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**FROM CONTROL-MASS SYSTEMS TO CONTROL-VOLUME SYSTEMS**

The simplest thermodynamic formulation is for an isolated system: during its evolution, mass, momentum and energy do not change, and entropy increases (exergy decreases). Further, we considered control mass systems that may exchange energy (and momentum) with the environment, but not mass. Now we are to analyse control volume systems, also known as open systems, i.e. systems that may exchange mass (and energy and momentum) with the environment.

It is just for our convenience that we introduce this new point of view (the choice of system is observer dependant), and all the formalism (except balance equations) developed in the previous chapters applies, i.e. the same thermodynamic relations apply (e.g.  $dU=TdS-pdV+\sum\mu_i dn_i$ ), and the same thermodynamic data are needed to fully know the system behaviour at equilibrium (usually  $\rho(T,p)$  and  $c_p(T,p_0)$ ).

It is helpful to write down the evolution laws in the form of balance equations, ascribing every input to the following common-language budgetary terms: accumulation, production, and flux, such that accumulation = production + net flux in. Trying to better identify the fluxes through the frontier, we take the liberty of ascribing the volumetric force to a production term.

We first recall that for a control mass (closed system) the balance equations in differential form are:

<u>Magnitude</u>	<u>accumulation</u>	=	<u>production</u>	<u>flux</u>
mass	$dm$	=	0	+0
momentum	$d(m \vec{v})$	=	$m\vec{g} dt$	$+\vec{F}_A dt$
energy	$d(me)$	=	0	$+dW+dQ$
entropy	$d(ms)$	=	$dS_{gen}$	$+dQ/T$
exergy	$d(m\phi)$	=	$-T_0 dS_{gen}$	$+dW_u+(1-T_0/T)dQ$

and now we want to deduce the following control volume balance laws (open systems) in differential form, i.e. per unit differential volume:

<u>Magnitude</u>	<u>accumul.</u>	<u>production</u>	<u>impermeable flux</u>	<u>permeable flux</u>
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mass	$dm$	=	0	+0	$+\sum dm_e$	(5.1)
momentum	$d(m\vec{v})$	=	$m\vec{g} dt$	$+\vec{F}_A dt$	$+\sum \vec{v}_e dm_e - \sum p_e A_e \vec{n}_e$	(5.2)
energy	$d(me)$	=	0	$+dW+dQ$	$+\sum h_{te} dm_e$	(5.3)
entropy	$d(ms)$	=	$dS_{gen}$	$+dQ/T$	$+\sum s_e dm_e$	(5.4)
exergy	$d(m\phi)$	=	$-T_0 dS_{gen}$	$+dW_u + (1-T_0/T)dQ$	$+\sum \psi_e dm_e$	(5.5)

To begin with, we divide the control volume frontier in two parts: the impermeable part and the permeable part, and separate the flux term accordingly. The permeable flux term must account for the magnitudes transported by the mass flowing through the openings (the convective flux), plus the diffusive flux through the openings. The summation in the permeable flux term extends to all the openings (gaps in the frontier). We indicate by subindex 'e' the variables at the entry or exit points, which are assumed at equilibrium (i.e. no gradients are allowed along an opening).

The permeable flux term in the mass balance equation is just the mass flowing through the openings. Notice that  $dm$  in the accumulation term is the differential of a state function, whereas  $dm_e$  is a small increment of a path function (same notation as in  $dE=dW+dQ$  in Chapter 1).

The permeable flux term in the momentum equation shows the momentum transported by the mass flowing through the openings, plus the diffusive flow through the openings due to the forces there applied, that, as no gradients are allowed, can only be the pressure force, assumed uniform over an opening, and directed along the normal to the surface at the entry/exit point,  $\vec{n}_e$ . It is very important to realise that the exit pressure from a system to the atmosphere (or to a plenum chamber), immediately adjust to the pressure in the plenum downstream because pressure fluctuations are quickly propagated at the speed of sound (this is, of course, not valid for supersonic exit). Contrary to pressure, the temperature at the exit is that of the fluid inside close to go out, because temperature fluctuations are very slowly propagated at the speed of heat transfer between the out-going fluid and the ambient fluid.

The permeable flux term in the energy equation introduces a new function, the total enthalpy,  $h_t$ , sum of the enthalpy plus the mechanical energy:

$$h_t \equiv h + e_m = e + pv = u + pv + e_m \quad (5.6)$$

that includes the convective flux of energy,  $e_e dm_e$ , and the diffusive flow of energy through the openings due to heat transfer (negligible) and work transfer (that according to what has been said for the forces applied it is just the work of pressure,  $p_e dV_e = p_e v_e dm_e$ ).

The permeable flux term in the entropy equation is trivial, but that on the exergy equation introduces another new function, the exergy flow (sometimes named flow availability), that per unit mass of flow is:

$$\psi \equiv h_t - T_0 s = e + pv - T_0 s \quad (5.7)$$

The relation  $\Phi = E + p_0 V - T_0 S$  still holds for the exergy content within a control volume (distinct to the exergy flow).

And the rest of the chapter is devoted to analyse more deeply some general applications of control volume systems.

## MASS FLOW RATE IN THERMODYNAMICS AND IN FLUID MECHANICS

Although Thermodynamics is not a time-rated science (it only deals with too-slow and too-quick processes), there are circumstances where energy flow rates and mass flow rates naturally appear in the analysis, as when an electrical resistance is switched on, or a fluid flow is established at system openings.

Thermodynamics uses a black-box approach, renouncing to know the internal details of systems; it is not a field theory, the temperature field is properly analysed in Heat Transfer, the velocity field in Fluid Mechanics, and so on. All real systems are out of equilibrium because we live in a non-equilibrium world, but the thermodynamic modelling of equilibrium systems is the most effective for the effort demanded.

In this context, the mass flow rate at an opening in a control-volume system,  $\dot{m}_e \equiv dm_e/dt$  (5.1), which is computed in Fluid Mechanics as a weighted integral of the normal projection of the velocity field, is either a given data in Thermodynamics, or a global result of the mass-conservation principle, velocities just showing up as average speeds defined by the relation  $\dot{m} \equiv \rho v A$ . Moreover, so common the use of average speeds and their symbol,  $v$ , is in this chapter, that we will avoid using the same symbol for specific volumes, using the inverse (densities) instead.

## ONE-OPENING SYSTEMS. APPLICATION TO THE PRESSURISATION AND DEPRESSURISATION OF RESERVOIRS

A problem of great interest is what happens when a reservoir with some gas inside is put quickly in contact, through an opening, with a larger mass of the same gas at different pressure. If the process occurred slowly enough, the temperature would be that of the environment all the time, and the pressure would be directly related to the mass exchanged, by means of the equation of state.

For the quick discharge of a reservoir (rigid or not), there would be little frictional losses in the inside, since the movement would be mainly an expansion without inner mixing, and the process would be adiabatic on reasons of the little time available for heat exchange, thence we can model the depressurisation as isentropic, and thus (1.11) or:

$$\frac{T_2}{T_1} = \left( \frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = \left( \frac{m_2}{m_1} \right)^{\gamma-1} \quad (5.8)$$

solves the problem of knowing the conditions at any instant 2 coming after the initial state 1.

### [Exercise 1. Discharge of a pressurised bottle](#)

For the quick charge of a rigid reservoir through a small opening, there would be great frictional losses in the inside, since the movement would be mainly a jet, forcing a vigorous inner mixing. The process is adiabatic on reasons of the little time available for heat exchange, but not isentropic because of the dissipation in the interior. The equation describing the evolution, assuming a representative temperature,  $T_2$ , exists at any time in the interior (what is based on the quick mixing, since there is no time for heat transfer) is derived from (5.3) with  $e=u$  (negligible mechanical energy variation),  $Q=0$  (quick),  $W=0$  (rigid) and  $dm_e=dm$  (mass balance), what yields:

$$d(mu)=h_{te}dm \quad (5.9)$$

where the total enthalpy at the entrance point is given. If that can be assumed constant (intake from a much larger room), (5.9) can be integrated to yield:

$$m_2u_2-m_1u_1=h_{te}(m_2-m_1) \quad (5.10)$$

A warning on reference values is most appropriate here: only one reference for all energetic functions can be arbitrarily chosen, either  $u_0$  or  $h_0$  but not both; i.e. choosing  $u_0$ , it must be verified for any  $u_0$  that:

$$m_2(u_2-u_0)-m_1(u_1-u_0)=(h_{te}-u_0)(m_2-m_1) \quad (5.11)$$

A usual choice of reference is  $u_0=u_{(T=0)}=0$ , and with the perfect gas model (5.10) reads:

$$m_2c_vT_2-m_1c_vT_1=c_pT_{te}(m_2-m_1) \quad (5.12)$$

showing that if the reservoir was initially evacuated ( $m_1=0$ ), the temperature inside would be  $T_2=\gamma T_1$  all the time during the (quick) filling, obviously just an approximation. For typical high-pressure loading, gas flow is choked most of the time (i.e. the mass flow rate is fixed by the throat area and the fixed upstream conditions, and independent of internal conditions), greatly simplifying the computation of the filling time.

[Exercise 2. Charge of an evacuated flask](#)

[Exercise 3. Heating in a two-flask assembly](#)

Liquid reservoirs are not considered here because they show different problems. First, rigid reservoirs are never full of liquid: a small gaseous phase is always left to avoid the huge pressure variations associated to liquid thermal expansion at constant volume. Containers with flexible or elastic walls can be fully liquid-filled, as for syringe or bellow devices, but the charging and discharge in them is controlled by the displacement of the wall, or by a two-phase process with cumbersome priming. Large liquid reservoirs cannot be hermetic because the walls are designed to support only small pressure differences (<10 kPa usually), and just thermal expansion of the gas above the liquid, assuming constant mass and volume, will cause larger variations with changing ambient temperature,  $\Delta p/p=\Delta T/T$ , exacerbated by the mass of gas increase due to increasing evaporation, and the volume of gas decrease due to liquid expansion.

Charging and discharge of a liquid-and-vapour reservoir, as for typical butane or carbon-dioxide bottles, is studied under [Phase Change](#).

## STEADY FLOWS

By far the wider application of control volume systems is for steady problems (i.e. where any variable at a point does not change with time). In fact, unsteady running is usually a sign of malfunction, although any practical process needs a start and an end (in any case, most failures of equipment do occur at starts and ends).

The steady equations for a control volume (the volume is fixed, if steady) are just (5.1-5) without the accumulation terms, but are written here by unit of time:

<u>Magnitude</u>	<u>accumul.</u>		<u>production</u>	<u>impermeable flux</u>	<u>permeable flux</u>	
mass	0	=	0	+0	+ $\sum \dot{m}_e$	(5.13)
momentum	0	=	$m\vec{g}$	+ $\vec{F}_A$	+ $\sum \vec{v}_e \dot{m}_e - \sum p_e A_e \vec{n}_e$	(5.14)
energy	0	=	0	+ $\dot{W} + \dot{Q}$	+ $\sum h_{te} \dot{m}_e$	(5.15)
entropy	0	=	$\dot{S}_{gen}$	+ $\dot{Q}/T$	+ $\sum S_e \dot{m}_e$	(5.16)
exergy	0	=	$-T_0 \dot{S}_{gen} + (1 - T_0/T) \dot{Q}$	+ $\dot{W}_u$	+ $\sum \psi_e \dot{m}_e$	(5.17)

The flow rate at an opening,  $\dot{m}$ , can be substituted in terms of the averaged speed,  $v$  (not to be confused with the intensive volume, in spite of using the very same symbol), in the way:

$$\dot{m} \equiv \rho v A \quad (5.18)$$

that, although valid in general (it can be viewed as the definition of  $v$ ), is most used in steady problems. Velocity fields are the real of Fluid Mechanics, whereas only one-dimensional flows with an average speed are considered in Thermodynamics.

### [Exercise 4. Combustion chamber](#)

#### TWO-OPENING SYSTEMS. STEADY FLOW WITH ONE INLET AND ONE EXIT

The simplest steady state configuration (apart from the trivial case of no evolution) has one inlet and one outlet. Most of the practical equipment used in thermodynamic facilities corresponds to this configuration: pipes, pumps, fans, compressors, expanders, filters, some valves, one side of heat exchangers,.... For 3-way valves, mixing chambers, separators and contact heat exchangers, the many-entries equations above mentioned must be used. The usual form of the balance equations, for one-inlet one-exit steady systems, is:

<u>Magnitude</u>	<u>balance equation</u>	
mass	$\rho_1 v_1 A_1 = \rho_2 v_2 A_2$	(5.19)
momentum	$\vec{F}_A + m\vec{g} = \dot{m}(\vec{v}_2 - \vec{v}_1) + (p_2 A_2 \vec{n}_2 - p_1 A_1 \vec{n}_1)$	(5.20)
energy	$h_2 - h_1 = w + q$	(5.21)

entropy	$s_{gen}=s_2-s_1-q/T$	(5.22)
exergy	$w_u=\psi_2-\psi_1+T_0s_{gen}-(1-T_0/T)q$	(5.23)

The mass balance shows, for instance that in a constant cross-section pipe, a liquid exits with the same averaged speed as it entries, no matter how much viscous it is.

The momentum balance is the key to understand engine thrust in propulsive systems, where  $\vec{F}_A$  is the force exerted by the outside on the impermeable wall of the engine (the drag of the medium and the reaction in the supports. For instance, for a jet engine in steady horizontal flight with horizontal thrust, the support for the engine must carry the loads of the weight of the engine and the thrust, that mainly corresponds to the mass flow rate times the jump in air speeds from inlet to exit, since the term of pressures is small (why?).

The energy balance (5.21) is perhaps the most frequently used thermodynamic equation in engineering, The similarity between this energy balance for a control volume,  $h_{t_2} - h_{t_1} = w + q$ , and the energy balance for a control mass,  $E_2 - E_1 = W + Q$ , is only apparent; in the former, 1 and 2 refer to the two openings of a system at the same time, whereas in the latter refer to two states of the same system at different times; additionally,  $w$  is the work input through the impermeable part of the frontier (by unit of mass flow rate), whereas  $W$  accounts for the work through the whole frontier. Similarly,  $q$  is the heat input through the wall by unit of mass flow rate.

The entropy balance is used to compute the entropy generated inside, although a detailed integration of the entropy flow would be required if the process is not adiabatic or isotherm.

The exergy balance is used to evaluate the minimum work needed to force a process (or the maximum work obtainable). To notice that the last two terms in the right-hand-side of (5.23) both add to the minimum work needed, or decrease the maximum work obtainable.

[Exercise 5. Hair dryer](#)

[Exercise 6. Cost of producing compressed air](#)

### Generalised Bernoulli equation

One of the most frequently used equation of Fluid Mechanics in engineering is Bernoulli equation, in the form  $p + (1/2)\rho v^2 + \rho g z = \text{constant}$ . Here we are to develop a generalisation of that, including dissipation, for any kind of fluid, but still in steady flow. If the friction-dissipated mechanical energy for a control mass,  $E_{mdf} = W + \int p dV - \Delta E_m$  (1.7) is applied to a control volume, recalling that  $W_{CM} = W_{CV} + \sum \int p_e v_e dm_e$ , one obtains by unit of mass flow rate  $e_{mdf} = w - \int v dp - \Delta e_m$ , showing that the energy balance  $h_{t_2} - h_{t_1} = w + q$  may be split in two parts (the second one by difference between the total energy and the mechanical balance):

$$\text{- mechanical energy balance: } w = \int \frac{dp}{\rho} + \Delta e_m + e_{mdf} \quad (5.24)$$

$$\text{- thermal energy balance: } q + e_{mdf} = \Delta u + \int p d\left(\frac{1}{\rho}\right) \quad (5.25)$$

The mechanical energy balance, also known as the generalised Bernoulli equation, teaches that when work is put into a system (through the impermeable wall, i.e. shaft work) in steady state, it may be used to rise the pressure (first term; the device may be a pump or compressor), to rise the mechanical energy (second term; the speed or the head, may be risen), and to be dissipated and incorporated to the internal energy (the last term in (5.24)).

The thermal energy balance is usually simplified to  $q=\Delta u$ , because the dissipated energy is always negligible (although it is the cause why pumps and fans are needed to force fluid flow), and the compressible term is also negligible.

### [Exercise 7. Thermo-siphon](#)

#### **Stagnation or total conditions**

We realise that all the time in the energy balance mechanical and internal energy add up. In most ordinary problems, mechanical energies are important in the flow of liquids, but negligible in the flow of gases because the speeds are not so high and the change in potential heads neither. But there is a bunch of problems with high speed gas flow, particularly inside rotatory engines (gas and steam turbines), nozzles, high speed flight, etc. Most of these problems fall in the category of steady flows with one inlet and one exit: In order to decouple the overall energy balance, from the internal mechanical details (as speeds, areas and the like), it is advantageous to define a virtual thermodynamic state called total state,  $(T_t, p_t)$ , with the same total enthalpy and entropy but without mechanical energy, that is the state the fluid would acquire if it become stagnant at the same head and isentropically. The total state, for the two most simple substance models, is defined as:

$$\text{For incompressible liquids: } p_t \equiv p + \frac{1}{2} \rho v^2 + \rho g z, \quad T_t \equiv T \quad (5.26)$$

$$\text{For perfect gases: } \left( \frac{p_t}{p} \right)^{\frac{\gamma-1}{\gamma}} = \frac{T_t}{T} = 1 + \frac{v^2}{2c_p T} \quad (5.27)$$

### [Exercise 8. Leading edge temperature in an aircraft](#)

#### **EFFICIENCIES IN COMPRESSION AND EXPANSION**

The compression and expansion of fluids may be done in 'displacement' machines (where a certain volume of fluid is taken from the inlet and discharged at the outlet, e.g. a reciprocating device) or in 'rotodynamic' machines (where the fluid is accelerated and decelerated, i.e. they cannot work slowly; e.g. centrifugal and axial devices). In order to evaluate the irreversibility of those processes, one may use the entropy generation or the irreversibility itself, but both being extensive, it is general practice to introduce new path variables known as efficiencies of the devices, or more properly adiabatic efficiencies (since the heat transfer is negligible because of the short residence time of the fluid inside the machine) or even better isentropic efficiencies (since they compare the actual adiabatic performance with the isentropic one).

Consider a turbo-compressor (centrifugal or axial) as a control volume of given entry conditions  $(T_1, p_1)$  and a prescribed output pressure  $(p_2)$ . We will use total conditions notation just in case the speed are high enough. From the mere thermodynamic point-of-view, the ideal process would be that of minimum work, i.e. the isothermal one if the entry is at ambient temperature, but it would be extremely slow and thus of no practical interest; the process must be rapid and thus the best model would be isentropic, and it is with respect to the isentropic process that real performances are compared to define the efficiency of the compressor,  $\eta_C$ , as the ratio between the isentropic work needed and the real work spent:

$$\text{For a compressor: } \eta_C \equiv \frac{w_s}{w} = \frac{h_{2'_t} - h_{1_t}}{h_{2_t} - h_{1_t}} = \frac{{}_{PGM} c_p (T_{2'_t} - T_{1_t})}{c_p (T_{2_t} - T_{1_t})} = \frac{\left(\frac{p_{2'_t}}{p_{1_t}}\right)^\gamma - 1}{\frac{T_{2_t}}{T_{1_t}} - 1} \quad (5.28)$$

where the auxiliary isentropic temperature  $T_{2'_t}$  (point  $2'_t$  in Fig. 5.1), is put as a function of its pressure (what is a data), thus relating the efficiency to the exit temperature. In a similar way, for given entry conditions and output pressure, the efficiency of a turbine,  $\eta_T$ , is defined by:

$$\text{For a turbine: } \eta_T \equiv \frac{w}{w_s} = \frac{h_{1_t} - h_{2'_t}}{h_{1_t} - h_{2_t}} = \frac{c_p (T_{1_t} - T_{2'_t})}{c_p (T_{1_t} - T_{2_t})} = \frac{1 - \frac{T_{2'_t}}{T_{1_t}}}{1 - \left(\frac{p_{2_t}}{p_{1_t}}\right)^\gamma} \quad (5.29)$$

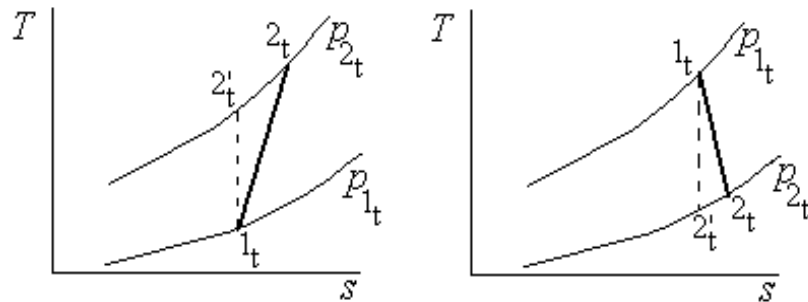


Fig. 5.1. Sketch of the real and isentropic processes: a) in a compressors, b) in a turbine.

Of interest in high speed flow, but still subsonic (less than the local speed of sound at all states), is the dynamic compression / expansion in a variable area convergent / divergent duct (without shaft work), known as diffusers / nozzles, respectively, which processes are represented in Fig. 5.2 and the corresponding efficiencies defined as:

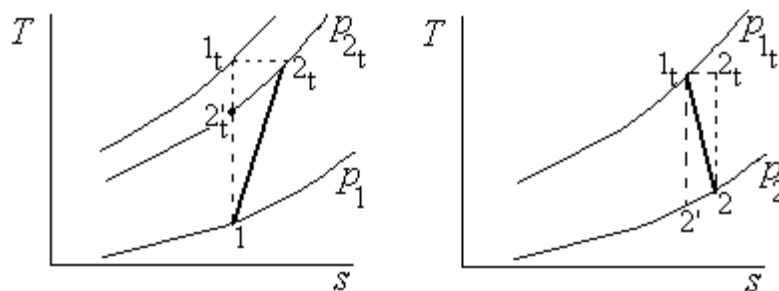


Fig. 5.2. Sketch of the real adiabatic processes and their isentropic references, in a dynamical compressors (diffuser), a), and in a dynamical expander (nozzle), b).



$$\text{For a diffuser: } \eta_D \equiv \frac{\Delta e_s}{\Delta e} = \frac{h_{2_i} - h_1}{h_{2_e} - h_1} \stackrel{PGM}{=} \frac{c_p (T_{2_i} - T_1)}{c_p (T_{2_e} - T_1)} = \frac{\left(\frac{p_{2_i}}{p_1}\right)^\gamma - 1}{\frac{T_{2_e}}{T_1} - 1} \quad (5.30)$$

$$\text{For a nozzle: } \eta_N \equiv \frac{\Delta e}{\Delta e_s} = \frac{h_1 - h_{2_e}}{h_1 - h_{2_i}} \stackrel{PGM}{=} \frac{c_p (T_1 - T_{2_e})}{c_p (T_1 - T_{2_i})} = \frac{1 - \frac{T_{2_e}}{T_1}}{1 - \left(\frac{p_{2_e}}{p_1}\right)^\gamma} \quad (5.31)$$

Sometimes, instead of the efficiency of a process, another variable is used to measure irreversibilities in real processes: the polytropic exponent,  $n$ , such that for perfect gases the evolution is modelled as  $p v^n = \text{constant}$ . The concept of polytropic transformations may be widened, and introduce a polytropic thermal capacity  $c_n = T ds/dT|_n$  to include all other processes:  $n = \gamma$  for isentropic,  $n = 1$  for isothermal,  $n = 0$  for isobaric, etc.

### Exercise 9. Dynamic air compressor

#### **Multistage compression and expansion: intercooling and reheating**

An isobar for a gas in the  $T$ - $s$  diagram is a curve with exponential growth; with the perfect gas model,  $T/T_0 = \exp((s-s_0)/c_p)$ , and different isobars are just the same curve horizontally displaced, thus isobars are divergent. That means that compression between two given isobars is more costly (more vertical separation) the warmer it is performed, and it is advantageous to initiate the compression from a cold state and do it in a stepped manner, with intermediate cooling between compression steps, i.e. intercooling (Fig. 5.3a). Similarly, to get maximum work, it is advantageous to initiate an expansion from a hot state and do it in a stepped manner, with intermediate heating between expansion steps, i.e. reheating (Fig. 5.3b). Notice that when using the environment as a heat source, some  $\Delta T$  must be allowed, as shown in Fig. 5.3a, to have practical-size heat exchangers, whereas if a different heat source was used for the first heating, the same temperature may be obtained in the reheating.

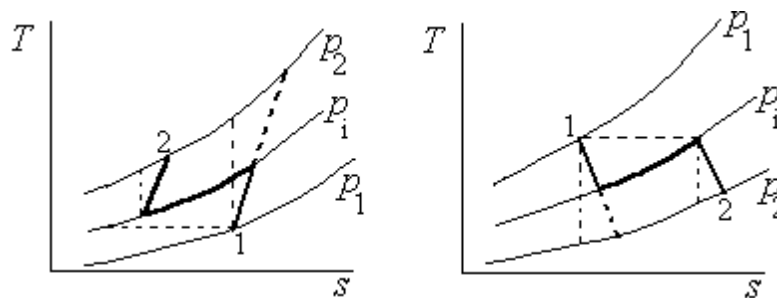


Fig. 5.3. Stepped compression and expansion.

It is easy to demonstrate that, for a perfect gas, the optimum pressure for [stepped compression /expansion](#) is the geometric mean of the extreme values, corresponding to equal share of the work between the steps.

In multistage compression and expansion, besides the isentropic efficiencies (5.28) and (5.29), it may be convenient to compare the actual work consumed/produced relative to the minimum work in the limit of

perfect cooling, i.e. in the limit of isothermal compression, what defines the isothermal efficiency of multistage compression,  $\eta_{C,\text{isot}}$ , and expansion,  $\eta_{T,\text{isot}}$ , respectively as:

$$\text{For multistage compressor: } \eta_{C,\text{isot}} \equiv \frac{w_{\text{isot}}}{w} = \frac{(h_{2t,\text{isot}} - h_{1t}) - T_0 (s_2 - s_1)}{(h_{2t} - h_{1t}) - q_{12}} \stackrel{\text{MGP}}{=} \frac{RT_0 \ln \left( \frac{p_{2t}}{p_{1t}} \right)}{\sum_{\text{adiabatic stages}} c_p \Delta T} \quad (5.32)$$

$$\text{For a multistage turbine: } \eta_{T,\text{isot}} \equiv \frac{w}{w_{\text{isot}}} = \frac{(h_{1t} - h_{2t}) + q_{12}}{(h_{1t} - h_{2t,\text{isot}}) - T_0 (s_1 - s_2)} \stackrel{\text{MGP}}{=} \frac{\sum_{\text{adiabatic stages}} c_p \Delta T}{RT_0 \ln \left( \frac{p_{1t}}{p_{2t}} \right)} \quad (5.29)$$

## TYPE OF PROBLEMS

Besides housekeeping problems of how to deduce one particular equation from others, the types of problems in this chapter are the typical single-stage (or multi-stage) engineering applications:

1. Find the conditions prevailing after the quick pressurisation and depressurisation of reservoirs.
2. Find the exit temperature in single components like compressors and expanders, heaters and coolers, and so on.
3. Exergy calculation for control volumes. In particular, analysis of thermodynamic feasibility.
4. And in general all kind of processes of a pure substance without phase change (air is the typical working substance).

It is good time now to summarise the steps in solving typical Thermodynamics problems:

1. Extract relevant data (explicit in the statement or implicitly assumed) and make a sketch as simple as possible, introducing notation.
2. Choose the appropriate system (usually a gas, but sometimes a universe).
3. Choose the appropriate states and constraints to study the system and its interactions.
4. Choose the appropriate thermodynamic model for the substance (constitutive equations).
5. Establish the balance equations for the evolution (this is the only difference between a control mass and a control volume system).
6. Solve the equations (more and more often relegated to a computer package).
7. Check (double check!): Does the answer corresponds to the question?, is it physically consistent (dimensions and units)?, is it numerically consistent (order of magnitud and significan figures)?, is it relevant to other tasks?

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