ANALYSIS AND CALCULATION OF FORCES ACTING ON IT IN HOT ROLLING PROCESS

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Abstract: The steel rolling and forging is the part of metal working processes in which rolling machines are used for constructing long and continuous sections of metal. In this paper we are discussing steel rolling use for hot metal working process. In this paper the concentration is given on analysis and the calculation of rolling load and the forces acting on the gears of steel rolling machine. After that power and torque required is calculated for the rolling load and the gear forces simultaneously which is further used for designing and analysis the gear used in gearbox of hot rolling machine.

Keywords: Hot rolling; Gear forces; Herringbone

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INTRODUCTION

The Steel Rolling Machine is used for making angles, rods, and billets. This machine uses hot rolling process for making the required product. The Steel Rolling Machine is run by the giant 1000HP motor. Motor runs at 737 rpm. The motor is connected to the flywheel with the help of belt drive. The flywheel is further connected to reduction gearbox. The reduction gearbox reduces the speed up to 112 rpm. The reduction gearbox is connected to the transmission gearbox from which the input power is given to the system. The length of the angles is 30 to 36 meters long and 800 to 900 angles are made per shift. In this machine a frequent failure is occurred in the DOUBLE HELICAL (HERINGBONE) GEAR of transmission gear box. In this paper we have suggested the methodology for calculation of forces acting on the gear by using theory of hot rolling. In this paper the analysis is done using solid works. For obtaining calculations we have calculated the motor power available at the input. For actual power requirement we have consider the HOT ROLLING THEORY and calculated the required power and torque for rolling process.

2. GEAR FORCE CALCULATIONS

2.1 Gear Profile Calculation:

- Material of Gear – EN9 (BS 970)
- Helix angle ψ – 26.6 degree
- Pressure angle Ø – 20 degree
- No. of Teeth – 22 teeth 2(D.P.)
- Thickness of teeth – 27 mm

2.2 Power Available at Input:

Motor power (P) = 745.69E3 watt

Efficiency of Gearbox = 85% (for Double Helical Gear)

Power at Input = 633.83E3 watt

Speed (N) = 112 rpm

2.3 Torque of Motor:

(T) = P*60/2Πn

= 633.83e3*60/2π112 Torque = 54.04 KNm

2.4 Force Calculations:

Pitch Circle Dia = 290mm

Pitch Circle Redius = 290/2

= 145mm

=0.145m

Motor force (Fm) = Torque/Distance

= 54041.34/0.145

Fm = 372698.93 N

2.5 Forces acting on Gear:

Compressive force (Fc) = Fm *sin Ø

Ø=Pressure angle

=372698.93*sin(20̊)

Ø=20̊

Fc = 127470.54 N

Tangential force (Ft) = Fm*cos Ø

= 372698.93*cos (20̊)

Ft = 350222.43N

Force acting Normal to the Teeth (Fn)

= Fm*cos ψ ψ= Helix angle

=372698.93*cos(26.5̊) ψ= 26.5̊

Fn = 333541.07N

Note: We are not considering Axial Thrust because of opposite angle of Helix in case of Double Helical Gear.

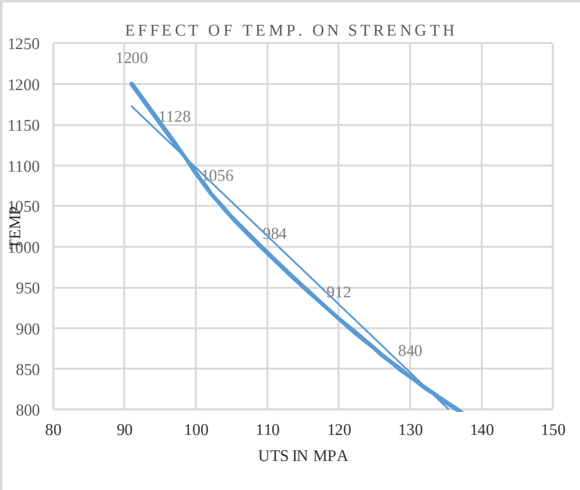
3. ROLLING LOAD CALCULATIONS

3.1 Properties of Material to be used for Hot Rolling:

- Material – A36 Mild Steel
- UTS – 400 Mpa
- Yield Strength – 250 Mpa
- Elongation – 20%
- Carbon – 0.26
- Density – 7800 kg/m^3
- Poisons ratio – 0.26
- Shear Modulus – 79.3 Gpa.

TABLE 1: Effect of Temp on UTS

PASS	*TEMP(̊C)*	*U T S (Mpa)*
1	1200	91
2	1128	97
3	1056	103
4	984	111
5	912	120
6	840	130
7	768	142



Graph 1: Effect of Temp on Strength

3.2 Analysis of Rolling Load (P) using Hot Rolling Theory:

$$P = \sigma o\left[\frac{1}{Q}(e^{Q}-1)b\sqrt{R\Delta h}\right]$$

Where: P = rolling load

σ_o = mean stress between entrance and exit

Q = complex function of reduction in thickness

b = width of material

R = radius of the roller

Δh= difference between input and output thickness

Now,

Mean Thickness: $\bar{h}$

$$\bar{h} = \frac{ho+hf}{2}$$

Where:

ho = input thickness

hf = output thickness

$\bar{h}$ = mean thickness

Complex func^{n} of reduction in thickness: Q

Where: $Q = \mu \frac{Lp}{\bar{h}} = \frac{(\mu \cdot \sqrt{R\Delta h})}{\bar{h}}$

Lp = Projected length of arc of contact

$L_p = \sqrt{R\Delta h}$

μ = coefficient of friction = 0.3

R = radius of roller = 165 mm

Mean stress:

$\sigma_O = (\sigma_{entrance} + \sigma_{exit}) / 2$

Maximum possible reduction in thickness:

$(\Delta h)_{max} = \mu^2 R$

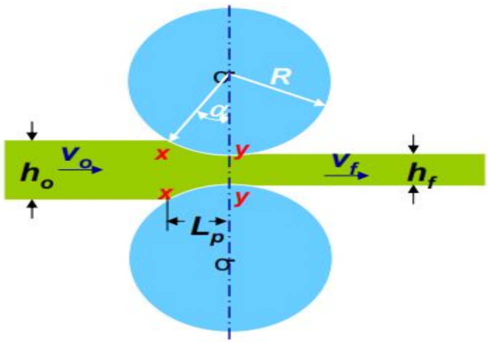


Fig: 1 Input and Output parameters for Rolling 1

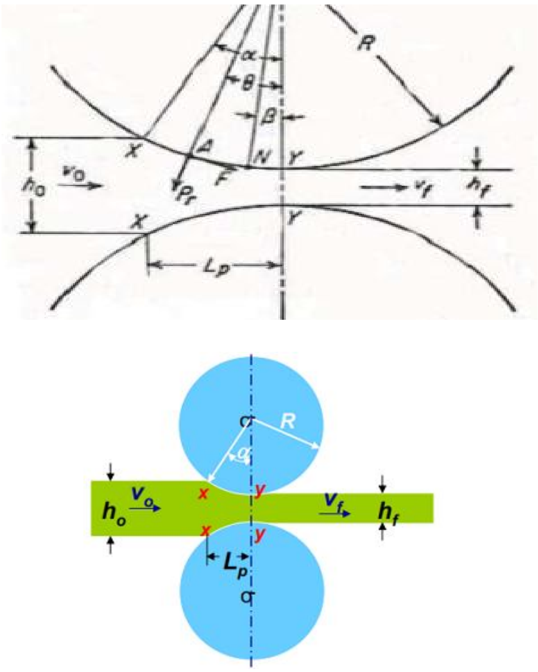


Fig: 2 Input and Output parameters for Rolling 2

TABLE 2: Rolling Load (P)

PASS	Ho mm	Hf mm	Δh mm	$\bar{h}$ Mm	σ_O mpa	σ in mpa	σ out mpa	Q	P (KN)
1	100	85	15	92.5	94	91	97	0.16	507.03
2	85	70	15	77.5	100	97	103	0.19	547.79
3	70	55	15	62.5	107	103	111	0.23	598.39
4	55	40	15	47.5	115.5	111	120	0.31	673.50
5	40	25	15	32.5	125	120	130	0.45	785.21
6	25	10	15	17.5	136	130	142	0.85	1066.14

The ratio of moment of arm to the projected length = λ

$\lambda = a / L_P$ $\lambda = 0.5$ (for hot rolling)

$a = 0.5*49.74$

$a = 24.87$ mm

Where: a - Effective moment arm, Lp - Projected length of arc of contact

3.3 Torque required for Hot Rolling:

$M_T = 2Pa$

Torque = 53.02 kn-m

3.4 Power required for Hot Rolling:

Work done = Power =W

$W = 4\pi aPN$, Power = 621.96Kw

4. ANALYSIS

The values obtain from the above calculation are taken into consideration for the analysis of the existing gear of the transmission gearbox. The analysis is done using solid works simulation express. For this purpose the factor of safety is consider as 3 which is best suitable for the design of herringbone gear. From the results we can say that at the certain areas the factor of safety is below safe limit. This area is indicate by the red colour. The von mises plot shows the maximum value 384.3 Mpa which is below the yield limit of 415 Mpa but since the FOS plot is not in range the gear need to make certain geometry changes in particular areas. Taking the material of higher strength and geometry change in certain area can solve the problem.

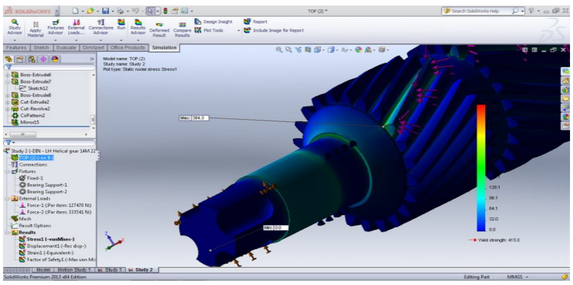


Fig.: 3 Von Mises Stress Plot.

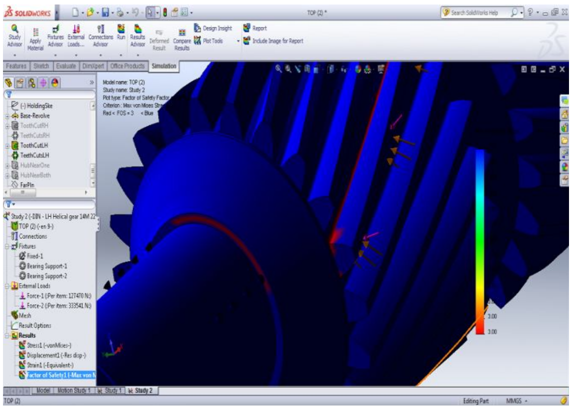


Fig.: 4 FOS plot.

5. CONCLUSION

From above calculations and analysis we can say that the torque obtain for rotating the transmission gearbox of rolling machine and the torque required for the rolling of metal are approximately same. The calculation obtain for the power required to operate the transmission gear box and the power required for the rolling of metal is found out to be approximately same. Using these calculations the analysis is done on the gear and it is found that in certain sections the factor of safety is below safe limit. There for changes in geometry or change in the material is suggested to solve the problem.

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