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# Analysis and Optimization of Centrifugal Casting Machine Shaft

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**ABSTRACT**

**In centrifugal casting machine, a drive shaft is the most important component to run the application, because of shaft failure, factory production line will be stopped. This paper could help the industries working in these areas to improve the life and functional ability of the unit which would lead them in terms of higher productivity, less failure, weight reduction and cost reduction of machine manufacturing.**
**A drive shaft is a rotating shaft that transmits power from the drive motor to the driven shaft with the help of a chain and sprocket mechanism. The centrifugal casting machine operates using the above principle is an old process which some companies still follows. As a result, the optimization is inevitable for reduction of the size and weight of the shaft. In this paper, the study of the existing shaft design analysis and analyze the failure point, in order to develop a new design for the centrifugal casting machine shaft. In this paper, comparison of existing machine shaft and new optimized shaft, has been carried out in following sequences - first the model is prepared in the ANSYS 15.0 workbench after that the analysis has been done in ANSYS 15.0 for comparing the difference such as shear stress, and deflection of the shaft for numerical as well as the software results. This paper can be very useful for optimization of the shaft or the modification of the machine. The proposed works not only carry out design and analysis, but also optimizes, implanting computerized techniques to evaluate deflection and stress analysis. This, in turn will decide the criteria for material selection and dimensional parameters. Thus, the work contributes to reduction of weight of casting machine, reduction of the manufacturing cost and delivering better results.**

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## I. Introduction

Centrifugal casting is an advanced slip casting technique; which can be used for a casting body which has axial symmetry. This method is commonly used in casting of molten metal, also shall be used in casting of slurry which contains ceramic powder. The method involves pouring of molten metal into a cylindrical mould rotating about its axis of symmetry. In this method casting involves rotating the mould in a horizontal axis. Horizontal centrifugal casting is preferred for the tube geometry as the diameter for such geometry is less than their length. In centrifugal casting machine drive shaft is most important component to run the application, because of shaft failure it tends to stop the production line of the components.

A drive shaft is a rotating shaft that transmits power from the motor to the shaft with the help of chain and sprocket mechanism. Drive shaft must operate in high initial torque. The centrifugal casting machine is an old process and many companies carry this process, so the optimization is necessary and reduction of the weight of the shaft is very important. In this paper, existing shaft design analysis and the failure point for the new optimised design is studied.

## I. PROBLEM STATEMENT

Layout of centrifugal casting machine shaft carries two rollers which are fixed to shaft at mention distances B, C and supported on bearings A, D. Power is supplied to the shaft by means of vertical chain at pulley E which is further transmitted to rollers B and C which carries horizontal transmission, the maximum torsion in chain sprocket at E is 962 N. The angle of wrap is 60˚ & coefficient of friction 0.24. The shaft is made of plain carbon steel 30 C8 (Syt=400 N/mm²) factor of safety is 1.2, according to existing design major diameter of shaft is 95 mm, above data we can cross check the shaft diameter by design is safe, according to ASME code optimized shaft design having less length than existing shaft length has been done, also comparison of existing machine shaft and new optimized shaft on ANSYS 15.0 has been done and difference such as bending stress, shear stress, and deflection of the shaft for numerical as well as the software result are shown.

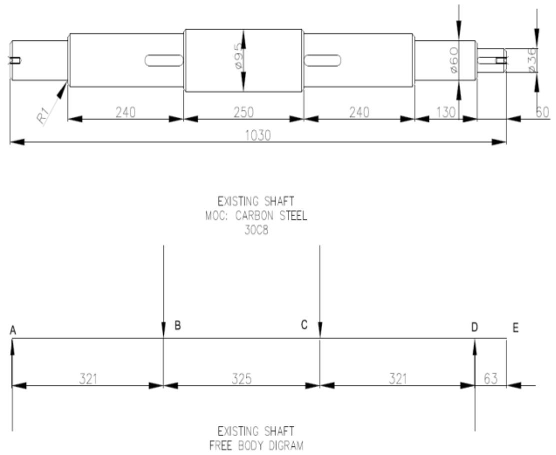



Fig 01: Existing Shaft design with free body diagram

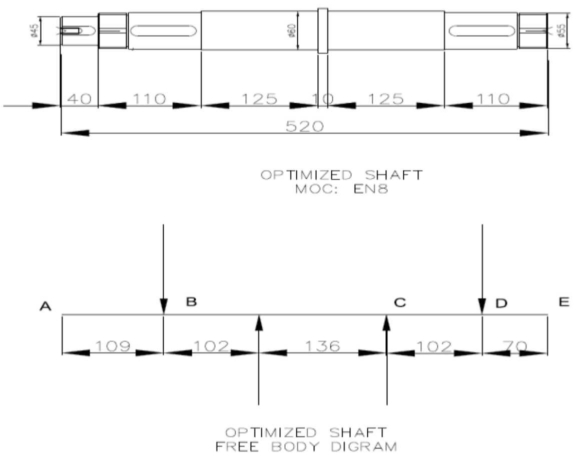



Fig 02: Optimized Shaft design with free body diagram

## II. DESIGN CONSTRAINTS

The shaft rotates at a constant speed about its longitudinal axis. It has a uniform, circular cross section. The shaft is perfectly balanced. The input revolution per minute is 980 and length is 1030mm major diameter is 95mm; main motor speed is 1440 rpm. Torsion is the primary load carried by the drive shaft. The shaft must be cross checked for torsion strength to carry the torque without failure.

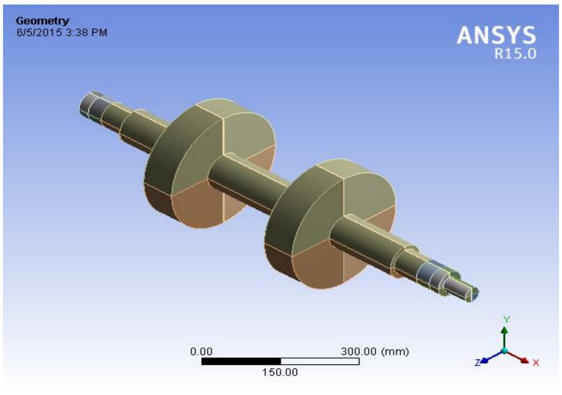



Fig 03: Existing Shaft model-ANSYS R15.0

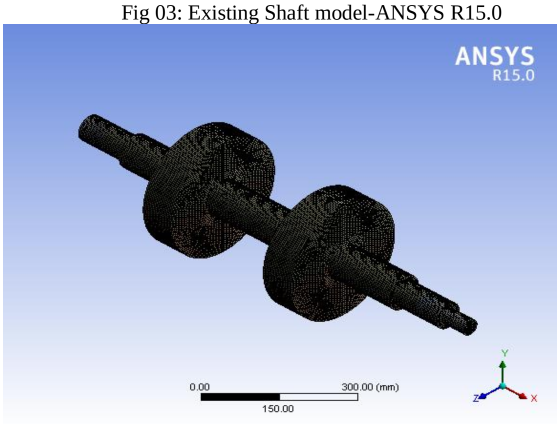



Fig 04: Existing Shaft model Meshing -ANSYS R15.0

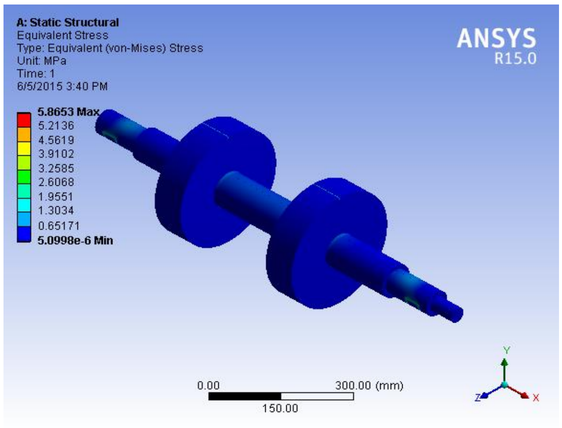



Fig 05: Existing Shaft Equivalent Stress (Von –Mises)

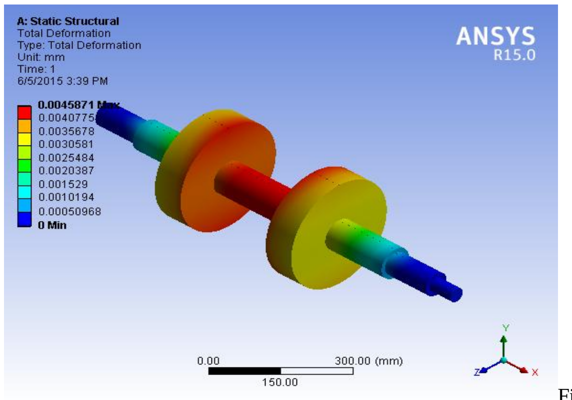



Fig 06: Existing Shaft Deformation

## III. THEORETICAL AND ANSYS RESULTS CALCULATION

First we analyse the existing shaft design diameter. The shaft diameter is calculated on the bases of two types here we can apply according to the maximum shear stress theory, Shaft Design under ASME Code:

TABLE I
Material Properties Carbon Steel 30C8

| Mechanical Properties | Symbol | Units | Value |
|---|---|---|---|
| Density | E | g/cc | 7.85 |
| Yield Stress | Syt | N/mm² | 400 |
| Poisson's ratio | μ | | 0.287 |
| Hardness | H | HBW | 120 |
| Elongation | E | % | 16 |
| Thermal conductivity | k | W/m | 58.6 |

The permissible shear stress is given by

$= \left(\frac{Ssy}{FOS}\right)\ \tau = \left(\frac{0.5\ Syt}{FOS}\right)$ τ=166.66 N/mm²

Bending Factor (kb) =1.5

Tensional Factor (kt) = 1

Maximum Shear Stress, τ (max) =85.20N/mm²

Maximum bending Moment (Mb) ref to fig 01 = 465127N/mm

Maximum torsional Moment (Mt) = 1153.65; FOS Consider is 2

Shaft Diameter calculated as per maximum shear stress theory $\tau(\max) = \left(\frac{16}{\pi d3}\right)\sqrt{(Mb)2 + (Mt)2}$

Diameter of the Shaft: 63mm

Existing shaft diameter is 95mm.

The Existing shaft design is safe as per the above data and numerical calculation.

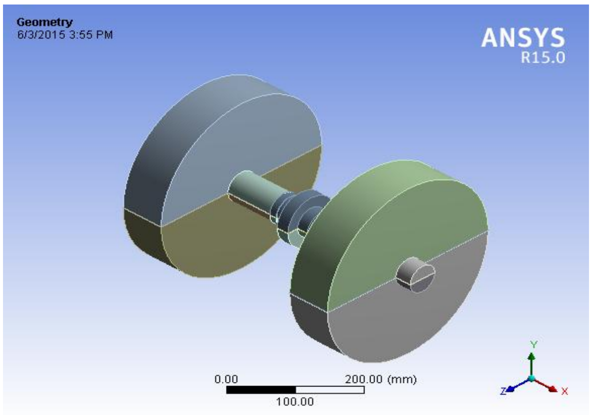



Fig 07: Optimized Shaft Model

Fig 08: Optimized Shaft Model Meshing

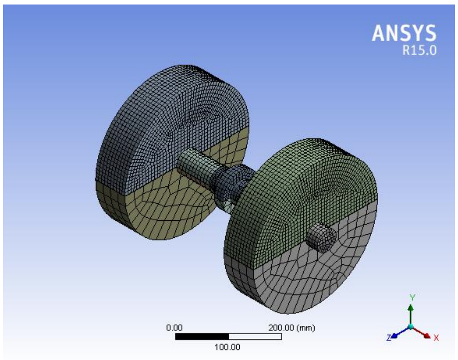



Fig 09: Optimized Shaft Equivalent Stress (Von –Mises)

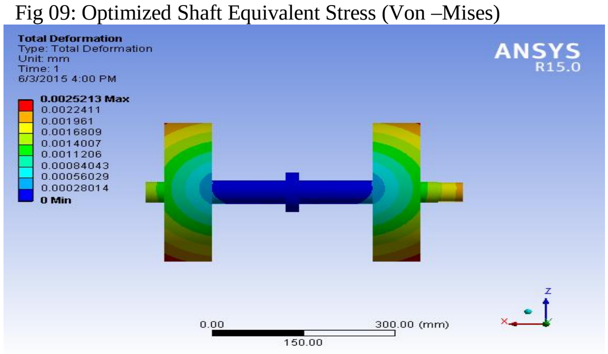
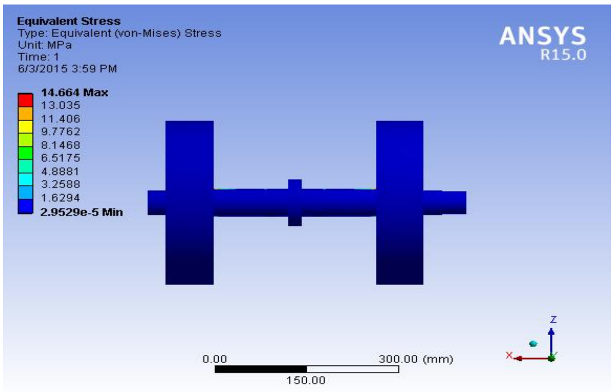



Fig 10: Optimized Shaft Deformation




Optimized shaft diameter calculation, Here we calculate shaft diameter as per above given data. Our objective is to reduce the shaft weight by considering dimension analysis

TABLE II
Material Properties Carbon Steel EN8

| Mechanical Properties | Symbol | Units | Value |
|---|---|---|---|
| Density | E | g/cc | 7.85 |
| Yield Stress | Syt | N/mm² | 465 |
| Poisson's ratio | μ | | 0.295 |
| Hardness | H | HBW | 178 |
| Elongation | E | % | 16 |
| Thermal conductivity | k | W/m | 58.6 |

Yield Strength (Syt) =465 N/mm²

The permissible shear stress is given by= $\left(\frac{Ssy}{FOS}\right)$ $\tau = \left(\frac{0.5\ Syt}{FOS}\right)$ τ=193.75N/mm²
Bending Factor (kb) =1.5
Tensional Factor (kt) = 1

TABLE III
Value of shock and fatigue factor

| Application | Bending Factor (kb) | Tensional Factor (kt) |
|---|---|---|
| Load Gradually Applied | 1.5 | 1 |
| Load Suddenly Applied (Minor Shock) | 1.5-2.0 | 1.0-1.5 |
| Load Suddenly Applied (Heavy Shock) | 2.0-3.0 | 1.5-3.0 |

Maximum Shear Stress, τ (max) =125 N/mm²
Maximum bending Moment (Mb) ref to fig 02 = 215969 N/mm,
Maximum tensional Moment (Mt) = 1153.65; FOS Consider is 2
Shaft Diameter calculated as per maximum shear stress Theory $\tau(\max) = \left(\frac{16}{\pi d3}\right)\sqrt{(Mb)2 + (Mt)2}$
Diameter of the Shaft: 53mm
Optimized shaft diameter is 60mm.The shaft design diameter is safe as per the above data and numerical calculation.
By comparing both the results, according to maximum shear stress theory the deflection of optimized shaft have better results as compared to existing shaft.

## IV. Meshing and ANSYS simulation:

Meshing: We have selected area mesh for the meshing with the element size of 8, which will provide us fine meshing. We have selected hex mesh element for accurate and uniform meshing of component. The meshing is the method in which the geometry is divided in small number of elements. This meshing of shaft is as shown in fig 04, fig 08.

Static analysis: A static analysis can be either linear or non-linear. All types of non-linearity's are allowed such as large deformations, plasticity, creep, stress stiffening, contact elements etc. this chapter focuses on static analysis. A static analysis calculates the effects of steady loading conditions on a structure, while ignoring inertia and damping effects, such as those carried by time varying loads. A static analysis is used to determine the displacements, stresses, strains and forces in structures or components caused by loads that do not induces significant inertia and damping effects. A static analysis can however include steady inertia loads such as gravity, spinning and time varying loads. If these values exceed the above allowable values then component is going to fail. Hence static analysis is necessary. The results show in fig 05 fig 06, fig 09, and fig 10.

## Conclusions

By comparing both the results, according to the maximum shear stress theory and deflection theory the optimized shaft have better results as compared to existing shaft.

TABLE IV
Result Comparison of Existing Shaft

| Description | Sym | UOM | Existing 30C8 | |
|---|---|---|---|---|
| | | | Analytic | FEA |
| Diameter | D | mm | 95 | 95 |
| Length | L | mm | 1030 | 1030 |
| Weight | W | Kg | 57 | 57 |
| Deflection | δ | mm | 0.0086 | 0.00458 |
| Equ. Stress (Von-Miss) | ϭ | Mpa | 7.65 | 5.8653 |
| Maximum Shear Stress | τ | Mpa | 3.857 | 3.2258 |

TABLE V
Result Comparison of Optimized Shaft

| Description | Sym | UOM | Optimized EN8 | |
|---|---|---|---|---|
| | | | Analytic | FEA |
| Diameter | D | mm | 95 | 60 |
| Length | L | mm | 1030 | 520 |
| Weight | W | Kg | 11.54 | 11.54 |
| Deflection | δ | mm | 0.0038 | 0.00252 |
| Equ. Stress (Von-Miss) | ϭ | Mpa | 14.55 | 11.66 |
| Maximum Shear Stress | τ | Mpa | 10.5 | 9.88 |

So optimized shaft diameter is 60 mm. Comparing tables IV and V, the EN8 is beneficial then the existing 30C8 shaft material. Also 80% weight reduction is done. Shaft length is optimized by 50%.

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