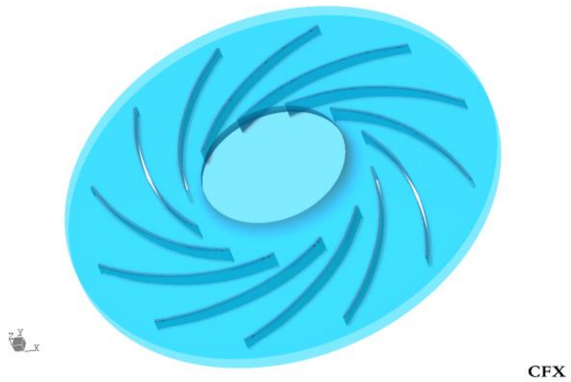
5. TURBOBLOWERS AND TURBOCOMPRESSORS

5.1. Blowers [30]

5.1.1. General remarks

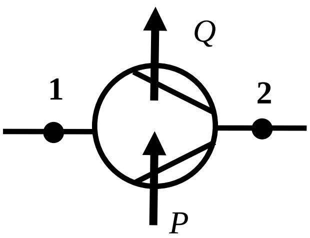
- Structure: similar to the fans (Fig. **5.1**), with constructional modifications. I. e. the requirements are strict regarding rotor balancing and bearing, due to the increased rotor speed.
- Multistage machinery: rotor and guide vane blade rows situated in series.
- Rotor speed: typically in the range of 6000 – 20000 RPM ⇔ fans: up to 3000 RPM
- The compression is usually moderate within one stage. Therefore, the design and flow studies can be carried out similarly to fans, with consideration of mean density within the stage.
- Temperature change: cca. 50 – 80 °C.



Q [l/s]	*n [1/min]*	*Δpt [Pa]*	*ψt*
32.5	32500	24386	1.16

Fig. 5.1. Blower rotor developed by the Dept. of Hydrodynamic Systems, BUTE [37]

5.1.2. Work process



For ideal case (no losses):

$$\frac{P}{q_m}-\frac{Q}{q_m}=\left[\frac{v^2}{2}+gh+U+\frac{p}{\rho}\right]_1^2=i_{t2}-i_{t1} \quad (5.1)$$

Q is the heat flux extracted by cooling.

For blowers $Q \approx 0$, (no artificial cooling; the system can be considered as thermally isolated), and *gh* plays role only if the densities of the ambient air and the transported gas are different.

Therefore, for blowers in general

$$P=q_m\left[\frac{v^2}{2}+U+\frac{p}{\rho}\right]_1^2 \quad (5.2)$$

Where

$$U=c_V T \quad (5.3)$$

$$\frac{p}{\rho}=RT=\left(c_p-c_V\right)T \quad (5.4)$$

Substitution to Eq. (5.2) reads:

$$P=q_m c_p\left[\frac{v^2}{2c_p}+T\right]_1^2 \quad (5.5)$$

Since

$$\frac{v^2}{2c_p}+T=T_{din}+T=T_t \quad (5.6)$$

$$\boxed{P=q_m c_p\left(T_{t2}-T_{t1}\right)} \quad (5.7)$$

Usually the change of dynamic temperature and dynamic pressure is negligible compared to the changes in the static characteristics. Therefore, the following modification of Eq. (5.7) will be used further on:

$$\boxed{P=q_m c_p\left(T_2-T_1\right)} \quad (5.8)$$

Applying the universal gas law and the equation of isentropic change of state (heating the gas from state 1 to state 2, with assumption of inviscid, thermally insulated gas), it can be pointed out that Eq. (5.8) also expresses the rise of static pressure. The increase of temperature is an unwanted but unavoidable consequence of the compression process. (Usually it is not aimed to heat the gas by the machinery but to increase its pressure.)

5.1.3. Energetic aspects

Let us consider the compression process as a series of elemental processes. During an elemental compression process, *dq* heat develops due to the fluid mechanical losses of the machinery. This heat is stored in the gas. Therefore, the fluid friction can be considered as a thermal source.

An elemental process is characterised by an elemental efficiency, which will be considered as constant during the entire process (because the fluid mechanical circumstances do not vary significantly during the compression process).

The introduced specific power input is equal to the increase of specific energy in ideal case. According to Eq. (5.8), this is $c_p dT$ for the gas of unit mass. It is not useful for us that dq heat input occurs due to fluid friction. Therefore, the useful specific power is only $c_p dT - dq$. The elemental efficiency is as follows:

$$\eta_e = \frac{c_p dT - dq}{c_p dT} \tag{5.9}$$

According to the 1st law of thermodynamics, the heat input turns partly to the increase of internal energy, and is converted partly to mechanical work related to the expansion of the gas:

$$dq = c_V dT + p d\left(\frac{1}{\rho}\right) \tag{5.10}$$

Considering that $RT = p/\rho$, $R\,dT = p\,d\left(\frac{1}{\rho}\right) + dp\left(\frac{1}{\rho}\right)$, the 1st law of thermodynamics is as follows:

$$dq = c_V dT + R\,dT - dp\left(\frac{1}{\rho}\right) = c_P dT - dp\left(\frac{1}{\rho}\right) \tag{5.11}$$

Substituting to Eq. (5.9):

$$\eta_e = \frac{dp\frac{1}{\rho}}{c_p dT} \tag{5.12}$$

Considering that $\frac{1}{\rho} = \frac{RT}{p}$,

$$\eta_e = \frac{dp\;R\,T}{c_p dT\;\;p} \tag{5.13}$$

Separation of the variables:

$$\frac{dT}{T} = \frac{R}{c_p \eta_e}\frac{dp}{p} \tag{5.14}$$

Integration of the differential equation, rearrangement, and consideration of $\frac{R}{c_p} = \frac{\kappa - 1}{\kappa}$ leads to the following expression of the efficiency, termed herein as **polytropic efficiency** (the terminology indicates that the process is not isentropic due to fluid friction):

$$\eta_p = \frac{\kappa - 1}{\kappa}\frac{\ln\frac{p_2}{p_1}}{\ln\frac{T_2}{T_1}} \tag{5.15}$$

In another form:

$$\frac{T_2}{T_1}=\left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa\,\eta_p}} \qquad (5.16)$$

If $\eta_p = 1$ would occur, Eq. (5.16) would correspond to the well-known equation of isentropic change of state [3]. If the mass flow rate and the shaft power input is fixed, the left-hand side of Eq. (5.16) can also be considered as fixed, based on Eq. (5.8). The fact that $\eta_p < 1$ notifies the user of the machinery, that due to the losses, the pressure does not increase as much as it could be expected in frictionless (isentropic) case.

5.1.4. Relationship between polytropic efficiency and total efficiency

The relationship between the polytropic efficiency – introduced for blowers – and the total efficiency – introduced for fans – is investigated herein, forming a "transition" between the two families of turbomachinery.

Eq. (5.16), in another form:

$$1+\frac{\Delta T_t}{T_1}=\left(1+\frac{\Delta p_t}{p_1}\right)^{\frac{\kappa-1}{\kappa\,\eta_p}} \qquad (5.17)$$

Within one stage, due to the moderate pressure ratio, $\frac{\Delta p_t}{p_1} << 1$, and $\frac{\Delta T_t}{T_1} << 1$. The expansion to Taylor series reads

$$(1+X)^C \approx 1+CX \text{ , if } X << 1 \qquad (5.18)$$

Applying this to Eq. (5.17) reads

$$1+\frac{\Delta T_t}{T_1} \approx 1+\frac{\kappa-1}{\kappa\,\eta_p}\frac{\Delta p_t}{p_1} \qquad (5.19)$$

Since $\frac{R}{c_p}=\frac{\kappa-1}{\kappa}$, Eq. (5.19) reads

$$\frac{\Delta T_t}{T_1} \approx \frac{R}{c_p}\frac{1}{\eta_p}\frac{\Delta p_t}{p_1} \qquad (5.20)$$

A rearrangement and consideration of Eq. (5.7) reads

$$c_p \Delta T_t = \frac{P}{q_m} \approx \frac{R T_1}{p_1}\frac{\Delta p_t}{\eta_p} = \frac{1}{\rho}\frac{\Delta p_t}{\eta_p} \qquad (5.21)$$

From which

$$P \approx \frac{q_m}{\rho}\frac{\Delta p_t}{\eta_p} = \frac{q_V \Delta p_t}{\eta_p} \qquad (5.22)$$

For fans, it was considered that

$$P = \frac{q_V \Delta p_t}{\eta_t} \qquad (5.23)$$

A comparison of Eqs. (5.22) and (5.23) reads

$$\eta_p \approx \eta_t \qquad (5.24)$$

Therefore, in a transitional state (a blower operated as a fan at low speed, or a fan operated as a blower at high speed), the polytropic efficiency reflects the same effect as the total efficiency: the effect of fluid mechanical losses.

The energetic aspects can be illustrated in a temperature – entropy diagram (**Fig. 5.2**):

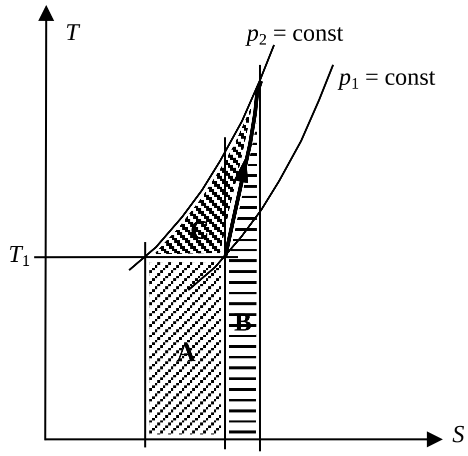


Fig. 5.2

The isobar (constant pressure) lines show the changes of state as if both the temperature and entropy would change (e.g. due to friction) but no change of pressure would occur. According to Eq. (5.15), this means zero polytropic efficiency: all the power input is converted to friction heat, and no useful power appears. The friction heat is proportional to the area below the isobar lines.

The area „**A”** corresponds to the work due to isothermal compression (i.e. compression at constant temperature).

The area **„B”** shows the work to be put in for covering the friction losses. This area would be zero in the case of no friction: according to isentropic change of state, the state of the gas would get from isobar line p1 to isobar line p2 along a vertical line.

Area **„C”** is due to the absence of heating.

The polytropic efficiency:

$$\eta_p = \frac{A+C}{A+B+C} \qquad (5.25)$$

If $B \to 0, \quad \eta_p \to 1, \quad \Delta S \to 0$, isentropic compression.

In the case of a blower, the designer can improve η_p (reduction of B), but it is not his/her fault that extra energy is to be out in due to the absence of cooling.

In a pessimistic aspect, the increase of temperature is not utilised at all. For such cases, the isothermal power factor is used for energetic characterisation of the machinery. This factor is also used for compressors.

It is usually specified in the documentation of the machinery, for economical calculations.

$$\lambda = \frac{H_{isoth}}{\sum P/q_m} \quad (5.26)$$

Where H_{isoth} is the (useful) specific power necessary for the pressure rise at constant temperature, and $\sum P/q_m$ is the overall specific power input, including all energetic effects (such as power covering the bearing friction losses, extra power necessary for the cooling process, e.g. water circulation pump, etc.).

The power is considered useful if it increases the pushing work of the fluid. An elemental increase of the pushing work is $\frac{1}{\rho}dp$, and with this, assuming T_1 constant temperature

$$H_{isoth} = \int_1^2 \frac{1}{\rho}dp = RT_1\int_1^2 \frac{1}{p}dp = RT_1 \ln\frac{p_2}{p_1} \quad (5.27)$$

5.2. Compressors [30]

Application of cooling, **Fig. 5.3**.

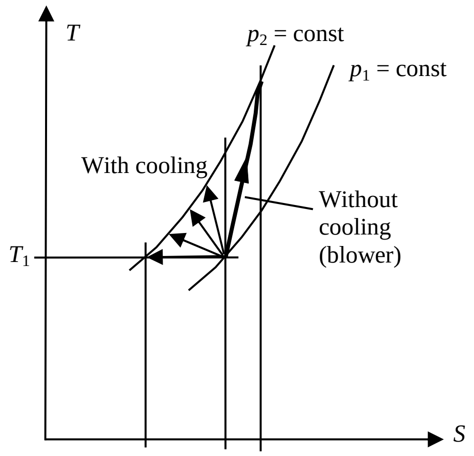


Fig. 5.3

Multistage cooling, **Fig. 5.4**.

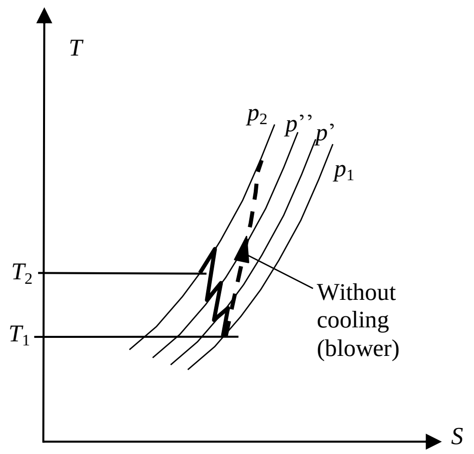


Fig. 5.4.

Reasons for cooling:

- The hot compressed air is usually disadvantageous from technological point of view
- Metallurgical, structural problems with high temperature

Machines of moderate size: internal cooling: internal passages, circulation, heat exchanger, or outer mantle.

Outer cooling (Fig. 5.4): the desired pressure rise is developed in more stages. The air is led intermediately through a heat exchanger, and then is led back to the next stage.

In the case of cooling, the power demand is limited. However, the heat exchanger causes extra investment costs.

General remarks:

- The performance is distributed nearly uniformly among the stages.
- Contrarily to fans, the performance depends on the inlet pressure.
- A compressor serves to increase the pressure on the pressure side, i.e. it is not to be confused with a vacuum pump.
- The inlet air is filtered, but the water content remains in the air and is usually condensed on the pressure side. In the case of compressed air networks, water separators are applied.
- The pressure ratio depends on the Mach number. The efficiency is nearly constant below a Mach number of 0.3 – 0.4, and decreases above this range.
- Compressors usually load the air into reservoirs.
- For compressors, surge (“pumping”) is a significant problem, due to the large pressure fluctuations. To avoid surge, rotor speed control can be used.

Dimensionless quantities: for the stages, for the entire machinery.

Temperature rise factor (stage and/or overall):

$$\chi = \frac{\Delta T}{u^2 / 2c_p} \qquad (5.28)$$

Where u is the circumferential speed of the rotor blade tip.

5.3. Problems

1. A blower sucks air from the surroundings (motionless air, $p_1 = 10^5$ Pa, $t_1 = 20$ °C). The pressure ratio is $p_2/p_1 = 1.6$. The mass flow rate is $q_m = 1$ kg/s. The outlet air temperature is $t_2 = 70$ °C. The isobar specific heat is $c_p = 1000$ J/(kg K). The isentropic exponent is $\kappa = 1.40$. The specific gas constant is R = 287 J/(kg K). The total and static temperatures can be taken as equal.

1A. Calculate the polytropic efficiency η_p.

1B. What would be the outlet temperature in case of isentropic compression ($\eta_p = 1$), with retainment of the same pressure ratio?

1C. Calculate the shaft power input P.

1D. How many turbomachine stages must be applied within the blower, if the flow within one stage is intended to be considered incompressible, ie. the stage pressure ratio is not higher than $\varepsilon = 1.1$?

2. A compressor sucks air from the surroundings (motionless air, $p_1 = 10^5$ Pa, $t_1 = 20$ °C). The pressure ratio is $p_2/p_1 = 4.0$. The mass flow rate is $q_m = 10$ kg/s. The outlet air temperature is $t_2 = 80$ °C. The isothermal power factor is $\lambda = 0.68$. The isobar specific heat is $c_p = 1000$ J/(kg K). The isentropic exponent is $\kappa = 1.40$. The specific gas constant is R = 287 J/(kg K). The total and static temperatures can be taken as equal.

2A. What would be the outlet temperature in case of isentropic compression, with retainment of the same pressure ratio?

2B. Calculate the overall power consumption P.

2C. How many turbomachine stages must be applied within the compressor if the stage pressure ratio is not higher than $\varepsilon = 1.2$?

3. An industrial hall has dimensions of 50 m x 20 m x 6 m. The hall must be ventilated with air. The flow rate Q must be sufficient to exchange the air 2 times per hour. An overpressure of 10 Pa must be maintained in the hall. The air is supplied in a circular duct of diameter d = 1 m, length L = 30 m. A filter is built in the duct, with a loss coefficient $\zeta_f = 20$. An axial fan is to be selected for the air supply. The fan sucks the air from the surroundings. ($p_1 = 10^5$ Pa, $t_1 = 20$ °C). At the air inlet, a protection grill is located, with a loss coefficient $\zeta_f = 10$.

3A. Calculate the volume flow rate, the total pressure rise and the static pressure rise for the axial fan.

3B. The total efficiency of the fan is estimated as $\eta = 0.75$. Calculate the nominal shaft power of the driving electric motor.