DESIGN AND VIBRATION ANALYSIS OF SCREW COMPRESSOR

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ABSTRACT

The screw compressor is widely used in refrigeration and air conditioning industry due to it high efficiency. The screw compressor are said to have an efficiency of about 75% to 80%. The one percent of efficiency of the screw compressor is lost due to some of the losses which restrict it from being 100% efficient. The project aim is to analyze that 20% to 25% losses which cause due to vibration and heat losses. Hence analytical approach is used to find these vibration by performing vibration analysis and to get an idea of temperature distributions alone the rotor temperature analysis is being done.

Keyword: - *Modal analysis, Vibration Analysis, Holzer method*

Introduction

The twin screw air compressor is a positive displacement compressor and has been widely used in gas industries. One of the major advantages of the twin screw compressor is its flexibility under various operation conditions. It utilizes the continual variations of the space formed between rotor grooves and case of the compressor. In this project, the finite element method (FEM) is used for computing the deformation of rotors of twin screw compressor. The male and female rotors are mapped to 3D elements. Finite element method, it is well known numerical method used to solve multi physics problem were governing differential equations are available. FEM consists of discretization process. In this the infinite degree of freedom is converted into finite degree of freedom for solving multi physics problems in an approach that solution can be obtained with minimum error. This method is widely used and is easy for solving the problems whose analytical solution is difficult to obtain such as of complex geometry, load and boundary condition.The modeling method for analyzing the rotor deformation of twin screw compressor by using the CAE method instead of CFD method with considering the working temperature and gaseous pressure as well as the contact force between two screw rotors. The thermal-structural coupling method is applied to analyze the rotor deformations and loads as there are many changes on operating clearance due to thermal loadings.

Modal Analysis

Modal analysis is the study of the dynamic behavior of structures under external exertion. Vibration is about frequencies. By its nature, vibration involves repetitive motion. Each involves of a complete motion sequence is called a cycle. Frequency is defined as complete cycles in a given time period. One cycle per second is equivalent to one Hertz. Then the results from a Natural Frequency or Modal analysis include displacements, these displacements will be used only to view the mode shape. That is, the magnitude of the displacements is relative to each other. The natural frequency is a theoretical result due to unspecified dynamic loads, so the results cannot obtain for loads.

Finite element method, it is well known numerical method used to solve muti physics problem governing differential1equations are available. Since FEM is a piece wise approximation it consists of discretization process in order to convert infinite degree of freedom to finite degree of freedom for solving multi physics problems in an approach that solution can be obtained with minimum error. This method is extremely easy for the problems with complex geometry, load and boundary condition for which the analytical solution may be difficult to obtain.

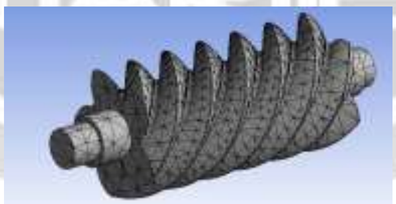
I. METHODOLGY

- Creation of CAD Model of the screw rotor used in rotary screw compressor using CATIA v5.
- Assigning Materials and properties for the generated cad modelling ANSYS.
- Meshing will be done using ANSYS tool.
- Modal analysis of the FEA Model with Loads and BCs using ANSYS.
- Extraction of results after completion of analysis and documentation using ANSYS.
- Theoretical calculation will be done related to analysis.
- Finally compare the results.

II. MODELLING

- In this section, modelling of screw rotor has been done using CATIA v5 modelling software.

Geometric 3d model of screw rotors

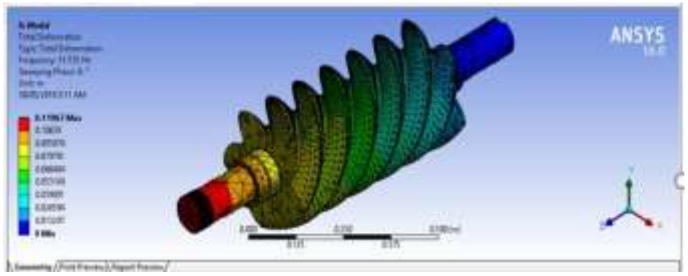


FEM model of screw rotor

Figure 2.3 shows the Finite Element model generated for the geometric model of the bearing structure using 3D Tetra elements. These elements are selected because the complex size of the structure.

I. ANALYSIS RESULTS

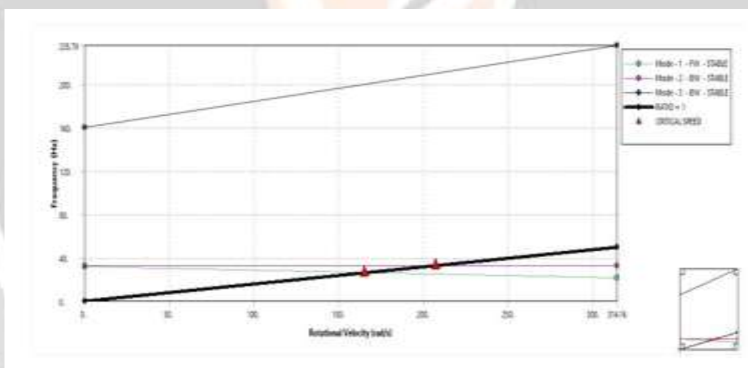
1. MODAL ANALYSIS



Tabular Data

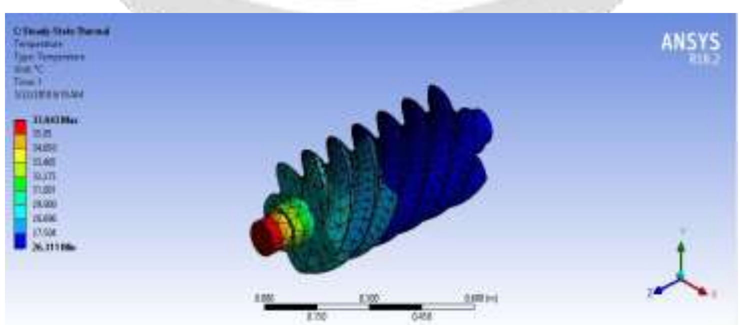
	Set	Solve Point	Mode	Damped Frequency [Hz]	Stability [Hz]	
1	1.	1.	1.	31.572	0.	0
2	2.	1.	2.	32.932	0.	0
3	3.	1.	3.	160.38	0.	0
4	4.	2.	1.	31.243	0.	0
5	5.	2.	2.	33.279	0.	0
6	6.	2.	3.	160.38	0.	0

Table of deformation



Graph of modal damping versus rotational velocity

2.THERMAL ANALYISIS



Theoretical Calculations

In general, rotating elements driven by conventional drives (Electric motors, engines, etc.) are susceptible to torsional resonance if any of the the natural torsional frequencies of the system is excited. It is therefore critical to determine the torsional frequencies of such systems and ensure that they do not lie within the input speed range of the machine and do not coincide with any excitation frequency. If they do, stiffness or inertia tuning can be used to shift the torsional frequencies of the system to the left or the right. Two general drive trains are encountered in practice. In one, the torque is transmitted from the drive through all the rotating elements without branching. In the other case, two or more rotating elements are connected by parallel shafts connected to the same gear stage. In such a case the torque is divided in the shafts at the branched point. To determine the system natural torsional frequencies, the structural dynamic equation

[*I*]*f*_□*g* + [*C*]*f*__*g* + [*K*]*f*_*g* = *fT*(*t*)*g*

is used, in which *T*(*t*), the forcing torque or excitation, is set to zero, and *C*, which is the damping coefficient, is usually negligible in torsional systems and therefore is set to zero. *I* is the moment of inertia and *K* is the torsional stiffness. Lumped mass models, in which inertias are concentrated at gears and rotors are developed, with the shafts inertias as well as the stiffness of the gears neglected. Gears are assumed to be infinitely stiff. Each inertia is treated as a node, N, and the number of frequencies depends on N. For N nodes, we have N-1 natural torsional frequencies *!*1, *!*2,....*!N*□1, including rigid body modes. The corresponding relative angular deflection, _, is a function of the angular frequencies. That is, *_N* = *f*(*!N*)

Leq=L1(deq/d1)^4 + L2 (deq/d2)^4

Where,

L1 & L2= length of shaft A & B

d1 & d2= diameters of shaft A & B

Leq & deq= length & diameter of equivalent shaft

Assuming deq=0.095m

Leq=0.633m

Now, considering three rotor system considering inertia of gear & rotors

Calculate natural frequency of torsional vibration

Condition (i) $Ip_1 * L_a = Ip_3^1 * L_c$

Where, $Ip_1 = 400\ kgm^2$

$Ip_2 = 0.158\ kgm^2$

$Ip_3 = 15.30\ kgm^2$

$Ip_2^{\ 1} = Ip + (Ig/GR^2) = 0.843\ kgm^2$

$Ip_3^{\ 1} = (Ig/GR^2) = 66.41\ kgm^2$

Condition (ii) ωna=ωnb

1/(Ia*La) = 1/(I’b)[1/(Leq1-La) + 1/(Leq2-Lc)]

Where,

Leq1= 0.5m

Leq2=0.133m

Ip_2= 0.158 kg/m

Ip_1=400 kg/m

Now,

Therefore, the value of Lc_1 = 0.8049 m & Lc_2 = 0.0622 m gives the position of two nodes

Whereas, La_1= 0.1336 m & La_2=0.01032 m give the position of one node on equivalent shaft

As, $F_{a1} = \frac{1}{2\pi} [\sqrt{\left(\frac{GJ}{Ip1*La1}\right)}]$

F_{n1}=22.10 Hz

F_{n2}=62.62 Hz

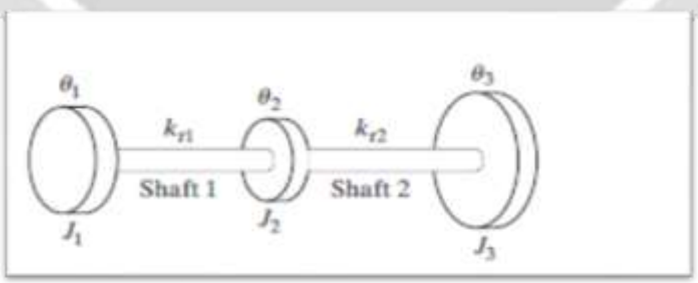
Holzer's Method

Holzer method is essentially a trial-and-error scheme to find the natural frequencies of

Undamped, damped, semidefinite, fixed, or branched vibrating systems involving linear and angular displacements. The method can also be programmed for computer applications. A trial frequency of the system is first assumed, and a solution is found when the assumed frequency satisfies the constraints of the system. This generally requires several trials. Depending on the trial frequency used, the fundamental as well as the higher frequencies of the system can be determined. The method also gives the mode shapes.

Torsional Systems

Consider the undamped torsional semidefinite system shown in figure below.



Torsional semidefinite system

The equations of motion of the discs can be derived as follows:

$J1\ddot{\theta}1+Kt1(\theta1-\theta2)=0$

$J2\ddot{\theta}2+Kt1(\theta2-\theta1)+ Kt2(\theta2-\theta3)=0$

$J3\ddot{\Theta}3+Kt2(\Theta3-\Theta2)=0$

Since the motion is harmonic in a natural mode of vibration, we assume that $\Theta1=\cos(\omega t-\phi)$in above equation

$\omega^2 J1\ \Theta1=Kt1(\Theta1-\Theta2)$

$\omega^2 J2\ \Theta2=Kt1(\Theta2-\Theta1)+ Kt2(\Theta2-\Theta3)$

$\omega^2 J3\ \Theta3=Kt2(\Theta3-\Theta2)$

Summing these equations gives

$$\sum_{i=1}^{3} \omega^2 Ji\ \Theta i=0$$

Above equation states that the sum of the inertia torques of the semidefinite system must be zero. This equation can be treated as another form of the frequency equation, and the trial frequency must satisfy this requirement

In Holzer s method, a trial frequency ω is assumed, and is arbitrarily chosen as unity.

Next, is computed from Eq.equations and then Θ3 is found from above equation. Thus we obtain

$\Theta1=1$

$\Theta2= \Theta1- \omega^2 J1\ \Theta1/Kt1$

$\Theta3= \Theta2- \omega^2 /Kt1(J1\Theta1-J2\Theta2)$

These values are substituted in above Eq.to verify whether the constraint is satisfied. If above Eq. is not satisfied, a new trial value of is assumed and the process repeated.

Above equations can be generalized for an *n*-rotors system as follows:

$$\sum^{n} \omega^2 Ji\ \Theta i=0$$

Thus the method uses in above Eqs. repeatedly for different trial frequencies. If the assumed trial frequency is not a natural frequency of the system, then above Eq.is not satisfied.

The resultant torque in above Eq. represents a torque applied at the last rotor. This torque is then plotted for the chosen ω. When the calculation is repeated with other values of the

resulting graph appears as shown in Fig. From this graph, the natural frequencies of the system can be identified as the values of at which The amplitudes corresponding to the natural frequencies are the mode shapes(1, 2,…. , n) of the system. Holzer's method can also be applied to systems with fixed ends. At a fixed end, the amplitude of vibration must be zero. In this case, the natural frequencies can be found by plotting the resulting amplitude (instead of the resultant torque) against the assumed frequencies. For a system with one end free and the other end fixed, above eq. can be used for checking the amplitude at the fixed end. An improvement of Holzer s method is presented.

After tabulation by holzer method, we select,

ω=200 rad/sec,where T_3 becomes negative

$F= \frac{w}{2\pi} = \frac{200}{2\pi} = 31.83$ Hz

Vibration analysis using FFT

A vibration FFT (Fast Fourier Transform) spectrum is an incredibly useful tool for machinery vibration analysis. If a machinery problem exists, FFT provide information to help determine the source and cause of the problem. FFT allow us to analyse vibration amplitudes at various component frequencies on the FFT spectrum. In this way, we can identify and track vibration occurring at specific frequencies. Since we know that particular machinery problems generate vibration at specific frequencies, we can use this information to diagnose the cause of excessive vibration. The measurement of motion s accomplished by motion by sensor. The sensor can be fixed to measure the displacement, velocity, angular velocity, strain level, temperature & other quantities.

The Anatomy of the FFT Analyser

The FFT Analyser can be broken down into several pieces which involve the digitization, filtering, transformation and processing of a signal.

Several items are important here:

Digitization and Sampling

Quantization of Signal

Aliasing Effects

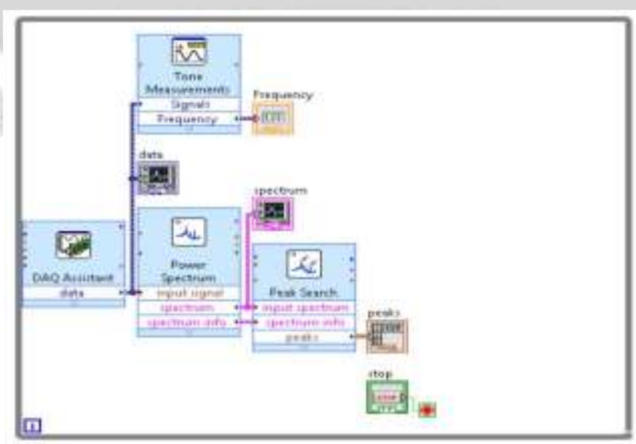
Leakage Distortion

Windows Weighting Functions

The Fourier Transform

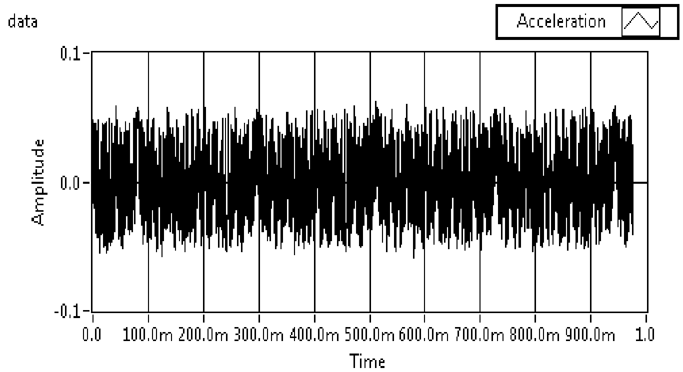
Measurement Formulation

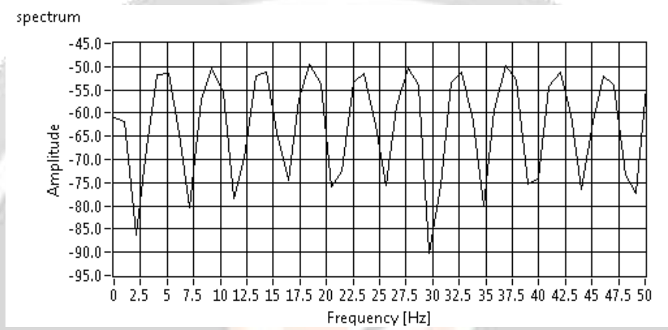
LAB View



Block diagram

Results of LAB view





Results

Sr. no.	frequency	Theoretical Result	Analytical Result	Exp.Result
1	f_{n1}	25.40	31.243	18.57
2	f_{n2}	62.60	160.38	125

4. CONCLUSIONS

Natural freuency of the screw compressor is obtained by experimental, theortical, and analytical method.It is found that the natural frequency lies in the range of 17 Hz to 160 Hz. Thes vibrations produced decreases the life of the parts of the compressor.These vibrations produced can be reduced can using accessorirs such as silencer, air-ment valve at the inlet or exhaust valve.

6. REFERENCES

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