Design And Modification of Leveller Bar With Rack And Pinion Mechanism

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Abstract- *The Leveller bar with rope and pulley mechanism is used in a steel manufacturing industry for leveling of coal inside a oven of height 7m,during the working condition of the bar some practical problems are occurred and they are observed. In this paper the practical problem caused by rope and pulley mechanism is replaced with the rack and pinion and its components are designed by using the theoretical relations used in Design of machine members and design data book.With the modified dimensions of the leveller bar is designed using CATIA software.*

Keywords- CATIA,Leveller bar,rope and pulley,rack and pinion mechanism

I. INTRODUCTION

The leveller bar is used in Steel manufacturing industry for levelling of coal inside an oven of height 7m, after charging of coal in the oven it is levelled by leveller bar moving to and fro motion in horizontal direction from outside to inside of the oven to make it as a flat surface of the uneven coal.The leveller bar is fitted on a pusher car and the mechanism is connected with an open drive rope and pulley mechanism. These two pulleys are situated at the front end and back end of the bar connected with the rope. The two ropes are connected at the same point with the rope fasteners ,with the rotation of the rope drum, the rope is moved in forward and backward direction. The rope drum is coupled with a gear box and the gear is rotated with a motor which is coupled to a gear shaft.The motor rotates in clockwise and anti-clockwise direction provides to and fro motion of leveller bar. The exact length of leveller bar is 24m. During levelling of coal 15.5m length of bar enters into the oven and temperature inside the oven is around 1100^0 C, the levelling is done.The leveller bar is fixed to pusher car used for pushing of coke time to time.

The name coke oven itself explains that oven which consists of a coke. The coke oven consists of a series of oven named battery. In this coke oven,Coal is converted into coke in a closed chamber with the absence of air,coal blend which is charged into the oven for making metallurgical coke.The process of coke making is basically dumping the coal into oven then levelling of coal then closing of heating the coal blend to a temperature of approximately 1000^0 C-1100^0C in the absence of air inside the oven. This process of conversion of coal to coke is called carbonisation.

1.1.Stagesin coke preparation

Stage 1:During this first stage (or) process the coal is charged into the empty oven from the top side of the coal with the coal charging car From the top of the oven the coal charging dumps the coal inside the oven through the lids provided on the oven top. After the coal is charged into the oven the both doors of the oven on the pushing side and coke discharging side are closed with the door lifting and snatching mechanism. At this stage the oven temperature is at (1000^0C - 1100^0C), when the coal is charged.

Stage 2: During this stage the coal which is dumped in the oven formed into an uneven form. The uneven form of this coal is levelled with a leveller bar. The leveller bar works on from a separate window, with the help of window lifter mechanism, the window lifting mechanism works with hydraulic principle the window is lifted and in order to level the uneven coke leveller bar is used. The leveller bar moves in horizontal to and fro direction .The bar levels the uneven coal and the excessive coal is collected in the bunker. Again the collected coal is recycled for charging. Length of leveller bar is about 24m.Total length of the bar entering into the oven is about 15.5m.

Stage3**:**Duringthis stage the application of leveller bar come into play it is used for levelling the coal inside the oven then the window is closed.The coal is heated in the oven for 1000^0C - 1100^0C and it is heated for 16hours in the absence of oxygen.

Stage4:In this stage the coke side door front door of the oven and back side of oven the door is removed using a guide car then with pusher the coke is pushed into quenching car further the coke is sent for cooling purposes.

1.2. Leveller bar and its construction:

The leveller bar consists of three major portion such as Nose end,middle portion and tail end .The present mechanism using for the leveller bar is rope and pulley.During the working of the present mechanism some of the problems are identified which causing the damage of the rope, pulley and the bar.The identified problems are and divided into minor and major problems.Minor Problems identified during the working condition of leveller bar and the mechanism

1. It is observed that while the leveller bar entering into the oven at the entrance there is problem due to the bar front end part.
2. It is observed that tail portion is slanted edges during full forward, generally the leveller bar gets tilted (sloped)and the front rope is rubbing with box portion of at tail piece.

For minor type of problems, the modification can be done by modifying the leveller bar design in the below figure as shown.

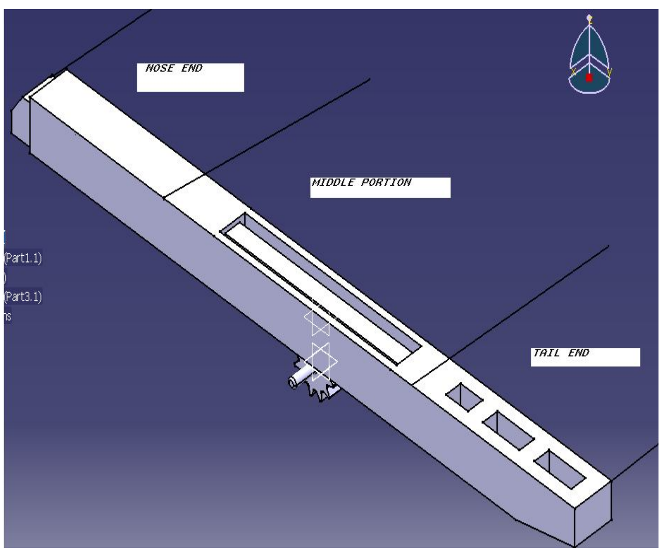

Fig1.Parts of Leveller bar

Major Problems identified during the working condition of leveller bar and the mechanism

1. It is observed that back rope clamps may damage the edges of the back pulley as a result ,failure of leveller bar backward limit switch or may be over back of leveller bar.
2. It is observed that while the bar is moving the lower portion of the bar contact with the rope due to this the rope strength is reduced ,the diameter of the rope reduced so it cannot bare the load due to which the rope may break
3. During the process of bar in levelling of coke in the oven, if the rope is broken the leveller bar cannot move in backward direction causing the bar to be struck inside the oven as a result pushing is stopped and due to the high temperature inside the oven causes damage to the bar.

For major problems like damage of back pulley edges, breakdown of rope on frequent rubbing with the lower surface of the bar cannot be removed but can be minimised,the frequent occurring problems of the rope breakdown during the working condition,the rope drive mechanism is replaced with rack and pinion mechanism to minimize the problems and to increase the life span of the mechanism, so that the problem of rope damage and pulleys can be avoided.

A rack and pinion is a pair of gearswhich convert rotational motion into linear motion.The circular pinion engages teeth on a flat bar, the rack rotational motion applied to the pinion will cause the rack to move to the side, up to the limit of its travel.

1.3.Materials:

The material selected for the pinion is medium carbon steel (c=0.55%) for the high strength gears and pinion,the steel gears or pinions are usually heat treated to combine properly the toughness and tooth hardness.Medium carbon steel C=0.55% normalized, hardened and tempered.

1.4.Mechanical properties:

Brinell hardness number = 220
Tensile strength = 300N/mm^2
Allowable static stress =100 N/mm^2
Endurance limit =480MPa
Modulus of elasticity =202×10^3 N/mm^2
Modulus of Rigidity =78.5×10^3 N/mm

II. DESIGN PROCEDURE FOR MODIFIED LEVELLER BAR MECHANISM PROCEDURE

The problem is practically observed during the working condition and due to some photographic restrictions with the steel manufacturing industry based on the dimensions the leveller bar is designed using CATIA software.

The actual rpm of motor is 1400 rpm with 60kw using speed reduction gear box the rpm is reduced to 400 rpm .Our design is based on speed 400rpm with 60kw.During the design required data , values and tables all are choosen from

Machine Design hand book by K. Mahadevan, K. Balaveera Reddy.

Let, N_P = 400 rpm , P = 60kw

Design of gear teeth or tangential tooth load is obtained from the power transmitted and the pitch line velocity by using the following relation:

F_T = $(p/v)\times C_S$

Where ,
F_T = permissible tangential load in newtons
,P = power transmitted in watts,
V = pitch line velocity in m/s.
D = pitch circle diameters.
N_P = speed of pinion , C_S = service factor.

Apply Lewis equation : $F_T = \sigma_a \times C_v \times b \times \pi \times m \times y$

Calculate the dynamic load on the tooth by using Buckingham equation

Where,F_D = $F_T + F_I$ = F_T + [21v + (b×c + F_T)] / [21 v +(b×c+ F_T)$^{(1/2)}$]

k = A factor depending upon the form of the teeth
= 0.107, for 14 $_{1/2}$composite and full depth involute
= 0.111, for 20° full depth involute system.
= 0.115 for 20° stub system

Where,

E_P = Young's modulus for the material of pinion in N/mm^2.
E_G = Young's modulus for the material of gear in N/mm^2
e = Tooth error action in mm.
F_D = Total dynamic load in newtons.
F_T = Steady transmitted load in newtons,
v = Pitch line velocity in *m/s.*
b = Face width of gears in mm, and
C = A deformation or dynamic factor in *N/mm.*

Adeformation factor (C) depends upon the error in action between teeth, the class of cut of gears, the tooth form and the material of the gears
.

Find the static tooth load by using the relation

F_S = $\sigma_e \times b \times \pi \times m \times y$

For safety against breakage F_S should be greater than the $F_{D.}$
Finally the wear tooth load by using the relation,

F_W = $D_P \times b \times Q \times K$

The wear load (F_W) should not be less than the dynamic load (F_D)
As per the above procedure the pinion is designed.

2.1.Design of rack and pinion

From the gear design principles the minimum number of teeth for 20^0 full depth involute system to avoid interference between rack and pinion is 18 .

But to obtain the least velocity for the bar to move the teeth is assumed to 28.So let the number of teeth assumed T_p = 28,

From the Machine design data book the face width of gear in terms of module can be taken from

9.5m ≤ b ≤ 12.5m.where ‘m’ is the module.

So consider face width b= 12m,Under some conditions the space width can be taken from 6m to 20m.

Velocity factor:

C_V = 3.05 / (3.05+V) ,For ordinary cut gears running with a pitch line velocity up to 8m/s (Barth’sformula)

Pitch line velocity

V = $(\pi \times D \times N_P) / 60$
= $(\pi \times m \times T_p \times N_P)/(60\times1000)$
= 0.586m /sec

Here we have to determine module ‘m’ and Torque,from the design data hand book,

Form factor
Y = 0.154 - (0.192 / T_P) for 20^0 involute system
Y = 0.154-(0.192/T_p)
T = number of teeth on the pinion (Assumed teeth),Therefore
Y = 0.154-(0.912 / 28) = 0.154 - 0.032
= 0.122

Consider the formula,

$P = (2 \times \pi \times N_P \times T) / 60.$

$T = (P \times 60) / (2 \times \pi \times N_P)$

$= (60 \times 60 \times 1000) / (2 \times \pi \times 400)$

T = 1432.39N-m, Where,

T = Torque in N-m,

P = power in Kw

Np = speed in rpm

2.2. Design stress calculation

Design stress =(endurance limit stress) / (factor of safety) $= (\sigma_e / Fs)$

Factor of safety =5(should be taken according to design principles for dynamic loading)

So, endurance limit stress =480 MPa for the considered material

Endurance limit stress = 480/5 N/mm^2 = 96 N/mm^2

Design stress = 96N/mm^2

Strength factor for the pinion = $\sigma_d \times Y = 96 \times 0.122 = 11.71$

Assume module 'm' can be as 8mm considered from the first choice,Therefore module m = 8mm

Face width b = 12m (taken from the design data book)

b = 12×8 = 96mm ,q = 0.394

Pitch diameter of the pinion

$D_P = m \times T = 8 \times 28 = 224$ mm

Diameter can be taken as 230mm,Consider D_P as 230mm

Pitch line velocity

$V = (\pi \times D \times N_p) / 60$

$= (\pi \times 230 \times 400) / (60 \times 1000) = 4.8$m/s

Therefore the pitch line velocity is dependent on the module and the assumed tooth, so for less pitch line velocity to be obtained the module should be considered as less as possible for the leveller bar working

From the equation $V = (\pi \times D \times Np) / 60$

If number of teeth is taken as minimum (i.e.18) for m=8mm,Then $V = (\pi \times D \times N_P) / 60 = 3$m/sec. The pitch line velocity depends on the module and the number of teeth.

Circular pitch for pinion

$P = (\pi \times D_P) / T = (\pi \times 230) / 28 = 25.80$mm

=26mm

Addendum circle or outside diameter, $D_o = D_r + 2h$

Deddendum circle (or) root diameter, $Dr = Dp - 2(t_f + t_c - k)$ m

T_f = tooth factor

=1 for standard tooth

t_c = tooth clearance factor

= 0.15 to 0.25

K = correction factor

= 0.01

D_r = 230-2(1+0.2-0.01)

= 230-2(1.19)

= 230-2.38

= 227.62

Consider D_r=227mm

$D_0 = D_r + 2h$

$h = (2t_f + t_c)$ m ,From design data hand book =2(1+0.2) ×8=19.2mm

Do = 270+19.2

= 289.2mm

= 290mm

From standard proportion of gear system for 20^0 involute full depth:

Working depth in terms of module = m

= 2×8 =16mm

Minimum total depth = 2.25m = 18mm

= 2.25 × 8 = 18mm

'm' is modulus

Tooth thickness = 1.5708m

= 1.5708×8

= 12.5664mm

Minimum clearance or working depth circle

= 0.25m= 0.25×8mm= 2mm

Fillet radius at root = 0.4m= 3.2mm

Tooth space= circular pitch – tooth thickness

= 26mm-12.5mm = 13.5mm,

Consider it as 14mm,Tooth space = 14mm

2.3.Rack design:

For 20 degree full depth involute system, the face width b=96mm

2m = 2×8 = 16mm

P = π×m = 26mm

1.25m+m = 1.25(8) +8 = 18.2mm =19mm

Required length of rack is 17m,

Number of teeth on the rack =(perimeterofpinion)/ (number of teeth)

= (2π×r) / 28 = 25.8 = 26mm

Number teeth on the rack = (length of the rack) / 26mm

= 17000mm/26mm

= 654 teeth

For 17m length of rack, the required number of teeth is 654.

2.4.Design of shaft for pinion wheel

Normal load F_N , acting on the tooth surface

F_N = F_T / cos Φ

F_T = tangential tooth load = 15569.8N

Φ = 20^0

F_N = 15569.8/ cos 20^0

= 16569.03

Weight of the pinion

W_P = $0.00118 \times T_P \times b \times m^2$

= $0.00118 \times 28 \times 96 \times 8^2$

= 203N

W_R Resultant load acting on pinion,

$\sqrt{(1659.03 \times 1659.03) + (203 \times 203) + 2 \times 1659.03 \times 203 \times \cos 20}$

=1851.03N

Since weight of the pinion is small compare to normal load so therefore it may neglected. Thus the resultant load acting on the pinion W_R may equal to the F_N. Assuming the pinion is over hung on the shaft taking overhung as 60mm, there fore bending moment on the shaft due to resultant load

M =W_R×60 = 111061 N-mm

Twisting moment on the shaft

T_w = F_T×(D_P/2)

= 15569.8N×115

= 1790527N-mm

Equivalent twisting moment

T_E = $\sqrt{M^2 + T^2}$

= $\sqrt{111061^2 + 1790527^2}$

= 179.3×10^3 N-mm

T_E = $\pi/16 \times \tau \times D_1^3$

Assume τ as 40N/mm^2

179.3×10^3 = $\pi/16 \times 40 \times D_1^3$

Consider, D_1 = 30mm

2.5.Check for load*s*

Tangential tooth load:

F_T = (P/V) ×C_S

F_T = tangential tooth load in newtons

P = power transmitted in watts =60kw

V = pitch line velocity in m/s

= (π×D×N_P)/60

= 4.817m/s

Cs = service factor

Service factor depends on type of load and working hours per day.For medium load (10-18hrs) per day = 1.25

F_T = ((60×1000) / 4.817) ×1.25 = 15569.8N

2.6.Dynamic load on tooth by using Buckingham equation

F_D = F_T + F_I

Where,

F_I = (21×V (b × C+ F_T)) / (21V+$\sqrt{(bc + FT)}$)

V = 4.8m/sec

B = 96mm

Cv = 0.39

C = 680N-mm

F_I =(21×4.8(96×680+15569.8))/ (21×4.8+$\sqrt{(96 \times 680 + 15569.8))}$

= 8149659.8/385.14

= 21160.25 N

F_D = F_I+ F_T

= 21160.25 + 15569.8

= 36730.05 N

C =(e×K_1)/ ((1/E_1) + (1/E_2))

K_1= 0.11 for 20 degree full depth teeth

e = maximum error depend on ‘m’
e = 0.0778mm (industrial gears)

$(1/E_P) + (1/E_G)$ or 1/E
For steel,E = 220KN/mm^2
C = $(8.55\times10^{(-3)}) / (4.54\ N/mm^2)$
C = 0.15

Static tooth load (i.e. beam strength or the endurance strength of the tooth)

$F_S = \sigma_e\times b\times \pi\times m\times v$
$= 400\times96\times\pi\times8\times0.122$
= 117741.86N

σ_e= flexural endurance limit for carbon steel
= 400MPa
=400N/mm^2

Static tooth load (F_S) is greater than dynamic tooth load (F_D) so the design is safe.

III. RESULTS

The loads are checked and the design is safe, under the loading conditions.

Pitch diameter of the pinion
=230mm
Addendum circle or outside diameter=290mm
Deddendum circle or root diameter = 227mm
Tooth thickness= 12.5mm
Working depth =18mm.
Minimum clearance
= 2mm
Fillet radius at root =3.2mm
Tooth space
= 14mm
Backlash = tooth space – tooth thickness
1. = 14mm-12.5m
2. = 1.5mm.

Module = 8mm

Diameteral pitch = 25.80mm
Face width
= 96mm

Pressure angle or angle of obliquity = 20^0

Depth of the tooth =19mm

No.of teeth on rack=654
Rack length= 17000mm

Weight of the pinion $=0.00118\times T_P\times b\times m^2$
=203N
Diameter of the pinion shaft = 30mm

Modified leveler bar with rack and pinion mechanism drawn using CATIA

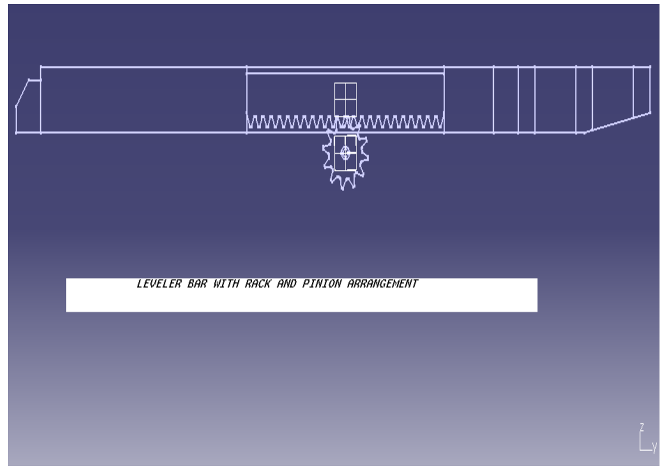

Fig.2.Leveller bar with rack and pinionarrangement

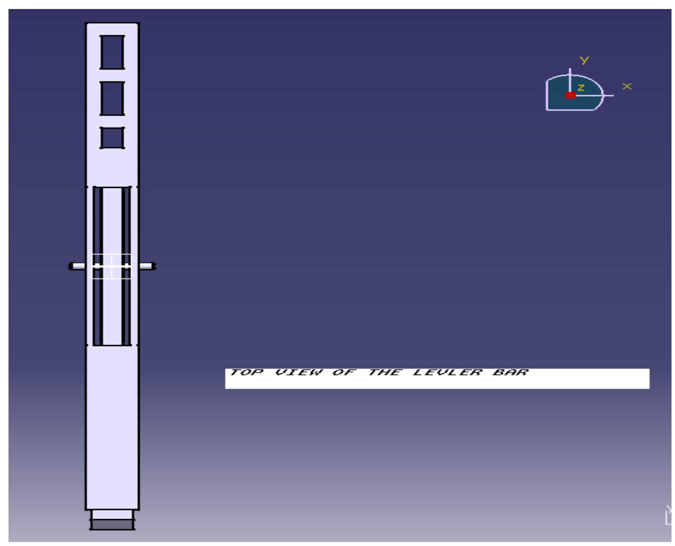

Fig.3.Top view of the Leveller bar with rack and pinion arrangement

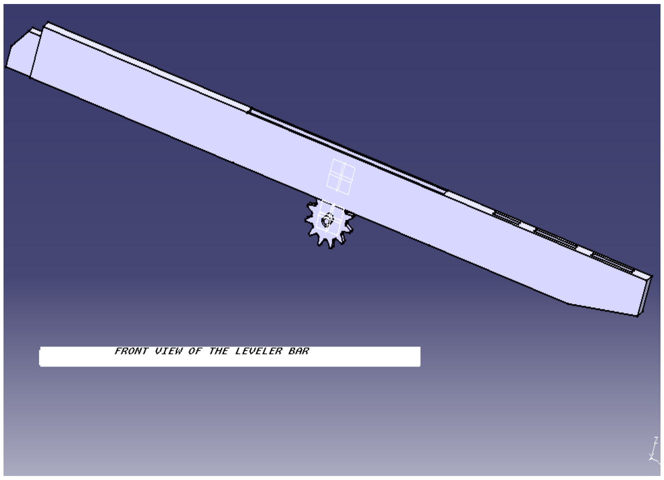

Fig.4.Top view of the Leveller bar with rack and pinion arrangement

IV. CONCLUSION

The theoretical design of the modified leveller bar mechanism using the Machine Design hand data book, the obtained results which are calculated are checked for the design safety and the design is safe as per the choosen values.The present work can be analyzed using the analysis software like ANSYS the stress values can be determined and can compared with the theoretical values and the redesign of the leveller bar can be done and the vibration analysis during the working condition with the replaced rack and pinion can be done.The rack and pinion with leveler bar can be replaced with the hydraulic application for minimization of the practical problem

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