Ministry of Higher Education and
scientific research
Middle Technical University
Engineering Technical College-Baghdad
Fuel and Energy Engineering Department

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Dr. Kadhum Audaa Jehhef
B. Sc., H. Dip., M.Sc. and Ph.D.
Mechanical Engineering.

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Bachelor Degree, FS302
Fuel and Energy Engineering Department
Engineering Technical College-Baghdad
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Chapter 5

STEAM NOZZLES

We have seen in Chapter 3 that water vapor at moderate or high pressures deviates considerably from ideal-gas behavior, and thus most of the relations developed in this chapter are not applicable to the flow of steam through the nozzles or blade passages encountered in steam turbines. Given that the steam properties such as enthalpy are functions of pressure as well as temperature and that no simple property relations exist, an accurate analysis of steam flow through the nozzles is no easy matter. Often it becomes necessary to use steam tables, an *h-s* diagram, or a computer program for the properties of steam.

A further complication in the expansion of steam through nozzles occurs as the steam expands into the saturation region, as shown in Fig. 17–59. As the steam expands in the nozzle, its pressure and temperature drop, and ordinarily one would expect the steam to start condensing when it strikes the saturation line. However, this is not always the case. Owing to the high speeds, the residence time of the steam in the nozzle is small, and there may not be sufficient time for the necessary heat transfer and the formation of liquid droplets. Consequently, the condensation of the steam may be delayed for a little while. This phenomenon is known as **supersaturation,** and the steam that exists in the wet region without containing any liquid is called **supersaturated steam.** Supersaturation states are nonequilibrium (or metastable) states.

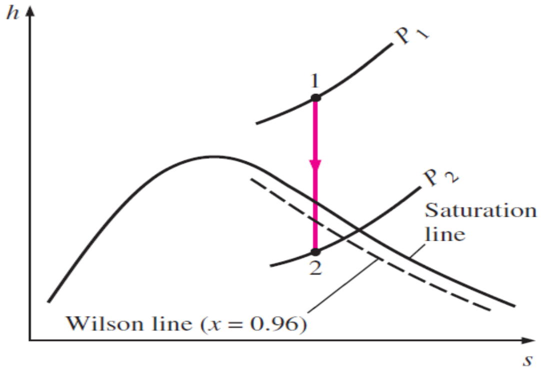


FIGURE 17–59

The *h-s* diagram for the isentropic expansion of steam in a nozzle.

During the expansion process, the steam reaches a temperature lower than that normally required for the condensation process to begin. Once the temperature drops a sufficient amount below the saturation temperature corresponding to the local pressure, groups of steam moisture droplets of sufficient size are formed, and condensation occurs rapidly. The locus of points where condensation takes place regardless of the initial temperature and pressure at the nozzle entrance is called the **Wilson line.** The Wilson line lies between the 4 and 5 percent moisture curves in the saturation region on the h-s diagram for steam, and it is often approximated by the 4 percent moisture line. Therefore, steam flowing through a high-velocity nozzle is assumed to begin condensation when the 4 percent moisture line is crossed.

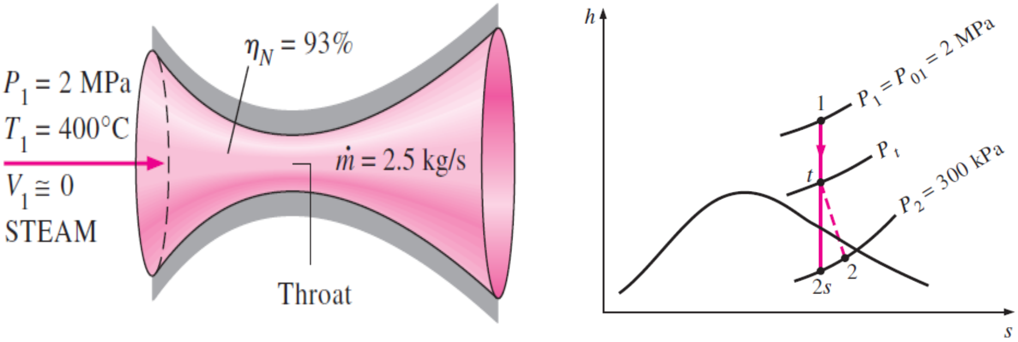
The critical-pressure ratio P^*/P_0 for steam depends on the nozzle inlet state as well as on whether the steam is superheated or saturated at the nozzle inlet. However, the ideal-gas relation for the critical-pressure ratio, Eq. 17–22, gives reasonably good results over a wide range of inlet states. As indicated in Table 17–2, the specific heat ratio of superheated steam is approximated as $k = 1.3$. Then the critical-pressure ratio becomes

$$\frac{P^*}{P_0} = \left(\frac{2}{k+1}\right)^{k/(k-1)} = 0.546$$

When steam enters the nozzle as a saturated vapor instead of superheated vapor (a common occurrence in the lower stages of a steam turbine), the critical-pressure ratio is taken to be 0.576, which corresponds to a specific heat ratio of $k = 1.14$.

EXAMPLE 17–16 Steam Flow through a Converging–Diverging Nozzle

Steam enters a converging–diverging nozzle at 2 MPa and 400°C with a negligible velocity and a mass flow rate of 2.5 kg/s, and it exits at a pressure of 300 kPa. The flow is isentropic between the nozzle entrance and throat, and the overall nozzle efficiency is 93 percent. Determine (*a*) the throat and exit areas and (*b*) the Mach number at the throat and the nozzle exit.



Analysis We denote the entrance, throat, and exit states by 1, *t*, and 2, respectively, as shown in Fig. 17–60.

(*a*) Since the inlet velocity is negligible, the inlet stagnation and static states are identical. The ratio of the exit-to-inlet stagnation pressure is

$$\frac{P_2}{P_{01}} = \frac{300 \text{ kPa}}{2000 \text{ kPa}} = 0.15$$

It is much smaller than the critical-pressure ratio, which is taken to be $P^*/P_{01} = 0.546$ since the steam is superheated at the nozzle inlet. Therefore, the flow surely is supersonic at the exit. Then the velocity at the throat is the sonic velocity, and the throat pressure is

$$P_t = 0.546P_{01} = (0.546)(2 \text{ MPa}) = 1.09 \text{ MPa}$$

At the inlet,

$$\left.\begin{aligned} P_1 &= P_{01} = 2 \text{ MPa} \\ T_1 &= T_{01} = 400°\text{C} \end{aligned}\right\} \quad \begin{aligned} h_1 &= h_{01} = 3248.4 \text{ kJ/kg} \\ s_1 &= s_t = s_{2s} = 7.1292 \text{ kJ/kg} \cdot \text{K} \end{aligned}$$

Also, at the throat,

$$\left.\begin{aligned} P_t &= 1.09 \text{ MPa} \\ s_t &= 7.1292 \text{ kJ/kg} \cdot \text{K} \end{aligned}\right\} \quad \begin{aligned} h_t &= 3076.8 \text{ kJ/kg} \\ v_t &= 0.24196 \text{ m}^3/\text{kg} \end{aligned}$$

Then the throat velocity is determined from Eq. 17–3 to be

$$V_t = \sqrt{2(h_{01} - h_t)} = \sqrt{[2(3248.4 - 3076.8)\ \mathrm{kJ/kg}]\left(\frac{1000\ \mathrm{m^2/s^2}}{1\ \mathrm{kJ/kg}}\right)} = 585.8\ \mathrm{m/s}$$

The flow area at the throat is determined from the mass flow rate relation:

$$A_t = \frac{\dot{m}v_t}{V_t} = \frac{(2.5\ \mathrm{kg/s})(0.2420\ \mathrm{m^3/kg})}{585.8\ \mathrm{m/s}} = 10.33 \times 10^{-4}\ \mathrm{m^2} = \mathbf{10.33\ cm^2}$$

At state 2*s*,

$$\left.\begin{aligned} P_{2s} &= P_2 = 300\ \mathrm{kPa} \\ s_{2s} &= s_1 = 7.1292\ \mathrm{kJ/kg \cdot K} \end{aligned}\right\} \quad h_{2s} = 2783.6\ \mathrm{kJ/kg}$$

The enthalpy of the steam at the actual exit state is (see Chap. 7)

$$\eta_N = \frac{h_{01} - h_2}{h_{01} - h_{2s}}$$

$$0.93 = \frac{3248.4 - h_2}{3248.4 - 2783.6} \longrightarrow h_2 = 2816.1\ \mathrm{kJ/kg}$$

Therefore,

$$\left.\begin{aligned} P_2 &= 300\ \mathrm{kPa} \\ h_2 &= 2816.1\ \mathrm{kJ/kg} \end{aligned}\right\} \quad \begin{aligned} v_2 &= 0.67723\ \mathrm{m^3/kg} \\ s_2 &= 7.2019\ \mathrm{kJ/kg \cdot K} \end{aligned}$$

Then the exit velocity and the exit area become

$$V_2 = \sqrt{2(h_{01} - h_2)} = \sqrt{[2(3248.4 - 2816.1)\ \mathrm{kJ/kg}]\left(\frac{1000\ \mathrm{m^2/s^2}}{1\ \mathrm{kJ/kg}}\right)} = 929.8\ \mathrm{m/s}$$

$$A_2 = \frac{\dot{m}v_2}{V_2} = \frac{(2.5\ \mathrm{kg/s})(0.67723\ \mathrm{m^3/kg})}{929.8\ \mathrm{m/s}} = 18.21 \times 10^{-4}\ \mathrm{m^2} = \mathbf{18.21\ cm^2}$$

(*b*) The velocity of sound and the Mach numbers at the throat and the exit of the nozzle are determined by replacing differential quantities with differences,

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left[\frac{\Delta P}{\Delta(1/v)}\right]_s^{1/2}$$

The velocity of sound at the throat is determined by evaluating the specific volume at $s_t = 7.1292$ kJ/kg · K and at pressures of 1.115 and 1.065 MPa ($P_t \pm 25$ kPa):

$$c = \sqrt{\frac{(1115 - 1065)\text{ kPa}}{(1/0.23776 - 1/0.24633)\text{ kg/m}^3}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kPa}\cdot\text{m}^3/\text{kg}}\right)} = 584.6\text{ m/s}$$

The Mach number at the throat is determined from Eq. 17–12 to be

$$\text{Ma} = \frac{V}{c} = \frac{585.8\text{ m/s}}{584.6\text{ m/s}} = \mathbf{1.002}$$

Thus, the flow at the throat is sonic, as expected. The slight deviation of the Mach number from unity is due to replacing the derivatives by differences.

The velocity of sound and the Mach number at the nozzle exit are determined by evaluating the specific volume at $s_2 = 7.2019$ kJ/kg · K and at pressures of 325 and 275 kPa ($P_2 \pm 25$ kPa):

$$c = \sqrt{\frac{(325 - 275)\text{ kPa}}{(1/0.63596 - 1/0.72245)\text{ kg/m}^3}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kPa}\cdot\text{m}^3/\text{kg}}\right)} = 515.4\text{ m/s}$$

and

$$\text{Ma} = \frac{V}{c} = \frac{929.8\text{ m/s}}{515.4\text{ m/s}} = \mathbf{1.804}$$

Thus the flow of steam at the nozzle exit is supersonic.

Problems

17–113C What is supersaturation? Under what conditions does it occur?

17–114 Steam enters a converging nozzle at 3.0 MPa and 500°C with a negligible velocity, and it exits at 1.8 MPa. For a nozzle exit area of 32 cm^2, determine the exit velocity, mass flow rate, and exit Mach number if the nozzle (*a*) is isentropic and (*b*) has an efficiency of 90 percent. *Answers:* (*a*) 580 m/s, 10.7 kg/s, 0.918, (*b*) 551 m/s, 10.1 kg/s, 0.865

17–116 Steam enters a converging–diverging nozzle at 1 MPa and 500°C with a negligible velocity at a mass flow rate of 2.5 kg/s, and it exits at a pressure of 200 kPa. Assuming the flow through the nozzle to be isentropic, determine the exit area and the exit Mach number. *Answers:* 31.5 cm^2, 1.738