4.1. DESCRIPTION OF THE GAS TURBINE CYCLE AND THERMAL ANALYSIS

The power output of gas turbine can be increased by inter-cooling. The compressed air from low pressure compressor during delivery to high pressure compressor is cooled in the intercooler. Therefore the compressionis performed in two stages. The compressed cooled air has lesser volume, enabling air to be compressed in asmaller compressor with less expenditure of energy.

Clearly the work required for compression is reduced withintercooler. The heat supplied with inter-cooling is more than that with the heat supplied in single stagecompression. The net output is also increased but thermal efficiency falls due to increased heat supply.

The system diagram, which was analyzed, is depicted in fig.4.1. The system consists of a low-pressurecompressor, intercooler, high-pressure compressor, combustion chamber and turbine.

Consider fig. 4.2 and assume that the compressor is working between the thermodynamic states 1 and 2. If theair is cooled from state 2 to 3 the required compressor power is decreased and the net cycle power delivered isincreased if the inlet temperature is reduced.

Efficient compression of large volumeof air is essential for a successful gas turbine power plant. It is assumed that the turbine efficiency ($\boldsymbol{\eta_T}$),compressor efficiency ($\boldsymbol{\eta_C}$) and effectiveness of intercooler($\boldsymbol{\varepsilon}$).

4.2 MODEL DESCRIPTION:

Intercooling is a way to reduce the power consumption for compression of an air. Thus, the inlet temperature of the second compressor stage can be kept low. For a given compression ratio, the power consumed in a compressor is directly proportional to the inlet temperature. Consider Figure 4.1 and assume that the compressor is working between the thermodynamic states 1 and 2. If the air is cooled from state 2 to 3 the required compressor power is decreased and the net cycle power delivered is increased if the inlet temperature is reduced Figure 4.2 shows a gas turbine power plant with intercooler has a single shaft gas turbine. In this gas turbine cycle with intercooler, air after compression in the first stage compressor enters into an intercooler where it is cooled. The cooled air then enters into second stage compressor to compress to required pressure, and then goes to combustion chamber, after additional heating to maximum permissible temperature in the combustion chamber. The network output of the cycle is thus proportional to the temperature drop in the turbine.

The actual processes and ideal processes arerepresented in dashed line and full line, respectively, on the T-S diagram (fig. 4.1).

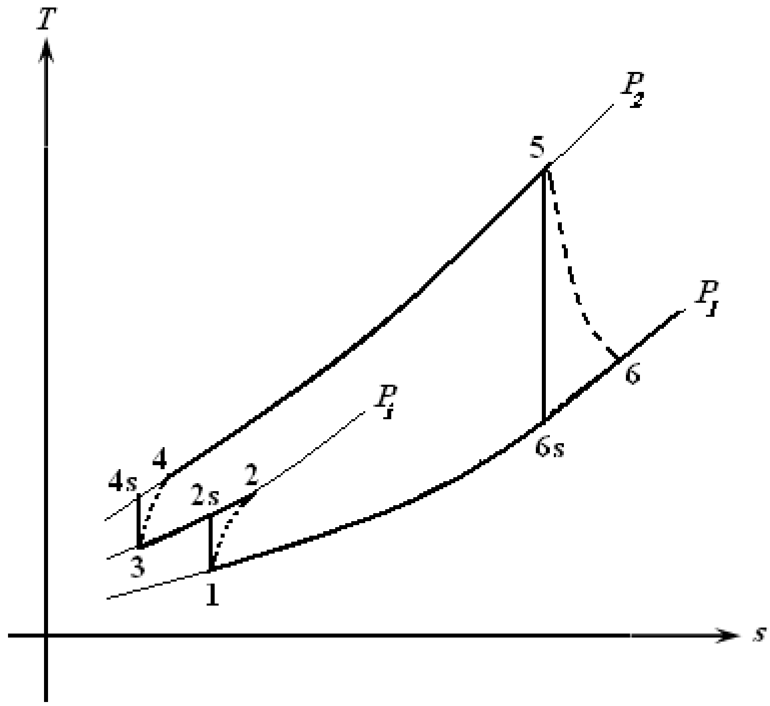

Fig. 4.1: T-s diagram for actual Brayton cycle with intercooler

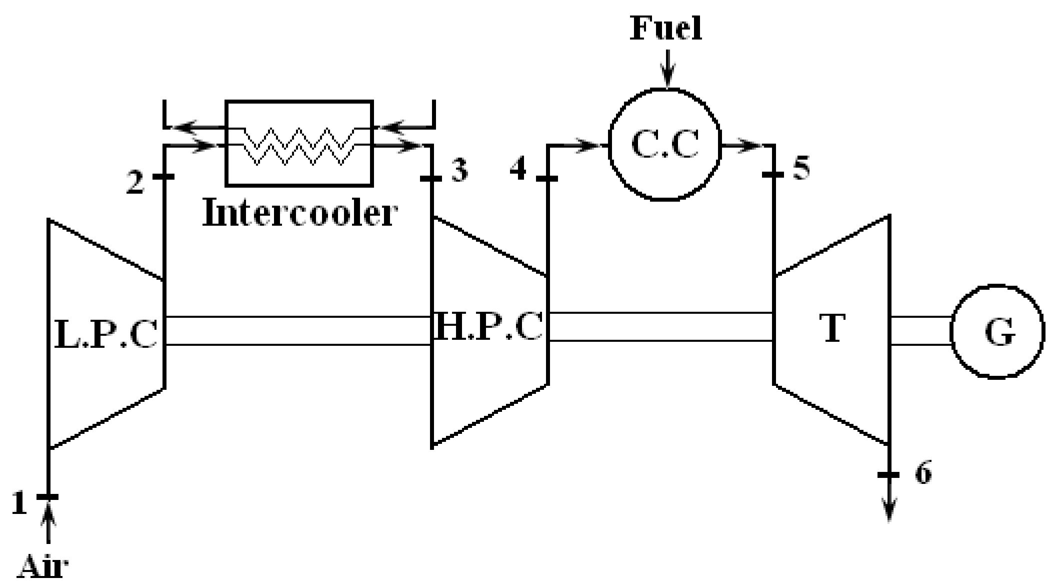

Fig. 4.2: Schematic diagram for actual Brayton cycle with intercooler

To calculate the temperature in any point uses these equations:

$$\frac{T_{2s}}{T_1} = \left(r_{p_L}\right)^{\frac{\gamma_a - 1}{\gamma_a}} \qquad \dots \dots (4-1)$$

Where:

T_{2s} ≡Theoretical low pressure compressor outlet temperature (K).

T_1 ≡ Ambient temperature (K)

r_{p_L} ≡First stage pressure ratio.

$$\frac{T_{4s}}{T_3} = \left(r_{p_H}\right)^{\frac{\gamma_a - 1}{\gamma_a}} \qquad \dots \dots (4-2)$$

Where:

T_{4s} ≡Theoretical high pressure compressor outlet temperature (K).

T_3 ≡Intercooler outlet temperature (K).

r_{p_H} ≡Second stage pressure ratio.

$$\frac{T_{6s}}{T_5} = \left(\frac{1}{r_{p_T}}\right)^{\frac{\gamma_g - 1}{\gamma_g}} \qquad \dots \dots (4-3)$$

T_{6s} ≡Theoretical turbineoutlet temperature (K).

T_5 ≡Turbine inlet temperature (K).

r_{p_T} ≡Total pressure ratio.

The effectiveness for intercooler can be defined as:

$$\boldsymbol{\varepsilon = \frac{T_2 - T_3}{T_2 - T_1}} \qquad \ldots\ldots(\mathbf{4-4})$$

After calculate all temperatures and effectiveness of inter cooler we can calculate isentropic efficiency of first and second stages of compression as following:

$$\boldsymbol{\eta_{C1} = \frac{T_{2s} - T_1}{T_2 - T_1}} \qquad \ldots\ldots(\mathbf{4-5})$$

Where:

$\boldsymbol{\eta_{C1}}$ ≡Isentropic efficiency of first stage compressor.

$$\boldsymbol{\eta_{C2} = \frac{T_{4s} - T_3}{T_4 - T_3}} \qquad \ldots\ldots(\mathbf{4-6})$$

Where:

$\boldsymbol{\eta_{C2}}$ ≡Isentropic efficiency of second stage compressor.

Then the thermal efficiency can be calculated from this formula

$$\boldsymbol{\eta_{T} = \frac{T_5 - T_6}{T_5 - T_{6s}}} \qquad \ldots\ldots(\mathbf{4-8})$$

$\boldsymbol{\eta_{T}}$ ≡Isentropic efficiency of turbine.

The work required to run the low pressure compressor is:

$$\boldsymbol{w_{C1} = C_{p_{a1}}(T_2 - T_1)} \qquad \ldots\ldots(\mathbf{4-9})$$

The work required to run the high pressure compressor is:

$$\boldsymbol{w_{C2} = C_{p_{a2}}(T_4 - T_3)} \qquad \dots \dots (\mathbf{4-10})$$

The work of gas turbine is:

$$\boldsymbol{w_T = C_{p_g}(T_5 - T_6)} \qquad \dots \dots (\mathbf{4-11})$$

The net work can give by:

$$\boldsymbol{w_{net} = w_T - (w_{C1} + w_{C2})} \qquad \dots \dots (\mathbf{4-12})$$

Heat addition:

$$\boldsymbol{q_{add} = C_{p_g}(T_5 - T_4)} \qquad \dots \dots (\mathbf{4-13})$$

The combustion energy balance equation used:

$$\boldsymbol{\dot{m}_f \times C.V = \dot{m}_a \times C_{p_g}(T_5 - T_4)} \qquad \dots \dots (\mathbf{4-14})$$

$C.V$ ≡Calorific value of fuel (kJ/kg).

To calculate specific fuel consumption we use the following formula

$$\boldsymbol{S.F.C = \frac{\dot{m}_f \times 3600}{w_{net}}} \qquad \dots \dots (\mathbf{4-15})$$

Where:

$\boldsymbol{\dot{m}_f}$ ≡Mass flow rate of fuel (kg/sec).

We assume the specific heat of air is given by

$$\left.\begin{aligned} \boldsymbol{C_{p_{a1}}} &= \mathbf{1.0189 \times 10^3 - 0.1378\,}\boldsymbol{T_a}\mathbf{ + 1.9843 \times 10^{-4}}\boldsymbol{T_a^2} \\ &\quad \mathbf{+4.2399 \times 10^{-7}}\boldsymbol{T_a^3}\mathbf{ - 3.7632 \times 10^{-10}}\boldsymbol{T_a^4} \end{aligned}\right\} \qquad \dots \dots (\mathbf{4-16})$$

Where;

$\boldsymbol{T_a}$ ≡The logarithmic mean temperature for compression processes (K).

For first stage compression, the logarithmic mean temperature can be given as follow:

$$\boldsymbol{T_a = \frac{T_2 - T_1}{\ln \frac{T_2}{T_1}}} \qquad \ldots\ldots(\mathbf{4-17})$$

Similarly, the logarithmic mean temperature for second stage compression can be given as follow:

$$\boldsymbol{T_{a2} = \frac{T_4 - T_3}{\ln \frac{T_4}{T_3}}} \qquad \ldots\ldots(\mathbf{4-18})$$

The specific heat of gas is given by

$$\boldsymbol{C_{p_g} = 1.8083 - 2.3127 \times 10^{-3} T_g + 4.045 \times 10^{-6} T_g^2 - 1.7363 \times 10^{-9} T_g^3} \quad \ldots\ldots(\mathbf{4-19})$$

Where;

$\boldsymbol{T_g}$ ≡The logarithmic mean temperature for compression processes (K).

The logarithmic mean temperature for expansion can be given as follow:

$$\boldsymbol{T_g = \frac{T_6 - T_5}{\ln \frac{T_6}{T_5}}} \qquad \ldots\ldots(\mathbf{4-20})$$

For simplification, we will assume the gas constant for air equal the gas constant for exhaust gases and both of them are equal 0.287 kJ/kg.K.

Also we will assume the specific heat ratio ($\boldsymbol{\gamma}$) for air is equal 1.4 and for exhaust gases is equal 1.33.

4.3 DETERMINING INTERCOOLER PRESSURE FOR MINIMUM COMPRESSOR WORK

For fixed inlet state and exit pressure, use a cold-air standard analysis to show that minimum total work input for a two-stage compressor is required when the pressure ratio is the same across each stage. Assume steady-state operation and the following idealizations: Each compression process is isentropic, there is no pressure drop through the intercooler, temperature at the inlet to each compressor stage is the same, and kinetic and potential energy effects can be ignored.

Assumptions:

1. The compressor stages and intercooler are analyzed as control volumes at steady state.
2. The compression processes are isentropic.
3. There is no pressure drop for flow through the intercooler.
4. The temperature at the inlet to both compressor stages is the same.
5. Kinetic and potential energy effects are negligible.
6. The working fluid is air modeled as an ideal gas.
7. The specific heat C_p and thus the specific heat ratio γ are constant.

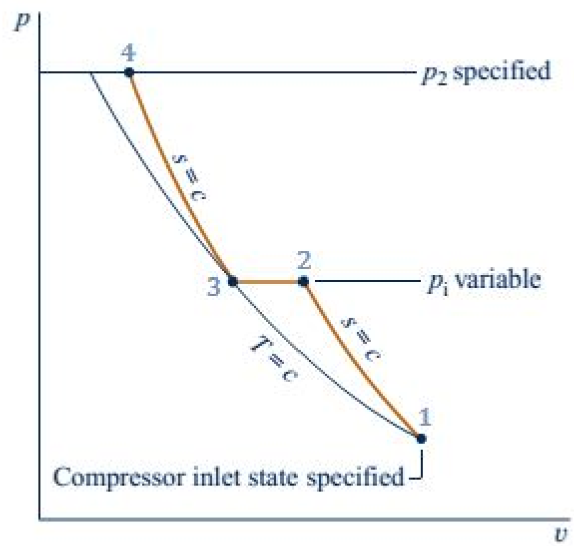

Fig. 4.3: *P-v* diagram for two stage compression with intercooler

The total compressor work input per unit of mass flow is

$$w_c = (h_2 - h_1) + (h_4 - h_3) \qquad \dots\dots (4-21)$$

Since C_p is constant

$$w_c = C_p(T_2 - T_1) + C_p(T_4 - T_3) \qquad \dots\dots (4-22)$$

With $T_3 = T_1$ (assumption 4), this becomes on rearrangement

$$w_c = C_p T_1 \left(\frac{T_2}{T_1} + \frac{T_4}{T_1} - 2\right) \qquad \dots\dots (4-23)$$

Since the compression processes are isentropic and the specific heat ratio γ is constant, the pressure and temperature ratios across the compressor stages are related, respectively, by

$$\frac{T_2}{T_1} = \left(\frac{P_i}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \qquad \dots\dots (4-24)$$

$$\frac{T_4}{T_3} = \frac{T_4}{T_1} = \left(\frac{P_2}{P_i}\right)^{\frac{\gamma-1}{\gamma}} \qquad \ldots\ldots(4-25)$$

Substituting in the equation

$$w_c = C_p T_1 \left[\left(\frac{P_i}{P_1}\right)^{\frac{\gamma-1}{\gamma}} + \left(\frac{P_2}{P_i}\right)^{\frac{\gamma-1}{\gamma}} - 2\right] \qquad \ldots\ldots(4-26)$$

Hence, for specified values of T_1, P_1, P_2 and C_p, the value of the total compressor work input varies with the intercooler pressure only. To determine the pressure P_i that minimizes the total work, form the derivative

$$\frac{\partial w_c}{\partial P_i} = \frac{\partial}{\partial P_i}\left\{C_p T_1 \left[\left(\frac{P_i}{P_1}\right)^{\frac{\gamma-1}{\gamma}} + \left(\frac{P_2}{P_i}\right)^{\frac{\gamma-1}{\gamma}} - 2\right]\right\} \qquad \ldots\ldots(4-27)$$

$$= C_p T_1 \left(\frac{\gamma-1}{\gamma}\right)\left[\left(\frac{P_i}{P_1}\right)^{-\frac{1}{\gamma}}\left(\frac{1}{P_1}\right) + \left(\frac{P_2}{P_i}\right)^{-\frac{1}{\gamma}}\left(-\frac{P_2}{P_i^2}\right)\right] \qquad \ldots\ldots(4-28)$$

$$= C_p T_1 \left(\frac{\gamma-1}{\gamma}\right)\frac{1}{P_i}\left[\left(\frac{P_i}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - \left(\frac{P_2}{P_i}\right)^{\frac{\gamma-1}{\gamma}}\right] \qquad \ldots\ldots(4-29)$$

When the partial derivative is set to zero, the desired relationship is obtained

$$\frac{P_i}{P_1} = \frac{P_1}{P_i} \qquad \ldots\ldots(4-30)$$

or

$$P_i = \sqrt{P_1 P_1} \qquad \ldots\ldots(4-31)$$

4.4 CALCULATIONS METHODOLOGY

After mention all equation we will use MATLAB program to shows the result of effect of parameter in graphics in chapter five. We study the changes in inter-cooler effectiveness with other parameter such as ambient temperature and turbine inlet temperature.