Design and Analysis of Conical Curved Agitator for Medium Density Fibre Processing

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Abstract— Agitator is use in various processing industry for fluid mixing. Mixing is the intermingling of two or more dissimilar portions of materials resulting in attainment of a desired level of uniformity either physically or chemically. This work gives approach for performing stress analysis of an agitator of a large mixing vessel used in medium density fiber process plant. The analysis is carried out for total deformation and natural frequencies in agitator body. The agitator is subjected to vibration due to multi-axial forces resulting from bending and tensional loading imposed by the mixing operation. To optimize the mixing percentage of mixture here in this dissertation two special type of paddle type impellers are suggested which are double helical ribbon impeller and anchor impeller. In this dissertation ANSYS software is used for computing structural analysis as well as modal analysis. From structural analysis it is found that all the type of impellers has deformation in acceptable limits i.e. Below 0.025 mm and form modal analysis it is found that double helical ribbon impeller can be resonated by only single operational frequencies i.e. Double helical ribbon impeller blade pass frequency (6.67 Hz) as it come in the six natural frequency band of double helical ribbon impeller i.e. 4.6 Hz 23.61Hz. Since this single frequency affects the system, it is very mush easy to counter that one.

Key words: Agitator, Agitation System, Fibre Processing

I. INTRODUCTION

Agitator is use in various processing industry for fluid mixing. Mixing is the intermingling of two or more dissimilar portions of materials resulting in attainment of a desired level of uniformity either physically or chemically. Perfect mixing is that state in which any sample removed from the mixture will have exactly the same composition as any sample taken from any point of the mixture. The objectives of mixing are to attainment of complete and mutual distribution of the constituent materials and to increasing the contact surface thus promoting chemical and physical reactions. The agitator generally used to fulfil following purpose:

a) Suspending solid particles.
b) Blending miscible liquids.
c) Dispersing a gas through the liquid.
d) Dispersing a second liquid to form an emulsion or suspension.
e) Promoting Heat Transfer.

The selection of agitator is depends on the viscosity of the liquid, time of mixing and the best mixer is the one that mixes in the required time with the smallest amount of Power. The performance for impeller mixer depends on: Creation of a current or stream of liquid which penetrate to all point of mixing vessel or tank and turbulence movement to carry the current in all places of the tank.

A. Basic Components in Agitation System

Following are the basic components which lead to develop an agitator:

1) The impeller is the most important part.
2) The motor rotates the shaft or impeller.
3) Baffles are many times added to improve mixing.
4) Circular tip.
5) Inlet outlet pipe.

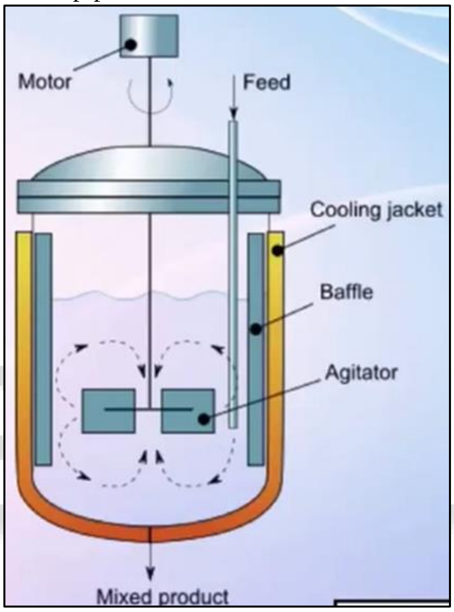


Fig. 1.1: Agitation Systems

B. Common Equipment

1) Impeller

Impellers can be broadly classified into two types, those producing radial flow and axial flow:

C. Axial Flow Impeller- Parallel Direction

Axial impellers pump axially and hence are intended to produce a single circulation loop throughout the vessel as shown in Fig. Hence top to bottom mixing is better than with radial flow impellers especially when multiple agitators are used.

D. Radial Flow Impeller- Tangential Direction

Paddles or flat blade turbines are the simplest designs of radial flow agitators and have the benefits of being cheap and non-propriety. They are often preferred for batch duties. Simple 2-blade designs are used in paint mixers at low transitional Reynolds numbers and have the benefit of being easy to clean. The basic types of impellers are:

a) Propellers
b) Paddles
c) Turbines

II. LITERATURE REVIEW

To accomplish the requirement of agitator design optimization, several studies of researcher are studies and the detail of the same are given below:

Mixing is very important operation in any process industry. All operations involving liquid phase reactions, blending homogenization, emulsion preparation, dissolution, extraction, etc., need mixing in one form or the other. Mixing of powders, pastes, paints jellies and many other products is needed to be done in many industries for many applications; it can be done by the rigorous shaking and creating turbulence in the contents. The process is called as agitation for which directionally reversible mixtures are utilized. These reversible mixers can give partial homogenization for more effective homogenization it is needed to design of a special purpose machine which gives cyclic reversal of the rotor and create more effective agitating turbulence.

In a paper by Dattatraya P. Patil et. al. a special purpose dynamic mixer is designed and developed for a paint manufacturing industry. In this paper, the design and development of this machine is discussed in details, the results of this work are encouraging and giving better agitating performance over conventional method.

In 2012 a researcher Prof. Hyoung Woo Oh published a paper, in this paper, a coupled computational fluid dynamics (CFD) code with a computational structural dynamics (CSD) code by using an efficient coupling interface for solving fluid-structure interaction (FSI) problems in the stirred tank equipped by a radial turbine. Specific techniques of rearrangement of grid and treatment of the boundary conditions are used to follow the behavior of the system. This method takes advantage of the parallel process involved within each analysis code. This allows both parts of the fluid structure interaction problem to be solved in the best possible way: a Finite Volume Method for the fluid dynamics and a finite element method for the structure. The CFD results obtained allow a visualizing of the velocity field, the turbulent kinetic energy, the dissipation rate of the turbulent kinetic energy, the turbulent viscosity.

III. DESIGN OF AGITATOR

Following are the some dimensionless number which is used in the design calculations:

A. Power Number (Po):

The agitator power number Po is analogous to the friction factor in pipe flow. It represents the ratio of the pressure differences producing flow to inertial forces and takes the form:

$$Po = \frac{P}{\rho * N^3 * D^5} \text{……………………. (1)}$$

Power Number (Po) = 1.4 (as per graph)

B. Reynolds Number (Re):

Agitator Reynolds Number, Re In mixing the Reynolds number is an important group. It is the ratio of inertial to viscous forces. For an agitator the Reynolds number is defined as:

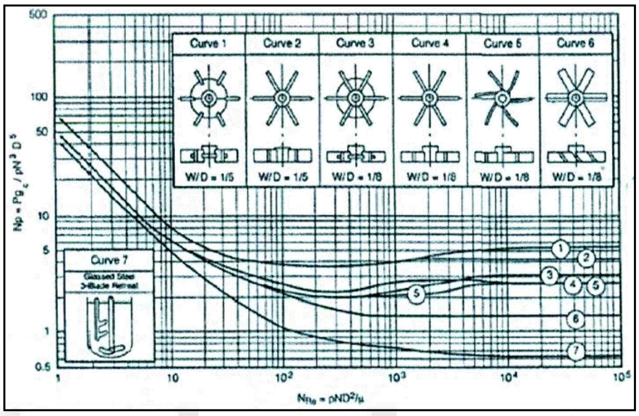
$$Re = \frac{N * D^2 * \rho}{\mu} \text{…………………….. (2)}$$

Laminar, Re ≤ 10 three agitation regimes are described by the agitator Reynolds number,
Transitional, 10 > Re <104
Fully turbulent, Re ≥ 104

The impellor Reynolds number in a non-Newtonian fluid uses an apparent viscosity μ a calculated using the Metzner and Otto (1957) approach. They proposed that for the laminar regime, the average shear rate was related to the agitator speed according to: γ &

$$\gamma = k_s * S \text{………… (3)}$$

Where k_s is an agitator specific constant.



Graph 3.1: Relations between Powers Number and Reynolds Number

The design inputs for existing agitator are:
Liquid Density (p) = 1200 kg/m 3
Liquid Viscosity (μ) = 50 CP
= 0.05N.s/m2
Required Rotation (N) = 200 RPM
=3.3333333 RPS

C. HP Calculation

1) Reynolds Number for Agitated Liquids

$$Re = \frac{N * D^2 * \rho}{\mu} = 28800$$

Since Re is in between 10 & 10000. Hence fluid is Transition Flow.

2) Impeller Tip speed / Peripheral Speed of an impeller:

Peripheral Speed of an impeller = π x Da x N = 6.283185 m/s

3) Power Required for Baffled Tank P

= Po x p x N^3 x Da^5 = 4838.4 watt
= 4.8384 kw
= 6.488400845 HP

4) Power Loss

a) Losses in mechanical seal (assumed) = 0.5 HP
b) Drive Losses:
1) Gear box is used = 0.836 HP
2) Gear box is not used = 0.209 HP

Approximately power loss in the agitation = 1.336 HP

Total Power required for agitation = 7.82440084 HP

Hence this power is rounded to the standard electric motor rating i.e. 8 HP

D. Shaft Diameter Calculation

Yield Stress (Fy) = 21.09 Kg/Mm2

Hp = 8
Out speed (rpm) = 200
Free Length = 3.55m
Sweep Diameter = 0.6m
Service Factor = 1.5
YIELD STRESS (Fy)
SS304 = 21.09 kg/mm2
SS316 = 20.9
C22 = 31.6
CS = 26.51
SS316L = 17.33
SS304L = 17.33
Continuous Torque (Tc) = (716.00 *8)/ 200 = 28.64 kg.mm
Design Torque (Tm) = 28.64 X 1.5 = 42.96 kg.mm
Maximum Force at Point of Jamming, Fm
Fm = 42.96/(0.75*0.3) = 190.9333 kg.s
MAXIMUM BENDING MOMENT, Mm
Mm = 190.9333333333 x 3.55 = 677.8133 kg.m
EQUIVALENT BENDING MOMENT, Meq.

$$Meq. = \sqrt{459430.9148444 + 0.75 * 1849.5616}$$

Meq.= 678.8358314 kg.m

$$ds = \sqrt[3]{(32 \times 678.835831 \times 1000)/(3.142 \times 21.09)}$$

ds = 68.9515434205 mm

Tank diameter	T	-
Baffles	B	4
Baffle width	W_b	T/10
Baffle clearance from wall	S_b	T/60
Impeller diameter	D	T/3
Blade length	L	D/4
Blade width	W	D/5
Disc diameter	D_d	3D/4
Impeller clearance	C	T/3
Submergence	S	2T/3

Table 3.1: Dimensions of the Standard Tank Configuration

E. New Suggested Design

1) Conical Agitator Tank Design

While replacing a cylindrical agitator a must be taken about the volume and height of both the new and old system should have been same.

The calculations regarding the same are as follows:
Cross sectional area of cylindrical agitator = (Di*H)
Cross sectional area of conical agitator = ((D1+D2)*H)/2
For same volume,
Di*H= ((D1+D2)*H)/2
Take, D1=2*D2
Therefore,
D1= 4*Di/3
D2= 2*Di/3
The CAD of new conical agitator is given below:

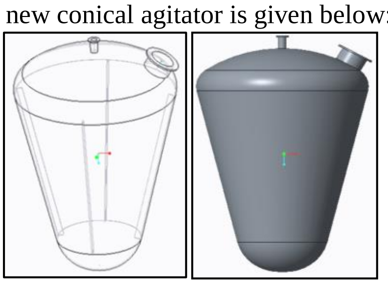


Fig. 3.1: Conical Curve Agitator Tank

2) Impeller

A rotating impeller in a fluid imparts flow and shear to it, the shear resulting from the flow of one portion of the fluid past another. Limiting cases of flow are in the axial or radial directions so that impellers are classified conveniently according to which of these flows is dominant. By reason of reflections from vessel surfaces and obstruction by baffles and other internals, however, flow patterns in most cases are mixed.

Following are the two newly suggested and available type of impellers which can be implementing in the existing application so that the improvisation in results may get.

3) Double Helical Ribbon Impeller

Helical ribbon normally functions by pumping liquid from the bottom of a tank to the liquid surface. The liquid then returns to the bottom of the tank to fill the void created when fresh liquid is pumped to the surface.

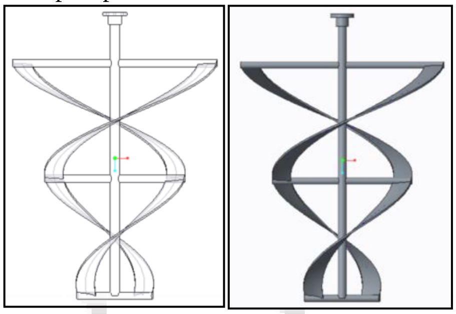


Fig. 3.2: Double Helical Ribbon Impeller

IV. STATIC STRUCTURAL ANALYSIS

Static analysis calculate the effects of steady loading condition on a structure, while ignoring inertia and damping effects, such as those caused by time varying loads. A static analysis can include steady inertia loads such as gravity and rotational velocity and varying loads that can be approximated as static loads.

Static analysis determines the displacements, stresses, strains, and forces in structure or components caused by loads that do not induce significant inertia and damping effects. Steady loading and response conditions are assumed; that is the loads and structures responses are assumed to vary slowly with respect to time.

A. Meshing Generation

Mesh generation is a process of dividing the structure continuum into a number of discrete parts or finite elements. If the mesh is finer, the results are also better but the analysis time is longer. Therefore, a compromise between accuracy & solution speed is usually made.

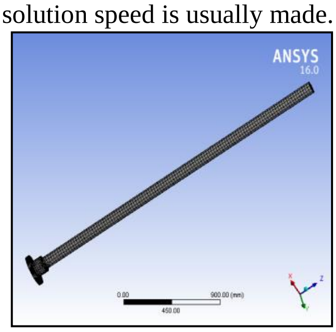


Fig. 4.1: Bottom Shaft

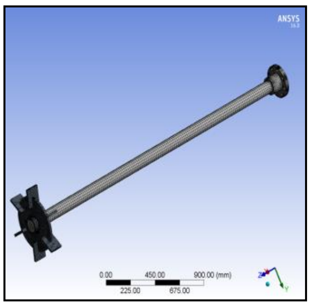


Fig. 4.2: Rushton Impeller

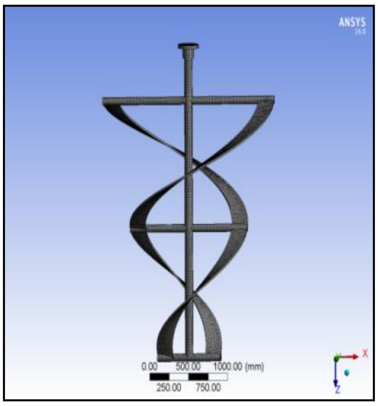


Fig. 4.3: Double Helical Ribbon Impeller

B. Applying Boundary Conditions

Some types of load are usually applied to the analysis model. The loading may be in the form of a point load, pressure or displacement in a stress analysis. If you apply a load to the model, then in order to stop it accelerating through the computer's virtual ether (mathematically known as a zero pivot), at least one constraint or boundary condition must be applied. Structural boundary conditions are usually in the form of zero displacements.

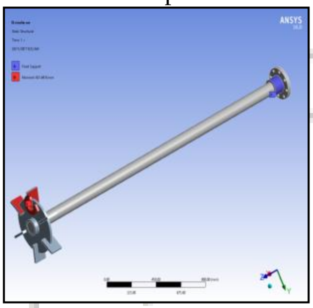


Fig. 4.4: Bottom Shaft

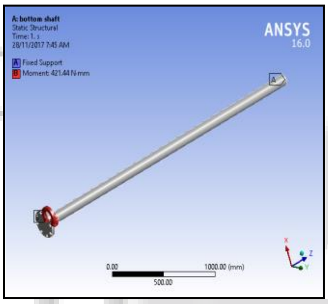


Fig. 4.5: Rushton Impeller

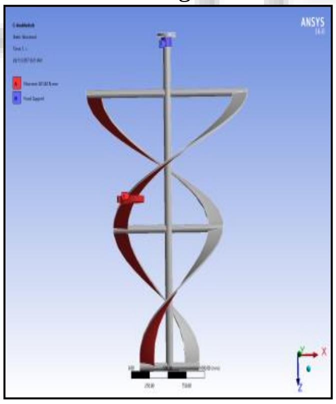


Fig. 4.6: Double Helical Ribbon Impeller

V. MODAL ANALYSIS

Now a day's most of the mechanical systems are subjected to dynamic loading which causes & shortens of the usable time, crack, noise and fatigue, in general the total effect of work for the mechanical system is lowered. Reasons for such behavior are type of loading, construction and conditions of work where the mechanical systems operate. The problems created the need of Vibration testing. Modal testing is the form of vibration testing of an object where the natural (modal) frequencies, modal masses, modal damping ratios and mode shapes of the object under test are determined.

A. Mesh Setting

In ANSYS workbench, the three impeller designs are imported in IGES format same as did in structural analysis. In modal analysis using FEM, each agitator component model was meshed with hexahedral and tetrahedron elements with average element size of 20 mm. The model consists of approximately 31 k nodes and 25 k elements.

B. Boundary Conditions

ANSYS workbench program was used to apply the motion constraints and contact conditions. There are two different conditions being analyzed in conducting modal analysis of the agitator components. The first six natural frequencies and their mode shapes were analyzed on the agitator component without pre-stress state involving constraints only.

C. Natural Frequency and Mode Shapes

Natural frequency is the force free frequency by which the structure will vibrate if once excited and the mode shapes gives the idea about the response shape taken during corresponding each natural vibration.

The mode shapes of the agitator component in FEM were calculated independently of the excitation, which means that the structure is only mass and stiffness distribution dependent. According to Berlioz and Trompette, low resonance modes (first few modes) correspond to higher global modal response (higher amplitude of excitation) compared to higher resonance modes. Following figure shows mod shapes for all three shaft and impeller assemblies.

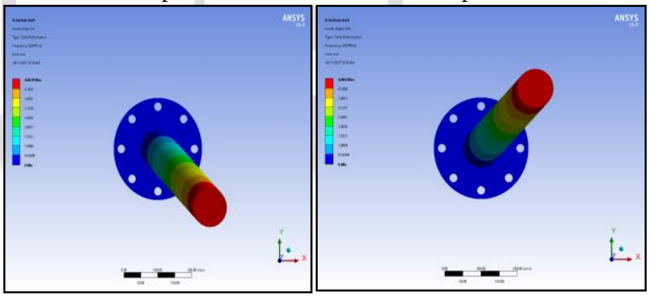


Fig 5.1: Modes Shapes of Bottom Shaft

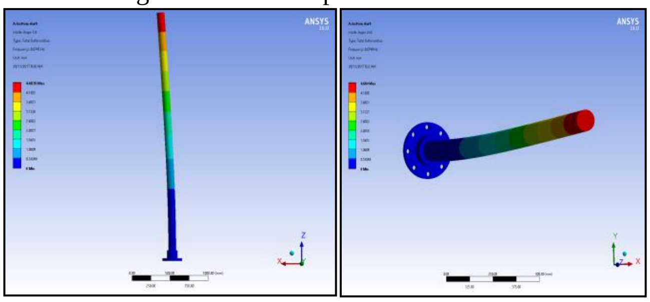


Fig. 5.2: Mode Shapes of Bottom Shaft

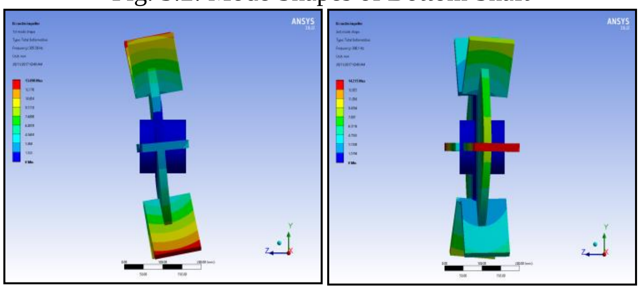


Fig. 5.3: Modes Shapes of Rushton Impeller

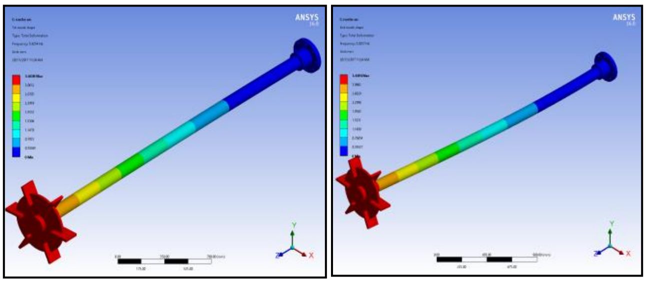


Fig. 5.4: Mode Shapes of Rushton Impeller Assembly

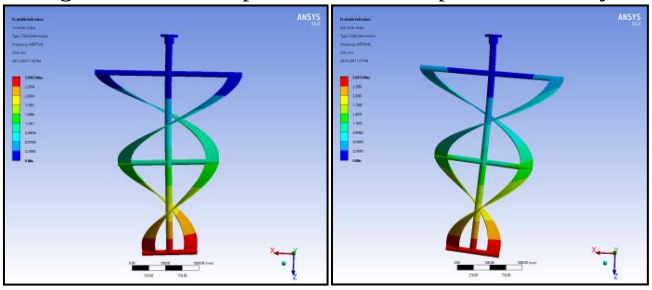


Fig. 5.5: Mode Shapes of Double Helical Ribbon Impeller

VI. Result and Discussion

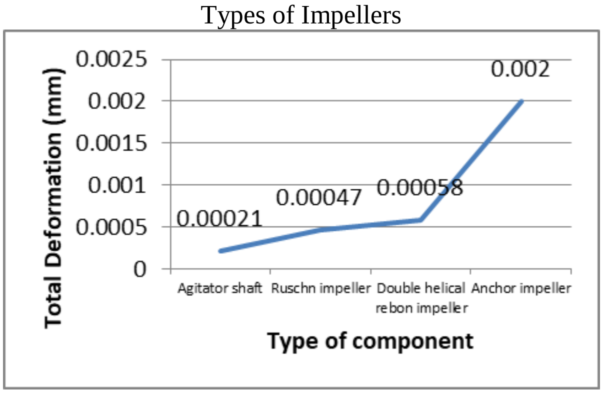
A. *Results and Discussion on Structural Analysis*

1) *Results*

As mention in chapter first the deformation of the impeller should be restricted to 0.025 mm, the total deformation of the impeller is chosen as the strength comparison parameter. Hence the impeller with lesser total deformation has more strength and should have first preference. In Static structural analysis the result of total deformation are shown and the same data in tabulated in the following table and graph.

Sr. No.	Component	Total Deformation (mm)
01	Agitator shaft	0.00021
02	Rushton impeller	0.00047
03	Double helical ribbon impeller	0.00058

Table 6.1: Result of Total Deformation in mm for Three Types of Impellers



Graph 6.1: Total Deformation Vs Type of Component

From the results of static structural analysis it is found that the double helical ribbon impeller i.e. 0.00058 has total deformation in between the deformation of the rushton impeller i.e. 0.00047 mm and anchor i.e. 0.002 mm type of impeller and all the impellers are in safe level of deformation.

B. *Results and Discussion on Modal Analysis Results:*

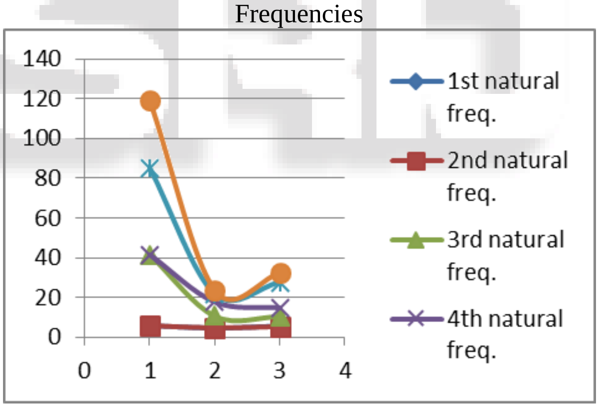
The result in above chapters shows the variation of mode shapes with type of impeller. In this study we used six mode shapes to compare the effect of impeller type on natural frequency. The first six mode result for all impeller types are plotted in tabular format and are shown in table. As we know resonance needs to be avoided when system is running at high frequency and to do the same we consistently work to find the methods to avoid this situation. To get the above stated goal here we find the effect of impeller type on resonance frequencies.

Operating frequencies (Hz)	
Shaft rotation	3.34
Rushton Impeller blade pass frequency	20
Rushton Impeller blade-baffles pass frequency	80
Double helical ribbon impeller blade pass frequency	6.67
Double helical ribbon impeller blade-baffles pass frequency	26.67
Anchor impeller blade pass frequency	6.67
Anchor impeller blade-baffles pass frequency	26.67

Table 6.1: Impellers and Corresponding Operational Frequencies

Sr. No.	Type of Impeller	Natural Frequency (Hz)	
		1st	*2nd*
01	Rushton Impeller	5.8254	5.8257
02	Double Helical Ribbon	4.6075	4.6754

Table 6.2: Type of Impeller and Corresponding Natural Frequencies



Graph 6.2 Comparisons of All Six Natural Frequencies of Three Types of Impellers

Now if the analysis on the operation frequencies and first six natural frequencies is to be carried out then the following observations can be made:

1) If observation on Rushton impeller operation frequency and natural frequencies are made, it is found that this impeller can be resonated by two operational frequencies i.e. Rushton Impeller blade pass frequency (20 Hz) and Rushton Impeller blade-baffles pass frequency (80 Hz) as they come in the six natural frequency band of Rushton impeller i.e. 5.82 Hz-119.3 Hz.
2) Similar to the same, If observation on double helical ribbon impeller operation frequency and natural frequencies are made, it is found that this impeller can be resonated by only single operational frequencies i.e. Double helical ribbon impeller blade pass frequency (6.67 Hz) as it come in the six natural frequency band of

double helical ribbon impeller i.e. 4.6 Hz-23.61Hz. Since this single frequency affect the system, it is very mush easy to counter that one and it simply could be done by adding the number of impellers in the system.

3) Again the same observations on anchor impeller operation frequency and natural frequencies are made, it is found that this impeller can also be resonated by two operational frequencies i.e. anchor Impeller blade pass frequency (20 Hz) and anchor Impeller blade-baffles pass frequency (80 Hz) as they come in the six natural frequency band of Rushton impeller i.e. 5.62 Hz-32.26 Hz.

VII. CONCLUSIONS

In this dissertation work a methodology to analysis and design of Agitator is presented. From this work following conclusions are found,

1) By replacing standard cylindrical agitator with conical agitator the mixing percentage will get improved.
2) In addition to conical curved agitator it is found that provision of baffle plates also gives uniform mixing since in new design of agitator tank/ vessel 4 baffle plates are attached to improve mixing percentage.
3) Replacing Rushton impeller by anchor or double helical ribbon impeller will improve the performance of impeller and from structural analysis it is found that both the impellers has total deformation are in the acceptable limit i.e. 0.025 mm.
4) If observation on double helical ribbon impeller operation frequency and natural frequencies are made, it is found that this impeller can be resonated by only single operational frequencies i.e. Double helical ribbon impeller blade pass frequency (6.67 Hz) as it come in the six natural frequency band of double helical ribbon impeller i.e. 4.6 Hz-23.61Hz. Since this single frequency affects the system, it is very much easy to counter that one and it simply could be done by adding the number of impellers in the system.

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