Performance studies on shrink-fit radial fan assembly using finite element analysis

R.Vinayagamoorthy, S. Madhavan
1Department of Mechanical Engineering, SCSVMV University, Enathur – 631561
2Department of Mechanical Engineering, Sri Venkateswara College of Engineering, Sriperumbudur

Abstract
Radial/axial fans were used in all critical areas of Indian industry like air-conditioning industry, automobile industry, boiler industry, marine-ventilation, nuclear and conventional power station, pollution control, dust collection etc. During design, engineers must decide how to connect each involved power transmission component to shafting. Traditional connection methods fall into three categories. Shrink fitted mounting where the hub-bore is expanded using heat, mounted on shaft and allowed to cool; Material fit: Typically entails the welding or soldering of the mounted component to the shaft and using keyways where a key is used as a lock to hold the shaft and hub together. In this work, an attempt has been made to analyze and study the performance difference between a shrink fitted radial fan and the same fitted with a keyway. The analysis is done using “solid works-supported by Geostar”. The analysis has been conducted on four different parameters such as stress, strain, displacement and factor of safety. However, the results show that the key fitted assembly is more efficient with a high factor of safety. Practical performance study of the shrink and key fitted radial fans (of the same material – AISI steel) show increase in all aspects such as speed, sound and vibration. Both the fans undergo the testing procedure where the speed, vibration and sound are measured. The shrink fit shows improved speed and efficiency with considerably less noise and vibration as compared to the one assembled with the keyway.

Key Words: Shrink fit, Key fit, factor of safety

1. Shrink Fitting

Most materials expand when heated and shrink when cooled. Enveloping parts are heated with torches or gas ovens and assembled into position while hot, then allowed to cool and contract back to their former size, except for the compression that results from each interfering with the other. Railroad axles, wheels, and tires are typically assembled in this way. Alternatively, the enveloped part may be cooled before assembly such that it slides easily into its mating part. Upon warming, it expands and interferes. Cooling is often preferable as it is less likely than heating to change material properties, e.g. assembling a hardened gear onto a shaft, where heating the gear would alter its hardness. Shrink fitting is the procedure of heating a hub to a particular temperature which reduces the hub diameter so that it can be fitted to a shaft of the same diameter. It is then allowed to cool down to room temperature. This makes the fit stronger.

1.1 Methodology

Different methods and techniques can be employed to accomplish a shrink fit. Most frequently, the retainer is heated to a temperature sufficient to cause the expansion and allow assembly of the insert. Sometimes, the process is done by deep freezing the retainer to contract the insert for an easier shrink fit. Deep freezing used to be done with dry ice at -84°C. In fact, in some shrink fitting operations, the retainers are not even heated. Instead, the liquid nitrogen is used directly which causes the contraction.

2. Boundary conditions

Displacement:
The rotating motor shaft is fixed in the impeller so the displacement on impeller hole is zero in all degree of freedom.

Angular velocity and angular acceleration:
The impeller rotates about the “z” axis at a speed of 1000rpm. Therefore

\[ \text{Angular velocity} = \frac{2\pi N}{60} = 104.7 \text{ rad/sec} \]

The angular acceleration is also given as 9810 rad/sec².
3. Selection of Material and Element type

Once a mesh has been built to describe the domain occupied by the structure, the rest of the computer model could be built. It is only at this stage that the description of the physical problem generated in the initial stage of the analysis can be related to the computational geometry described by the mesh of nodes and elements. For each element, its material properties must be defined together with the boundary conditions on the faces of the elements, or at the nodes, which form the exterior of the mesh. It is not necessarily a straightforward task to define precisely about the material properties and frequently, they must be approximated when compiling the model data for an analysis.

For this model, the constant isotropic material has been used and their values are
- Young’s modulus \( E_X = 21000 \text{ kg/mm}^2 \)
- Density \( \text{DENS} = 8.002 \times 10^{-10} \text{ mN/mm}^3 \)
- Poisson’s ratio \( \text{NUXY} = 0.3 \)

For the analysis a 4-noded area element (SHELL 63) is used. SHELL 63 element is well suited for mapped meshing for this model. Usually for any area can be meshed using 4-noded area element (SHELL 63) in a uniform manner (mapped meshing). It is a kind of mesh in which the points of the mesh are arranged in a regular way all through the continuum and can be stretched to fit a given geometry so that the results will be more accurate when compared to free mesh results. Fan work can be equated to the system resistance. Fan pressure has the dimension of work per unit volume. Thus the system resistance may also be regarded as the work required per unit volume of gas. The ratio of this air power to the power required to drive the fan is the fan efficiency. The pressure may be total including the velocity pressure or static and resulting efficiencies may also be “total” or “static”. Selecting a fan of higher efficiency normally results in higher first cost, but in lower operating cost.

4. Estimation of fan type, size and speed:

Once the flow rate and pressure are known it is possible to derive some idea of feasible options for the type of fan required. Simple formula will allow initial estimates to be made of the probable type(s) and size(s), which are optimum for a particular installation.

The specific speed, \( N_s \), of a fan is a measure of the fan shape or type. \( N_s \) is defined as \( N_s = \left[ \frac{w \cdot (Q)^{0.5}}{(gH)^{0.75}} \right] \)

Where,
\( w \) is the rotation speed of the fan (rad / sec)
\( Q \) is the volume flow rate (m\(^3\) / sec)
\( gH \) is the specific energy (J / Kg)
\( gH = \frac{p}{\rho} \)

Where,
\( p \) is the fan pressure raise – pa
\( \rho \) is the fluid density – Kg / m\(^3\)

Knowing \( Q \) and \( gH \), a range of rotational speeds can be assumed. Typically these will correspond to 2, 4, or 6 pole motor speeds with a wide choice available for belt driven fans. The value of \( N_s \) defines the optimum fan type for the duty. If \( N_s \) is less than about 1.5 the fan will be a centrifugal machine; if \( N_s \) is greater than about 2.5 the optimum fan will be an axial, between 1.5 and 2.5 the optimum unit would be mixed flow type. The next stage is to determine the approximate impeller diameter. For a centrifugal type fan the diameter can be estimated from the relationship,
\[ V^2_{\text{tip}} = gH / (0.8 - 0.23 N_s) \]
And for an axial or mixed flow machine from
\[
D = 2\left[\frac{gHQ}{w^3kL}\right]^{0.2}
\]
Where the loading coefficient, \(k_L\), has a value typically in the range 0.01 to 0.08.

It is thus possible to calculate the potential speed and diameter of the fan best suited to the duty. If either the speed or diameter appears impractical this may well point to the need to consider multistage fans or a series of fans in parallel. For multistage fans the head per stage reduces, thus raising the specific speed per stage. For fans in parallel the flow per fan decreases thus increasing the specific speed per unit.

5. Results and Discussion

![Fig.1 Meshed component.](image1)

**Connecter definitions:**
Pin connectors on 2 faces: no pin rotation: with axial stiffness 1000kgf/cm. The loading type is sequential loading.

![Fig.2 Static Displacement-Key fit](image2)
Fig.3 Static Nodal Stress - key fit

Fig.4. Static Strain – key fit

Fig.5 Factor of Safety - key fit
The procedure for creating the model of a shrink fitted shaft – hub assembly is however similar to that of the one fitted with the keyway. The shaft of diameter 28mm is created. The difference between the keyway and the shrink fit is the absence of the key. Here the hub created has the exact inner diameter of the shaft. The amount that a particular metal will expand can be calculated using the coefficient of thermal expansion:

\[ d = aL(dt) \]

Where,
- \(d\): Total deformation desired (in mm)
- \(a\): Coefficient of thermal expansion (in mm/mm °C)
- \(L\): Nominal length of the part being heated (diameter for a cylinder in mm)
- \(dt\): Temperature Difference.

Thus, as per the calculations, it was observed that for AISI1020 steel, there is a deformation of 0.1mm when heated at a temperature of 320°c. In order to obtain the deformation, the temperature is applied on the hub separately. When fitted to the shaft, the hub diameter decreases 0.1mm thus making the fit tight. Finally the analysis is done by applying the same boundary conditions and input data and the solution is obtained.
The stress distribution, displacement, strain and the factor of safety for both the types of assembly are analysed. The minimum factor of safety of the shrink fit is less than that of the key shaft assembly. However, the performance study of both the assemblies at various levels show an increase in the performance of the shrink fitted assembly. Both the assemblies are studied by adjusting the inlet duct and keeping them at three different levels: 1. Full close 2. Duty point and 3. Full open. The vibration along the three moments: Vertical, horizontal and axial are also measured. The sound is measured using a decibel metre where the device is kept at an angle of 45° to the axis of the shaft on all four directions. The results are as follows:

**Keyshaft assembly:**
MOTOR RATING: 1.5KW; 3.1amps
2pole; Direct drive.
Decibel level at 1m distance: 78db.
Decibel level at 2m distance: 77db.
Table -1

<table>
<thead>
<tr>
<th>Shrink fit assembly:</th>
<th>Voltage (volts)</th>
<th>Current (amps)</th>
<th>Speed (rpm)</th>
<th>Vibration(mm/sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full close</td>
<td>415</td>
<td>2.00</td>
<td>2928</td>
<td>Vertical 1.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Horizontal 3.1</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Axial 2.4</td>
</tr>
<tr>
<td>Half close</td>
<td>414</td>
<td>2.68</td>
<td>2881</td>
<td>Vertical 1.44</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Horizontal 2.70</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Axial 1.44</td>
</tr>
<tr>
<td>Full open</td>
<td>414</td>
<td>3.35</td>
<td>2849</td>
<td>Vertical</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Horizontal</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Axial</td>
</tr>
</tbody>
</table>

The shrink fitted assembly varies with high speed and low sound and vibration as compared to the keyshaft assembly. This is due to the weight of the shrink fitted assembly which is lesser than the keyshaft. It is also due to the absence of the keyway which makes the shrink fit as one whole component as if it is welded together.

6. Conclusion

In this work, a study has been made on both the keyway and the shrink fit assembly. The analysis of shrink fit results as a failure because of the low factor of safety. This is because of loss in tensile strength due to frequent use of the component. However practical readings show that the shrink fit assembly is more efficient. Thus it was found that the shrink fitted radial fans can be use for light duty applications and for a short time period. The only advantage of the shrink fit assembly is that they are more efficient in factors like speed, noise and vibration. Thus the choice of using a shrink fit or a keyway depends on the purpose.

7. References