FAILURE ANALYSIS AND REDESIGN OF FORGING HAMMER SHAFT

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1.1. INTRODUCTION

Forging is one of the oldest known metal workingprocesses. Traditionally, forging was performed by a smith using hammer and anvil, though introducing water power to the production and working of iron in the 12th century drove the hammer and anvil into obsolescence.

Forging is a manufacturing process involving the shaping of metal using localized compressive forces. The blows are delivered with a hammer or a die. Forging is often classified according to the temperature at which it is performed: cold forging, warm forging, or hot forging. For the latter two, the metal is heated, usually in a forge. Forged parts can range in weight from less than a kilogram to hundreds of metric tons. Forging has been done by smiths for millennia; the traditional products were kitchenware, hardware, hand tools, edged weapons, and jewellery. Since the Industrial Revolution, forged parts are widely used in mechanisms and machines wherever a component requires high strength; such forgings usually require further processing such as machining to achieve a finished part. Today, forging is a major worldwide industry.

In modern times, industrial forging is done either with presses or with hammers powered by compressed air, electricity, hydraulics or steam. These hammers may have reciprocating weights in the thousands of pounds. Smaller power hammers, 500 lb (230 kg) or less reciprocating weight, and hydraulic presses are common in art smithies as well. Some steam hammers remain in use, but they became obsolete with the availability of the other, more convenient, power sources.

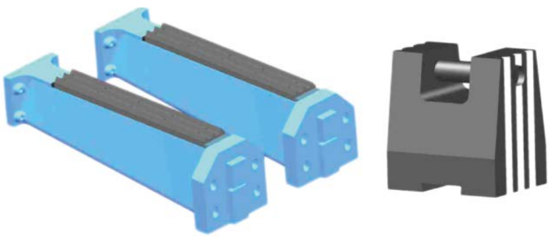
Forging hammer are most suitable for precision forging for automobile, railways, defence, aeronautical, hand tool, agriculture, bicycles and other engineering industries. These are basically designed to achieve better productivity, durability and economy.

1.1.1 MAIN COMPONENTS

COLUMN, SLIDES & RAM:-

Cast steel column is duly annealed and machined are very stiff and robust in construction to ensure longer stability. These are positioned by a large spigot on underside of each foot which fits into a machine recess in theanvil block the columns are locked with the help of tapered wedges of alloy steel toensure firm alignment of columns which helps in accurate guiding of ram. Clearance between the ram and guide ways attached to column is maintained with the help of tapered wedge which bring the column inward and drawback bolt provided pushes the column outward.

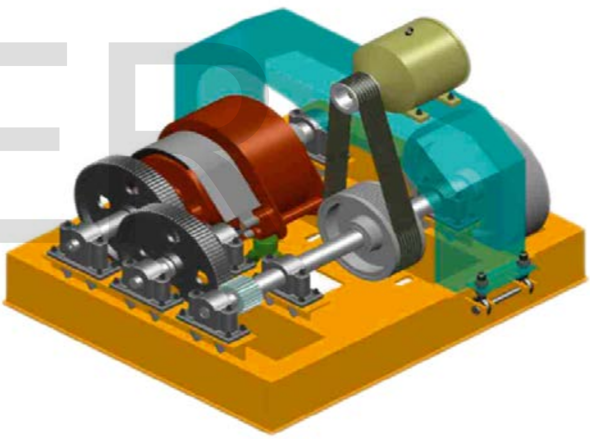
Synthetic rubber mat is provided on the anvil block on anvil rest. It resists the induced shock vibration which increases the life of ram and also ensures smooth working of machine for longer period.



No. 1:- Column, slides & ram

HEAD ASSEMBLY:-

Heavy duty head assembly fabricated from rolled steel sections is mounted on top of the columns. The drive is through v-belts from high torque A.C. electric motor via flywheel and reduction gears to the lifter shaft which runs on double ball bearings and one central phosphorus bronze bearing which also serves as support to lifters shaft.



No. 2:- Head assembly

The friction lifter consists of constantly rotating drum and break lined steel band. This band is anchored at one end to stud in lifter drum which is actuated by lever. The lever is operated with the help of pulling cord tied to lever at one end goes to operating point by passing through capstan bush attached to lifter shaft. When we pull the cord it tightens on to the rotating capstan bush which operate the lever resulting in tightening of brake around the brake drum with the help of cam shaft. At this stage, lifter drum is rotated and ram is lifted with help of nylon belt provided. An release of cord, the spring loaded arrangement help free fall of ram by disengaging friction band immediately from the friction drum.

FLYWHEEL:-

A flywheel is a rotating mechanical device that is used to store rotational energy. Flywheels have a significant moment of inertia and thus resist changes in rotational speed. The amount of energy stored in a flywheel is proportional to the square of its rotational speed. Energy is transferred to a flywheel by applying torque to it, thereby increasing its rotational speed, and hence its stored energy. Conversely, a flywheel releases stored energy by applying torque to a mechanical load, thereby decreasing the flywheel's rotational speed.

No. 3:- Flywheel

PINION:-

A pinion is a round gear used in several applications which is usually the smallest gear in a gear drive train, although in the case of John Blenkinsop'sSalamanca, the pinion was rather large.In many cases, such as remote controlledtoys, the pinion is also the drive gear, or the smaller gear that drives in a 90-degree angle towards a crown gear in a differential drive, or the small front sprocket on a chain driven motorcycle.

No. 4:- Pinion

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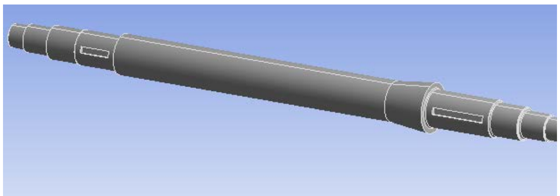
SHAFT:-

A shaft is a rotating member usually of circular cross-section (solid or hollow), which is used to transmit power and rotational motion in machinery and mechanical equipment in various applications. Most shafts are subjected to fluctuating loads of combined bending and torsion with various degrees of stress concentration. For such shafts the problem is fundamentally fatigue loading. Failures of such components and structures have engaged scientists and engineers extensively in an attempt to find their main causes and thereby offer methods to prevent such failures.

Eccentric Shaft is widely appreciated for its features like corrosion resistant, long service, effective performance and reliability.



No. 5:- Shaft

MOTOR:-

Type- 3 phase squirrel cage induction motor

Power- 3HP

Voltage- 440V, 50Hz

Make- Roto motive

Speed-1440rpm

1.3. LITERATURE SURVEY

A deep and profound literature survey is backbone of a successful project. We extensively searched for past and related work in this field. We used various resources available to us. Internet is of course a valuable source of information in this day and age. Especially for a concept like ours which is quite new and constantly evolving. However we did not restrict ourselves just to that.

We used various books and technical magazines to gather information related to our insightful literature survey has helped us to exactly find out where our project stands and we can explain how our project is novel.

The main aim is to recognize any vibration occurring in machine, misalignment of parts, environment effect (temperature and assembling of parts), internal effect like torsion, bending and tension-compression effect etc.

Osman Asi

This paper describes the failure analysis of rear axle shaft used in an automobile. Shaft is subjected to bending and torsional stress. Vehicle was 9 year old & failure leas to accident. Failure leads to breaking of shaft into two pieces close to wheel base. Failed shaft inspected microscopically, chemical analysis carried out also fractured surface examine using SEM (scanning electron microscope). There were two fillet welded region between axle shaft and bearing locking ring. Bearing locking ring was fixed to axle shaft by gas welding during repair and maintenance instead it should be pressed or shrink fitted. Fracture has initiated from welded area towards the center. Final fracture area was approximately 15% suggest high cycle low stress type. Chemical composition found ok with AISI 4140 steel.From observation crack was initiated at stress concentration point leading to fracturing of axle shaft. Fracture occurred at spline portion.Improper welding of hardened materials involves low ductility in the heataffected zone (HAZ), stress concentration points, and inclusions in the structure that served as nuclei for the fatigue cracks Pre & Post heating of shaft were not carried out. Result indicates that axle shaft fractured in reverse bending fatigue as a result of improper welding

M. Ristivojevic, R. Mitrovic, T. Lazovic

This paper present the failure analysis of air fan shaft used in boiler of thermal power plant. Microstructure examination carried out to determine material chemical & mechanical properties. Also design solution for bearing seating analyzed. Bearing on shaft was locked to prevent axial movement by lock nut which tends to expand bearing inner ring, which result in decreasing clearance between bearing element. The breakdown resulted in damage of fan shaft sleeve location due to permanent deformation and material melting. Heat source is generated due to friction between contacting parts of double pressed joint which result in change in interference clearance values between bush and sleeve. In service clearance is formed. Nut tightening tends to expand inner ring of bearing result in lowering radial clearance & weakening interference between bush & sleeve. This result in sleeve slipping in bushing. Consequently bearing friction is increased, the micro sliding joint resulting in sleeve material melting & bearing element damaged. By selecting narrower tolerance, assembly strict to actual dimensions of double pressed joint, insure firm joint elements failure can be prevented

Sandip Bhattacharyya

In this paper failure of gas blower shaft of blast furnace is analyzed. Earlier failure was due to fatigue at fillet radius. Latest failure is on uniform diameter shaft. Microstructure study shows that deformation of shaft material near the surface .Compositional analysis shows high percentage of sulphur not confirming to the standard. Also hardness of shaft material measured 40Rc against 44Rc near the surface. From analysis it was found that lock plate loosening because of improper interference fit leads to groove formation on shaft and one of such groove leads to fracture. The fracture was perpendicular to the shaft axis. Examination shows rotary deformation marks and severe heat effects. Hardness at fracture region found reduced considerably. In conclusion loosening of bearing lock plate due to poor interference fit result in failure of shaft, the excess amount of MnS inclusions and delta-ferrite are generally not acceptable in the material since they promote fatigue crack initiation.

R.W. Fuller

In this paper failure analysis of mixer unit shaft made of AISI 303 stainless steel using conventional 14 step approach is carried out. Failed component is drive unit output shaft of 15 HP mixer unit.AISI304 shaft fails within 3 week of operations. During investigation it was found that shaft having a loose fit, weld plug were installed using same material AISI304,other factor such as weight of mixer, material & ambient temperature were not critical. Spectrometer reading was taken to determine materialcomposition, found not deviating from original values. Inter-granular cracks were observed under.Weld plug used to overcome loose fit result in failure AISI 304 not suitable for welding, instead.

1.4. PROBLEM DEFINITION

The problem faced in the machine was failure of shaft within two months of it's usage. This problem might have arisen due to following unavoidable circumstances:

1. Vibration caused due to hammering.
2. Misalignment during fitting.
3. Bending of shaft.
4. Shearing due to lifting, etc.

1.4.1. BACKGROUND OF FAILURE ANALYSIS:-

Failure analysis is the process of collecting and analyzing data to determine the cause of a failure and how to prevent it from recurring. It is an important discipline in many branches of manufacturing industry. Such as the electronics industry where it is a vital tool used in the development of new products and for the improvement of existing products. However, it also applied to other fields such as business management and military strategy. Failure analysis and prevention are important functions to all of the engineering disciplines. The materials engineer often plays a lead role in the analysis of failures, whether a component or product fails in service or if failure occurs in manufacturing or during production processing. In any case, one must determine the cause of failure toprevent future occurrence or to improve the performance of the device, component or structure. Failure analysis can have three broad objectives.

1. Determining modes of failure.
2. Failure Cause
3. Root causes.

Failure mode can be determined on-site or in the laboratory, using methods such as fractography, metallographic and mechanical testing. Failure cause is determined from laboratory studies and knowledge of the component and its loading and its environment. Comparative sampling or duplication of the failure mode in the laboratory may be necessary to determine the cause. Root failure cause is determined using knowledge of the mode, the cause and the particular process or system. Determining the root failure cause require complete information about the equipment's design, operation, maintenance, history and environment. A typical failure analysis might include fractography, metallographic and chemical analysis.

No. 8:- Actual Pic of failure of shaft

1.4.2. CAUSES AND ANALYSIS OF SHAFT FAILURE:-

Causes of failure

Austin H. Bonnett discussed the causes of shaft failures. He has focused on failures associated with fatigue. XU Yanhui says that shaft damaged can be induced by sub synchronous resonance (SSR). According to J. feller fatigue loading on wind turbine drive trains due to the fluctuating nature of wind is major cause of premature failure of gearboxes.

Cause of shaft failure	**Percentage**
Corrosion	2
Fatigue	25
Brittle fracture	16
Overload	11
High temperature Corrosion	7
Stress concentration fatigue	6
Creep	3
Wear, abrasion and erosion	3

Table no. 1- Causes of shaft failure

The shaft failed due to fatigue, which arises due to following reasons.

a. Presence of cyclic over-loads.

b. Stress concentration: They may be due to production or operation causes e.g. under cuts, machining, traces, notches etc.

c. Wrong adjustment of bearing, insufficient clearances.

In corrosion failures, the stress is the environment and there action it has on the shaft material. At the core of this problem is an electrochemical reaction that weakens the shaft. Corrosion is a process that occurs when oxygen, water, acids and salts mix together. The temperature must be above 0˚C, when the relative humidity is below 40% almost no corrosion from 40-60% (relative humidity) significant corrosion is to be expected.

The other causes of shaft failure which could be detected can be fatigue failure in shaft. Fatigue failure can be defined as the failure that occurs when stress is concentrated at a particular section when there is irregularities in surfaces or uneven surfaces.

The other easily detected cause of failure would be improper machining, human errors, poor assembly fits, etc.

1.5. METHODOLOGY

From above discussions it is clear that the failure of shaft occurs due to shear of shaft while lifting of ram. The lifting process which is through a rope occurs with a minimum amount of force. So, the entire load falls on the gear box assembly i.e., gear set. This leads to excess amount of load on the flywheel section of the shaft which is the main part which transmits power from motor to the machine.

Now, the shear occurring on the shaft is first calculated analytically and then it is verified with analysis software ANSYS 14.5.

Modeling software such as CATIA V5 is first used and then it is exported to the analysis software such as ANSYS 14.5.

After the analysis of the shaft, suggested remedies are taken into consideration and shaft is redesigned with proper constraints.

1.6. ANALYSIS OF SHAFT

1.6.1. Calculations of Shaft

Power, P =75HP =55.9275KW

Speed of motor, N=1440rpm

Diameter of pulley, d=280mm

Diameter of flywheel(big pulley), D=1000mm

Poisson's ratio, μ=0.28

Diameter of pinion, D_p=210mm

Speed ratio,

$$\frac{n}{N} = \frac{d}{D}$$

$$\frac{n}{1440} = \frac{280}{1000}$$

n = 403.2rpm

$$V_B = \frac{\pi d N}{60}$$

$$= \frac{\pi \times 0.28 \times 1440}{60}$$

=21.1115m/sec

$P = (T_{ft}-T_{fs}) \times V_B$

$55.9275\times10^3 = (T_{ft}-T_{fs}) \times 21.1115$

$T_{ft}-T_{fs}$= 2649.1482N……………………………………………………….(1)

$$\theta = \pi+2\times\sin^{-1}\left[\frac{D-d}{2\times C}\right]$$

$$= \pi+2\times\sin^{-1}\left[\frac{1000-280}{2\times1000}\right]$$

= 222.2° or 3.8781rad.

$$\frac{Tft}{Tfs} = e^{\frac{\mu\times\theta}{\sin 2\phi}}$$

Where, 2ϕ = Groove angle of pulley = 38°

$$= e^{\frac{0.28\times3.8781}{\sin 38}}$$

= 1.7637

$T_{ft}-1.7671\times T_{fs}$= 0……………………………………………… (2)

From eq^n (1) and (2),

T_{ft}= 6117.9817N

T_{fs} = 3468.8335N

Torque to be transmitted,

$$P = \frac{2\pi n T}{60}$$

$55.9275\times10^3 = \frac{2\times\pi\times403.2\times T}{60}$

$T = 1324.5741Nm$

Forces acting on pinion

1. Tangential force

$T = F_t\times\frac{dp}{2}$

$1324.5741\times10^3 = F_t\times\frac{210}{2}$

$F_t = 12614.9914N$

2. Axial force

$F_r = F_t\times\tan\alpha$

Where, α = pressure angle = 20^o

$F_r = 12614.9914\times\tan20^o$

$Fr = 4591.4813N$

Reaction forces on the shaft:-

In addition to the tangential and axial forces acting on the pinion and the belt tensions of the flywheel, the weight of the flywheel and the pinion are also considered.

Weight of flywheel, $W_F = 1000\times9.81 = 9810N$

Weight of pinion, $W_p = 20\times9.81 = 196.2N$

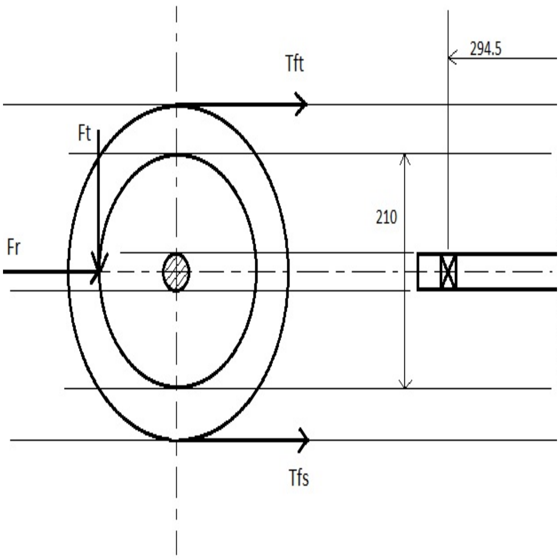


Fig No. 1:- Reaction forces on shaft

For Vertical Plane,

$\Sigma Fy = 0$

$R_{BV} = 9810N$

SF Calculations

L of A = 0

R of A = 0

L of B = 0

R of B = 9810N

L of C = 9810N

R of C = 9810-9810 = 0

At D = 0

BM Calculations

$M_A = 0$

$M_B = 0$

$M_C = R_{BV} \times 1307.5 = 12.8265 \times 10^6 Nmm$

$M_D = R_{BV} \times 1673.5 - 9810 \times 366 = 12.8265 \times 10^6 Nmm$

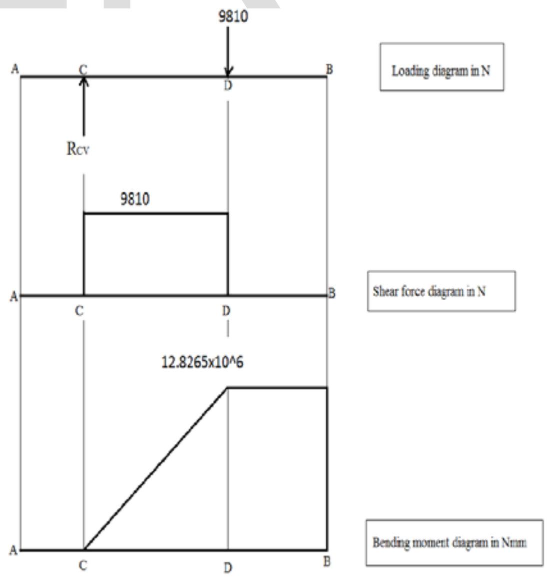


Fig No. 2:- Vertical plane SFD and BMD

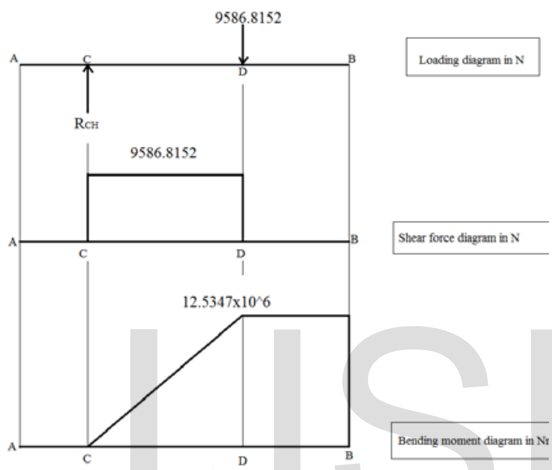


Fig No. 3:- Horizontal plane SFD and BMD

For Horizontal Plane,

$\Sigma F_H = 0$

$R_{BH} = 9586.8152N$

SF Calculations

L of A = 0

R of A = 0

L of B = 0

R of B = 9586.8152N

L of C = 9586.8152N

R of C = 0

At D = 0

BM Calculations

$M_A = 0$

$M_B = 0$

$M_C = R_{BH} \times 1307.5 = 12.5347 \times 10^6 Nmm$

$M_D = R_{BH} \times 1673.5 - 9586.8152 \times 366$

$= 12.5347 \times 10^6 Nmm$

Therefore,

$(M_{eq})_C = 17.9343 \times 10^6 Nmm$

$T_{eq} = \sqrt{(Km \times M)^2 + (Kt \times T)^2}$

$= \sqrt{(2 \times 17.9343 \times 10^6)^2 + (2.5 \times 1324.5741 \times 10^3)}$

$= 36.0211 \times 10^6 Nmm$

The shear stress induced in the shaft is given by,

$\tau = \frac{16Teq}{\pi \times d^3}$

Where, diameter of shaft is taken as 125mm.

$= \frac{16 \times 36.0211 \times 10^6}{\pi \times 125^3}$

= 83.502Mpa

As per ASME code for design of shaft,

τ_{per} = 0.3×S_{yt} OR

= 0.18×S_{ut}……………whichever is smaller.

For EN8 material,

S_{yt}	S_{ut}
450 MPa	600 MPa

= 0.3×450 OR 0.18×600

= 135MPa OR 108 MPa

Therefore, selecting τ_{per}= 108MPa

For keyway effect,

τ_{per} = 0.75×108

= 81Mpa

Therefore, it can be seen that,

$\tau > \tau_{per}$

So, the shaft fails in shear.

Design of shaft according to the permissible shear stress

$\tau_{per} = \frac{16T}{\pi d^3}$

$81 = \frac{16 \times 36.0211 \times 10^6}{\pi \times d^3}$

d = 132mm

1.6.2. BEARING DESIGN

Sr. no.	Bearing Type	Bearing No.	Qty	Shaft
1.	Deep Groove Ball Bearing	6318	2	Main Transmission Shaft

Table no. 2- Bearing Design

Nomenclature of Bearing

D= Outside Diameter

d= Bore Diameter

B= Width

DEEP GROOVE BALL BEARING 6318:-

d	D	B	C	C_o
90mm	190mm	43mm	11200kgf	9800kgf

Table no. 3- Bearing Specifications

$F_r = F_t x \tan(\alpha)$

Where,

F_r= Radial Forces

F_t = Tangential Forces= P/V

$\alpha = 20^o$ for spur gear

F_{r1}= 12614.9914 x tan20

= 4591.48 N

$F_r = F_t + F_{r1}$

= 12614.9914 + 45910.48

= 17206.47 N

F_a = 0 ......(for spur gear)

When the bearing are subjected to a purely radial load.

Then,

X=1, Y=0

P= (X xF_r) + (Y x F_a)

= 17206.47 N

L_h = 60000 hrs.......(PSG Design Data Book Page No. 4.20)

$$L = \frac{60 \text{ x N x Lh}}{10^6}$$

Where,

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L_h= Bearing Life

N = Revolution per min

L_h= 60000hrs, N = 1440rpm

$$L = \frac{60 \ x \ 1440 \ x \ 60000}{10^6}$$

L= 5184 million rev.

L= $(C/P)^a$

a= 3 for roller bearing

C= P x $L^{1/3}$

C= 4963.20 < 11200

Hence Bearing Design is safe.

1.6.3KEY DIMENSION

WIDTH(b) =22mm

HEIGHT(h) =18mm

LENGTH(l) =180mm

S_{yc}=S_{yt}=450 N/mm^2

$\sigma_c = \frac{Syc}{Nf}$

$= \frac{450}{2}$

σ_c= 225N/mm^2

$T = \frac{Ssy}{Nf}$

$T = \frac{340}{2}$

T = 170 N/mm^2

$T = \frac{2\ xT}{dbl}$

$T = \frac{2\ x\ 1324.5741\ x\ 10^3}{22\ x\ 180\ x\ 18}$

T = 37.165N/mm^2 < 170N/mm^2

$\sigma_c = \frac{4T}{dhl}$

$\sigma_c = \frac{4\ x\ 1324.574\ x 10^3}{20\ x\ 180\ x\ 18}$

σ_c = 74.33N/mm^2 < 225N/mm^2

Hence, Design of Key is safe.

1.6.4. Finite Element Analysis:-

The finite element method (FEM), sometimes referred to as finite element analysis (FEA), is a computational technique used to obtain approximate solutions of boundary value problems in engineering. Simply stated, a boundary value problem is a mathematical problem in which one or more dependent variables must satisfy a differential equation everywhere within a known domain of independent variables and satisfy specific conditions on the boundary of the domain. Boundary value problems are also sometimes called field problems. The field is the domain of interest and most often represents a physical structure.

Modelling of shaft:-

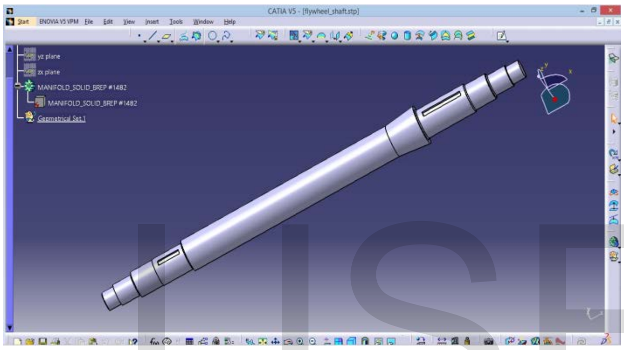

Fig No. 4:- Modelling of shaft

CatiaV5 is used for modelling of shaft. CAD software like Catia V5 is higher end software which is feature based solid modelling systems. It is the only menu driven higher end software. It provides mechanical engineers with an approach to mechanical design automation based on solid modelling technology.

Forces applied on the shaft:-

Figure shows the forces which are applied on the shaft.

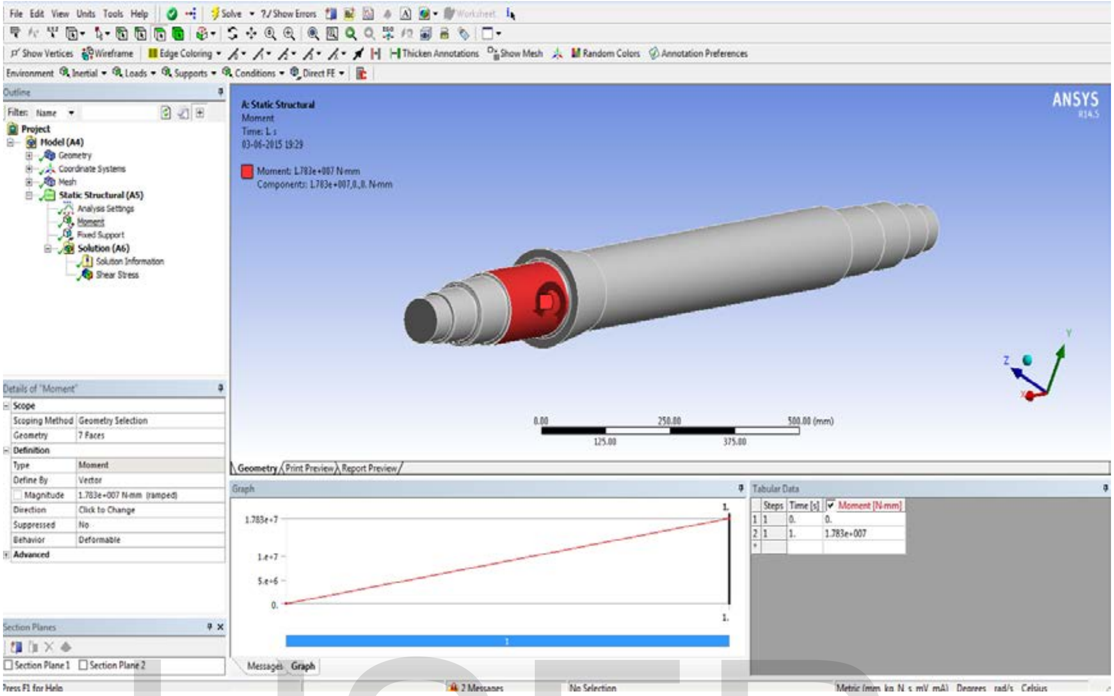

Fig No. 5:- Forces applied on shaft

The moment M= 17.9343KNm is applied to the flywheel(pulley) of the shaft.

Fixed support is considered on pinion side.

Maximum Shear Stress on shaft of EN8 material:-

Composition of EN8 material,

Carbon: 0.4 – 0.5%

Manganese: 0.62 – 0.9%

Nickel: 0.9 – 1.2%

Silicon: 0.1 – 0.25%

For En8 material

Ultimate Tensile Strength, S_{ut}= 600MPa

Tensile Yield Strength, S_{yt}= 450MPa

Density,ρ= 7850 kg/m^3

Temperature C	Young's Modulus MPa	Poisson's Ratio	Bulk Modulus MPa	Shear Modulus MPa
50	2.1e+005	0.29	1.6667e+005	81395

Table no. 4- Properties of EN8 material

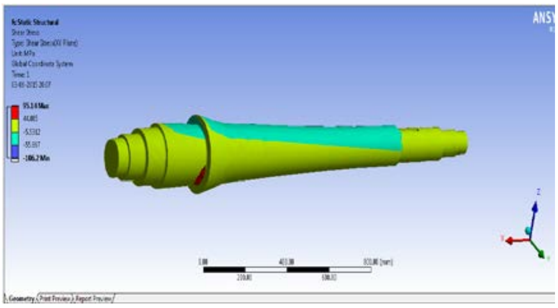

Fig No. 6:- ANSYS report on EN8

Figure shows the maximum shear stress on shaft and it is found 95.14MPa which is greater than the allowable shear stress at the location near the flywheel of the shaft.

Allowable shear stress of the shaft of EN8 material is 81MPa.

Stress	Allowable Stress	ANSYS Result
Shear stress	81MPa	95.14MPa

From above table it can be conclude that, the shear stress is greater than allowable stress.Shaft can be redesign by keeping diameter more than 125 mm.

But the client will not be interested because he has to redesign the complete system including the gear train and other devices such as brake drum dynamometer attached to it. So an alternate way to redesign the shaft is to change the material of the shaft.

Redesign of shaft by using various material:-

EN24 (SAE4340):-

EN24 steel is high tensile steel material renowned for its wear resistant properties and also where high strength properties are required. EN24 is used in components subjected to high stress and with large cross-section. This can include aircraft, automotive and general engineering applications for e.g. propeller or gear shafts, connecting rods, aircraft landing gear components and high strength machine parts like collets, spindle, bolts, gear, etc.

Composition of EN24 material,

Carbon: 0.32 – 0.36%

Silicon: 0.2 - 0.45%

Manganese: 0.4 – 0.7%

Nickel: 0.1 – 0.3%

Chromium: 1.5 – 1.8%

Molybdenum: 0.22 – 0.25%

For En24 material

Ultimate Tensile Strength, S_{ut}= 1000MPa

Tensile Yield Strength, S_{yt}= 680MPa

Density ,ρ= 9490 kg/m^3

Temperature C	Young's Modulus MPa	Poisson's Ratio	Bulk Modulus MPa	Shear Modulus MPa
50	2.06e +005	0.29	1.6349e +005	79845

Table no. 5- Properties of EN24 material

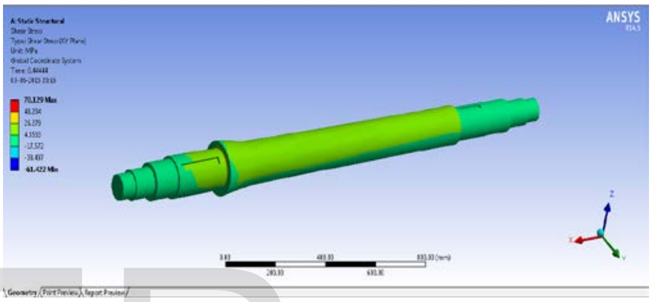

Fig No. 7:- ANSYS report on EN24

Figure shows the maximum shear stress on shaft and it is found 70.129MPa which is less than the allowable shear stress at the location near the flywheel of the shaft.

Allowable shear stress of the shaft of EN24 material is 135MPa.

SAE 6145 (Chromium Vanadium steel):-

SAE 6145 is a fine grained, highly abrasion resistant carbon-chromium alloy steel. Very good shock resistance and toughness are also key properties of this alloy in the heat treated condition. It is used for torsion springs and spring for truck, engine, vehicle parts and shaft.

Composition of SAE 6145 (Chromium Vanadium Steel)

Carbon: 0.43 – 0.48%

Chromium: 0.8 – 1.1%

Manganese: 0.7 – 0.9%

Silicon: 0.2 - 0.35%

Vanadium: >1.5%

Phosphor: <0.4%

For SAE6145 material

Ultimate Tensile Strength, S_{ut}= 1570MPa

Tensile Yield Strength, S_{yt}= 1430MPa

Density ,ρ= 7850 kg/m^3

Temperature C	Young's Modulus MPa	Poisson's Ratio	Bulk Modulus MPa	Shear Modulus MPa
50	2.09e+005	0.29	1.6587e+005	81008

Table no. 6- Properties of SAE6145 material

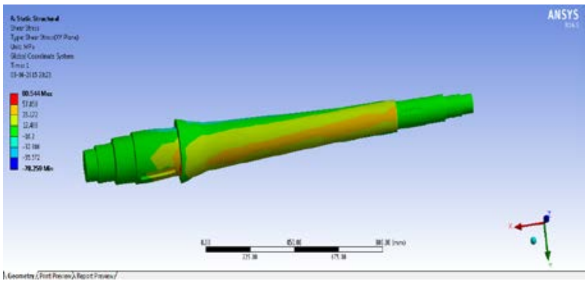

Fig No. 8:- ANSYS report on SAE6145

Now the same moment are applied on the shaft with SAE 6145 and it is observed that maximum shear stress is 80.544MPa which is less than allowable shear stress 211.95MPa.

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SAE 6150 (Chromium Vanadium Steel):-

SAE 6150 is a fine grained, highly abrasion resistant carbon chromium alloy steel and very good shock resistance and toughness also key properties of this alloy in heat treated condition. It is commonly employed in stressed machinery part including shaft, gears, and pinions and also in hand tools components, etc.

Composition of SAE 6150 (Chromium Vanadium Steel)
Carbon: 0.45 – 0.5%
Silicon: 0.12 – 0.35%
Manganese: 0.5 – 0.8%
Chromium: 0.9 – 1.2%
Vanadium: >1.5%

For SAE6150 material
Ultimate Tensile Strength, S_{ut}= 1690MPa
Tensile Yield Strength, S_{yt}= 1200MPa
Density ,ρ= 7700 kg/m^3

Temperature C	Young's Modulus MPa	Poisson's Ratio	Bulk Modulus MPa	Shear Modulus MPa
50	1.9e+005	0.29	1.5079e+005	73643

Table no. 7- Properties of 6150 material

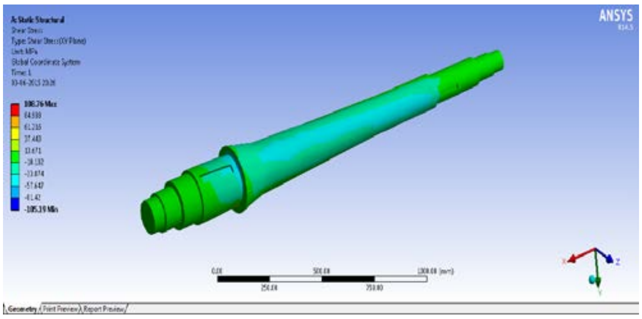

Fig No. 9:- ANSYS report on SAE6150

It is observed that the maximum shear stress value of SAE 6150 is 108.76MPa from ANSYS result and allowable shear stress limit of SAE 6150 is 228.4 MPa.

SAE 4140 (Chromium Molybdenum Steel)

SAE 4140 alloy Steel is chromium, molybdenum alloy steel. It has high fatigue strength, abrasion and impact resistance, toughness and torsion strength etc. It is used extensively in most industry sectors for a wide range of application such as axle shaft, bolts, crankshaft and part lathe, spindle, motor shaft, nut, pinions, pump shaft, worm, etc.

Composition of SAE 4140 (Chromium Molybdenum Steel)

Carbon: 0.36 – 0.44%

Silicon: 0.12 – 0.4%

Manganese: 0.65 – 1.1%

Chromium: 0.75 – 1.2%

Phosphor: 0 – 0.04%

Sulphur: 0 – 0.04%

For SAE4140 material

Ultimate Tensile Strength, S_{ut}= 1300MPa

Tensile Yield Strength, S_{yt}= 1130MPa

Density ,ρ= 7750 kg/m^3

Temperature C	Young's Modulus MPa	Poisson's Ratio	Bulk Modulus MPa	Shear Modulus MPa
50	2.1e+005	0.29	1.6667e+005	81395

Table no. 8- Properties of SAE4140 material

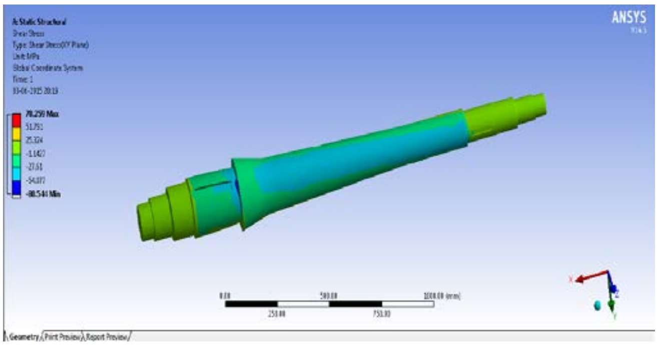

Fig No. 10:- ANSYS report on SAE4140

It is observed that maximum shear stress value of SAE 4140 is 78.259MPa which is less than allowable shear stress limit of SAE 4140 i.e. 175.5MPa.

1.7. RESULT AND DISCUSSION

Failure of shaft is mainly due to the corrosion, fatigue, overload, creep, wear, abrasion, erosion. The diameter of shaft is very less as compared to stress developed on the tooth load of the shaft. So we conclude that shaft failure occurred due to minimum diameter of shaft as compared to stress developed on the shaft.

1.7.1. Comparison of material

Specifications	EN 8	EN 24	SAE 6145	SAE 6150	SAE 4140
Young's Modulus, MPa	210	206	209	190	210
Poisson's ratio	0.26-1	0.29-1	0.27-0.3	0.27-0.3	0.27-0.3
Density, kg/m^3	7850	9490	7850	7700	7750

Table no. 9

1.7.2. Result from ANSYS

Specifications	EN 8	EN 24	SAE 6145	SAE 6150	SAE 4140
Maximum Shear Stress, MPa	95.14	70.129	80.554	108.76	78.259
Allowable shear stress, MPa	81	135	211.95	228.40	175.5

Table no. 10

From above analysis we can conclude that SAE 6150 has minimum shear

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stress value than SAE6145 and SAE4140. Hence SAE 6150 material is safer and it can be used for the manufacturing of the shaft.

The SAE 6150 (Chromium Vanadium Steel) has minimum shear stress value than SAE6145, SAE4140 and existing material. So SAE 6150 best material suggested for manufacturing of shaft because its low shear stress value than allowable shear stress value.

1.9. ADVANTAGES

1. The material SAE 6150 can sustain at high temperature.
2. Heat Treatment process which is annealing makes grains closer, which gives better hardness.
3. The productivity of the plant increases as shaft needs to be replaced by long duration.
4. Time losses occurring due to failure of shaft during maintenance is reducedwhich increases considerable profit.
5. As by employing proposed SAE6150 material rather than earlier EN8 material life cycle of shaft increases.
6. The design of shaft is based on ASME code which is the best theory to design the shaft.
7. System reliability increased with increase in life of the shaft, aand thus the production increases.
8. Maintenance cost is reduced due to avoidance of shaft

failure which is occurring during two, months of repair.

9. The new design proposed reduces the overall stresses induced at flywheel side and the head assembly.

1.10. CONCLUSION

Analysis of shaft is carried out by using analytical method and using ANSYS-14.5 software. Both these methods showed that maximum stresses are generated near the portion of gear. Static loading for equivalent stress is safe and develops bending stress up to 650.243MPa. But maximum shear stress is 95.14MPa which is exceeding allowable shear stress of 81MPa. The diameter of shaft is less as compared to stress acting on tooth. So it can be conclude that shaft is failed due to less diameter.

Materials of shaft are selected from data book and shear stresses acting on the shaft calculated. SAE6150 (Chromium Vanadium Steel) is bestmaterial suggests for manufacturing of shaft because its shear stress value is very less than allowable shear stress.

Alternate ways of solving the above mentioned problem would be to add an extra bearing near the flywheel section which would carry heavy load occurring in that side.

The other way to redesign is to make alternations in the design of the shaft near the taper provided on the flywheel side of the shaft.

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