International Journal of Current Research in Multidisciplinary (IJCRM)
ISSN: 2456-0979
www.ijcrm.com Volume 1 Issue 1 ǁ May. 2016 ǁ PP 06-10

Fabrication of rotating bending type fatigue testing machine

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Abstract:
This project is centred on the design of a low–cost cantilever loading rotating bending fatigue testing machine using locally sourced materials. The design principle is based on the adaptation of the technical theory of bending of elastic beams. Design drawings are produced and components/materials selections are based on functionality, durability, cost and local availability. The major parts of the machine: the machine main frame, the rotating shaft, the bearing and the bearing housing, the specimen clamping system, pulleys, speed counter, electric motor, and dead weights; are fabricated and then assembled following the design specifications. The machine performance is evaluated using test specimens which are machined in conformity with standard procedures. It is observed that the machine has the potentials of generating reliable bending stress – number of cycles data; and the cost of design was lower in comparison to that of rotating bending machines from abroad. Also the machine has the advantages of ease of operation and maintenance, and is safe for use.

Keywords: *Fabrication, rotating bending, fatigue testing machine.*

I. Introduction

This project is centered on the design of a low–cost loading rotating bending fatigue testing machine using locally sourced materials. The design principle is based on the adaptation of the technical theory of bending of elastic beams. Design drawings are produced and components/materials selections are based on functionality, durability, cost and local availability. The major parts of the machine: the machine main frame, the rotating shaft, the bearing and the bearing housing, the specimen clamping system, pulleys, speed counter, electric motor, and dead weights; are fabricated and then assembled following the design specifications. The machine performance will be evaluate using test specimens which are machined in conformity with standard procedures. It is observed that the machine has the potentials of generating reliable bending stress – number of cycles data; and the cost of design was lower in comparison to that of rotating bending machines from abroad. Also the machine has the advantages of ease of operation and maintenance, and is safe for use. Fatigue cracks once initiated often grow in an insidious manner resulting in failures with serious implications. The technical problems, economic and potential human losses which accompany fatigue failures make its consideration during materials design of utmost importance if the challenges associated with its occurrence are to be mitigated. A lot of research interest has been devoted to studying the fatigue behaviour of engineering materials with a view to arriving at ways to effectively design against the failure mode.

II. Prime Mover Selection

Motor is a Single phase AC motor, Power 60 watt, Speed is continuously variable from 0 to 6000 rpm. The speed of motor is variated by means of an electronic speed variator . Motor is a commutator motor ie, the current to motor is supplied to motor by means of carbon brushes. The power input to motor is varied by changing the current supply to these brushes by the electronic speed variator, thereby the speed is also is changes. Motor is foot mounted and is bolted to the motor base plate welded to the base frame of the indexer table.

III. Motor Selection

Thus selecting a motor of the following specifications
Single phase AC motor
Commutator motor
TEFC construction
Power = 1/15hp=60 watt
Speed= 0-6000 rpm (variable)

Motor is an Single phase AC motor, Power 60 watt , Speed is continuously variable from 0 to 6000 rpm. The speed of motor is variated by means of an electronic speed variator . Motor is an commutator motor ie, the current to motor is supplied to motor by means of carbon brushes. The power input to motor is varied by changing the current supply to these brushes by the electronic speed variator, thereby the speed is also is changes. Motor is foot mounted and is bolted to the motor base plate welded to the base frame of the indexer table

Motor Torque
P= 2 Π N T/60
T = (60 x 60)/(2 Π x 6000)
T = 0.095 N-m
Power is transmitted from the motor shaft to the input shaft of drive by means of an open belt drive,
Motor pulley diameter = 20 mm
IP _ shaft pulley diameter = 110 mm
Reduction ratio = 5
IP shaft speed = 6000/5 = 1200 rpm
Torque at IP shaft = 5 x 0.095 = 0.475 Nm

IV. Design of Open Belt Drive

Motor pulley diameter = 20 mm
IP _ shaft pulley diameter = 110 mm
Reduction ratio = 5
Coefficient of friction = 0.23
Maximum allowable tension in belt = 200 N
Center distance = 120
α= 180 – 2{sin-1 (D-d)/2C}
α= 180 – 2{sin-1 (110-20)/2x120}
α= 1360
α= 2.37c
Now,
eμα /sin(θ/2) = e0.2 x 2.37/sin (40/2) = 4
width (b2) at base is given by
b2 = 6-2(4 tan 20) = 3.1
Area of cross section of belt = ½{6 + 3.1}x 4
A = 18.2 mm2
Now mass of belt /m length = 0.23 kg/m
V = ΠDN/(60 x 1000) = 4.188m/sec
Tc = m V2
Tc = 4.034 N
T1 = Maximum tension in belt – Tc
T1= 195.966 = 196 N
T1 / T2 = eμα /sin(θ/2) =4
T2 = 49 N

Result Table

Tension in tight side of belt (T1) = 196 N
Tension in slack side of belt (T2) = 49 N

V. Design of Input Shaft

Motor Torque
P= 2 Π N T/60
T = (60 x 60)/(2 Π x 6000)
T = 0.095 N-m
Power is transmitted from the motor shaft to the input shaft of drive by means of an gear box 1:3 ratio standard from 4" DC grinder: Dewalt make
Reduction ratio = 3
IP shaft speed = 6000/3 = 2000 rpm
Torque at IP shaft = 3 x 0.095 = 0285 Nm
T Design = 2 x T =0.57 Nm. FOS =2
= 0.57x 103 N.mm

Selection of Shaft Bearing : In selection of ball bearing the main governing factor is the system design of the drive ie; the size of the ball bearing is of major importance; hence we shall first select an appropriate ball bearing first select an appropriate ball bearing first taking into consideration convenience of mounting the planetary pins and then we shall check for the actual life of ball bearing.

6.1 ball bearing selection.

ISI NO	Brg.Basic Design No (SKF)	D	D1	D	D2	B	Basic capacity	
							C kgf	Co Kgf
20A C02	6004	20	23	42	12	9	4500	7530

P = X Fr + Yfa.
Where;
P=Equivalent dynamic load (N)
X=Radial load constant
Fr= Radial load(H)
Y = Axial load contact
Fa = Axial load (N)
In our case;
Radial load FR= weight of system + tangential tooth load
Fa = 0
Diameter of standard bevel gear used is 46mm hence
Pt = 0.57x x 103 / 23 =24.7
P= 24.7 + 2x 9.81 (assuming 2 kg load applied on the system)
P= 44.32 N
⇒L= (C/p) p
Considering 4000 working hours
$L = 60\ n\ L\ h/10^6$ = 480 mrev….assuming 2000 rpm wheel speed

C = 345 N
AS; required dynamic of bearing is less than the rated dynamic capacity of bearing;
⇒Bearing is safe.

VI. Design of lh shaft

Selection of RH shaft material

Designation	Ultimate Tensile Strength	Yield strength N/mm2
EN 24 (40 N; 2 cr 1 Mo28)	720	600

Using ASME code of design;
Allowable shear stress;
Fsall is given stress;
Fsall = 0.30 syt = 0.30 x 600
=180 N/mm2
Fsall = 0.18 x Sult = 0.18 x 720
= 130 N/mm2
Considering minimum of the above values;
fsall = 130 N/mm2
As we are providing dimples for locking on shaft;
Reducing above value by 25%.
⇒ fsall = 0.75 x 130
= 97.5 N/mm2
a) Considering pure torsional load;
Tdesign = Π fsall d3/16
⇒d3 = 16 x 0.57 X 103/Πx 97.5

d = 7.8 mm
Shaft diameter is taken to be 16mm as muff coupling available standard has standard bore of 16mm.

VII. Selection of LH Shaft Bearing

In selection of ball bearing the main governing factor is the system design of the drive i.e. the size of the ball bearing is of major importance; hence we shall first select an appropriate ball bearing first select an appropriate ball bearing first taking into consideration convenience of mounting the planetary pins and then we shall check for the actual life of ball bearing .

VIII. Ball Bearing Selection

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Diameter of standard bevel gear used is 46mm hence
Pt = 0.57x x 103 / 23 =24.7
P= 24.7 + 2x 9.81 (assuming 2 kg load applied on the system)
P= 44.32 N
$\Rightarrow$L= (C/p) p
Considering 4000 working hours
L = 60 n L $h/10^6$ = 480 mrev….assuming 2000 rpm wheel speed
$\Rightarrow 480 = (C/44.32)^3$
C= 345N
AS; required dynamic of bearing is less than the rated dynamic capacity of bearing;
$\Rightarrow$ Bearing is safe.

IX. Conclusion

We have observed that materials fail under fluctuating stresses at a stress magnitude which is lower than the ultimate tensile strength of the material. Sometimes the magnitude is even lower than the yield strength.The fatigue failure is sudden and total. It is relatively easy to design a component for static load. The fatigue failure, however, depends upon a number of factors, such as the number of cycles, mean stress, stress amplitude, stress concentration, corrosion creep. This makes the design of component subjected to fluctuating stresses more complex.

X. Acknowledgement

We are thankful to my friend mr.Nagwe Rishikesh and for their valuable guidance and encouragement. It gives us immense pleasure and sense of satisfaction to present this report on FABRICATION OF ROTATING BENDING TYPE FATIGUE TESTING MACHINE and expressing sincere. We also would like to thank all the staff members of Mechanical department for providing us with required facilities and support towards the completion of this project. We thank to all my friends who helped us directly or indirectly to submit this project report on time.

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