Validation of Failure of Beater Shaft of Double Roller Ginning Machine using Mathematical Failure Analysis Approach

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Abstract

A cotton gin is a machine that quickly and easily separates cotton fibers from their seeds, allowing for much greater productivity than manual cotton separation. The fibers are processed into clothing or other cotton goods, and any undamaged seeds may be used to grow more cotton or to produce cottonseed oil and meal. The cotton gin is a machine used to separate cotton fibers from the seed. The double roller ginning machine consists of various parts such as beater shaft, leather roller, moving knife, fixed knife, feeder, etc. During the ginning operation the shaft fails at certain location. The actual failure position of shaft shown in fig no A was studied and the failure analysis using theories of failure was used to identify and validate the place or point of failure.
Keywords: Ginning machine, beater shaft, theories of failure, SFD, BMD, failure analysis

I. INTRODUCTION

Ginning, in its strictest sense, refers to the process of separating cotton fibers from the seeds. The cotton gin has as its principal function the conversion of a field crop into a salable commodity. Thus, it is the bridge between cotton production and cotton manufacturing. Ginning is the first and most important mechanical process by which seed cotton is separated into lint (fiber) and seed and machine used for this separation is called as gin. It consists of two spirally grooved leather roller pressed against a fixed knife, are made to rotate at about 90-120 rpm. Two moving blades combined with seed grids constitutes a central assembly known as beater which oscillates by means of a crank or eccentric shaft, close to the fixed knife. When the seed cotton is fed to the machine in action, fibers adhere to the rough surface of the roller are carried in between the fixed knife and roller in such a way that the fibers are partially gripped between them. The oscillating knife beats the seed and separates the fibers. This process is repeated for number of times and due to push-pull-hit action the fibers are separated from the seed, carried forward on the roller and dropped out of machine. The ginned seeds drop down through the grid which is oscillating along with beater.

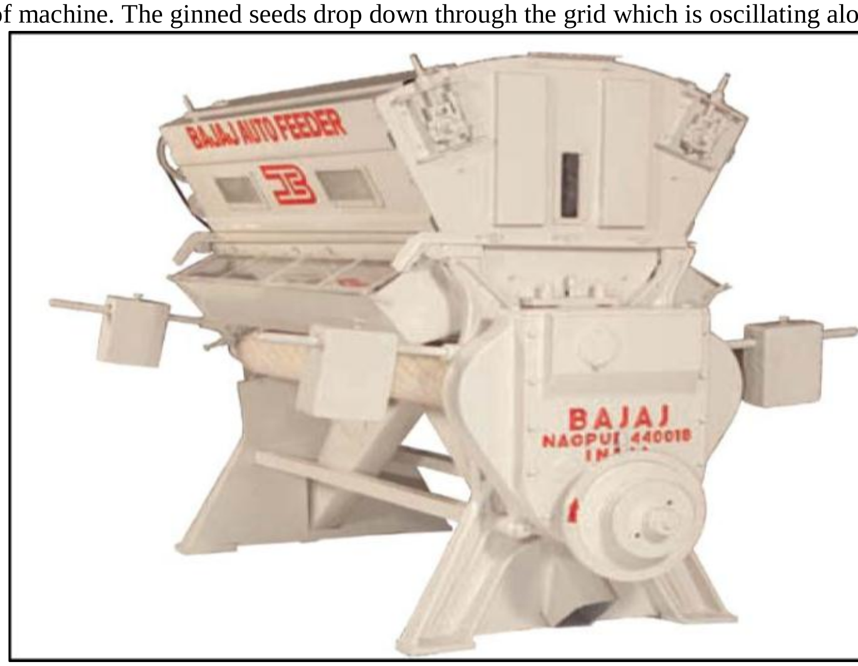

Fig. 1: Double Roller Gin machine

The beater assembly is the innermost and major part of the double roller gin and is sandwiched symmetrically between the stationary knives as shown in fig. It is composite unit consisting of moving knives and seed grids situated on the either side of the beater shaft. It has anchor shaped cross section and its axle is situated 100cm above ground and 15cm from each roller. Beater trough with perforations having concave edge with radius of 565cm and angle of 144 degree between two arms is provided to stop unginned cotton to fall down into seed chute.

Fig. 2: Beater Shaft Assembly

In DR gins the shaft of the oscillating knife is coupled to three beater arms. The motion of the reciprocating knives is symmetric about the fixed knives. The alignment of the beater and moving knives should be such that it should not touch the roller or the stationary knife and back knife at any working position. The driving mechanism in the DR gin is fully controlled from the gearbox. At present in the most of the double roller gins, the roller speed is around 90 to 100 rpm while the beater oscillates with the frequency of 900 to 1000 per minute. Uniform spacing should be maintained between the two moving knives throughout the length of the beater. This is to be effected by inserting or removing thin packing between the knife arms and beater knives.

II. PROBLEM IDENTIFICATION

After visiting two industries i.e. CIRCOT (Central Institute for Research on Cotton Technology, Nagpur) and Bajaj Steel Industries Ltd., Nagpur. It was found that there are various problems occur during operation and one of the problem is the shaft fails at particular point due to wet cotton and after some duration (fatigue failure). The reasons are:

- Moisture content in the cotton.
- Improper setting of machine.
- Maximum load at one end of the shaft that is connected to the gear train.
- Fluctuating load/ Reversing load

Fig. A. Broken Beater Shaft

Hence the shaft failure analysis is done by using mathematical analysis and theories of failure.

III. MATHEMATICAL ANALYSIS OF BEATER SHAFT

A. Design Calculation:

1) Power Required

$P=2\pi NT/60$

Where,

P=5h ,N=Speed=5.93rpm,T=torque

$5*0.736*10^3 = 2*\pi*N*T/60$

[T=6000 N-m]

2) Torque Transmitted

$T=\pi/16 * d^3 * \tau$

Where,

$6000*10^3= \pi/16* 58^3* \tau$

[τ =156.61 MPa]

3) Force Calculation

Force on first arm,

T= F1*L

$6000*10^3$=F1*62

[F1 = 96.774*10^3N]

Force on second arm,

T= F2*L

$6000*10^3$=F2*800

[F2 = 7.5*10^3N]

Force on third arm,

T= F3*L

$6000*10^3$=F3*1404

[F3 = 4.273*10^3N]

$\sum F=0$,

R1- 96.774-7.5-4.273+ R2 = 0

[R1 + R2 = 108.547 KN]

Moment @ A ($\sum$MA) = 0,

$-96.774*10^3*62-7.5*10^3*800-4.273*10^3*1404+R2*1600= 0$

[R2 = 11.249*10^3N]

[R1=97.299*10^3N]

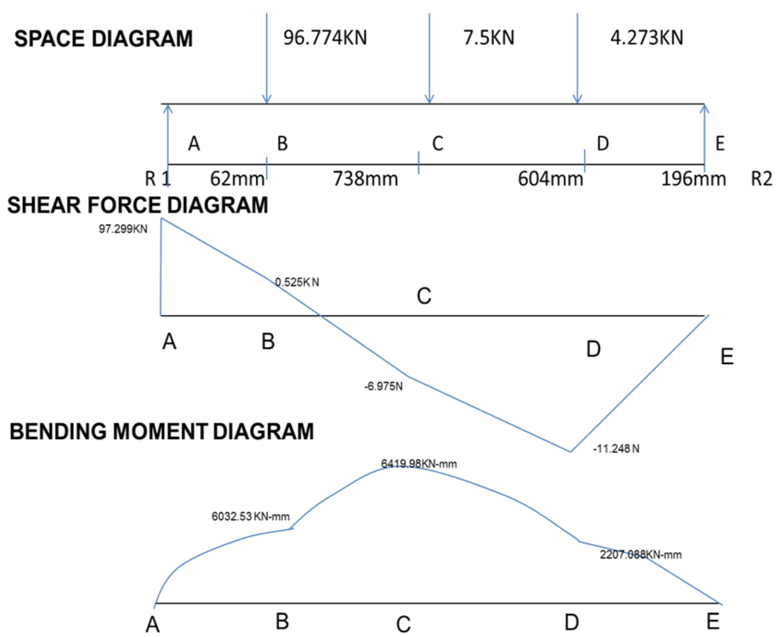

Fig. 3: SFD and BMD of beater shaft

4) Shear Force Calculation
S.F.at A = 97.299K N
S.F. at B = 97.299 96.774=0.525K N
S.F.at C = 0.525-7.5 = -6.975 KN
S.F.at D = -6.975-4.273= -11.248K N
S.F.at E = -11.248+11.248= 0 KN
5) Bending Moment Calculation
B.M. at A & E = 0, end support
B.M. at B = 97.299*62=6032.538KN-mm B.M. at C= 97.299*800-96.774*738=6419.98KN-mm
B.M. at D= 97.299*1404-96.774*1342-7.5*604 =2207.088KN-mm

B. Material Properties:
The material of the beater shaft is bright steel and the properties are as given below:
BRIGHT STEEL (B250)
Youngs Modulus = 204 GPa
Poission Ratio = 0.33
Tensile Yield Strength = 230 MPa
Tensile Ultimate Strength = 410 GPa
1) Factor Of Safety in Torsion,
from PSG data book,
$\eta t= \tau_{-1} / kt * Bsize* \tau$
Where,
τ_{-1}=Endurance limit stress in torsion
τ_{-1}= 6e * Ks
6e=Endurance limit
For steel,
6e=0.8 to 0.9 6yt ,(From Machine design by khurmi & gupta)
Ks= load correction factor
For ductile =0.8
τ= shearing stress
Bsize=size factor =1.5 (from PSG databook)
Kt=stress concentration factor=1(from D.B.)
2) For Bright Steel Material,
Sut=410 Mpa, Syt= 230 MPa, E=204 GPa
Factor of safety,
$\eta t= \tau_{-1} / kt * Bsize* \tau$
τ = 128.21 Mpa
τ_{-1}=6e*ks
6e=0.9 Syt=0.9*230=207 Mpa
τ_{-1}=207*0.8=165.6 Mpa
F.O.S., ηt=165.6/1*1.5*128.21
[$\eta t=0.861\approx1$]

IV. USING THEORIES OF FAILURE

- Maximum principal or normal stress theory ,

Considering, F.O.S.=2
6t= Syt / F.O.S.
6t= 230/2
[6t=115 MPa]

- Maximum shear stress theory ,

τmax=6yt/ 2*F.O.S
=230/2*2
[τmax=57.5 MPa]

- Maximum principal strain theory,

εmax=6yt/E*F.O.S.
=230/204*10^3*2
[εmax=0.002254]

V. RESULTS OF BEATER SHAFT ANALYSIS

Table 1.1:
The stresses generated in the Beater shaft

Sr N	*Material*	*Allowable stress (MPa)*	*Design stress (MPa) (Assuming, FOS=2)*	*Max Principle stress MPa*	*Max shear stress Mpa*	*Shear stress MPa*	*Max principle strain*
1	*Bright steel*	*230*	*115*	*115*	*57.5*	*156.61*	*0.002254*

VI. ANALYSIS

After carefully study of the result tabulated in table no1.1 and result of mathematical and theories of failure analysis . The conclusions are as below:

1) The maximum value of stress in torsion is 230MPa and of the presently material being used is bright steel and the maximum stress obtained using theories of failure is 115MPa.
2) The place of failure actually seen in Fig A and the place of maximum concentration of stress shown by SFD and BMD (Fig 3) coincides.

VII. CONCLUSION

1) The failure which is occurring in the shaft was because of the stress generation were maximum as comparing to the allowable stress. Hence, the failure is occurring.
2) The point of failure seen in the actual specimen coincides by the failure indicated by the mathematical analysis. Failure indicated by the mathematical analysis that justifies correctness of the approach.

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