Analytical Calculations for Design Optimization of Stirrer Shaft of Butter Extraction Machine

Ms. P. A. Borkar[1] S. N. Gandhare[2] M. S. Jibhakate[3] A. N. Ingole[4]
[1]M. Tech. Student [2,3,4]Assistant Professor
[1,2,3,4]Department of Mechanical Engineering
[1,2,3]Datta Meghe Institute of Engineering, Technology & Research Salod (H), Wardha, MH, India
[4]Bapurao Deshmukh College of Engineering & Technology Sewagram, Wardha. MH, India

Abstract— In this paper, shaft employed as a Butter extractor which is rotated at 75 to 100 rpm is studied. Considering the system, torque acting on rotating stirrer shaft which is rotated clockwise and anticlockwise direction and reversible direction stresses is induced on shaft. The One end of stirrer shaft is fixed and other end is freely rotating (over hanging) during butter extraction, due to centrifugal force jerk acted on shaft.

Key words: RPM, Torque, Overhanging, Centrifugal Force

I. Analytical Calculation of Stirrer Shaft

A. *Design of Shaft*

1) *Condition: (Design is not safe)*

a) Input

Speed N= 75 rpm

Diameter of Pulley Dp1=152mm

Kl =1.25 for driven machinery is line shafts or transmission shafts

Motor Power P =700 W Ratio=3

Distance from bearing 1 to bearing 2,

L= 240 mm

Distance from bearing 1 to pulley 1, x=120 mm

Shaft diameter D =25 mm

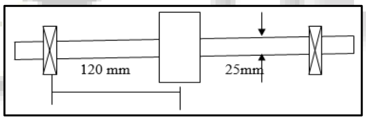

Fig. 1: Dimensions

$$P = \frac{2\,\pi NT}{60}$$

$$T= 167.113 \text{ N-m}$$

– Force acting by pulley

$$T= (F_1-F_2)\ \frac{Dp}{2}$$

$$\text{Belt tension ratio (3)} =\frac{F1}{F2},$$

$$F_1= 3\ F_2 \text{ N}$$

$$167.113 = (3\ F_2 - F_2)\ \frac{152}{2}$$

$$F_2= 1.099 * 10^3 \text{ N}$$

$$F_1 = 3.298 * 10^3 \text{ N}$$

– Total force acting

$$F=F_1 + F_2$$

– Force summery

– Force acting due to pulley

$$F_h = F = 4.398*10^3$$

– Calculation for horizontal bending moment

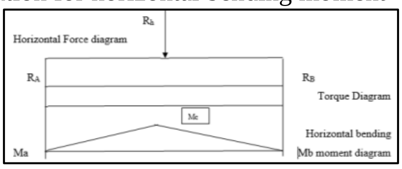

Fig. 2: horizontal force diagram

– Taking moment at A

$$R_B= 2.199*10^{3}$$

$$R_A+R_B - F_h$$

$$R_A= 2198.85 \text{ N}$$

– Bending moment at A & B =0

– Bending moment at pulley

$$Mc =2.693 * 10^{5}\text{N}$$

$$M=2.639* 10^{5}\text{N}$$

– Material of rotating shaft is SAE 1030.

(D. D. B. - B.D. Shiwalkar)

$$Syt = 218\text{Mpa}$$

$$Sut = 379\text{Mpa}$$

$$E = 206*10^3 \quad \mu =0.3$$

Consider the values of shock and fatigue factors

Where, Kt = Shock and fatigue factor for Twisting Moment

Kb= Shock and fatigue factor for Bending Moment

Kt=1.25 and Kb=1.75 (D.D.B – B. D. Shiwalkar)

τ_1 =0.3 Syt and τ_2 =0.18 Sut

τ max= 68.22 Mpa

τ min= 65.4 Mpa

Due to Keyway stress is reduced by 25%

$$\tau_{max}= \tau*0.75$$

$$\tau \text{ max}= 51.165 \text{ Mpa}$$

– Equivalent Torque

$$Te =\sqrt{(Kt * T)^2 + (Kb * M)^2}$$

$$Te= 5.068 *10^5\text{N-mm}$$

$$T\,e = \frac{\pi}{16} * Zmax * d3$$

– Diameter of shaft is 25 mm

$$\tau_{max}= 165.19 \text{ N/mm}^2$$

$$51.165 \text{ Mpa} \leq 165.19 \text{ Mpa}$$

Therefore, τ max allowable $\leq \tau$ max

Hence design is not safe

2) *Design of shaft (Design is safe)*

a) Input

– N=75 rpm
– belt tension ratio=3
– kl =1.25 (D.D.B – B. D. Shiwalkar)
– P=700 rpm
– Dp =152 mm
– L=240 mm
– X=160 mm

b) Design

$$P =\frac{2\,\pi NT}{60*Kl}$$

$$T=111.408 \text{ N-m}$$

– Force acting due to pulley

$$T=(F_1-F_2)\ \frac{Dp}{2}$$

$$F_1= \text{belt tension ratio} * F_2$$

$$F_2=7.329*10^3 \text{ N}$$

$$F_1 = 2.199*10^3\ N$$

– Total force acting

$$F = F_1 + F_2$$
$$F = 2.932*10^3\ N$$

– Force Summery

$$F_h = 2.932*10^3\ N$$
$$R_A = 977.267\ N$$
$$R_B = 1.955*10^3\ N$$

– Moment at bearing is zero $M_A = M_B = 0$

$$M_c = 1.564*10^5\ N$$

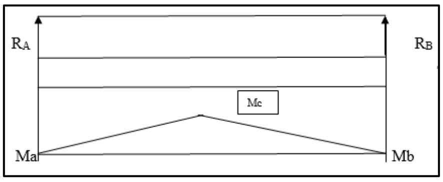

Fig. 3: Horizontal Force Diagram

– Resultant bending Moment

$$M = 1.564*10^5\ N$$

Define Material for Modified shaft (SAE 1030) (D.D.B – B. D. Shiwalkar)

$$Syt = 296\ Mpa$$
$$Sut = 527 Mpa$$
$$E = 205*10^3$$
$$\mu = 0.3$$
$$\tau_1 = 0.3\ Syt \text{ and } \tau_2 = 0.18\ Sut$$
$$\tau_{max} = 94.86 Mpa$$
$$\tau_{min} = 88.88 Mpa$$

Due to keyway shear stress is reduced by 25%

$$\tau_{max} = \tau*0.75 = 71.145\ Mpa$$

– Equivalent Torque

$$Te = \sqrt{(Kt*T)^2 + (Kb*M)^2}$$
$$Te = 3.07*10^5 N\text{-}mm$$
$$Te = \frac{\pi}{16}*\tau*d^3$$
$$d = 28.012\ mm$$
$$d = 30\ mm$$

– Radius of gyration

$$K = \frac{D}{4} = 7.5$$

– Distance from critical section

$$\frac{x}{k} = 21.33 \qquad \alpha = 1.28$$
$$Te = \frac{\pi}{16}*\tau*d^3$$
$$\tau_{max} = 57.91\ Mpa$$
$$\tau_{max} < \tau_{allowance}$$

Hence Design is Safe

– Deflection in shaft

$$I = \frac{\pi}{16}*\tau*d^4 = 3.976*10^4\ mm^4$$
$$\frac{d2y}{dx2} = \frac{M}{EI}\,dx = 0.25$$

– Slope of shaft

$$\frac{dy}{dx} = \frac{M}{EI}\,dx = 0.0031$$
$$\sigma_t = \frac{32\ M}{\pi*D3} = 58.989\ Mpa$$
$$\sigma_t = 58.989\ Mpa$$
$$\tau = \frac{16*Te}{\pi*d3} = 57.915\ Mpa$$
$$\tau = 57.915\ Mpa$$
$$\tau_{eq} = \sqrt{\sigma t^2 + \tau^2} = 82.667\ Mpa$$
$$\tau_{eq} = 82.667\ Mpa$$

– Maximum principal tensile stress

$$\sigma_{t\ max} = \frac{\sigma t}{2} + \frac{1}{2}\left(\sqrt{\sigma t^2 + 4\tau^2}\right) = 94.487\ Mpa$$

– Maximum principal Shear stress

$$\tau_{max} = \frac{1}{2}\left(\sqrt{\sigma t^2 + 4\tau^2}\right) = 64.993\ Mpa$$

– Endurance Strength

$$A = 1 \quad Kr = 0.702 \quad K\theta = 1 \quad Kz = 1\ Ks = 0.45 \quad Kf = 1.73$$
$$Sef = 225$$
$$Se = \frac{A*Kr*K\theta*Kz*Ks}{Kf}*Seb = 41.085$$

– Factor of Safety

$$\eta 1 = \frac{Syt}{\sigma t} = 3.133$$
$$\eta 2 = \frac{Syt}{2*\tau\ max} = 2.08$$

– Infinite Life = 50 rpm

$$Ne = 5*365*8*60*60*\frac{50}{60} = 4.38*10^7$$
$$\sigma_t = Sut + 122.697 = 649.697\ Mpa$$
$$b = \frac{\log(\frac{\sigma t}{Se})}{\log(2Ne)} = -0.151$$
$$f = \frac{\sigma t}{Sut}*((2.10)^3) = 0.881$$
$$a = \frac{f(Sut)2}{Seb} = 958.32$$
$$Na = \left(\frac{\sigma eq}{a}\right)\frac{1}{b} = 1.118*10^7$$
$$Fr = \frac{Ne}{Na} = 3.918$$

II. CONCLUSION

Based on FEM results it is found that Min stresses found at the End junction of stirrer shaft because of change of Diameter and material of shaft. After rechecking, stress levels reduced by major amount and fatigue life are calculated by Simulation and they found to be very much within the limits.

In order to eliminate the problem of Stirrer Shaft of Traditional Butter Extraction Machine due to the stresses developed during its operation, the Stirrer Shaft had been analyzed considering different parameters like Diameter, materials and tool life of stirrer shaft using ANSYS and analytical calculation it is found that;

Diameter of stirrer shaft should be increased from the end (from 25mm to 30 mm) to minimize the deflection in shaft.

So, it is concluded that analytical method and change in dimension of stirrer shaft, shows the improvement in efficiency of butter extraction process.

REFERENCES

[1] V. S. Khangar1, Dr. S. B. Jaju2" A Review Of Various Methodologies Used For Shaft Failure Analysis" International Journal of Emerging Technology and Advanced Engineering, Volume 2, Issue.6, June-2012.

[2] Sandeep Gujaran1 and Shivaji Gholap2, "Fatigue Analysis of Drive Shaft" International Journal Of Research In Aeronautical And Mechanical Engineering, Vol.2 Issue.10, Pgs: 22-28, October 2014.

[3] Sumit P.Raut1, Laukik P.Raut2 "A Review Of Various Techniques Used For Shaft Failure Analysis." IJSRET, Volume 2, Issue 2, Feb-Mar 2014

[4] Yézouma Coulibaly1, Stéphane Ouédraogo1 and Nathalie Niculescu2 "experimental study of shea butter extraction efficiency using a centrifugal process." ARPN Journal of Engineering and Applied Sciences, Vol.4 Issue.6, August- 2009.

[5] Tomáš Jirout, František Rieger "Impeller Design For Mixing Of Suspensions" Czech technical university in

prague, faculty of mechanical engineering department of process engineering (2011)1144-1151.

[6] T. Lassen1 and A. Spagnoli2, "Fatigue Crack Paths in Shafts Subjected to Bending and Torsion" University of Stavanger, Norway – tom.lassen@uis. No 2 University of Parma, Italy – spagnoli@unipr.it.