DESIGN ANALYSIS OF COMPONENTS OF CONICAL DUAL BLADE SINUSOIDAL MIXER

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Abstract : Mixing or Stirring process constitutes an important part in process engineering especially in paint manufacture industry. After the sizing , design development of conical dual blade sinusoidal mixer , the critical components of the dual blade sinusoidal mixer have being designed and analysis of the components is done The paper deals with design development of the components as the worm shaft , worm , worm gear, main shaft and the secondary agitator shaft. The theoretical design of the components has been done after appropriate selection of materials, followed by solid modelling of the components using Unigraphix Nx. The static structural analysis of the components has been done using Ansys work bench 16.0. The design of the component suing the theoretical method is thus validated using the ansys results. The structural strength of the critical components of the dual blade sinusoidal mixer is thus ensured.

Keywords : Ionic paint , dual blade sinusoidal cone mixer , worm , worm shaft , worm gear, secondary agitator shaft

I. INTRODUCTION

The sizing, design of conical dual blade sinusoidal mixer is successfully done in the previous work and the dimensions of the stirrer and the tank have being determined. Estimation of the power requirements of the mixer and the motor power is selected with factor of safety of 1.5 and 50watt power motor is selected and the gear ratio of worm gear box is revised to 1:80-. The arrangement of the blades of the constituting of an worm gear pair. As the designed rpm of 20 the system components are designed for the design torque corresponding to the designed speed

Input data :

Motor power = 50 watt , 8000 rpm

Reduction ratio of pulley system = 5

Worm shaft torque = 0.3 N-m

Worm gear ratio = 1:80

Worm gear torque = 23.9 N-m

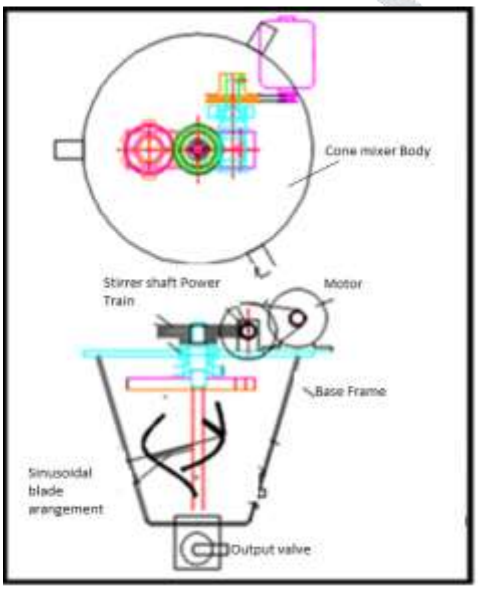

Fig. 1 : Schematic of the Conical dual blade sinusoidal mixer

II. DESIGN OF WORM SHAFT

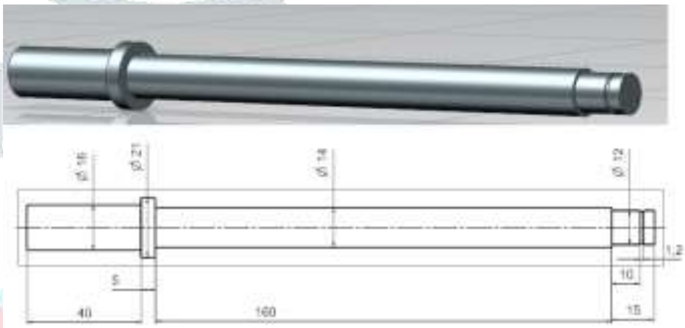

Fig. 2 : Design of Worm Shaft

Table 1. MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.24)

DESIGNATION	ULTIMATE TENSILE STRENGTH (MPa)	YEILD STRENGTH (MPa)
EN24	800	680

As per ASME code $fs_{allowable}$= 104 Mpa

Check for torsional shear failure of shaft

$$Te = \frac{\Pi fs\, d^3}{16}$$

fs_{act} = 0.884079496/mm^2 As; $fs_{act} < fs_{all}$

The worm shaft is safe under torsional load

A. Analysis of Worm shaft

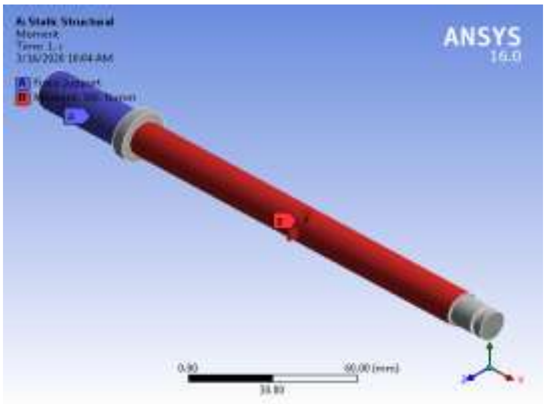

Fig. 3 : Analysis of Worm shaft

The component geometry was developed using Unigraphix NX and step file was used as input to ansys workbench, the boundary conditions and loading was done as shown in figure above.

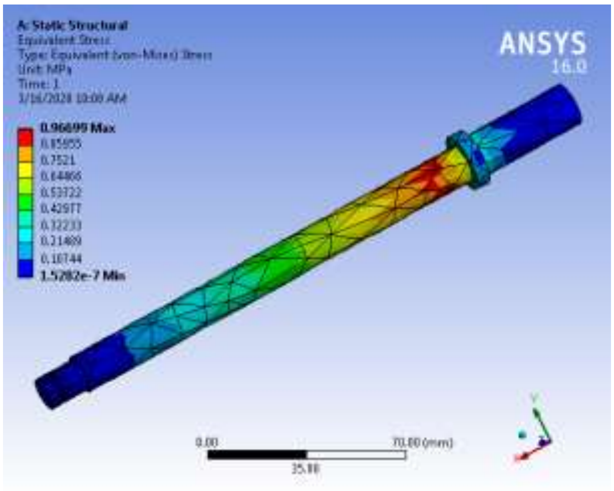

Fig. 4 :

The maximum von mises stress developed is 0.969 Mpa which is well below the allowable stress hence the worm shaft is safe under torsional load

III. DESIGN OF WORM

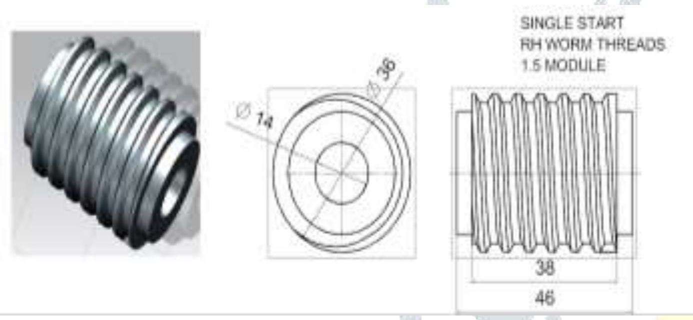

Fig. 5 : Design of Worm

Table 2. MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.18)

DESIGNATION	ULTIMATE TENSILE STRENGTH (MPa)	YEILD STRENGTH (MPa)
20MnCr1	1000	760

As per ASME code $fs_{allowable}$= 110Mpa

Check for torsional shear failure of shaft

$Td = \Pi/16 \times fs_{act} x(D^4 - d^4) /D$

fs_{act} = 0.233236 Mpa As; $fs_{act} < fs_{all}$

The worm is safe under torsional load

B. Analysis of Worm:

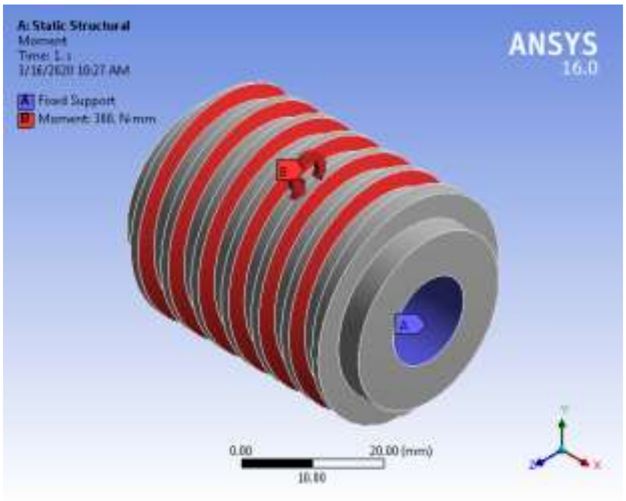

Fig. 5 : Analysis of worm

The component geometry was developed using Unigraphix NX and step file was used as input to ansys workbench, the boundary conditions and loading was done as shown in figure above.

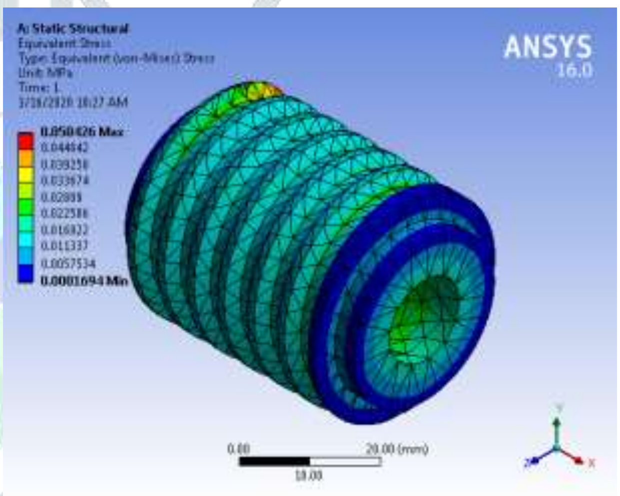

Fig. 6 :

The maximum von mises stress developed is 0.056 Mpa which is well below the allowable stress hence the worm is safe under torsional load

IV. DESIGN OF WORM GEAR

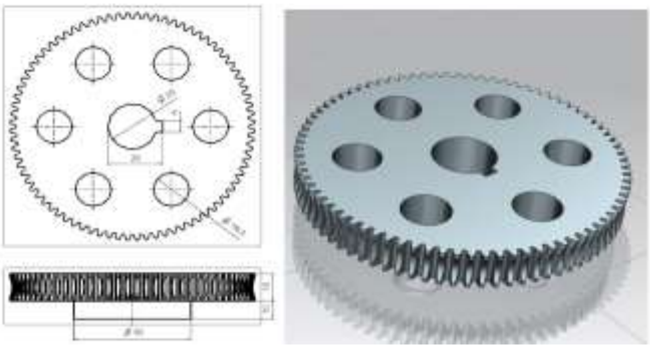

Fig. 7 : Design of worm gear

Table 3. MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.40)

DESIGNATION	ULTIMATE TENSILE STRENGTH (MPa)	YEILD STRENGTH (MPa)
Cast steel	550	460

As per ASME code $fs_{allowable}$= 110Mpa
Check for torsional shear failure of shaft
$Td = \Pi/16 \times fs_{act} x(D^4 - d^4) /D$
fs_{act} = 1.826 Mpa As; $fs_{act} < fs_{all}$
The worm gear is safe under torsional load

C. Analysis of Worm Gear:

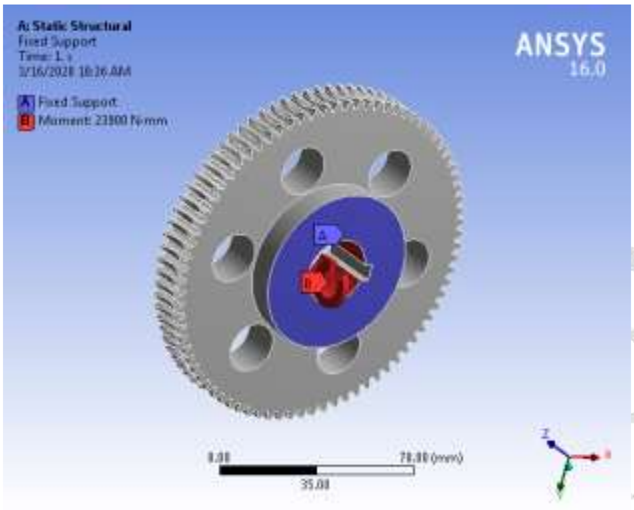

Fig. 8 : Analysis of Worm Gear

The component geometry was developed using Unigraphix NX and step file was used as input to ansys workbench, the boundary conditions and loading was done as shown in figure above.

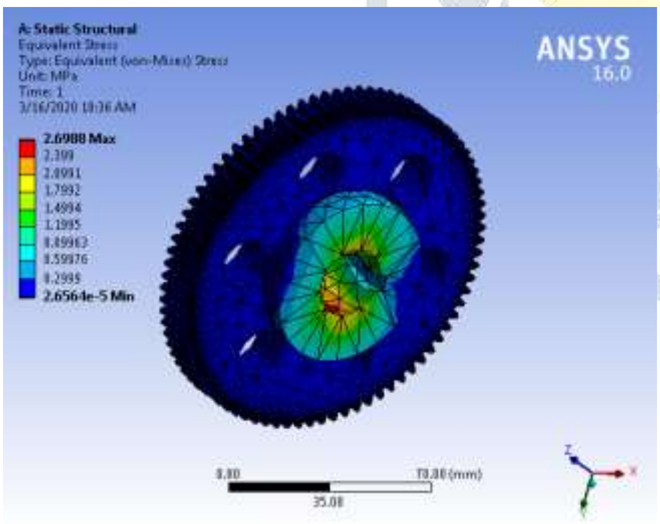

Fig. 9 :

The maximum von mises stress developed is 0.056 Mpa which is well below the allowable stress hence the worm is safe under torsional load

V. DESIGN OF MAIN SHAFT

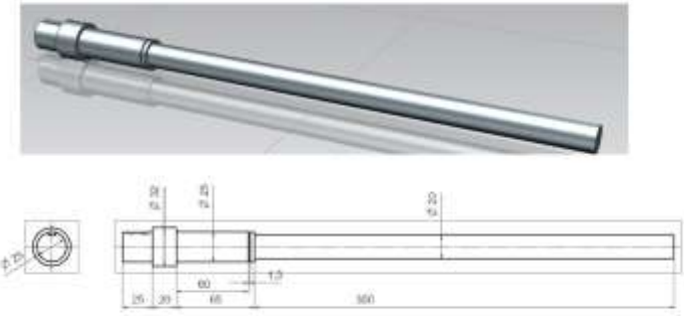

Fig.10 : Design of main shaft

Table 4. MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.24)

DESIGNATION	ULTIMATE TENSILE STRENGTH (MPa)	YEILD STRENGTH (MPa)
EN24	800	680

As per ASME code $fs_{allowable}$= 104 Mpa
Check for torsional shear failure of shaft

$$Te = \frac{\Pi fs\, d^3}{16}$$

fs_{act} = 15.21323997 MPa As; $fs_{act} < fs_{all}$
The worm shaft is safe under torsional load

D. Analysis of Main shaft :

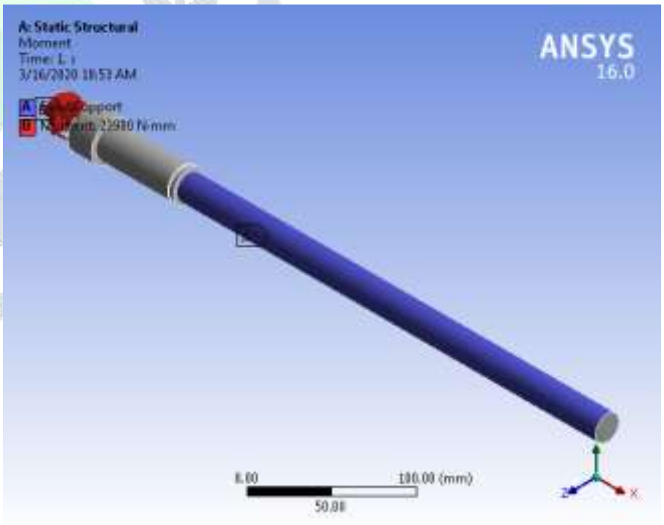

Fig.11 : Analysis of Main shaft

The component geometry was developed using Unigraphix NX and step file was used as input to ansys workbench, the boundary conditions and loading was done as shown in figure above.

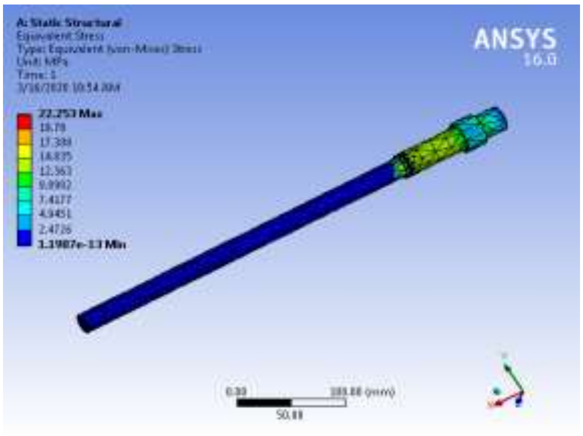

Fig.12 :

The maximum von mises stress developed is 0.969 Mpa which is well below the allowable stress hence the worm shaft is safe under torsional load

VI. DESIGN OF SECONDARY AGITATOR SHAFT

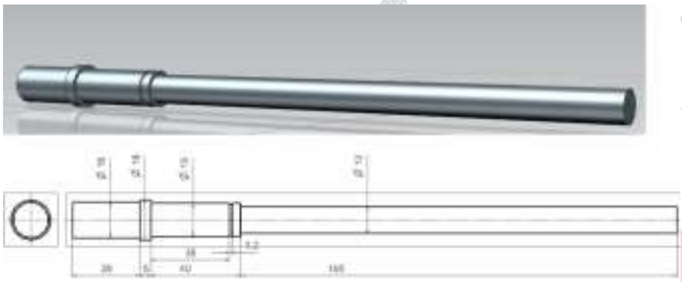

Fig.13 : Design of Secondary Agitator Shaft

Table 5. MATERIAL SELECTION : -Ref :- PSG (1.10 & 1.24)

DESIGNATION	ULTIMATE TENSILE STRENGTH (MPa)	YEILD STRENGTH (MPa)
EN24	800	680

As per ASME code $fs_{allowable}$= 104 Mpa

Check for torsional shear failure of shaft

$$Te = \frac{\Pi fs\, d^3}{16}$$

fs_{act} = 5.89386331= MPa As; $fs_{act}<fs_{all}$

The secondary agitator shaft is safe under torsional load

E. Analysis of Secondary agitator shaft :

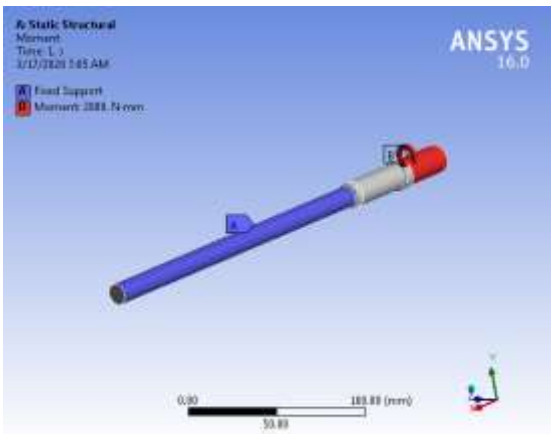

Fig.14 : Analysis of secondary agitator shaft

The component geometry was developed using Unigraphix NX and step file was used as input to ansys workbench, the boundary conditions and loading was done as shown in figure above.

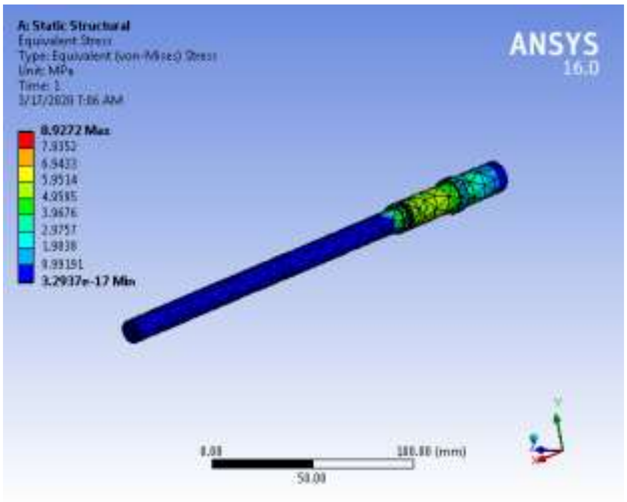

Fig.15 :

The maximum von mises stress developed is 0.969 Mpa which is well below the allowable stress hence the worm shaft is safe under torsional load

Result and Discussion

The following results for design and analysis of critical components of the Conical Dual blade sinusoidal mixer

1. Maximum stress induced by theoretical method in the worm shaft is 0.88 Mpa and the maximum Von-mises stress determined using Ansys Workbench is 0.0.05 Mpa , both are well below the allowable limit hence the worm shaft is safe under torsional load.
2. Maximum stress induced by theoretical method in the worm is 0.23 Mpa and the maximum Von-mises stress determined using Ansys Workbench is 0.05 Mpa , both are well below the allowable limit hence the worm shaft is safe under torsional load.
3. Maximum stress induced by theoretical method in the worm gear is 1.86 Mpa and the maximum Von-mises stress determined using Ansys Workbench is 2.692 Mpa , both are well below the allowable limit hence the worm shaft is safe under torsional load.
4. Maximum stress induced by theoretical method in the main shaft is 15.23 Mpa and the maximum Von-mises stress determined using Ansys Workbench is 22.283 Mpa , both are well below the allowable limit hence the worm shaft is safe under torsional load.
5. Maximum stress induced by theoretical method in the secondary agitator shaft is 5.89 Mpa and the maximum Von-mises stress determined using Ansys Workbench is 8.92 Mpa , both are well below the allowable limit hence the worm shaft is safe under torsional load.

Conclusion

The sizing, design analysis critical components of conical dual blade sinusoidal mixer is successfully done and the dimensions of the components have being determined. Estimation of the maximum stress induced in the components of the mixer have being determined by both theoretical method as well as using Ansys Work bench and the results indicate that the maximum stress values are well below the permissible limit hence the parts are safe under given system of loads.

Future Scope

The sinusoidal blades geometry determination and structural analysis of the twin blade in individual system as well as in the integrated form to the main shaft will be done in the subsequent work. Similarly the geometry of the secondary agitator blade in the individual system as well as in the integrated form to the secondary agitator shaft will be done in the subsequent work

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