Applications of Thermodynamics to Flow Processes

Flow processes inevitably result from pressure gradients within the fluid. Moreover, temperature, velocity, and even concentration gradients may exist within the flowing fluid. This contrasts with the uniform conditions that prevail at equilibrium in closed systems. The distribution of conditions in flow systems requires that properties be attributed to point masses of fluid. Thus we assume that intensive properties, such as density, specific enthalpy, specific entropy, etc., at a point are determined solely by the temperature, pressure, and composition at the point, uninfluenced by gradients that may exist at the point. Moreover, we assume that the fluid exhibits the same set of intensive properties at the point as though it existed at equilibrium at the same temperature, pressure, and composition.

1. TURBINES (EXPANDERS)

The expansion of a gas in a nozzle to produce a high-velocity stream is a process that converts internal energy into kinetic energy. This kinetic energy is in turn converted into shaft work when the stream impinges on blades attached to a rotating shaft. Thus a turbine (or expander) consists of alternate sets of nozzles and rotating blades through which vapor or gas flows in a steady-state expansion process whose overall effect is the efficient conversion of the internal energy of a high-pressure stream into shaft work. When steam provides the motive force as in a power plant, the device is called a turbine; when a high-pressure gas, such as ammonia or ethylene in a chemical or petrochemical plant, is the working fluid, the device is often called an expander. The process for either case is shown in Fig. below

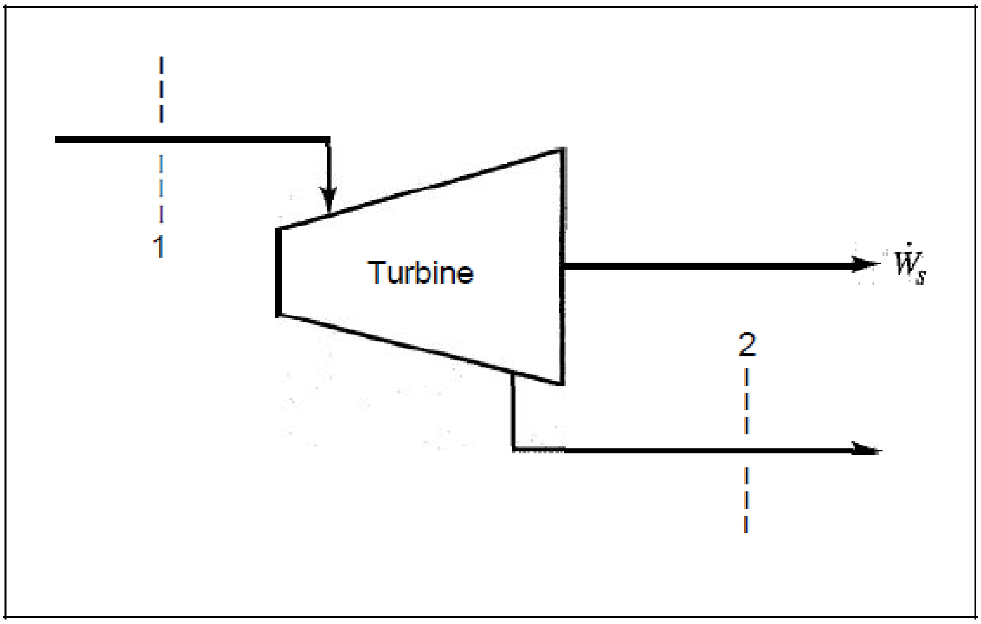

Figure 7.3 Steady-state flow through a turbine or expander

Equations (2.31) and (2.32) are appropriate energy relations. However, the potentialenergy term can be omitted, because there is little change in elevation. Moreover, in any properly designed turbine, heat transfer is negligible and the inlet and exit pipes are sized to make fluid velocities roughly equal. Equations (2.3 1) and (2.32) therefore reduce to:

$$\Delta\left(H + \tfrac{1}{2}u^2 + zg\right)\dot{m} = \dot{Q} + \dot{W}_s \qquad (2.31)$$

$$\boxed{\Delta H + \frac{\Delta u^2}{2} + g\,\Delta z = Q + W_s} \qquad (2.32a)$$

$$\dot{W}_s = \dot{m}\,\Delta H = \dot{m}(H_2 - H_1) \qquad (7.13)$$

$$W_s = \Delta H = H_2 - H_1 \qquad (7.14)$$

alone does not allow any calculations to be made. However, if the fluid in the turbine undergoes an expansion process that is reversible as well as adiabatic, then the process is isentropic, and $S_2 = S_1$. This second equation allows determination of the final state of the fluid and hence of H_2. For this special case, W, is given by Eq. (7.14), written:

$$W_s(\text{isentropic}) = (\Delta H)_S \tag{7.15}$$

The shaft work $|Ws\,,(isentropic)|$ is the maximum that can be obtained from an adiabatic turbine with given inlet conditions and given discharge pressure. Actual turbines produce less work, because the actual expansion process is irreversible. We therefore define a *turbine eficiency* as:

$$\eta \equiv \frac{W_s}{W_s(\text{isentropic})}$$

where W_s, is the actual shaft work. By Eqs. (7.14) and (7.15),

$$\eta = \frac{\Delta H}{(\Delta H)_S} \tag{7.16}$$

Values of η for properly designed turbines or expanders usually range from 0.7 to 0.8. Figure 7.4 shows an H_S diagram on which are compared an actual expansion process in a turbine and the reversible process for the same intake conditions and the same discharge pressure. The reversible path is a vertical line of constant entropy from point 1 at the intake pressure P_1 to point 2' at the discharge pressure P_2. The line representing the actual irreversible process starts also from point 1, but is directed downward and to the right, in the direction of increasing entropy. Since the process is adiabatic, irreversibilities cause an increase in entropy of the fluid. The process terminates at point 2 on the isobar for P_2. The more irreversible the process, the further this point lies to the right on the P_2 isobar, and the lower the efficiency η of the process.

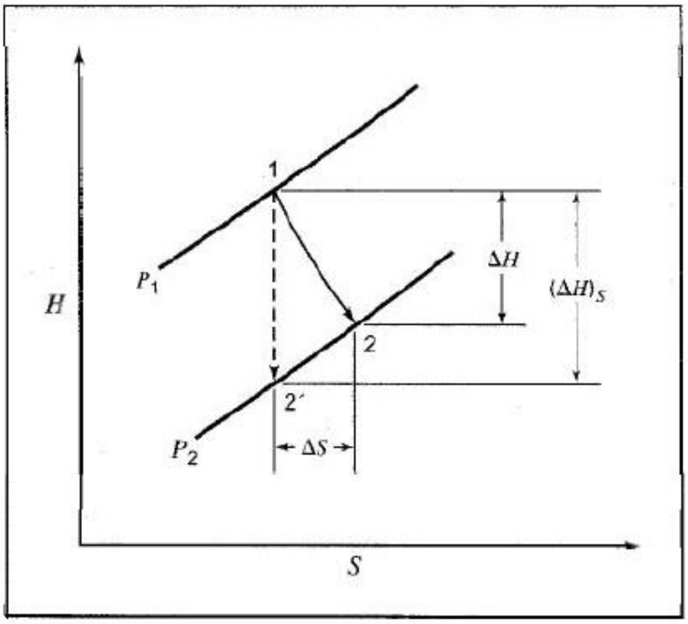

Figure 7.4 Adiabatic expansion process in a turbine or expander

Two-Phase Liquid / Vapor Systems

When a system consists of saturated-liquid and saturated-vapor phases coexisting in equilibrium, the total value of any extensive property of the two-phase system is the sum of the total properties of the phases. Written for the volume, this relation is:

$$nV = n^l V^l + n^v V^v$$

where V is the system volume on a molar basis and the total number of moles is $n = n^l + n^v$. Division by n gives:

$$V = x^l V^l + x^v V^v$$

where x^l and x^u represent the fractions of the total system that are liquid and vapor. Since $x^l = 1 - x^v$,

$$V = (1 - x^v)V^l + x^v V^v$$

In this equation the properties V, V^l, and V^v may be either molar or unit-mass values. The mass or molar fraction of the system that is vapor x^v is called the *quality*. Analogous equations can be written for the other extensive thermodynamic properties. All of these relations are represented by the generic equation:

$$M = (1 - x^v)M^l + x^v M^v \tag{6.73a}$$

where M represents V, U, H, S, etc. An alternative form is sometimes useful:

$$M = M^l + x^v \Delta M^{lv} \tag{6.73b}$$

Example 7.6

A steam turbine with rated capacity of 56,400 kW (56,400 kJ·s^{-1}) operates with steam at inlet conditions of 8600 kPa and 500°C, and discharges into a condenser at a pressure of 10 kPa. Assuming a turbine efficiency of 0.75, determine the state of the steam at discharge and the mass rate of flow of the steam.

Solution 7.6

At the inlet conditions of 8600 kPa and 500°C, the steam tables provide:

$$H_1 = 3391.6 \text{ kJ·kg}^{-1} \qquad S_1 = 6.6858 \text{ kJ·kg}^{-1}\text{·K}^{-1}$$

If the expansion to 10 kPa is isentropic, then, $S'_2 = S_1 = 6.6858$ kJ·kg^{-1}·K^{-1}. Steam with this entropy at 10 kPa is wet. Applying the "lever rule" [Eq. (6.96b), with $M = S$ and $x^v = x'_2$], the quality is obtained as follows:

$$S'_2 = S^l_2 + x'_2(S^v_2 - S^l_2)$$

Then, $6.6858 = 0.6493 + x'_2(8.1511 - 0.6493) \qquad x'_2 = 0.8047$

This is the quality (fraction vapor) of the discharge stream at point 2′. The enthalpy H'_2 is also given by Eq. (6.96b), written:

$$H'_2 = H^l_2 + x'_2(H^v_2 - H^l_2)$$

Thus, $H'_2 = 191.8 + (0.8047)(2584.8 - 191.8) = 2117.4 \text{ kJ·kg}^{-1}$

$(\Delta H)_S = H'_2 - H_1 = 2117.4 - 3391.6 = -1274.2 \text{ kJ·kg}^{-1}$

and by Eq. (7.16),

$$\Delta H = \eta(\Delta H)_S = (0.75)(-1274.2) = -955.6 \text{ kJ·kg}^{-1}$$

Whence, $H_2 = H_1 + \Delta H = 3391.6 - 955.6 = 2436.0 \text{ kJ·kg}^{-1}$

Thus the steam in its actual final state is also wet, with its quality given by:

$$2436.0 = 191.8 + x_2(2584.8 - 191.8) \qquad x_2 = 0.9378$$

Then $S_2 = 0.6493 + (0.9378)(8.1511 - 0.6493) = 7.6846 \text{ kJ·kg}^{-1}\text{·K}^{-1}$

This value may be compared with the initial value of $S_1 = 6.6858$.

The steam rate $\dot{m}$ is given by Eq. (7.13). For a work rate of 56,400 kJ·s^{-1},

$$\dot{W}_s = -56{,}400 = \dot{m}(2436.0 - 3391.6) \qquad \dot{m} = 59.02 \text{ kg·s}^{-1}$$

Table F.2 Superheated Steam, SI Units (Continued)

P/kPa T^{sat}/K(t^{sat}/°C)		sat. liq.	sat. vap.	TEMPERATURE: T kelvins (TEMPERATURE: t°C)							
				748.15 (475)	773.15 (500)	798.15 (525)	823.15 (550)	848.15 (575)	873.15 (600)	898.15 (625)	923.15 (650)
8200 569.85(296.70)	V	1.391	22.863	38.893	40.614	42.295	43.943	45.566	47.166	48.747	50.313
	U	1315.2	2569.5	3015.6	3063.3	3110.5	3157.4	3204.3	3251.1	3298.1	3345.2
	H	1326.6	2757.0	3334.5	3396.4	3457.3	3517.8	3577.9	3637.9	3697.8	3757.7
	S	3.2239	5.7338	6.6311	6.7124	6.7900	6.8646	6.9365	7.0062	7.0739	7.1397
8400 571.54(298.39)	V	1.398	22.231	37.887	39.576	41.224	42.839	44.429	45.996	47.544	49.076
	U	1324.3	2567.2	3013.6	3061.6	3108.9	3155.9	3202.9	3249.8	3296.9	3344.1
	H	1336.1	2754.0	3331.9	3394.0	3455.2	3515.8	3576.1	3636.2	3696.2	3756.3
	S	3.2399	5.7207	6.6173	6.6990	6.7769	6.8516	6.9238	6.9936	7.0614	7.1274
8600 573.21(300.06)	V	1.404	21.627	36.928	38.586	40.202	41.787	43.345	44.880	46.397	47.897
	U	1333.3	2564.9	3011.6	3059.8	3107.3	3154.4	3201.5	3248.5	3295.7	3342.9
	H	1345.4	2750.9	3329.2	3391.6	3453.0	3513.8	3574.3	3634.5	3694.7	3754.9
	S	3.2557	5.7076	6.6037	6.6858	6.7639	6.8390	6.9113	6.9813	7.0492	7.1153
8800 574.85(301.70)	V	1.411	21.049	36.011	37.640	39.228	40.782	42.310	43.815	45.301	46.771
	U	1342.2	2562.6	3009.6	3058.0	3105.6	3152.9	3200.1	3247.2	3294.5	3341.8
	H	1354.6	2747.8	3326.5	3389.2	3450.8	3511.8	3572.4	3632.8	3693.1	3753.4
	S	3.2713	5.6948	6.5904	6.6728	6.7513	6.8265	6.8990	6.9692	7.0373	7.1035
9000 576.46(303.31)	V	1.418	20.495	35.136	36.737	38.296	39.822	41.321	42.798	44.255	45.695
	U	1351.0	2560.1	3007.6	3056.1	3104.0	3151.4	3198.7	3246.0	3293.3	3340.7
	H	1363.7	2744.6	3323.8	3386.8	3448.7	3509.8	3570.6	3631.1	3691.6	3752.0
	S	3.2867	5.6820	6.5773	6.6600	6.7388	6.8143	6.8870	6.9574	7.0256	7.0919
9200 578.04(304.89)	V	1.425	19.964	34.298	35.872	37.405	38.904	40.375	41.824	43.254	44.667
	U	1359.7	2557.7	3005.6	3054.3	3102.3	3149.9	3197.3	3244.7	3292.1	3339.6
	H	1372.8	2741.3	3321.1	3384.4	3446.5	3507.8	3568.8	3629.5	3690.0	3750.5
	S	3.3018	5.6694	6.5644	6.6475	6.7266	6.8023	6.8752	6.9457	7.0141	7.0806
9400 579.59(306.44)	V	1.432	19.455	33.495	35.045	36.552	38.024	39.470	40.892	42.295	43.682
	U	1368.2	2555.2	3003.5	3052.5	3100.7	3148.4	3195.9	3243.4	3290.9	3338.5
	H	1381.7	2738.0	3318.4	3381.9	3444.3	3505.9	3566.9	3627.8	3688.4	3749.1
	S	3.3168	5.6568	6.5517	6.6352	6.7146	6.7906	6.8637	6.9343	7.0029	7.0695
9600 581.12(307.97)	V	1.439	18.965	32.726	34.252	35.734	37.182	38.602	39.999	41.377	42.738
	U	1376.7	2552.6	3001.5	3050.7	3099.0	3146.9	3194.5	3242.1	3289.7	3337.4
	H	1390.6	2734.7	3315.6	3379.5	3442.1	3503.9	3565.1	3626.1	3686.9	3747.6
	S	3.3315	5.6444	6.5392	6.6231	6.7028	6.7790	6.8523	6.9231	6.9918	7.0585