Thermodynamics
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Lecture 55
Tutorial Problem (3 Numbers).

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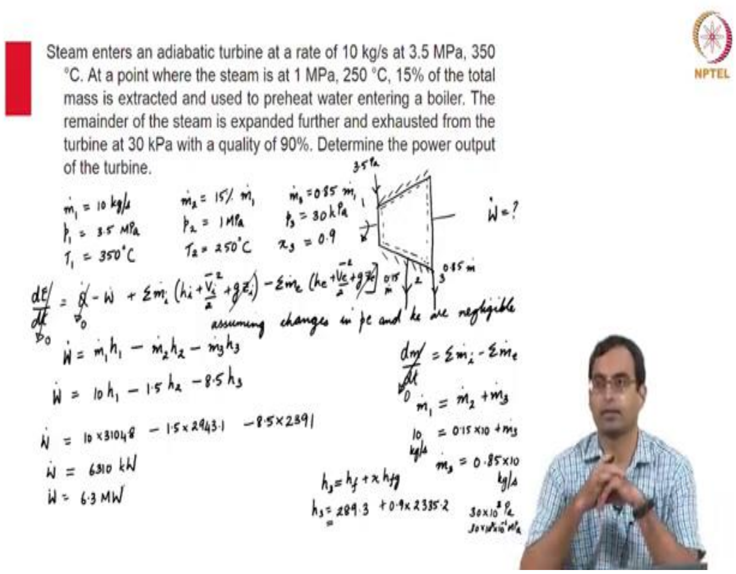
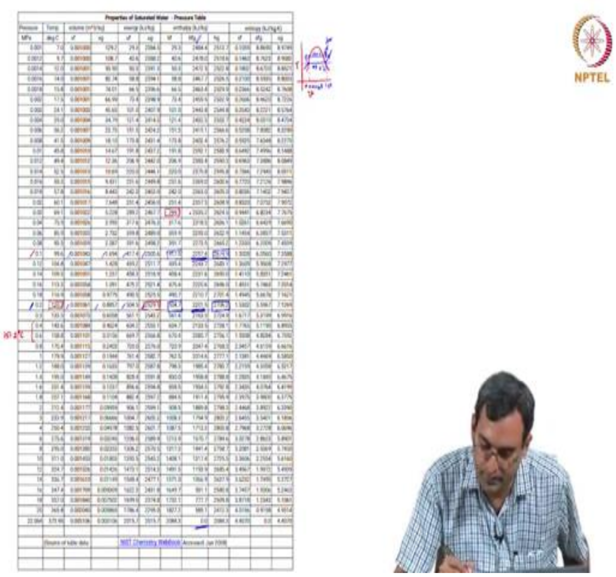


Figure 1.



Solution of the problem in Fig. 1:

$\dot{m}_1 = 10\frac{kg}{s}, p_1 = 3.5\ MPa, T_1 = 350\ ℃,$

$\dot{m}_2 = 0.15\dot{m}_1 = 1.5\frac{kg}{s}, p_2 = 1\ MPa, T_2 = 250\ ℃,$

$p_3 = 30\ kPa, x_3 = 0.9$

A schematic of the turbine is shown in Fig. 1.

The turbine is our control volume. The first law for a control volume,

$$\frac{dE}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \sum \dot{m}_i \left(h_i + \frac{1}{2}\bar{V}_i^{\,2} + gZ_i\right) - \sum \dot{m}_e \left(h_e + \frac{1}{2}\bar{V}_e^{\,2} + gZ_e\right) \ldots(1)$$

The process is adiabatic ($\dot{Q} = 0$). We assume that the process is steady $\left(\frac{dE}{dt} = 0\right)$. Since we are not given any information regarding the velocities and heights at inlet and outlets, we will assume that the changes in kinetic and potential energy are negligible. Hence, equation (1) implies,

$$\dot{W} = \dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3 \ldots.(2)$$

Now, $\dot{m}_1 = \dot{m}_2 + \dot{m}_3 \;\; \rightarrow \;\; \dot{m}_3 = 0.85\dot{m}_1 = 8.5\,\frac{kg}{s}$

We will obtain values of h_1, h_2, h_3 from steam tables.

At the inlet, at 350 °C, the saturation pressure p_{sat}=16 MPa. Since $p_{sat} > 3.5\ MPa,$ the steam is in superheated zone. From the tables for superheated steam, $h_1 = 3104.8\,\frac{kJ}{kg}$.

At the outlet 2, at 250 °C, $p_{sat} = 4\ MPa$. Since $p_{sat} > 1\ MPa$, the steam is in superheated zone. From the tables for superheated steam, $h_2 = 2943.1\,\frac{kJ}{kg}$.

At the outlet 3, $h_3 = \left[h_f + x_3\left(h_{fg}\right)\right]_{30\ kPa} = 2391\,\frac{kJ}{kg}$

(2) implies $\dot{W} = 10 \times 3104.8 + 1.5 \times 2943.1 + 8.5 \times 2391 = 6.3\ MW$.

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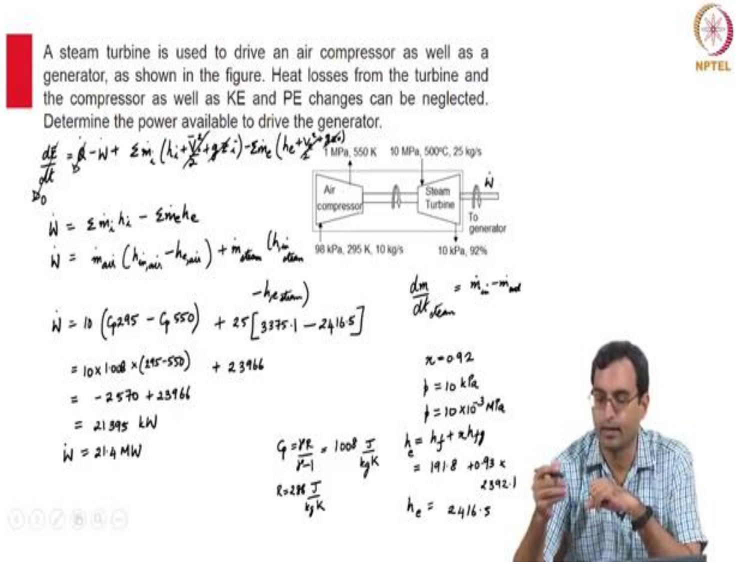

Figure 2.

Solution of the problem in Fig. 2:

Conditions at the inlet and the outlet of the air compressor and the turbine are shown in Fig. 1. The control volume includes both, the air compressor and the turbine.

The first law for a control volume

$$\frac{dE}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \sum \dot{m}_i \left(h_i + \frac{1}{2}\bar{V}_i^{\,2} + gZ_i \right) - \sum \dot{m}_e \left(h_e + \frac{1}{2}\bar{V}_e^{\,2} + gZ_e \right) \ldots (1)$$

Heat losses from the control volume and the changes in kinetic and potential energy can be neglected. Assume steady state conditions. Hence, equation (1) reduces to

$$\dot{W} = \sum \dot{m}_i h_i - \sum \dot{m}_e h_e$$

For the compressor and the turbine, $\frac{dm}{dt} = 0$. The air and steam do not mix. Hence, $\dot{m}_{i_{air}} = \dot{m}_{e_{air}} = \dot{m}_{air}$ and $\dot{m}_{i_{steam}} = \dot{m}_{e_{steam}} = \dot{m}_{steam}$

Hence,

$$\dot{W} = \dot{m}_{air}\left(h_{i_{air}} - h_{e_{air}}\right) + \dot{m}_{steam}(h_{i_{steam}} - h_{e_{steam}}) \ldots (2)$$

Now, at the inlet of turbine, at 10 MPa, $T_{sat} = 300\ ℃$. Since $500\ ℃ > T_{sat}$, the state is in superheated zone. From the tables for superheated steam, $h_{i_{steam}} = 3375.1 \frac{kJ}{kg}$.

At the exit of turbine, $x_2 = 0.92$. $h_{e_{steam}} = \left[h_f + x_2\left(h_{fg}\right)\right]_{10\ kPa} = 2416.5 \frac{kJ}{kg}$

Assuming air an ideal gas and C_p is constant, $h_{i_{air}} = C_p T_i$ and $h_{e_{air}} = C_p T_e$. $C_p = \frac{\gamma R}{\gamma - 1} = \frac{1.4 \times 288}{0.4} = 1008\ \frac{J}{kg \cdot K}$ ($\gamma = 1.4$ assuming air to be a diatomic gas)

(2) implies $\dot{W} = 10 \times 1008 \times (295 - 550) + 25(3375.1 - 2416.5) = -2570 + 23966 = 21395\ kW = 21.4\ MW$.

The turbine produces 23966 kW of power. The compressor consumes 2570 kW. 21395 kW is available for the generator.

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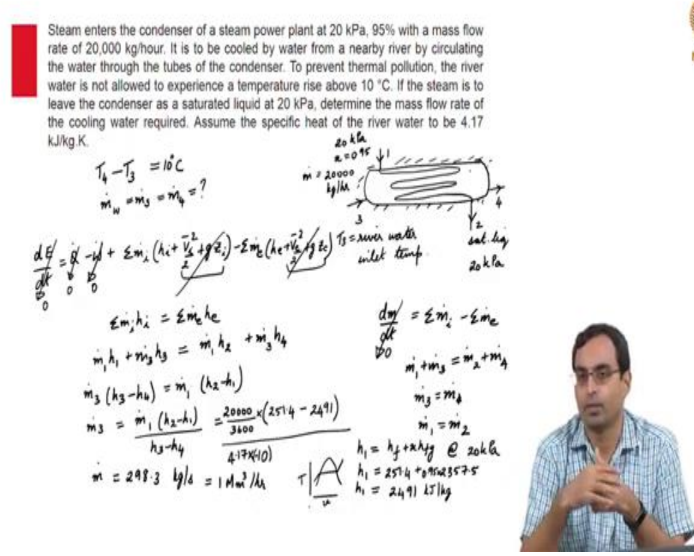

Figure 3.

Solution of the problem in Fig. 3:

Out control volume includes the condenser (heat exchanger). A schematic is drawn in Fig. 3. It has two inlets (1 and 3) and two outlets (2 and 4).

$T_4 - T_3 = 10$ ℃.

Assume that the operation is at steady state conditions. The river water and the steam do not mix.

Hence, $\dot{m}_1 = \dot{m}_2$ and $\dot{m}_3 = \dot{m}_4$.

The first law for a control volume,

$$\frac{dE}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \sum \dot{m}_i \left(h_i + \frac{1}{2}\bar{V}_i^{\,2} + gZ_i\right) - \sum \dot{m}_e \left(h_e + \frac{1}{2}\bar{V}_e^{\,2} + gZ_e\right) \ldots (1)$$

For a heat exchanger, $\dot{Q}_{cv} = \dot{W}_{cv} = 0$. Since the operation is at steady state conditions, $\frac{dE}{dt} = 0$. Also, the changes in the kinetic and potential energy are negligible. Hence,

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad \rightarrow \quad \dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4.$$

Since, $\dot{m}_1 = \dot{m}_2$ and $\dot{m}_3 = \dot{m}_4$, $\dot{m}_3(h_3 - h_4) = \dot{m}_1(h_2 - h_1) \rightarrow \dot{m}_3 = \frac{\dot{m}_1(h_2 - h_1)}{(h_3 - h_4)} =$ mass flow rate of the cooling water.

Now, $h_1 = \left[h_f + x_1\left(h_{fg}\right)\right]_{20\ kPa} = 2491\,\frac{kJ}{kg}$

$$h_2 = h_{f_{20\ kPa}} = 251.4\,\frac{kJ}{kg}$$

$$h_3 - h_4 = C_p(T_3 - T_4) = 4.17 \times (-10) = -41.7\,\frac{kJ}{kg}$$

Hence, $\dot{m}_3 = 298.3\,\frac{kg}{s}$ = mass flow rate of the cooling water.