International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 8 Issue 06, June-2019

Design and Fabrication of Portable Industrial Mixer

Mohtasim. A. Mapkar, Prasad. A. More, Shreyans. D. Mehta
Sinhgad Academy of Engineering,
Department of Mechanical Engineering, Kondhwa
(Bk), Pune, India

***Abstract*: In mixing industry, different mixers are required for mixing at various stations, for applications having minimum process time and involves use of big tanks for mixing, it is economical to equip a mixer that facilitates transfer from one station to another than the tank. This work is aimed at achieving the required portability without compromising the process requirement and achieving economical mixing in the industry. In this work, an industrial mixer is designed to ensure hassle free mixing at various stations and portability in operation. To facilitate portability, the industrial mixer is designed with certain degrees of freedom like longitudinal/axial motion of mixer for loading and unloading, traversing ability to operate between different mixing stations, facility to adjust the impeller location from the bottom, etc. In this dissertation, different components required for the industrial mixer such as shaft, im peller, motor mount and frame were modelled using CAD software (Solidworks). Stress analysis of the components was carried out using FEA software package (Solidworks Simulation).**

I. INTRODUCTION

Mixing equipment must be designed for mechanical and process operation. Although mixer design begins with a focus on process requirements, the mechanical design is essential for successful operation. Usually, a competent manufacturer of mixing equipment will take responsibility for the mechanical design. However, process conditions, such as impeller operation near a liquid surface, can impose severe mechanical loads. Similarly, the process environment will influence the selection of a motor enclosure. In many ways the process requirements can have a direct impact on the mechanical design. In other ways, such as the natural frequency of a mixer shaft, appropriate mechanical design must be determined by the equipment designer.

Although normally viewed as a single piece of equipment, like a pump, the typical mixer is composed of several individual components, such as a motor, gear reducer, seal, shaft, impellers, and tank, which is often designed and purchased separately. Although highly customized for many applications, most mixers are a combination of standard components, sometimes with modifications, and often with unique characteristics, such as shaft length.

Generalizations, especially for mixers, can misrepresent individual situations, but some features are common to the largest number of mixers built worldwide. The most common motive force for a mixer is an electric motor, so a knowledge of standard motor characteristics is useful. Most mixers operate at or below typical motor speeds, so some type of speed reduction is common. Speed reduction can be accomplished with several different types of gears, usually in enclosed housings, or with belts and sheaves. Besides speed reduction, antifriction bearings are found in all types of rotating equipment. Some type of seal around the rotating shaft is required for closed-tank operation and the type depends on degree of seal required, operating pressure, and operating temperature.

The shaft for a mixer, especially a large one, involves significant mechanical design, partly because of the myriad of shaft lengths, impeller sizes, and operating speeds, and partly because both strength and rigidity are necessary for a successful design. The combination of custom process and mechanical design necessary for mixers is unique for chemical process equipment. Mechanical design does not end with the shaft, since strength and practical issues remain for the impeller.

Another part of mixer design is the tank in which the mixer is used, since tank dimensions influence mixer features, especially shaft length. Conversely, a mixer requires tank features, such as baffles, support strength, and other tank internals. Materials of construction, although most commonly metal alloys for mixers, depend on process chemistry and operational requirements.

The second step in design sequence is mechanical design of mixer components. The fundamental approach is straightforward, design for power (torque and speed), then shaft loads, and finally mixer dynamics. For larger systems above 100 HP it may be prudent to perform a mixer system modal analysis (FEA) to avoid unexpected interactions. General test procedure and design methodology are based on the assumption that the loading on the mixer and vessel components are geometrically symmetric and temporally invariant- a condition that is not often met.

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 8 Issue 06, June-2019

The design of mixer consists of a prime mover, gear reduction unit, a shaft and impellers. Most of the installations have overhung shafts, i.e. without a steady bearing to support the free end of shaft. Followed by shaft design, different crucial components of the mixer such as motor mount, frame, impeller hubs, etc. are designed following standard procedures.

II. SELECTION OF FLUID

Viscosity is the property of a fluid which opposes the relative motion between two surfaces of the fluid that are moving at different velocities

$$\mathrm{Re} = \frac{\rho N D^2}{\mu} \tag{2.1}$$

Where,
ρ = Density of fluid
V = Velocity of tip
D = Diameter of impeller
μ = Kinematic viscosity
If Re is $< 10^4$ the flow is laminar
Whereas if Re $> 10^4$ the flow is turbulent.
Hence it is necessary to ensure that the flow is turbulent. To ensure mixing using a single phase motor for calculations purpose, fluids with viscosity 1 to 3 cP with densities ranging from 1000 to 1100 kg/m^3 were selected.

III. VESSEL DESIGN

Quantity of fluid to be mixed had to be decided before doing further calculations as vessel dimensions have a huge impact on impeller design, forces acting on the shaft and the setup as a whole. Also, the design of an impeller and its location in the tank primarily depends on the tank diameter and height. Thus, first the effects of vessel dimensions on the design were studied before selecting the tank dimensions.
After due considerations, the dimensions were selected which are stated below

Vessel Dimensions:
D = 590 mm
H = 950 mm
h = 767 mm (Fluid height)
Now,

$$V = \frac{\pi}{4} * D^2 * L$$

$$= \frac{\pi}{4} * 590^2 * 950$$

= 209.695 * 106 mm^2

From Solidworks model, the volume of mixing fluid was found to be 199.635 litres due to shape of the tank. Hence, capacity of tank was found to be approximately 200 litres.

IV. IMPELLER DESIGN

Different types of impellers are used for different mixing processes such as hydrofoil, Rushton turbine, pitched blade turbine, flat blade turbine, etc. A comparison is made of different types of impellers before selecting an appropriate type of impeller for the process. The most important factors governing the selection of impellers are power number and flow number which provide even grounds for comparison of different impellers

The power number gives the amount of the power consumed by the impeller whereas the flow number suggests the discharge or flow set up by the impeller.
The power number is one of the most widely used design specifications in the mixing operation and has proven to be a reliable predictor of a number of process results. The power number can be calculated from the formula:

$$Np = \frac{P}{\rho N^3 D^5} \tag{4.1}$$

Where,
Np = Power number
P = Power consumed
ρ = Density of fluid
d = Diameter of the impeller
N = Speed of impeller
Flow number is also an important parameter to be taken into consideration while selecting an impeller.
Flow number is calculated using formula:

$$N_q = \frac{Q}{N d^3} \tag{4.2}$$

Where Q is the radial discharge
Pitch blade turbine has the least power number of 1.5 and the highest flow number of 0.8. Hence, we have selected a two blade 45 pitch blade turbine.
Now, following empirical relations are used to find the impeller specifications as taken from [5]"Handbook of Industrial Mixing Science and Practice"

$$d = 0.3 * D$$
$$= 0.3 * 590$$
$$= 177$$

(For practical feasibility, d = 210 mm)

$$w = \frac{d}{10} = 20 \text{ mm}$$

t = (3 to 10 mm) = 3 mm

n = 2 (No. of impellers)

$$d_h = 1.3 * d_o$$
$$= 1.3 * 60$$
$$= 80 \text{ mm}$$

Stresses in Blade

$$R_b = \frac{d_b}{2} = \frac{210}{2} = 105 \text{ mm}$$

$$R_h = \frac{d_h}{2} = \frac{96.6}{2} = 48.3 \text{ mm}$$

Stresses in the blade are given by the equation[11]

$$\sigma = \frac{MaxB.M}{Z} = \frac{F_m (0.75 R_b - R_h)}{\frac{t * w^2}{6}}$$

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 8 Issue 06, June-2019

$$= \frac{202.2727 \quad (0.75*105-48.3)}{\frac{10*20^2}{6}}$$

$$= 5.4841 \ \text{N/mm}^2$$

$$= 5.4841*10^6 \quad \text{N/m}^2$$

$5.4841 * 10^6$ N/m2 < $80 * 10^6$ N/m^2 (Blade design stress)
Hence, Blade design is safe

V. POWER REQUIREMENTS AND MOTOR SELECTION

The power required to run the mixer is calculated using the power number for the selected impeller. But first, in order to ensure that mixing is possible, the flow must be turbulent which is confirmed by calculating the Reynold's number.
As mentioned earlier in equation 2.1, Reynold's number can be calculated as,

$$Re = \frac{1100 * 6.67 * 0.210^2}{0.003}$$

$$= 1.07 * 10^5$$

$$R_e > 10^4$$

Hence, mixing is possible.
Power number is obtained by plotting the point on $1.07 * 10^5$ on the power number v/s Reynold's number graph in [12]Joshi's Process Equipment Design.
Therefore, Np = 1.7
Now, from equation (4.1)
Total power consumed (P) is the sum of power required for driving the impellers

$$P_1 = N_p * \rho * N^3 * d^5$$ ….(Power required to drive 1st impeller)

$$P_1 = 1.27 * 1100 * 6.67^3 * 0.210^5$$
$$= 169.305 \text{ Watts}$$

$$P_2 = N_p * \rho * N^3 * d^5$$ ….(Power required to drive 2nd impeller)

$$P_2 = 1.27*1100*6.67^3*0.210^5$$
$$= 169.305 \text{ Watts}$$

Now,

$$P = P_1 + P_2$$
$$= 169.305 + 169.305$$
$$= 338.61 \text{ Watts}$$

Assuming 10 % gland losses,[11]

$$P = 1.1*338.61$$
$$= 372.471 \text{ Watts}$$
$$= 0.49954 \approx 0.5 \quad \text{HP}$$

Thus, a 0.5 HP motor is selected running at 400 rpm.

VI. SHAFT DESIGN MATERIAL SELECTION:

Material selection for shaft is done by taking into consideration the following factors: resistance against corrosion as liquid to liquid interface enhances pitting and crevice, adequate durability, toughness to endure radial and axial hydraulic forces, stiffness against bending moment due to motor vibration and fluctuating loads. Also, material should be easily available in the market and should be cost effective. Keeping in mind the above requirements, Aluminium 6063 T6 is the most suitable material used for this application.

MATERIAL PROPERTIES:

Aluminium 6063 T6, Mechanical Properties	Value
Ultimate Tensile Strength	240 MPa
Yield Strength	215 MPa
Density	2700 kg/m^3
Modulus of elasticity	68.9 GPa

DESIGN OF SHAFT BASED ON STRENGTH:

Length of shaft is the primary design parameter in design of shaft based on the tank and liquid height and also on the location of impeller. It is calculated as follows:

Minimum length of shaft = Tank height + Clearance from top of the tank - Off bottom clearance

$$= 1950 + 50 - 100$$
$$= 900$$

Therefore,
L = 900 mm
Torque acting on shaft (T),

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

$$= \frac{60 * 372.471}{2\pi * 400}$$

$$= 8.904 \text{ Nm}$$

To account for additional starting torque service factor is assumed to be 1.5

$$T_m = 1.5*T$$
$$= 1.5*8.904$$

$= 13.35$ Nm

Now,

$$z_p = \frac{16}{\pi d^3}$$ (Z_p = Sectional Modulus of Shaft)

According to Maximum Shear Stress Theory,

$$\sigma_s = \frac{T_m}{z_p}$$

$$z_p = \frac{13.35}{\sigma_s}$$

$$\frac{\pi * d^3}{16} = \frac{13.35}{\sigma_s}$$

$$d = \sqrt[3]{\frac{16 * 13.35}{\pi * \sigma_s}}$$ σ_s = Permissible Shear Stress (N/m^2)

$$= \sqrt[3]{\frac{16 * 13.35}{\pi * 40 * 10^6}}$$

$= 0.0119$ m

$$\boxed{d = 11.93 \text{ mm}}$$

According to Normal Stress Theory,

$$F_m = \frac{T_m}{0.75 R_b}$$

$$= \frac{13.35}{0.75 * 0.105}$$

$F_m = 169.523$ N

Now, for two impellers with their respective distances as

$L_1 = 0.517$ m

$L_2 = 0.880$ m

Bending moment acting on the shaft due to two impellers is calculated,

$M = F_m * L_1 + F_m * L_2$

$= 169.523*0.517 + 169.523*0.880$

$= 236.823$ Nm

Equivalent Bending (Me) moment which is cumulative resultant of bending moment and maximum torque is,

$$M_e = \frac{1}{2}\left[M + \sqrt{M^2 + T_m^2}\right]$$

$$M_e = \frac{1}{2}\left[236.823 + \sqrt{236.823^2 + 13.35^2}\right]$$

$= 237.0109$ Nm

Now,

$$\sigma_t = \frac{Me}{z_p}$$ Zp = Sectional Modulus of Shaft

$$z_p = \frac{Me}{\sigma_t}$$ σ_t = Permissible Tensile Stress(N/m^2)

$$= \frac{237.0109}{66.2 * 10^6}$$

$$\frac{\pi * d^3}{16} = 3.5802 * 10^{-6}$$

$d = 0.0263$ m

$d = 26.32$ mm

Selecting maximum value of the two as shaft diameter
d = 27 mm

A hollow shaft, made from pipe, can increase the stiffness and reduce the weight of mixer shaft in critical speed calculations. Such changes will increase natural frequency and extend the allowable shaft length or operating speed. When determining the appropriate shaft size for the strength of hollow shaft, we began with the dimensions of standard available pipe. Then the shear and tensile stress values were computed and compared with allowable values.[5]

Selecting nearest standard dimensions of hollow shaft,

$d_i = 33.4$ mm

$d_o = 27.8$ mm

Now,

Shear stress and tensile stress values are calculated using the relationship given in [5]Mechanical Design of Mixing Equipment's by D.S Dickey & J.B Fasano.

$$\sigma_s = \frac{16\sqrt{T_m^2 + M^2}}{\pi} * \frac{d_o}{d_o^4 - d_i^4}$$

$$= \frac{16\sqrt{13.35^2 + 236.823^2}}{\pi} * \frac{0.0334}{0.0334^4 - 0.0278^4}$$

$= 62.344 * 10^6$ N/m2

Also,

$$\sigma_t = \frac{16\left(M + \sqrt{T_m^2 + M^2}\right)}{\pi} * \frac{d_o}{d_o^4 - d_i^4}$$

$$= \frac{16\left(236.823 + \sqrt{13.35^2 + 236.823^2}\right)}{\pi} * \frac{0.0334}{0.0334^4 - 0.0278^4}$$

$= 124.5895*10^6$ N/m2

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 8 Issue 06, June-2019

The calculated stresses are greater than the permissible stress. Hence, design is unsafe.
By trial and error,

$d_i = 40$ mm

$d_o = 60$ mm

Now, the stresses calculated are

$\sigma_s = 6.969 * 10^6$ N/mm²

$\sigma_t = 13.928 * 10^6$ N/mm²

Now, the calculated stresses are within the permissible stresses. Hence, design is safe from shaft strength point of view.

VII. DESIGN OF SHAFT BASED ON CRITICAL SPEED:

Natural frequency is a dynamic characteristic of a mechanical system. Of primary concern to mixer design is the first lateral natural frequency, which is the lowest frequency at which a shaft will vibrate violently as a function of length and mass. This speed at which shaft vibrates violently is known as critical speed. It is recommended that the range of speed between 70% and 130% of the critical speed should be avoided. The shaft diameter should be so chosen that its normal working speed range does not fall within the critical speed range. The procedure to find critical speed is followed as given in [11] Joshi's Process Equipment Design.

δ1 = Deflection due to 1st impeller
δ2 = Deflection due to 2nd impeller
δs = Deflection due to shaft weight
w = uniformly distributed length per unit length
w1 = w2 = Fm = Impeller force on the shaft
L = Total length of shaft
L1 = Distance of 1st impeller from bearing
L2 = Distance of 2nd impeller from bearing
E = Young's Modulus
I = Moment of inertia

Moment of Inertia of the hollow shaft (I) :

$$I = \frac{\pi}{64}\left(d_o^4 - d_i^4\right)$$

$$= \frac{\pi}{64}\left(60^4 - 40^4\right)$$

$$= 510.508 * 10^3$$

Now, deflections of shaft due to the two impeller and self-weight are

$$\delta_1 = \frac{w_1 l_1^3}{3EI}$$

$$\delta_2 = \frac{w_2 l_2^3}{3EI}$$

$$\delta_s = \frac{w l^4}{8EI}$$

Therefore,

$$\delta_1 = \frac{169.523 * 517^3}{3 * 68.9 * 10^3 * 510.508 * 10^3}$$

$$= 0.2220 \text{ mm}$$

$$\delta_2 = \frac{169.523 * 880^3}{3 * 68.9 * 10^3 * 510.508 * 10^3}$$

$$= 1.0948 \text{ mm}$$

$$w = \frac{\pi}{4} * \left(d_o^2 - d_i^2\right) * \rho g$$

$$= \frac{\pi}{4} * \left(60^2 - 40^2\right) * 2700 * 10^{-9} * 9.81$$

$$0.04160 \text{ N/mm}$$

$$\delta_s = \frac{0.04160 * 900^4}{8 * 68.9 * 10^3 * 510.508 * 10^3}$$

$$= 0.09699 \text{ mm}$$

Now, Critical speed is calculated using the formula,[11]

$$N_c = \frac{946}{\sqrt{\delta_1 + \delta_2 + \left(\frac{\delta_s}{1.25}\right)}}$$

$$= \frac{946}{\sqrt{0.2220 + 1.0948 + \left(\frac{0.09699}{1.25}\right)}}$$

$$= 801.12 \text{ RPM}$$

Critical speed range i.e (70% - 130% Nc) is found to be 560.784 – 1041.456 rpm. Operating speed doesn't lie in critical speed range.
Actual speed of agitator = 400 rpm which is 49.93% of critical speed. Therefore, selected dimensions of shaft is satisfactory which is not between 70% to 130%.

Shaft Dimensions:

$d_i = 40$ mm

$d_o = 60$ mm

$L = 900$ mm

VIII. MOTOR MOUNT DESIGN:

Following the selection of motor, design of shaft & impeller, next step is to design a mount on which motor will be mounted to generate the necessary torque for mixing operation. The longitudinal motion of the whole motor and shaft assembly is crucial for loading & unloading at the mixing station. The primary function of the motor mount was to provide a rigid support to the motor and shaft assembly. The mount was to be designed based on the geometry of motor. The motor selected was vertical mount type with provision for mounting near the lower end in form of 4×M10 bolts and 4×M8 bolts below the integrated planetary gearbox.

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 8 Issue 06, June-2019

The mount was accordingly designed to accommodate the above mentioned provisions. The mount was designed using Solidworks and after trial and error thickness of the mount plate was chosen to be 3 mm taking into consideration the weight of the subassembly, minimum self-weight of the motor mount. A provision was made for axial motion of the sub assembly by sliding in the centre box pipe provided in the frame. Locking mechanism was designed to facilitate locking of the mount to provide safe assembly and disassembly of the shaft. The spring loaded retractable pin located laterally was used for ease in operation.

IX. FRAME DESIGN:

This step involved design of layout to permit mounting of motor, its vertical motion, mounting and dismounting of the vessel for mixing of fluid. Consequently, the frame for the setup was to be designed. The important factors considered during design involved height & width of tank, overall height of the frame, ergonomic location of control board, location of impeller from bottom, sliding length of the assembly required for loading and unloading, also the axis location of the shaft for concentric mixing. Due to ease of availability & ample strength a 1.5’×1.5’ mild steel box pipe was selected for design of frame. Loads acting on the frame were considered i.e. motor and shaft as well as impeller weight which were 12.755 Kg and 7.5 Kg respectively. Based on the location of impeller required according to the process characteristics, the mounting position on the frame was finalized. By taking into consideration the tank height which was 950 mm and to allow complete removal of shaft from the tank for detachment, the sliding length of the subassembly was taken to be 250 mm with provision for locking and holding the motor and shaft assembly at required position. The actual assembly of the industrial mixer is shown below.

Figure No.1 Industrial Mixer Assembly

X. ANALYSIS

A. IMPELLER PITCH BLADE

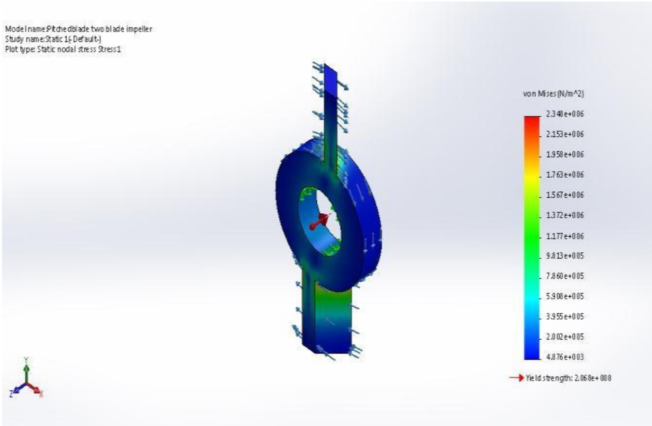

Figure No.2 Pitch blade Static Stress

The stress induced in impeller is in the range between 2.348*10^6 to 4.87*10^3 N/mm2. Yield strength of the material is 2.068*10^8 N/mm2, hence the stresses induced are safe. The stress concentration is maximum in between the connection of the hub and the blade, but the stress induced in that area is within the limits.

B. SHAFT AND IMPELLER SUB-ASSEMBLY

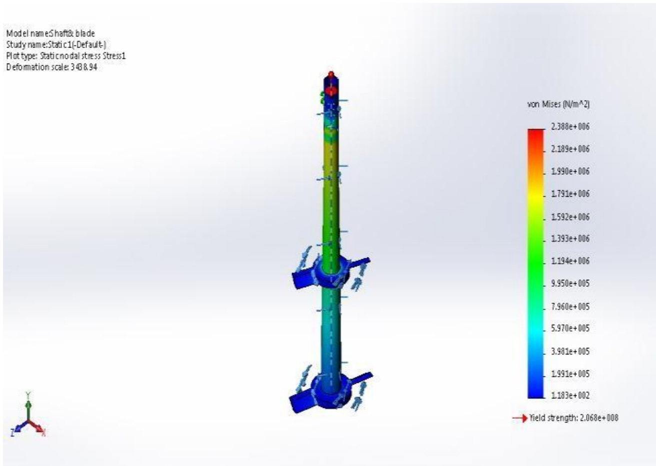

Figure No.3 Shaft and Impeller Sub-Assembly stress

This analysis is of Shaft and Impeller, where the range of which the stresses are induced due to load, torque and other factors are 2.3.88*10^6 to 1.183*10^2 N/mm2. The yield strength is 2.068*10^8 N/mm2 so the stress induced in the assembly is less than the yield strength.

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 8 Issue 06, June-2019

C. MOTOR MOUNT

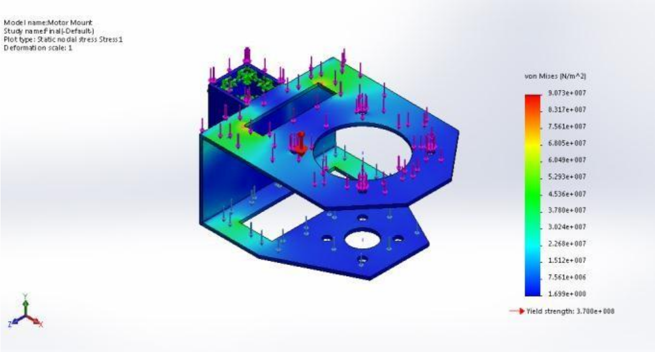

Figure No.4 Motor Mount-Static Stress

The motor mount experiences large amount load which is about 14 kg. Hence the maximum stress induced in motor mount is 9.073*10^7 N/mm2, which is very safe for application. Hence the assembly and individual parts are safe for design and manufacturing. But this analysis does not take into consideration the effect various fluid forces.

Solidworks Flow Simulation was used to ensure proper axial flow was setup by using the shaft and impeller sub assembly and the used dimensions of tank using two baffles at 180^0 angle. The results of the study are demonstrated using velocity vectors plot as shown in figure below.

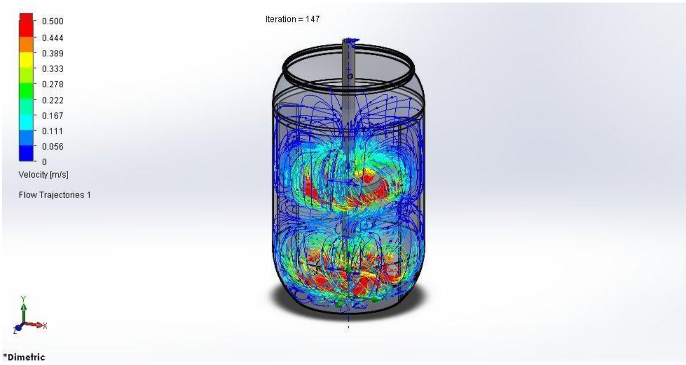

Figure No.5 Flow Simulation

XI. CONCLUSION

The aim of the project was to design and develop an industrial mixer for a range fluid viscosity (1-3cp). During the course of the project we studied the different configuration, impellers and layouts and have come to select the best possible outcome based on safety, availability and feasibility of manufacturing. Every component in the set up was designed and analyzed using computerized software packages such as Solidworks Simulations and ANSYS Workbench for safety requirements. On manufacturing the setup is expected to achieve maximum possible mixing efficiency while facilitating mixing at various stations. Through this project we have come to the conclusion that the design of an industrial mixer is a standard procedure that one needs to follow. Thus a mixing station can be designed based on the process requirements by following the standard procedure by having ample knowledge concerning the process. Also, an attempt was made to standardize the design procedure in form of an excel sheet for scaling based on the application.

XII. REFERENCES

[1] Kazuhiko Nishi, Naoki Enya, Kazufumi Sonoda, Ryuta Misumi, Meguru Kaminoyama, Potential of an asymmetrical agitation in industrial mixing, Internat. J. Sci. Eng., Vol. 5(2)2013:73-80, October 2013

[2] Saeed Asiri, Design and Implementation of Differential Agitators to Maximize Agitating Performance, International Journal of Mechanics and Applications 2012, 2(6): 98-112

[3] H.S. Pordal & C.J. Matice, Design analysis and scale up of mixing processes

[4] Ronald J Weetman, Bernd Gigas, Mixer Mechanical design – Fluid Forces, Proceedings of the 19 th international pump users symposium

[5] David Dickey & J. B. Fasano, Industrial Handbook Of Mixing - Science & Practice Pg no 1247 – 1329

[6] Kushare D.A , Dhepale D.P. Ukirde A.S Lande S.C., Prof. Dange B.S., Design & Fabrication Of Bi-Directional Mixer International Journal of Advance Research in Engineering, Science & Technology (IJAREST) Volume 4, Issue 3, March 2017, e-ISSN: 2393-9877, print-ISSN: 2394- 2444

[7] S. Saravanakumar, P. Sakthivel, S. Shiva Swabnil and S. Rajesh, 3037- 3056 International Journal of Pure and Applied Mathematics Volume 119 No. 12 2018,

[8] D Devkumar, K Saravanan, Study and design of multiphase reactors, Vol 2 Modern Applied Science, www.ccsenet.org

[9] Yeng- Yung Tsui, Jian-Ren Chou, Yu Chang Hu, Transactions of the ASME, Vol. 128, July 2006

[10] Uhl, V. W. and Gray, J. B., Eds. Mixing Theory and Practice, Volume 1, Academic Press Inc.,

[11] Sumit R.Desai, S.Y.Gajjal and Kulloli, International Journal of Current Engineering and Technology, E-ISSN 2277 – 4106, P-ISSN 2347 – 5161

[12] V.V Mahajani and S.B Umarji, Joshi's Process Equipment Design, 4th ed., pg no 370-438