KING FAHD
UNIVERSITY OF PETROLEUM & MINERALS

ME 316: Thermofluids Laboratory

Experiment # 5

CENTRIFUGAL AIR COMPRESSOR

1) OBJECTIVES

   a) To introduce the operational principle of a centrifugal compressor.
   b) To study the performance characteristics of a multi-stage centrifugal compressor.

2) INTRODUCTION

Fans and compressors are gas movers, however, compressors are characterized by high-pressure increase through the machine. It is widely accepted that if the density change through the machine is more than 5% the machine is called a compressor, otherwise it is called a fan. Centrifugal compressors are widely used in petrochemical industries, gas turbines, domestic gas piping systems, and many others. The main components of the centrifugal compressor are much the same as those for a centrifugal pump, however, the compressor impeller has a large number of vanes and equipped with an inducer section at inlet. Compressors are normally driven at higher speeds than pumps and the flow is treated as compressible because of the appreciable density changes. Figure 1 shows a sectional view of a multi-stage centrifugal compressor and Figure 2 shows a photograph of a typical impeller.
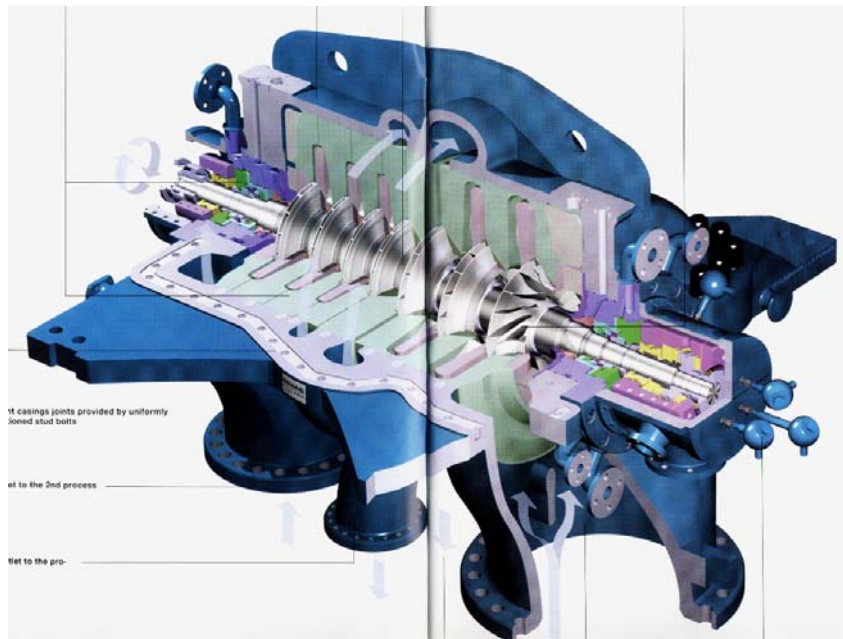

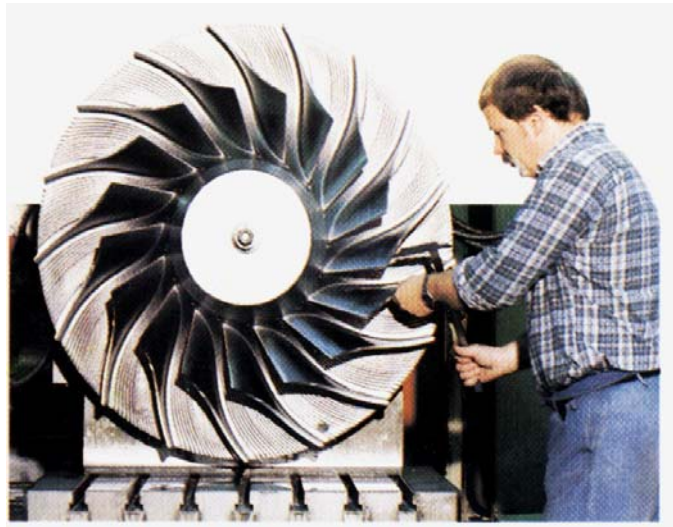Figure 1. A sectional view of a multi-stage centrifugal compressor.

Figure 2. A typical open-type impeller of a centrifugal compressor.

The main output of a compressor is the delivery of a certain mass flow rate of a gas at a pressure higher than the suction pressure. The stagnation pressure ratio for one stage (stagnation pressure at exit / stagnation pressure at inlet) may be as high as 7. Large centrifugal compressors are now manufactured to supply gases at pressures that may reach 150 bars or more and consume powers of 15 MW or more. Multiple compressor systems (more than one compressor driven by one prime mover) are also manufactured nowadays.

3) THEORETICAL BACKGROUND

When a gas is compressed using a centrifugal compressor, the main compressor output is increasing the pressure. Let the condition at inlet be labeled with subscript 1 and at exit be labeled with subscript 2. Since heat losses during the compression process are negligibly small, one can write the energy equation in the form (SSSF process)
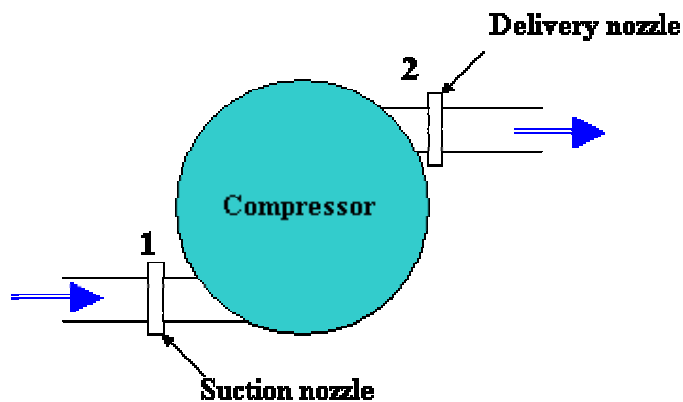
$$\cancel{q} + h_1 + \frac{V_1^2}{2} + \cancel{gz_1} = h_2 + \frac{V_2^2}{2} + \cancel{gz_2} + w$$

$$Or \quad w = \left(h_1 + \frac{V_1^2}{2} \right) - \left(h_2 + \frac{V_2^2}{2} \right) = h_{01} - h_{02} \tag{1}$$

where h_{01} and h_{02} are the stagnation enthalpies at inlet and exit respectively. The definitions of the stagnation state and all stagnation properties are given in Appendix A. The term w in equation (1) is defined as the work done by the compressor according to the thermodynamic convention and has a negative value. If we redefine the actual work done $w_{act.}$ to indicate the work input to the compressor, then equation (1) becomes

$$w_{act.} = h_{02} - h_{01} = C_p \left(T_{02} - T_{01} \right) \tag{2}$$

The compression process can be presented on the h-s diagram as shown below. This process is accompanied by entropy increase because of friction (irreversible process). On the same graph, the ideal compression process (isentropic process 01→02i) is also plotted.
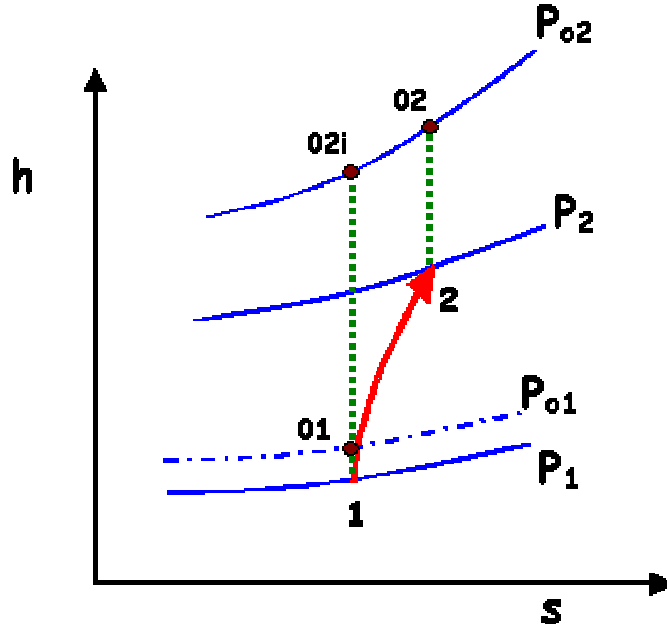

Figure 3. The thermodynamic processes for the actual and ideal compressors.

Similarly, the work input to an ideal compressor, w_{ideal}, is obtained from

$$w_{ideal} = h_{02i} - h_{01} = C_p \left(T_{02i} - T_{01} \right) \tag{3}$$

The thermodynamic efficiency of a centrifugal compressor is defined as follows:

$$\eta_c = \frac{w_{ideal}}{w_{act.}} = \frac{h_{02i} - h_{01}}{h_{02} - h_{01}} = \frac{T_{02i} - T_{01}}{T_{02} - T_{01}} \tag{4}$$

The compressor overall efficiency is defined as the ideal power required to drive the compressor divided by the actual brake power, thus

$$\eta_o = \frac{\dot{m}\, w_{ideal}}{B.P.} = \frac{\dot{m}\left(h_{02i} - h_{01}\right)}{B.P.} \tag{5}$$

The difference between the thermodynamic efficiency and the overall efficiency is mainly due to mechanical losses.

The ideal power required (sometimes called theoretical adiabatic power) can be expressed in terms of the stagnation pressure ratio as follows:

$$P_{ideal} = \dot{m}\, w_{ideal} = \dot{m}\left(h_{02i} - h_{01}\right) = \dot{m}\, C_p \left(T_{02i} - T_{01}\right)$$

$$= \dot{m}\, C_p T_{01} \left(\frac{T_{02i}}{T_{01}} - 1\right) = \dot{m}\, C_p T_{01} \left[\left(\frac{p_{02}}{p_{01}}\right)^{\frac{k-1}{k}} - 1\right] \tag{6}$$

The important variables needed to describe the compressor performance include the mass flow rate ($\dot{m}$), the stagnation pressure rise (Δp_o), brake power (B.P.), and the overall efficiency (η_o). The compressor performance is normally presented in three graphs when operating at a constant speed. These are Δp_o versus $\dot{m}$, B.P. versus $\dot{m}$, and η_o versus $\dot{m}$. Figure 4 represents typical performance curves for a centrifugal compressor.
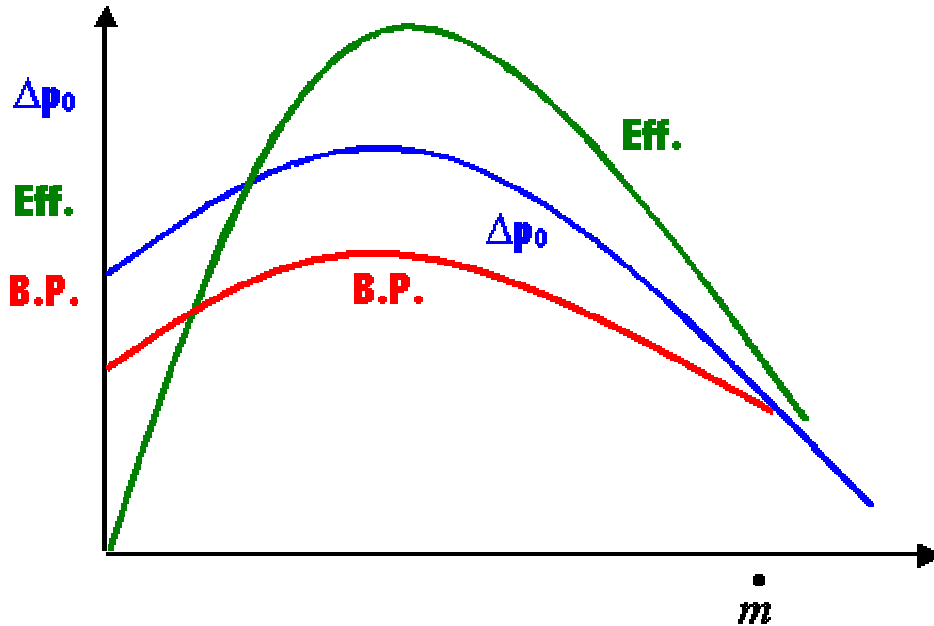

Figure 4. Typical performance curves for a centrifugal compressor.

4) APPARATUS

The apparatus to be used is the Armfield centrifugal compressor (model # FM12). Figure 4 represents a general view of the apparatus, which consists of the following:
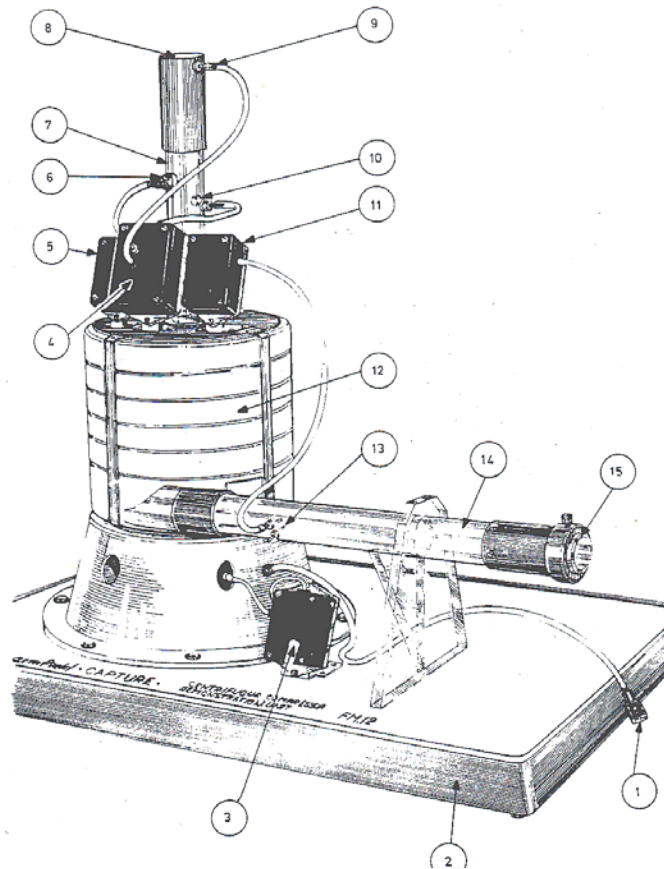

Figure 4. A general view of the centrifugal compressor showing the main components.

i) A seven-stage centrifugal compressor driven by 120W AC induction motor. The maximum pressure is 4.5kPa and the maximum flow rate is 0.02 m^3/s. The pressure increase across the machine is lower than typical compressors since this is a demonstration unit.

ii) Transparent inlet and outlet ducting to permit visual observation of the measurement system.

iii) Adjustable aperture device to permit variation of the airflow rate, which is monitored by an orifice plate.

iv) Measurement sensor for monitoring the pressure increase developed across the blower.

v) Sensors for measuring the pressure difference across the orifice plate for the determination of the flow rate and for measuring the temperature at the compressor inlet and the rotational speed of the impeller.

vi) A power supply unit equipped with speed control.

vii) Interface console (model # IFD6) to enable linking all sensors a PC.

viii) A PC equipped with special software for analysis of experimental data and also for plotting the compressor performance curves.

5) TEST PROCEDURE

a) Test Procedure

1. Place the Centrifugal Compressor Unit in a suitable location adjacent to a compatible PC and place the Interface Console IFD1/IFD1A alongside the PC. Also, place the integrating Wattmeter SWA1 on top of or alongside the IFD1 as convenient.

2. Make all electrical and tapping connections as described in the apparatus manual.

3. Switch on the mains supply and make sure that the compressor operates.

4. Adjust the throttle valve on the discharge duct to complete shutoff position.

5. Use the variable transformer speed control unit (SWA1) to adjust the compressor speed at 1200 rpm.

6. Use the computer interface to record the static pressure rise through the compressor (Δp_{static}), the pressure difference across the orifice plate ($\Delta p_{orifice}$), air inlet temperature, and the compressor input power (brake power). Data are recorded by pressing the SPACE BAR once readings have stabilized.

7. Slightly open the throttle valve on the discharge duct and repeat steps 5 and 6. Flow rate is monitored by measurement, of differential pressure across the orifice plate ($\Delta p_{orifice}$). As the relationship between $\Delta p_{orifice}$ and flow rate is non-linear care should be taken to select suitably spaced samples (e.g. 0, 0.02, 0.08, 0.18, 0.33, 0.51, 0.73, 1.00 kPa).

8. Repeat step 7 until the throttle valve reaches the fully open position. Up to 12 samples can be taken in sequence between minimum and maximum flow rates.

9. Repeat steps 5-8 for one more speed (2400 rpm).

b) Calculations

1. The diameters of the orifice, suction pipe and delivery pipe are as follows:

    D_s=32 mm, D_d=39 mm, $D_{orifice}$ =35 mm

2. Calculate the air density from the equation of state, $\rho = p/RT$.

3. Calculate the volume flow rate from the equation $Q = C_d \dfrac{\pi}{4} D_{orifice}^2 \sqrt{2\Delta p_{orifice}/\rho}$. The coefficient of discharge for the orifice, $C_d = 0.596$.

4. Determine the air velocity in the suction and the delivery pipes from the equation $V_s = Q/A_s$ and $V_d = Q/A_d$.

5. Determine the stagnation pressure rise across the compressor from the equation

    $$\Delta p_0 = \Delta p_{static} + \frac{1}{2}\rho\left(V_d^2 - V_s^2\right)$$

6. Determine the output fluid power from the equation $P_{fluid} = Q\Delta p_0$. This equation is used because of the small pressure through the nmachine.

7. Calculate the compressor overall efficiency from the equation $\eta_o = P_{fluid}/B.P.$

6) <u>PRESENTATION OF RESULTS</u>

1. It is required to plot the following graphs based on your own calculations for the low speed (1200 rpm):

   - Δp_0 versus Q

   - B.P. versus Q

   - η_o versus Q

2. Repeat the above for the high speed (2400 rpm).

3. Present also the curves obtained from the computer output for the two speeds and comment on any differences.


7) <u>IDEAS FOR FURTHER DISCUSSIONS</u>

1. What is the effect of speed of rotation on the stagnation pressure rise through the machine?

2. What causes the difference between your calculated results and the computer output?

3. Is there a need for calibrating the pressure and temperature sensors?

APPENDIX A

Stagnation Properties for an Ideal Gas

The stagnation state is defined as the state of the fluid that will be reached if the fluid is brought to rest through an isentropic process. The figure below shows a hypothetical isentropic process in which part of the fluid stream is taken to a large reservoir. In this case, the state in the reservoir is the stagnation state.
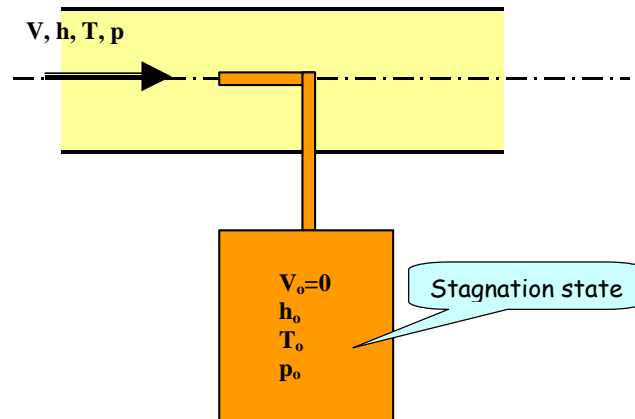
i) The Stagnation Enthalpy "h_o"

If we write the energy equation between any two points 1 and 2 along the same streamline, then

$$q + h_1 + \frac{V_1^2}{2} + gz_1 = h_2 + \frac{V_2^2}{2} + gz_2 + w \qquad \text{(A1)}$$

For the case of isentropic flow in a flow passage (no heat transfer, no friction, and with no work), the above equation reduces to

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} \qquad \text{(A2)}$$

If state 2 is replaced by the stagnation state (denoted by the subscript 0), then

$$h + \frac{V^2}{2} = h_0 \qquad \text{(A3)}$$

where h_o is the stagnation enthalpy.

ii) The Stagnation Temperature "T_o"

Using equation (A3) and assuming ideal gas behavior, then

$$h_0 = h + \frac{V^2}{2} \quad \Rightarrow \quad C_p T_0 = C_p T + \frac{V^2}{2} \qquad \text{but} \qquad C_p = \frac{kR}{k-1}$$

therefore, $\quad T_0 = T + \frac{(k-1)V^2}{2kR} = T[1 + \frac{k-1}{2}\frac{V^2}{kRT}]$ and since $kRT = C^2$

$$T_0 = T\left(1 + \frac{k-1}{2}\frac{V^2}{C^2}\right) \quad \text{and} \quad V/C = \text{Mach number "M", therefore,}$$

$$\frac{T_0}{T} = 1 + \frac{k-1}{2}M^2 \tag{A4}$$

iii) The Stagnation Pressure "p_o"

Since the process from the flow condition to the stagnation state is isentropic by definition, then

$$p_o / p = [T_o / T]^{\frac{k}{k-1}} , \quad \text{therefore, by using equation (A4), one can write}$$

$$\frac{p_0}{p} = [1 + \frac{k-1}{2}M^2]^{\frac{k}{k-1}} \tag{A5}$$

iv) The Stagnation Density "ρ_o"

The stagnation density can be obtained from the equation of state $\left(\rho_o = p_o / RT_o\right)$ and by using equations (A4 and A5), one can write

$$\frac{\rho_0}{\rho} = [1 + \frac{k-1}{2}M^2]^{\frac{1}{k-1}} \tag{A5}$$