Design of an Axial Flow Compressor Test Rig
**Objectives:**

The task at hand involves the design of an axial flow multi-stage compressor for one of the ME labs. In order to do so, many milestones must be achieved for the design to be successful. These include:

- Defining a number of design characteristics to start developing analyzing the machine.
- Constructing the appropriate velocity diagram for the design parameters previously defined.
- Applying basic thermodynamic and fluid mechanics principles to determine the missing design characteristics.
- Predicting the mechanical behavior of the machine.
- Evaluating the design to determine if all the components meet the requirements.
- Designing and providing a 3-D model of the gas angles for the blades using “Free Vortex.”
- Developing a test procedure for the lab.

**Introduction:**

The study of turbomachinery and its thermodynamic behavior is used to create rotating devices in order to provide the best design for various applications. Adhering to the mechanical design process and following some basic criteria lead to the creation of a compressor that could be used in the ME lab. The development of an appropriate velocity diagram starts out the magical process which gives birth to such a compressor. The design of an appropriate blade profile, leads to the thermodynamic analysis which determines if the desired performance is attained. After this process, the machine is ready for modeling and manufacture. A test rig procedure is also developed for future ME students to analyze the characteristics of this machine. Understanding the basic principles of the design, characteristics, and inner workings for various types of mechanical equipment is crucial to the proper development of a mechanical engineer. The design of an axial flow multi-stage compressor for the ME compressor lab provides this thorough understanding, and sets the minds of the students as engineers by providing the opportunity to work on a real life situation.

**Compressor Design Theory:**

To begin designing the compressor, some parameters must be fixed and chosen for later analysis.
**Reaction:**

A quick study and research on the reaction design of an axial flow compressor shows that the most effective choice is around a 50% reaction. For this reason, a reaction of R=0.5 is chosen as our first parameter.

**Blade profiles:**

The size of the blades and the hub must be determined. The purpose of this machine is to provide the students with a compressor to study its characteristics. It must also fit inside a lab, and be compact for other equipment and people to be inside the room. Because the actual compression and delivery output of the compressor is not one of the important and driving forces behind our design, small sizes are chosen as tabulated below.

<table>
<thead>
<tr>
<th>Tip Diameter</th>
<th>Hub Diameter</th>
<th>Blade Chord Length</th>
<th>Thickness</th>
<th>Mean Diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>400mm</td>
<td>300mm</td>
<td>25mm</td>
<td>2.5mm</td>
<td>350mm</td>
</tr>
</tbody>
</table>

*Table 1. Blade Sizes*

Based on the aforementioned, we can now decide on the flow angles of our compressor. For a compressor design, the velocity inlet velocity projected onto the blade velocity must be smaller than the outlet velocity projection ($C_{y2}>C_{y1}$). This implies that the inlet angle $\alpha_1$ must be greater than the outlet angle, $\alpha_2$. A quick search on the book and various examples provided lets us determine that some standard values for these angles are chosen as 30° and 60°. A chord length of 25mm is also chosen for our design, which we will base on the C4 blade profile type. We also need a set number of blades, to then determine the pitch. A choice of number between 11 and 130 is reasonable according to experience and the examples studied. The chosen values are tabulated below:

<table>
<thead>
<tr>
<th>$\alpha_1=\beta_2$</th>
<th>$\alpha_2=\beta_1$</th>
<th>Chord Length (C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>30°</td>
<td>60°</td>
<td>25mm</td>
</tr>
</tbody>
</table>

*Table 2. Flow Angles and Chord Length*

<table>
<thead>
<tr>
<th>Number of Blades</th>
<th>Rotor</th>
<th>Stator</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pitch (S)</td>
<td>21.9mm</td>
<td>20.7mm</td>
</tr>
<tr>
<td>S/C</td>
<td>0.876</td>
<td>0.828</td>
</tr>
</tbody>
</table>

*Table 3. Rotor and Stator Blade Design Parameters*
**Velocity Diagram:**

The rotational speed needs to be defined before an accurate velocity diagram can be constructed. Because this machine is fairly small sized and for study purposes only, the rotational velocity does not need to be enormous. Based on various other similar designs researched, a speed of 3000 rpm should suffice for our design. First, the blade speed can be found for any of the diameters (tip, hub or mean). The mean diameter speed (because all our angles correspond to the mean diameter) can be found through the following relation:

\[
U_{dm} = \frac{\pi d N}{60} = \frac{\pi (0.35m)(2500rpm)}{60} = 54.98 m/s
\]

Based on the flow angles defined above, the blade dimensions provided, and the mean diameter tangential (blade) velocity, the velocity diagram can now be accurately constructed. Using AutoCad to aid us in the development of our velocity diagram, we obtain the following:

![Figure 1. Velocity Diagram.](image)

From simple measurements, numerous velocities can be defined.
**Thermodynamic Calculations**

- **Mass Flow Rate:**

The velocity diagram presented above corresponds to the profile of the blades at the mean diameter. The design of this particular compressor is based on the free vortex flow, which means that the product of the tangential velocity and the radial distance from the center remains constant. For this reason, after a momentum equation has been solved, it is determined that the axial component of the velocity remains constant at all radii. From the velocity diagram below, it is very simple to find the axial velocity, but for the sake of checking, the axial velocity should be calculated to make sure that the velocity diagram is built correctly. For a Free vortex design, the axial component of velocity is calculated as:

\[
C_x = \frac{U_{T_{dm}}}{\tan(\alpha_{1_{dm}}) + \tan(\beta_{1_{dm}})} = \frac{54.98 \text{ m} / \text{s}}{\tan(30^\circ) + \tan(60^\circ)} = 23.81 \text{ m} / \text{s}
\]

Therefore, it is theoretically demonstrated that the velocity triangle is correct. The mass flow rate of the compressor is defined as the amount of mass per unit time that passes through the compressor. Due to thermodynamic equilibrium and basic principles, it is understood that this quantity remains constant throughout (matter cannot be created nor destroyed). Though the density of the fluid changes when it undergoes the various (in this case four) compression processes, it remains almost constant during one stage. Therefore it is safe to assume that at the first stage (air enters the compressor) the density of the flowing air is and remains constant during the whole stage. Thus, the design mass flow rate can be easily found as:

\[
\dot{m} = A_1 \rho_1 C_x = \pi (r_t^2 - r_h^2) \rho C_x = \pi (0.2^2 - 0.15^2) \times 1.2 \times 23.81 = 1.57 \text{ kg} / \text{s}
\]

- **Work (Power absorbed):**

The Work done on every stage of the compressor, or power absorbed per every stage of the compressor is another thermodynamic characteristic that is very important. By defining the power required by the stages, the total power necessary to drive the compressor can be easily found, and the motor chosen for this application can be validated or modified to meet the requirements. The design power absorbed by the first stage is defined as:

\[
\dot{W}_c = \dot{m} U_T (C_{\theta 2t} - C_{\theta 1t}) = \dot{m} U_T C_x (\tan \alpha_2 - \tan \alpha_1)
\]

\[
\Rightarrow \dot{W}_c = 1.57 \times 54.98 \times 23.81 \times (\tan 60^\circ - \tan 30^\circ) = 2.373 \text{ KW}
\]
The above quantity corresponds to the power required to drive the compressor through one stage. To find the total power absorbed, we need to multiply this by the number of stages and divide it by the mechanical efficiency. The mechanical efficiency is a measure of losses due to mechanical friction. This efficiency is usually very hard to measure and sometimes it is discarded, therefore, we will simply assume that the mechanical efficiency of the machine is one hundred. The blades must be smooth and the bearings in very good condition to allow minimal mechanical losses. Therefore, with the four stages, the total power necessary is:

\[
\dot{W} = 3.764KW \times 4 = 9.493KW = 12.72HP
\]

**Motor Selection:**

Now that we have thermodynamically defined the stages of the compressor and established the operating points, an appropriate choice can be made. The motor of choice needs to be able to handle a top speed of 3000 rpm. Another requirement of the motor is that, an estimated power needed to run the compressor will be of around 10 KW, or 13 HP. Therefore, performing a basic search on the web, it is determined that a Baldor-Dodge-Reliance Motor model RPM III-T25T1351 will do the work. This is a DC motor, so a change in current is used to drive it. The following information contains the basic motor information obtained from the vendor:

<table>
<thead>
<tr>
<th>Model Number</th>
<th>T25T1351</th>
<th>HP</th>
<th>15</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed (RPM)</td>
<td>500</td>
<td>Field Weakened Speed</td>
<td>3000</td>
</tr>
<tr>
<td>Frame Size</td>
<td>C2512ATZ</td>
<td>Enclosure</td>
<td>DPG-FV</td>
</tr>
<tr>
<td>Mounting</td>
<td>N/A</td>
<td>Armature Voltage</td>
<td>240</td>
</tr>
<tr>
<td>Field Voltage</td>
<td>240</td>
<td>Power Code</td>
<td>C</td>
</tr>
</tbody>
</table>

*Table 4. Motor Data.*

**Pressure Rise**

The pressure rise across a stage is defined as the change in pressure from the inlet to the outlet of the stage. This is a very important parameter to define whether or not the compressor is doing its work. The assumption is that for the stage the fluid is not compressed, and the stage is ideal (no stagnation pressure losses). The mathematical expression to derive this pressure rise for a stage is the following:
\[ P_3 - P_1 = \rho U (C_{y2} - C_{y1}) \]

From the velocity diagram, \( C_{y2} \) and \( C_{y1} \) for the first stage of this compressor can be found, and have been defined as:

\[
P_3 - P_1 = \rho U (C_{y2} - C_{y1}) = 1.2 \times 54.98 \times (41.23 - 13.74) = 1814 \text{ Pa}
\]

This change in pressure is assumed to be constant at all stages of the compressor. Because this compressor contains four stages, the total pressure rise through the compressor is of 7256 Pa.

**Design Model**

In order to finish the design of the blades, the appropriate flow angles need to be defined at the hub or tip. Therefore, due to simplicity when producing the 3-D model of the blades, the hub angles are defined. The reaction at the hub may be subject to change as well, so all of the flow angles need to be calculated because they may not be symmetric. Because the design is based on a free vortex, the product of the radial distance from the center and the tangential velocity of the fluid is constant. Therefore:

\[
rC_{\theta} = \text{const}
\]

\[
C_{\theta 1 dm} = C_x \tan \alpha_1 \Rightarrow C_{\theta h1} = C_x \tan(\alpha_1)(r_{dm} / r_h) = 23.81 \times \tan(30^\circ) \times (175 / 150)
\]

\[ C_{\theta h1} = 16.04 \text{ m} / \text{s} \]

Similarly,

\[
C_{\theta 2 dm} = C_x \tan \alpha_2 \Rightarrow C_{\theta h2} = C_x \tan(\alpha_2)(r_{dm} / r_h) = 23.81 \times \tan(60^\circ) \times (175 / 150)
\]

\[ C_{\theta h2} = 48.11 \text{ m} / \text{s} \]

The flow angles at the hub, are therefore given by:

\[
\alpha_{1h} = \tan^{-1}(C_{\theta h1} / C_x)
\]

\[
\beta_{1h} = \tan^{-1}\left(\frac{U_{Th}}{C_x} - \tan \alpha_{1h}\right)
\]

\[
\alpha_{2h} = \tan^{-1}(C_{\theta h2} / C_x)
\]

\[
\beta_{2h} = \tan^{-1}\left(\frac{U_{Th}}{C_x} - \tan \alpha_{2h}\right)
\]
These equations have been solved, and the results are tabulated below:

<table>
<thead>
<tr>
<th>α₁</th>
<th>33.97°</th>
</tr>
</thead>
<tbody>
<tr>
<td>β₁</td>
<td>52.54°</td>
</tr>
<tr>
<td>α₂</td>
<td>63.67°</td>
</tr>
<tr>
<td>β₂</td>
<td>-2.38°</td>
</tr>
</tbody>
</table>

*Table 5. Hub Gas Angles.*

It can be observed that at the hub, due to the effects of radial flow, the flow angles drastically change. Thus, this causes a change in the reaction and load coefficient as well. However, approximating them as constant for all the stages at the mean diameter still remains a reasonable assumption.

To finalize the details of the blade, a blade profile basis needs to be established. The blade profile to follow corresponds to the British C4 airfoil profile. Charts for various parameters and design dimensions are found and looked at to decide on an appropriate design. These airfoils can be of circular, ellipsoidal or parabolic camber. For simplicity, a circular camber is chosen, and following the dimensions established per the design criteria for C4 blades, it is determined that the maximum thickness is of 2.5mm (10%C) and it occurs right at the center (maximum deflection angle for a circular shape).

Aside from this dimensions, it is also important to know how to position these blades. The incidence and deviation need to be defined. These two are the angles that define the deviation of the blade from the inlet and outlet gas flow angles. From studies, it has been determined that for certain shapes, a 10 degree variation at the inlet (incidence) usually showed better performance. Therefore, a similar angle of incidence should be used. Because we would like to keep the blade angles as close to the gas angles as possible, an incidence angle of 7 degrees is chosen, which results in a 5 degree deviation angle. Using a computer aided design program the following drawing is created, and demonstrates the basic shape of the rotor blade. A 3-D model is then generated and can be observed for the blade as well.

![Fig 2. 2-D blade profile drawing.](image1)

![Fig 3. 3-D Blade profile.](image2)
Now that the blade profile has been modeled, the whole first stage rotor can be modeled as well. Using the design criteria defined at the beginning, the blades can be arranged along the hub circumference. There are fifty rotor blades for this design, and they correspond to the amount of blades around the hub. Using AutoCad to model, the following images are produced:
**Potential Test Procedure and Instrumentation**

The objective of the laboratory session is to study the compressor map. Also, the student has the opportunity to operate a compressor and observe some compressor phenomena. This equipment is very noisy, so the appropriate ear protection is necessary. Also, students should use goggles when operating any equipment on the lab, to protect their eyes.

This test rig should be similar to the already existing one; a compressor with a venturi at the discharge to allow smooth flow and nice readings. At the outlet, the mass flow rate is controlled by a moving plate. Some instruments need to be put up at the inlet and exhaust of the compressor to measure the performance variables. The following instrumentation is recommended:

- One set of digital differential pressure gages to read the pressure difference at the inlet and outlet of the compressor.
- A digital torquemeter to indicate the necessary mass to balance the torque.
- A digital tachometer to measure the speed of the compressor.
- A digital thermometer to measure ambient temperature.

After all the security measures have been taken, and the instruments checked and placed, the following procedure can be followed:

1. The backing plate needs to be located well away from the outlet of the rig. A temperature measurement of the room can be done at this point as well.

2. Turn the motor on, thus starting the process of testing. Stop the velocity of the motor at a point before 3000 rpm, to measure off-design parameters.

3. When the speed is steady, a torquemeter reading should be obtained.

4. Record the pressure gage differential readings.

5. Reduce the mass flow rate by bringing the blanking plate closer to the outlet of the rig, to test the effects of mass flow rate on the performance curve.

6. Now reduce the flow rate still further by bringing the blanking plate very slowly in towards the outlet until the compressor surges (a change in noise occurs). Read the pressure gages and observe the effects of this phenomenon.

7. Repeat the steps above for various other off-design points.

10. Increase the speed to 3000 rpm and repeat previous steps until surge occurs.

At this point, the experiment should be ready, and all the machinery should be shut down.
Conclusions:

The critical features of involved in the design of the compressor test rig for the ME lab have been determined through the use of fluids, thermodynamics, and velocity diagrams. For the selected blade design, the calculated performance characteristics predict a good mechanical behavior. The selected motor to drive the compressor is small in size, and provides a great fit for this design. The compressor itself is of a fairly small size. This choice is reasonable since a big performance is not required for the design, because the purpose is for the compressor to be studied.

The use of a computer aided design program, such as AutoCad, facilitated a lot of the calculations. By clearly defining the parameters on the velocity diagram and accurately drawing it, a lot of the work had been done. Basing the design on the British C4 blade profile also helped speed up the blade design process. These blades have been carefully studied, and thus the design should be safe and good. The three-dimensional model of the possible blade design foreshadows a successful one as well.

The creation of a possible testing procedure and instrumentation are specific to what the objectives of it are. Digital instruments were chosen over all other types of instrument because they smooth the progress of the laboratory. The desired outcome of this test rig is so that the student understands the behavior of a compressor. Therefore, complicating the rig with non-digital instrumentation would only add cumbersomeness. In the end, all the objectives were met successfully.