EXPERIMENT TITLE: CONSTANT SPEED CENTRIFUGAL COMPRESSOR

OBJECTIVE

Pump manufacturers and fan system design engineers require a method for indicating compressor performance to allow the correct compressor to be selected for any given system and performance requirement. Designers will wish to select a compressor that will be able to produce the required flow rate and pressure differential under typical operating conditions, and also to select a model that will operate efficiently at the expected flow rates. A chart or a graph is a common method of presenting the sort of information required, plotting characteristic indications of performance against discharge to illustrate the compressor performance across a range of possible duty points. The aims of this experiment are:

- to obtain the compressor characteristic curves for a multistage centrifugal compressor operating at constant speed.
- to predict the performance of a compressor at a given speed from data obtained at a different speed.

THEORY AND PRINCIPLES

The centrifugal compressor is a radial flow rotodynamic machine also sometimes called a radial blower, jet, or squirrel cage or squirrel wheel compressor. As in other rotodynamic machines, the centrifugal compressor consists of blades mounted on an axle. The working fluid (for example air) enters this rotor or impeller at one radius, and leaves at a larger radius. In doing so, energy is imparted to the air as kinetic energy, which is then converted to pressure energy as the flow velocity is decreased within the outlet. Theoretically compressors can be run in reverse, but the blades and case are usually designed for optimum operation in one direction, and this usually means that the compressor will show greatly reduced performance if run in the reverse direction. In the centrifugal compressor there are usually multiple stages (more than one set of blades with the air passing through the sets in series). A pressure differential is generated across each stage, and the use of multiple stages thus allows a greater pressure differential to be achieved than is possible with a single impeller[2].

The general relationship between the various forms of energy, based on the 1st Law of Thermodynamics applied to a unit mass of fluid flowing through a ‘control volume’ (such as the compressor itself) is expressed as:

$$-W_s = \Delta KE + \Delta PE + \Delta P + F \qquad (1)$$

where $-W_s$ is the mechanical shaft work performed on the fluid, ΔKE is the change in kinetic energy of the fluid, ΔPE is the change in potential energy of the fluid, ΔP is the change in pressure energy and F is the frictional energy loss as heat to the surroundings or in raising temperature of the fluid itself as it travels from inlet to outlet. Equation 1 can explicitly be written as:

$$-W_s = \left(\frac{v_2^2 - v_1^2}{2}\right) + g\left(z_2 - z_1\right) + \left(\frac{p_2 - p_1}{\rho}\right) + F \tag{2}$$

where v is the velocity of the fluid in m/s, , g is the gravitational acceleration in m/s^2, z is the elevation in m, p is the pressure in N/m^2 and ρ is the density of the fluid in kg/m^3. The subscripts 1 and 2 refers to the compressor inlet and outlet. The first three terms of the right hand side represent the useful work W_a:

$$W_a = \left(\frac{v_2^2 - v_1^2}{2}\right) + g\left(z_2 - z_1\right) + \left(\frac{p_2 - p_1}{\rho}\right) \tag{3}$$

The term W_a represents the actual work performed in changing energy states of a unit mass of the fluid. As the change in atmospheric pressure between the inlet and outlet is negligible the change in potential head of the fluid may be ignored. Friction losses may also be ignored and it is assumed that the fluid is incompressible. Thus, Equation 3 may be represented as the compressor total pressure (total pressure increase resulting from the work done by the compressor on the air passing through it), p_{tc}, by converting the units from work per unit mass to pressure.

$$p_{tc} = \left(\frac{\left(v_2^2 - v_1^2\right) \times \rho}{2}\right) + \left(p_2 - p_1\right) \tag{4}$$

The compressor output is expressed as:

$$P_u = Q_V \times p_{tc} \tag{5}$$

where Q_v is the volume flow rate in m^3/s. Q_v is calculated by utilising the pressure difference between the air at the entrance to the inlet duct (which is assumed to be at the same pressure and velocity as the ambient air in the room) and the air at the inlet orifice:

$$Q_v = \frac{C_d \pi d^2 \times \sqrt{2\rho . p_o}}{4\rho} \tag{6}$$

where d is the orifice diameter (d=35mm), p_o is the differential pressure across the orifice, C_d is the coefficient of discharge for the orifice (C_d=0.596). The average inlet velocity may be found by dividing the volume flow rate by the inlet area:

$$v_1 = \frac{Q_v}{\left(\frac{1}{4}\pi d_1^2\right)} \tag{7}$$

where d_1 =67 mm.

Similarly, the outlet velocity can be calculated by dividing the volume flow rate by the outlet area:

$$v_2 = \frac{Q_v}{\left(\frac{1}{4}\pi d_2^2\right)} \tag{8}$$

where d_2=79 mm.

The mechanical power input to the compressor (work done on the compressor by motor), P_m is expressed as:

$$P_m = \frac{2\pi.n.t}{60} \tag{9}$$

where n is the rotational speed of the compressor, t is the torque applied by the motor to the compressor in N.m. The overall efficiency of the compressor, E_{gr}, calculated as the useful energy absorbed by the air stream as a fraction of the electrical power supplied to the motor is expressed as a percentage:

$$E_{gr} = \frac{P_u}{P_m} \times 100 \tag{10}$$

When the size, rotational speed or gas density is changed, the affinity laws are most often used to calculate changes in flow rate, pressure and power of a compressor. Therefore in the following affinity laws the suffix “1” has been used for initial known values and the suffix “2” for the changed values.

$$
\begin{aligned}
Q_{v_2} &= Q_{v_1} \times \left(\frac{n_2}{n_1}\right) \times \left(\frac{D_2}{D_1}\right) \\
p_{tc_2} &= p_{tc_1} \times \left(\frac{n_2}{n_1}\right)^2 \times \left(\frac{D_2}{D_1}\right)^2 \times \left(\frac{\rho_2}{\rho_1}\right) \\
p_{u_2} &= p_{u_1} \times \left(\frac{n_2}{n_1}\right)^3 \times \left(\frac{D_2}{D_1}\right)^5 \times \left(\frac{\rho_2}{\rho_1}\right)
\end{aligned}
\tag{11}
$$

These laws can be simplified when variables remain unchanged (e.g. when only compressor speed is varied):

$$
\begin{aligned}
\frac{Q_{v_1}}{Q_{v_2}} &= \frac{n_1}{n_2} \\
\frac{p_{tc_1}}{p_{tc_2}} &= \frac{n_1^2}{n_2^2} \\
\frac{p_{u_1}}{p_{u_2}} &= \frac{n_1^3}{n_2^3}
\end{aligned}
\tag{12}
$$

EXPERIMENTAL SET-UP

The unit consists of a **seven-stage centrifugal compressor** which is driven by a three-phase electric motor, mounted on a support plinth. The compressor is fitted with **clear acrylic inlet and outlet ducts**. The inlet duct incorporates an **orifice plate** for flow measurement and the flow is regulated by a **variable aperture device** at the exit of the discharge duct. The system is fitted with **electronic pressure sensors** which measure the differential pressures across the inlet orifice and across the compressor, and a **temperature sensor** measures the temperature of the inlet air stream. Electronic sensors allow compressor performance to be measured.

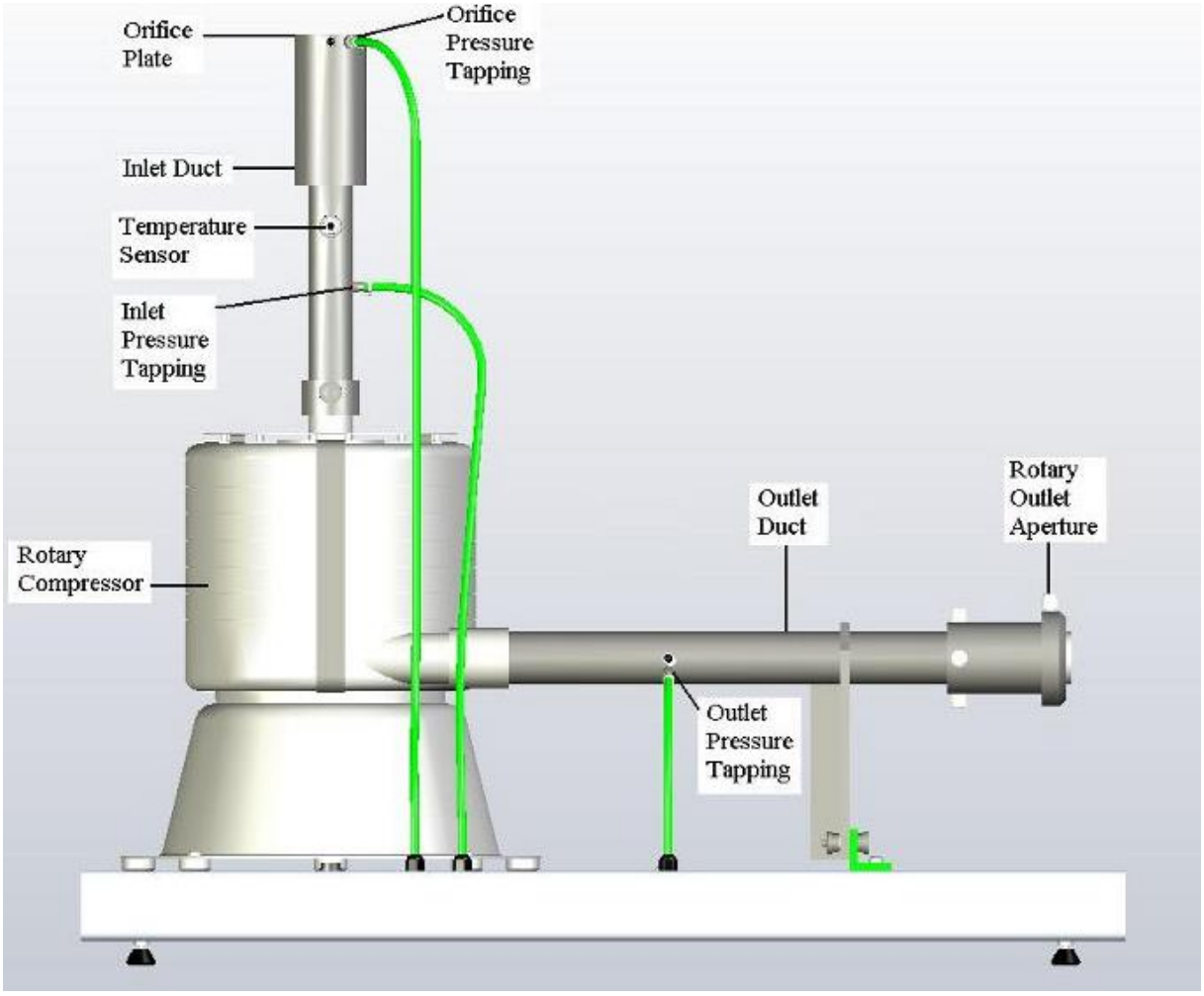

Figure 1. Centrifugal Compressor

The compressor consists of a series of seven stages (sets of blades) mounted about the axle, driven by an induction motor. Each stage is driven at the same rotational speed. The motor is controlled from an attached computer using Armfield software via the Armfield IFD7 interface device. Motor torque is measured and relayed back to the software and displayed on the software screen.

The vertical, clear acrylic inlet duct is mounted on top of the compressor. An orifice is positioned at the inlet of the duct for flow measurement. There are two sets of tapping points in the inlet duct. One is connected to one side of a differential pressure sensor, with the other side of the sensor connected to a similar tapping in the outlet duct, providing differential pressure measurement across the compressor. The second tapping is connected to a differential pressure sensor positioned at the orifice, to provide a reading for the orifice pressure relative to atmosphere. An electronic temperature sensor mounted on the inlet duct measures the temperature of the inlet air stream.

The horizontal, clear acrylic outlet duct is secured to the base. A tapping point in the duct is connected to a differential pressure sensor, with the other side of the sensor connected to a similar tapping in the inlet duct, providing a differential pressure measurement across the compressor. A variable aperture device is mounted at the exit of the duct. The aperture control consists of a rubber tube connected to a collar. The collar is rotated manually to twist the rubber and reduce or increase the cross-sectional area of the outlet, and thus to vary the head and/or flow produced by the compressor. Two small thumb nuts are used to secure the aperture device at the required setting.

PROCEDURE

- On the software interface set the compressor speed to maximum (100%) and note but do not record the flow rate.
- Fully close the outlet aperture to give minimum flow.
- Click the "GO" button on the toolbar to record a set of readings.
- Using the maximum air velocity as a guide, select incremental values for air velocity that will give 10-15 individual steps between minimum and maximum velocity.
- Open the aperture to increase the flow rate to the first increment. Click "GO" button to take a sample. Repeat for each increment until the aperture is fully open.
- Repeat the above procedure for compressor speeds of 80%, 60%, 40% and 20%.

RESULTS AND CALCULATIONS

Note the following variables for each increment:

	Inlet Temperature	p_o	p_2-p_1	n	t
1					
2					
3					
4					
5					
6					
7					
8					
9					
10					
11					
12					
13					
14					
15					

Calculate the following parameters:

- Air density
- Volume flow rate (discharge)
- Compressor total pressure
- Mechanical power (input)
- Compressor power (output)
- Efficiency

For compressor speed of 100% plot graphs of:

- Compressor total pressure against discharge
- Compressor efficiency against discharge
- Mechanical power against discharge
- Compressor power against discharge

For each speed setting plot a graph of compressor total pressure against flow rate (discharge) on the same set of axes.

Choose a speed setting (e.g.; 100%). Using the results obtained at the first setting (100%) calculate the following parameters for a second speed setting (e.g.; 40%):

- Predicted discharge
- Predicted compressor pressure
- Predicted power

Plot a graph of predicted compressor pressure against predicted discharge.

DISCUSSION QUESTIONS

- Compare the predicted results with the measured results. How well do the two graphs compare? Do the predicted values match across the performance range of the compressor? If not, then where in the range does the greatest deviation occur?
- Explain two basic principles of compression. Indicate which of these principles apply to centrifugal compressors.
- What are the applications of centrifugal compressors? Discuss the advantages and disadvantages of centrifugal compressors.

REFERENCES AND FURTHER READING

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