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Thermal and Fluid Machines

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and Building Engineering

UNIVERSITY OF DEBRECEN
FACULTY OF ENGINEERING
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AND BUILDING ENGINEERING

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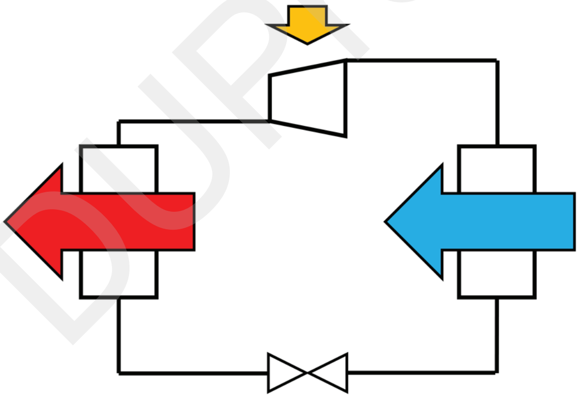

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1. Fluid machines introduction

Machines used in fluid flow systems can be broadly categorized as either turbomachines or positive-displacement machines. Turbomachines add or extract energy from a fluid utilizing a rotating component. When it extracts energy from the fluid it is a power machine and when it adds energy to it becomes a work machine. Turbomachines are also called centrifugal machines because their main part is rotating around an axis. In contrast to turbomachines, positive displacement machines move fluids by forcing fluid into and out of the chamber. A change in pressure is usually the dominant component of the change in mechanical energy between the inlet and outlet of a turbomachine, with pressure increases across pumps and decreases across turbines. All turbomachines include one or more rotating components, called rotors, between the inflow and outflow sections of the turbomachine. The three types of turbomachines commonly encountered in engineering practice are radial-flow machines (commonly referred to as centrifugal machines), axial-flow machines, and mixed- flow machines. [1]

In the case of centrifugal machines, the rotor is commonly called an impeller and is powered by an external motor, and the impeller does work on the fluid. The flow enters the housing along the axis of the impeller, called the eye of the impeller, and the flow is discharged radially by centrifugal action into a casing that encloses the impeller. The casing is shaped like a diffuser that decelerates the fast-moving fluid that exits the impeller, converts the velocity head to pressure head, and directs the flow to the outlet of the pump. [1]

In axial-flow pumps, the flow enters and leaves the flow chamber along the axis of the rotor. The rotor blades of axial-flow pumps are designed to maintain a constant axial-flow velocity as the fluid moves through the impeller. Some axial-flow pumps have adjustable blades and/or fixed diffuser vanes, called stator vanes, on the downstream side of the impeller to remove the swirl component of the velocity as it exits the impeller. Axial-flow pumps are commonly used to deliver high flows with little added head. In mixed-flow pumps, outflows have both radial and axial components. Mixed-flow pumps perform optimally at a state that is intermediate between centrifugal pumps axial flow pumps. [1]

In common usage, the term “pump" is usually applied to machines that move liquids, whereas machines that move gases are variously called fans, ventilators, blowers, and compressors, depending on their applications. Typically, if the pressure rise is below 5% of atmospheric pressure fan; if the pressure rise is between 5-30% blower; and for pressure rise above 30% compressor is used. The primary use of compressors is to increase the pressure of the gas, whereas the primary use of fans and blowers is to move gas. [1]

In case of turbines, the rotor is commonly called a runner. The fluid does work on the runner, which transmits mechanical energy to an external generator. The flow of fluid toward the rotor of a turbine is typically controlled by fixed or adjustable vanes or blades. For both centrifugal pumps and turbines, the assembly of the rotor and vanes is usually contained within a casing, which is called housing. Turbines also have a variety of uses; for example, hydraulic turbines are used to extract energy from flowing water in hydroelectric facilities, wind turbines and windmills are used to extract energy from wind, and gas turbines and steam turbines are used to extract energy from the expansion of gases. [1]

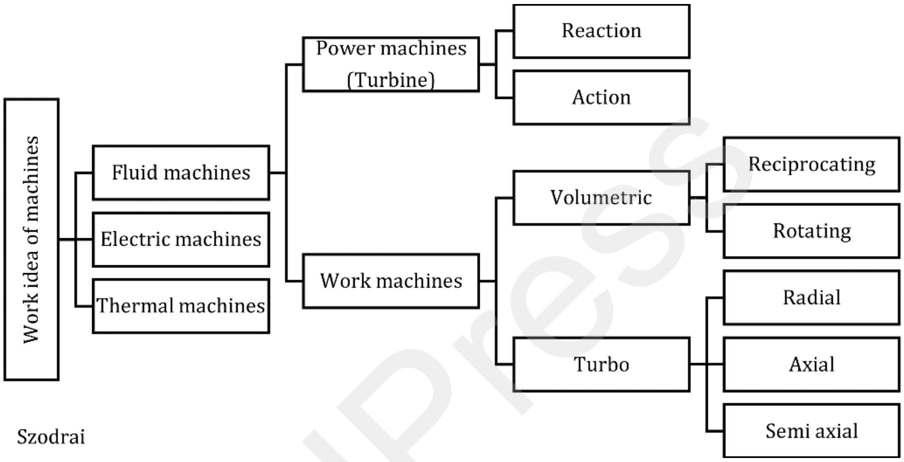

Figure 1.1. Types of machines

1.1.Turbo machine equation

1.1.1.Concept of Relative Velocity and Velocity Triangle

A turbomachine changes the kinetic energy of the fluid flow. In a turbomachine the inlet velocity, c_1 gets changed to the outlet velocity c_2. The velocity of fluid can be split into axial z, tangential u, and radial components r. The c absolute velocity is the vector sum of the impeller tangential velocity (u) and the relative velocity (v) of the fluid. On the figures and equations, the vector notations are missing because the name of the velocities are representing the directions. These three vectors can be drawn that can give us a velocity triangle. The angle between v and c is the β blade angle. This triangle can be drawn at any point of any type of turbomachine. Usually, it can be drawn to two significant points, one on the inlet and one on the outlet. As shown in figure 1.2; 1.3 and 1.4 for axial and centrifugal devices, respectively.

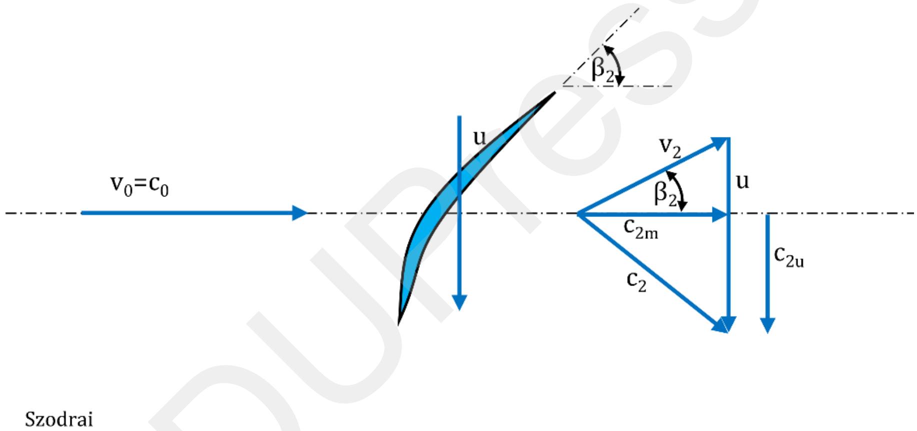

Figure 1.2. Axial machine velocity triangles for a cross-section

In figure 1.1 a rotor blade of a wind turbine or a ventilator can be seen, this is an axial machine, thus the direction of the flow has to be parallel to the rotation of the axis. Both the relative and the absolute velocity are not aligned with the rotation axis. It means that for a turbine more potential power could be harnessed or for a fan some unwanted radial flow is also generated. Without any enclosure, these devices are only effective in certain cases (at fixed flow). Placing an enclosure would not increase its effectiveness but only the generated pressure would be increased. With an enclosure, it is possible to apply guide vanes which could increase the efficiency. As in figure 1.3 due to the guide vanes, the absolute velocity now is aligned with the axis of rotation.

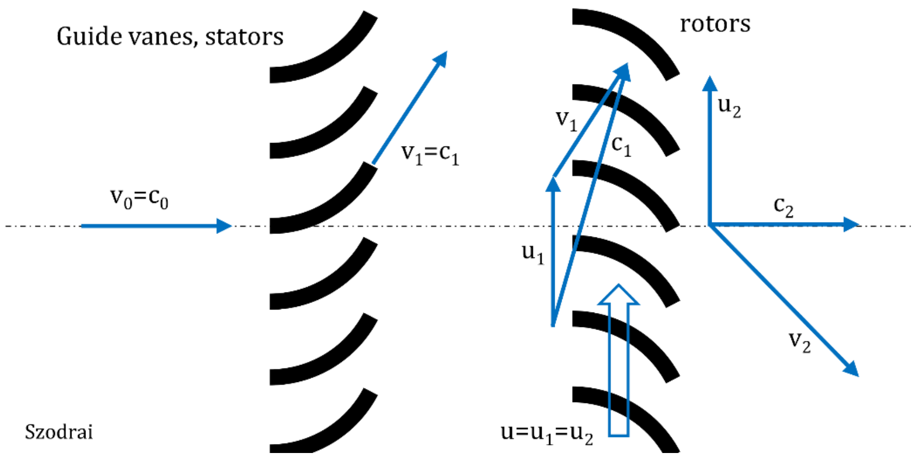

Figure 1.3. Velocity triangles of an axial machine with guide vanes

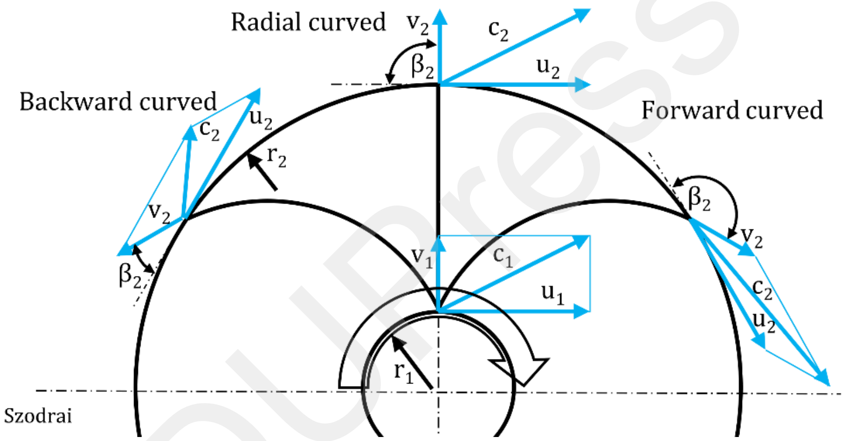

Figure 1.4. Radial machine velocity triangles

Radial impellers (see figure 1.3) require closed construction. For these kinds of ventilators or pumps guide vanes are not needed, furthermore for efficient devices it is assumed that the inlet flow is swirl-less. The magnitudes of the velocities can be calculated with trigonometry for axial and radial devices.

$$u_1 = 2 \cdot r_1 \cdot \pi \cdot RPM \tag{1.1}$$

$$u_2 = 2 \cdot r_2 \cdot \pi \cdot RPM \tag{1.2}$$

$$c_{u2} = c_{u2}/u_2 \cdot u_2 \tag{1.3}$$

$$c_1 = \sqrt{c_{u1}^2 + c_{m1}^2} \tag{1.4}$$

$$c_2 = \sqrt{c_{u2}^2 + c_{m2}^2} \tag{1.5}$$

$$v_1 = \sqrt{(u_1 - c_{u1})^2 + c_{m1}^2} \tag{1.6}$$

$$v_2 = \sqrt{(u_2 - c_{u2})^2 + c_{m2}^2} \tag{1.7}$$

c_m refers to meridian velocity. It is the velocity component that is normal the outlet of the impeller.

1.1.2.Euler Turbomachine Equation

To make the fluid flow in a turbomachine there should be an external torque acting on the impeller. This torque can be derived from Newton's 2nd law of motion, which acts as a fundamental equation of turbomachinery. The torque is given by the following equation, which is also called as Euler turbomachine equation and can be expressed by the following,

$$\vec{M} = \frac{d}{dt}\int_V (r \times \rho \cdot c) dV \tag{1.8}$$

where,

t is time (s)
r is the radial distance from the rotation axis (m)
ρ is the density of the fluid ($kg \cdot m^{-3}$)
c is the velocity of the fluid ($m \cdot s^{-1}$)
V is the volume of the control volume (m^3)
A cross-section area of the flow (m^2).

This torque can be divided into two parts: a local and a convective part, where the local part will not be generating any torque, due to steady operation.

$$\vec{M} = \underbrace{\int_V r \times \frac{\partial}{\partial t}(\rho \cdot c) dV}_{local} + \underbrace{\int_A (r \times c) \cdot \rho \cdot (c \cdot dA)}_{convective} \tag{1.9}$$

Simplifying with the stationary part the equation of the torque becomes:

$$\vec{M} = \int_A (r_0 \times c) \cdot \rho \cdot (c \cdot dA) \tag{1.10}$$

To calculate how much torque is generated on the axis the vector of the torque is divided into components.

$$M_z = \vec{M} \cdot e_z = \int_A e_z \cdot (r_0 \times c) \cdot \rho \cdot (c \cdot dA) \tag{1.11}$$

The multiplication of the e_z is the following:

$$e_z \cdot (r_0 \times c) = c \cdot (r_0 \times e_z) = c \cdot [e_z \times (z \cdot e_z + r \cdot e_r)] \quad (1.12)$$
$$= \underbrace{c \cdot z \cdot (e_z \cdot e_z)}_{0} + c \cdot r \cdot \underbrace{(e_z \cdot e_r)}_{e_u} = r \cdot c_u$$

where,

e_Z is the axial component (-)
e_r is the radial component (-)
e_u is the tangential component (-)
z is the distance along the axis which is 0m.
c_u is the velocity is the u-direction component ($m \cdot s^{-1}$)

The shaft torque now can be expressed with the following:

$$M_z = \int_A r \cdot c_u \cdot \rho \cdot (c \cdot dA) \quad (1.13)$$

Integrate between the inlet (1) and the outlet (2) boundary:

$$M_z = \int_1^2 r \cdot c_u \cdot \rho \cdot c \cdot dA \quad (1.14)$$

$$M_z = \rho \cdot c \cdot [(r_2 \cdot c_{u2} \cdot A_2) - (r_1 \cdot c_{u1} \cdot A_1)] \quad (1.15)$$

And by assuming the inlet and outlet surfaces are equally large:

$$M_z = \rho \cdot c \cdot A \cdot [(r_2 \cdot c_{u2}) - (r_1 \cdot c_{u1})] = \dot{m} \cdot (r_2 \cdot c_{u2} - r_1 \cdot c_{u1}) \quad (1.16)$$

M_Z is the torque of the fluid and the M is the torque of the shaft is the opposite.

$$M = -M_z = \dot{m} \cdot (r_1 \cdot c_{u1} - r_2 \cdot c_{u2}) \quad (1.17)$$

This equation expresses the relation between the fluid and the torque on the axis. We can see that the torque depends on the mass flow axial distances from the boundaries and the velocities occurring on the inlet and outlet boundaries.

By multiplying the RPM (round per minute) (min^{-1}) with the torque we get the power.

$$P = M \cdot RPM = \dot{m} \cdot RPM \cdot (r_1 \cdot c_{u1} - r_2 \cdot c_{u2}) \quad (1.18)$$

The multiplication of the axial distance and the angular speed can give us the u tangential velocity.

$$P = \dot{m} \cdot (u_1 \cdot c_{u1} - u_2 \cdot c_{u2}) = \dot{m} \cdot y_i \quad (1.19)$$

By dividing with the mass flow, we can get the specific energy change (y) for the Euler equation. If the power required by the fluid is positive, that means fluid is absorbing energy therefore the device is acting as a compressor. Otherwise, fluid is losing energy, so the device acts as a turbine.

In power machines, the velocities are decreasing in the device.

$$y_i = u_1 \cdot c_{u1} - u_2 \cdot c_{u2} \tag{1.20}$$

In work machines, the velocities are increasing in the device.

$$y_i = u_2 \cdot c_{u2} - u_1 \cdot c_{u1} \tag{1.21}$$

If we calculate a more ideal state, we can multiply the efficiency of the hydraulic losses.

$$y = \eta_h \cdot y_i \tag{1.22}$$

In this equation, there is no density therefore compressible and non-compressible fluids can be also applied.

$$\dot{P} = \dot{m} \cdot \eta_h \cdot y_i \tag{1.23}$$

$$\dot{P} = \Delta p \cdot \dot{V} \tag{1.24}$$

Finally, we can conclude that on every type of turbomachine the intake power is always equal with the multiplication of the Δp pressure drop that occurs before and after the device and the $\dot{V}$ volume flow that flows through the device.

1.2. Calculations

For a backward, radial, and forward-curved blade pump impeller following are known revolution $500 \mathrm{min}^{-1}$, diameters 30mm and 300mm, meridian velocity $4 \mathrm{m} \cdot \mathrm{s}^{-1}$ and the velocity ratios c_{u2}/u_2=0.6; 1; 1.8, gravity is $9.81 \mathrm{m} \cdot \mathrm{s}^{-2}$

- Draw its velocity triangles for the outlet and determine the missing velocities (for all three cases).
- Calculate the angle of attack at the outlet (β_2) and the head (H) (for all three cases).

Recommended literature for this section:

[1] Fluid Mechanics for Engineers in SI Units David A. Chin 2018

2. Centrifugal pumps

Operation of the pump creates a suction effect (a lower pressure) at the suction side so that the fluid can enter the pump through the inlet. Pump operation also causes higher pressure at the discharge side by forcing the fluid out at the outlet.

Centrifugal pumps provide the primary force to distribute and circulate fluid in a hydraulic system. In a centrifugal pump, an electric motor or other power source rotates the impeller at the motor's speed. Impeller rotation adds energy to the fluid after it is directed to the centre of the rotating impeller. The impeller is the most essential part of the pump as it adds pressure and kinetic energy to the fluid (see figure 2.1). The fluid is always pulled in through the eye of the impeller. regardless of the direction of the rotation. The impeller can modify the pump's performance: if the impeller is thicker (b) more water flows through it. If the impeller has a larger outer diameter (D_2) it can generate more pressure. With the increase of the RPM both the flow and pressure can be increased so the intake power will also be increased.

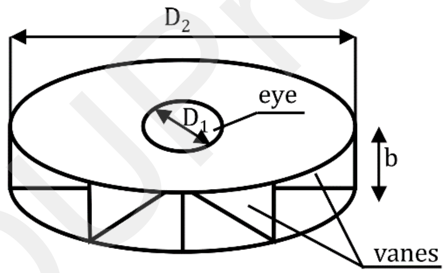

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Figure 2.1. Schematic of an impeller

2.1.Wet and dry runner pumps

During the operation of the pump, it generates heat. We must deal with that generated heat whether the heat is low or high. If the intake power is below 1kW, wet runner pumps can be used. In these pumps, the motor's rotor is encapsulated together with an impeller where the water is being pumped surrounds the rotor. It has several advantages: the pump has higher efficiency, lower starting torque, runs more silently, has no mechanical seal which means no leakage or maintenance is needed in contrast to the dry runner pump. However, if the intake power is much higher the fluid cannot cool it down the pump. This large intake power is often needed due to the high-pressure demand or as a result of high fluid temperature. Dry runner pumps have no direct contact between the pumped liquid and rotor of the motor, which leads to lower efficiency and louder operation. Another mentionable note is that the wet runner pump must be fitted vertically while the dry runner can be fitted both horizontal and vertical.

2.2.Pump curves

The relation between the pressure increases (Δp) or head pressure (H) and the volume flow ($\dot{V}$) gives the pump curve. Usually, with a pump curve, it can be decided whether that pump is capable to perform a given task or not. The shape of the pump curve is determined by the impeller. Every change in the delivery head always results in a change inflow.

2.3.Centrifugal pump curve equation

With the Euler turbomachine equation, the following relation can be expressed. Knowing the specific power change (y) ($W \cdot kg^{-1}$) equation and assuming that at the inlet of a centrifugal pump flow velocity is radial, the tangential component of velocity is zero. Thus, the inlet component can be disregarded.

$$y_i = \eta_h \cdot u_2 \cdot c_{u2} \tag{2.1}$$

So, the head developed by the pump simplifies to:

$$H = \Delta p \cdot (\rho \cdot g)^{-1} = \rho \cdot \eta_h \cdot u_2 \cdot c_{u2} \cdot (\rho \cdot g)^{-1} \tag{2.2}$$

Outlet blade angle β_2 (°) can be easily represented as follows.

$$cot\beta_2 = (u_2 - c_{u2}) \cdot {c_{m2}}^{-1} \tag{2.3}$$

You can substitute values of c_{u2} from this equation, into the head equation. After substituting the value of the effective flow velocity (c_{m2}), we get the most important performance equation of a centrifugal pump. How energy head is varied with flow rate:

$$H = u_2^2 \cdot g^{-1} - u_2 \cdot cot\beta_2 \cdot \dot{V} \cdot (2 \cdot \pi \cdot r_2 \cdot b_2 \cdot g)^{-1} \tag{2.4}$$

where b is the width of the impeller (m) and g is the gravity $9.81 m \cdot s^{-2}$. If β_2 is less than 90° pressure head decreases with an increase in flow. These kinds of impellers are called backwards curved (figure 2.2). If β_2 is 90°, with flow rate there is no change in pressure rise. They are called radial type (figure 2.3). If β_2 is more than 90°, pressure increases with an increase in flow rate. Such blades are called forward-curved blades (figure 2.4).

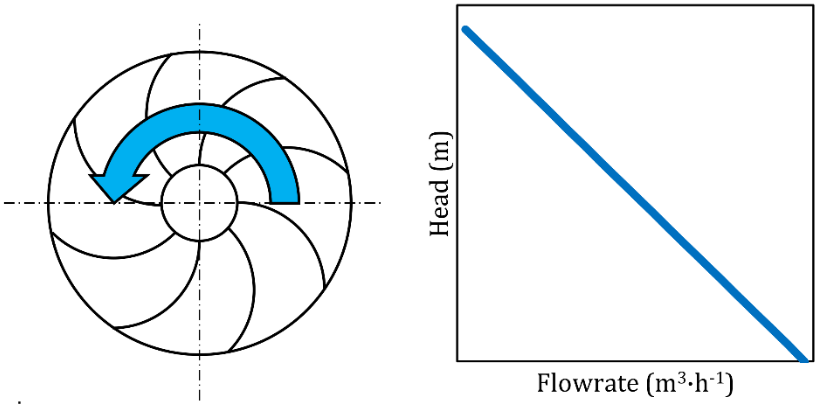

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Figure 2.2. Backward curved blade

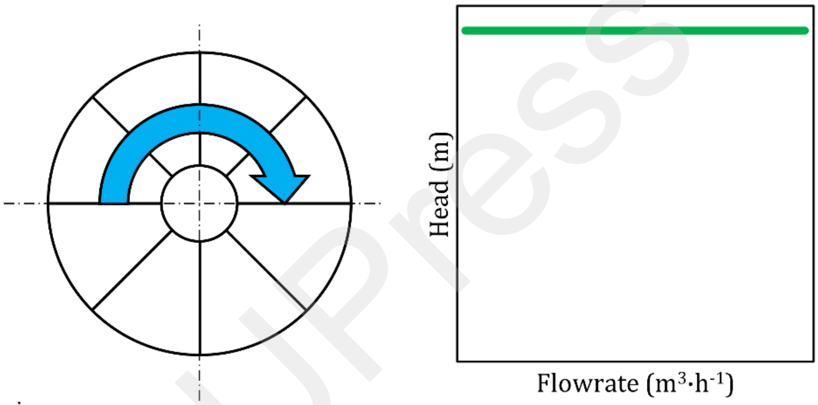

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Figure 2.3. Radial curved blade

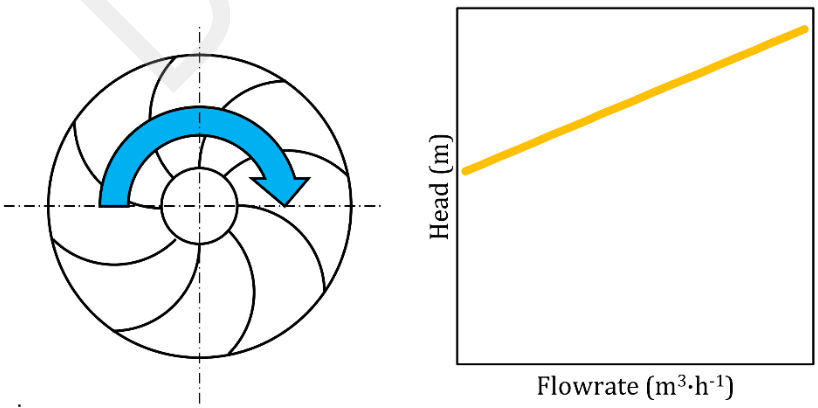

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Figure 2.4. Forward curved blade

For a backward curved bladed pump when the head decreases, the discharge power consumption stabilizes with the flow. Since the radial blades does not have any effect on flow rate, power consumption increases proportionally. When head increases for a forward curved bladed pump power consumption increase exponentially with the flow. This will make the operation unstable and will eventually lead to the burnout of the motor. Backward curved blades which have self-stabilizing characteristics in power consumption are the most preferred ones in the industry.

The pump characteristic curve can be flat or steep. Flat characteristic pumps are usually installed where the head deviation is not substantial. Steep characteristic pumps are usually installed where higher pressure and constant flow are usually desired.

In the pump curve, there are two mentionable parameters, one is the shut-off head, which is the maximum pressure that a pump can generate at that point the pump cannot generate fluid flow. The other one is the maximum flow that gives the largest volume flow that the pump can generate. At that point there is no head produced.

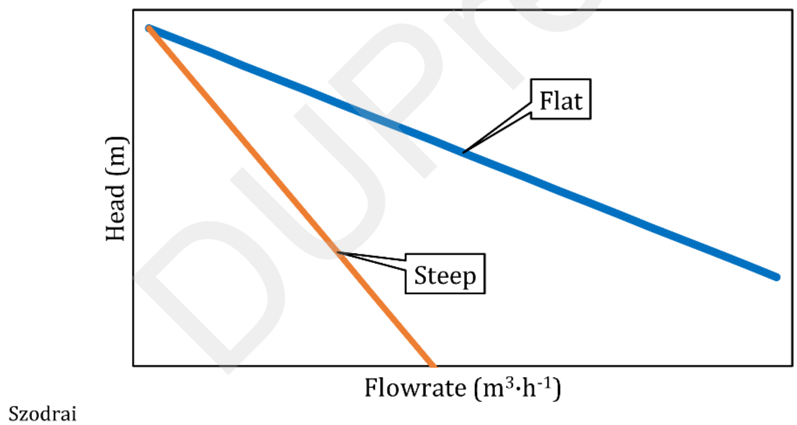

Figure 2.5. Flat and steep curves

2.3.1. Realistic curve

In theory, the pump curve characteristic is straight at any blade angle as figure 2.6 left side shows. However, significant pressure loss can occur in the housing of any pump; thus, curves will be deformed. figure 2.6. right side two of these losses are shown. The shock loss has a significant effect on the maximum head and the maximum flow rate, while friction is mostly decreasing the maximum flow rate. Shock loss can be reduced by optimizing the shape of the housing while friction can be decreased with a smoother housing surface. (A small reminder, most of the housing made by casting which has a rough surface.)

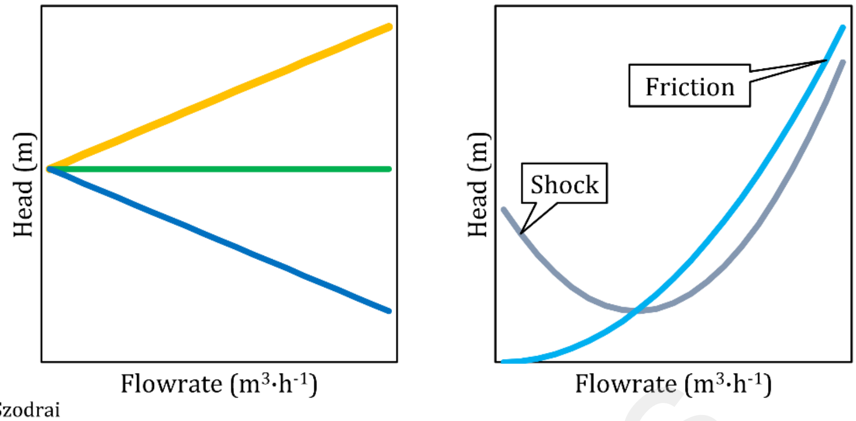

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Figure 2.6. Ideal and realistic pump curves (left diagram legend: blue: backward curved; green: radial curved; yellow forward curved blade impellers)

Both friction and shock losses are added up in the pump which decreases the head corresponding to the flow rate. When pressure losses are subtracted from the ideal pump curves we get realistic pump curves that will take up a shape similar to figure 2.7.

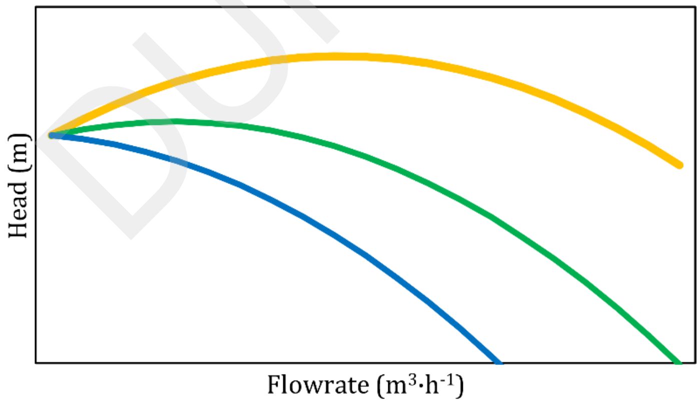

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Figure 2.7. Ideal and realistic pump curves (legend: blue: backward curved; green: radial curved; yellow forward curved blade impellers)

2.4.Cavitation and NPSH

When flow moves through an impeller its pressure increases as the radial distance grows from the centre. Liquid materials have high-pressure tolerance, yet low pressure could cause cavitation. Cavitation happens when due to low pressure the water reaches its saturation point and it starts to boil at low temperature, thus vapour is present in the impeller. Further away from the centre due to high pressure the vapour bubbles collapse on the impeller vanes. Frequent collapses damage the impeller. Damage due to cavitation is often only detected when the pump is dismantled. Furthermore, cavitation results in increased noise and vibrations (caused by the formation and collapse of vapour bubbles), which can consequently damage the bearings, and the shaft seals. This low pressure can appear on the vanes of the centrifugal pumps if the inlet pressure is low relative to the volume flow. This inlet pressure can be determined by knowing the saturation pressure at the temperature of the fluid. If we know the saturation pressure at a given volume flow can get the NPSH (Net Positive Suction Head) which is an expression for the suction capability of the pump. It is used to calculate the inlet pressure needed at the pump to avoid cavitation. The NPSH curve displays the minimum required head pressure to avoid cavitation inside the pump.

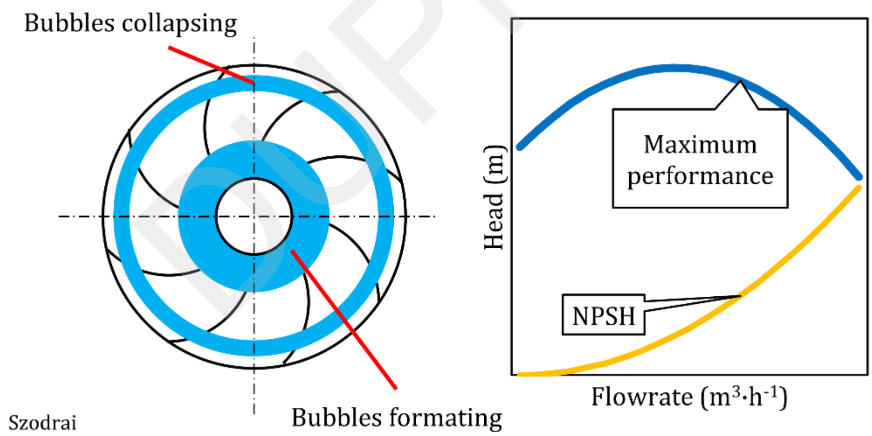

Figure 2.8. Cavitation in the pump

When a pump is operated the head corresponding to the flow rate (also known as duty point) must be above the NPSH curve. To be sure the safety of the operation NPSH+0.5m is used as a safety margin. If the duty point is below the NPSH curve the following measures can be done. Duty point can be moved in the x-direction by lowering the flow rate, and also in y-direction where the inlet pressure must be raised. These measures decrease the pump performance thus stronger or additional pumps are required to satisfy the demand.

2.5.Connection for pump and fan series and parallel

To enhance a turbomachine head or volume flow, multiple devices can be connected. The connection can have two options: series or parallel. If the turbomachines are connected in series their head pressure will add up at every single point of their performance curve (see figure 2.9).

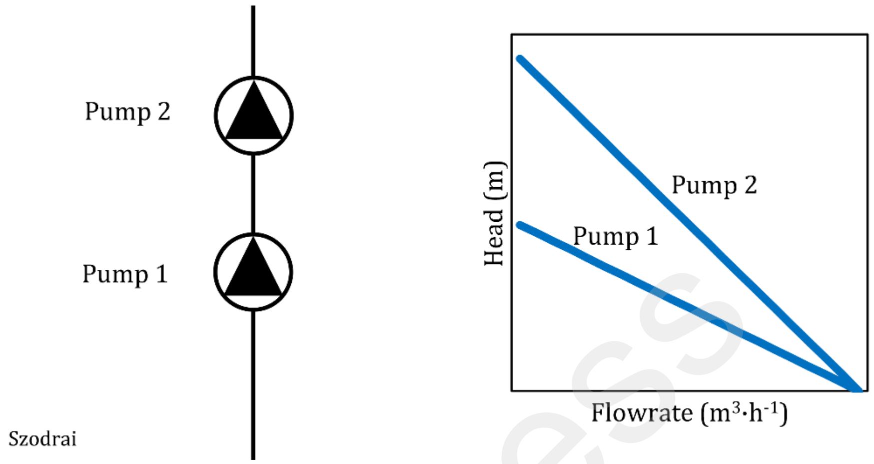

Figure 2.9. Ideal pump curves when connected in series.

The pumps are usually single-stage pumps, which means that they have only one impeller in the housing. In multistage pumps, two or more impellers are arranged in series in such a way that the discharge from one impeller enters the eye of the next impeller. If a pump has two impellers arranged in series, it is called a two-stage pump, and if the pump has three impellers arranged in series, it is called a three-stage pump, and so on. Multistage pumps are typically used when a large pumping head is required. Multistage or deep well pumps are used in the extraction of water from deep underground sources.

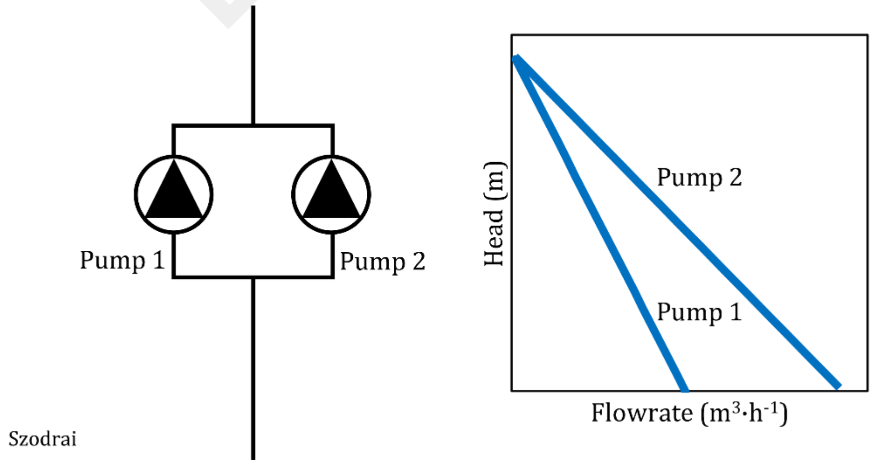

Figure 2.10. Ideal pump curves when connected in parallel.

When pumps are connected in parallel, they are usually required to operate on a wide volume flow range. The other reason for a pump to be placed in parallel is to keep two pumps in housing to alternate the two pumps and their load therefore their lifetime can be enhanced. When turbomachines are connected in parallel their volume flows are added up.

If two turbomachines connected in parallel their flow rate will only add up if they produce the same pressure. If it has a pressure difference, backflow will occur which can even prevent the flow in the whole hydraulic system. For this reason, a non-return valve is needed in front of every pump. In liquid fluid systems, the non-return valve can prevent the backflow, in gaseous fluid systems parallel-connected fans cannot be used.

Multiple pumps performance can be determined by adding the pump curve characteristics. It means that if the relationship between the head and flow rate is known for each pump, it can be used for function calculations and the coupled function can be determined. For example, when pumps connected in series the head and volume flow functions are added. The head at a given (V_x) flow rate can be calculated in function of the impeller geometries and volume flow; this equation assumes no pressure losses.

$$H_x = (D_2 \cdot \pi \cdot RPM)^2 \cdot g^{-1} - D_2 \cdot \pi \cdot RPM \cdot cot\beta_2 \cdot \dot{V}_x \cdot (\pi \cdot D_2 \cdot b_2 \cdot g)^{-1} \tag{2.5}$$

$$H_{coupled} = H_1(\dot{V}_x) + H_2(\dot{V}_x) \tag{2.6}$$

When it is connected in parallel the volume flow is added from the previous equation, if the volume flow is derived:

$$\dot{V}_x = (H_x - (D_2 \cdot \pi \cdot RPM)^2 \cdot g^{-1}) \cdot \pi \cdot D_2 \cdot b_2 \cdot g \cdot (-D_2 \cdot \pi \cdot RPM \cdot cot\beta_2)^{-1} \tag{2.7}$$

In this parallel-connected case, the volume flow is calculated at a given (H_x) head pressure, thus the flow rate for coupled pumps will be.

$$\dot{V}_{coupled} = \dot{V}_1(H_x) + \dot{V}_2(\mathrm{H}_x) \tag{2.8}$$

2.6.Calculation

Two pump outer diameters D_1=100mm, D_2=150mm, impeller widths b_1=30mm, b_2=40mm, exiting blade angles β_1=45°, β_2=30°, revolutions RPM_1=500min^{-1}, RPM_2=600min^{-1} and gravity 9.81·m·s^{-2} are given.

- Calculate the head (H) if the volume flow ($\dot{V}$) is half of the maximum volume flow and the pumps are connected in parallel.
- Calculate the head if the volume flow is half of the maximum volume flow and the pumps connected in series.
- Calculate the volume flow if the head is half of the maximum head and the pumps are connected in parallel.
- Calculate the volume flow if the head is half of the maximum head and the pumps connected in series.

Recommended literature for this section:

[1] Fluid Mechanics for Engineers in SI Units David A. Chin 2018

3. Pump efficiency

The performance of a pump is measured by how effectively it produces head at a flow rate. The head added by a pump, is equal to the difference between the total head on the discharge side of the pump and the total head on the suction side of the pump. The efficiency of a pump, (η) is defined by the ratio of the delivered power (Δp·V) and the absorbed power (P_{shaft}).

$$\eta_{overall} = \dot{P}_{useful} \cdot \dot{P}_{shaft}^{\;-1} = \Delta p \cdot \dot{V} \cdot \dot{P}_{shaft}^{\;-1} \tag{3.1}$$

Pumps are inefficient for a variety of reasons, such as frictional losses as the fluid moves over the solid surfaces, viscous dissipation within the fluid, separation losses, leakage of fluid between the impeller and the casing, mechanical losses in the bearings and sealing glands of the pump, and shock losses due to the inlet flow angle not matching the blade angle.

A pump must deliver a certain amount of flow rate with a certain amount of head. The flow rate is usually defined by another device demand and the head is equal to the hydraulic systems pressure losses. the flow rate and the head define a point, and it is called duty point. The duty point for a pump has different efficiencies. Though the efficiencies follow a pattern, and the values peak at a certain point. If the same efficiency points are connected an iso efficiency curve can be drawn (see figure 3.1).

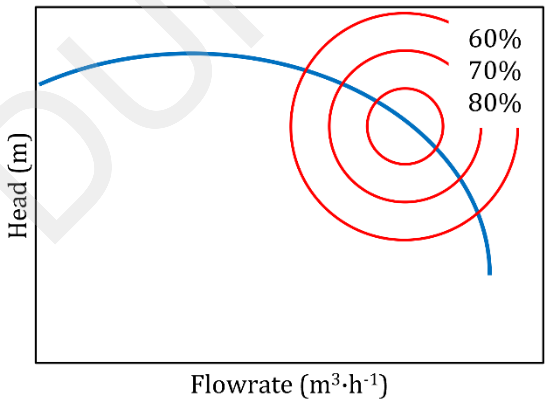

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Figure 3.1. Efficiency curves

The goal is to select and operate a pump near its peak efficiency. The result is a more efficient use of electricity thus reducing operating costs. However, sometimes it is not possible to select a pump that fits the optimum duty point because the requirements change over time. The dynamic behaviour can be attributed to the fluctuating flow demand or when head increases due to fouling.

Often, oversized pumps are selected for the system and it is necessary to limit the performance to adjust the pump to the changed requirements. The most common methods of changing pump performance are applying a throttle valve or using a bypass branch, changing the diameter of the impeller, modulating the revolution. All methods can be carried out continuously during operation apart from the modifying impeller diameter method. [1]

3.1. Throttle valve

If a throttle valve is placed in series with the pump it makes it possible to adjust the duty point. The throttling results in a reduced flow. The throttle valve adds resistance to the system and raises the system curve to a higher position. With the throttle valve connected in series to the pump, the flow is not changing. When the pump performance is adjusted by the throttling method, the pump will deliver a higher head than necessary for that system. The increased head leads to higher power consumption. This control method is one of the cheapest solutions, yet the most inefficient. Also, it shows that if necessary, the number of valves should be reduced in a hydraulic system.

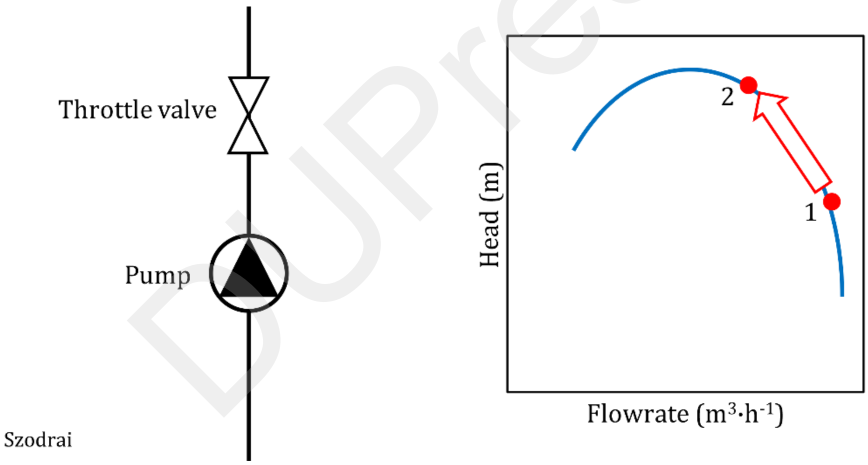

Figure 3.2. Throttle control (red dots are duty points)

3.2. Bypass control

Rather than connecting a valve in series with the pump, a bypass valve can be used to adjust the pump performance. Compared to the throttle control, installing a bypass valve head and flow rate will not be changed, yet on the bypass branch, a flow will flow. The power that is used to circulate the flow in the bypass branch is a loss. From figure 3.3 it can bees that duty point 2 is not on the pump curve, the reason is that the pump still operates on duty point 1 but due to the bypass branch the volume flow is reduced.

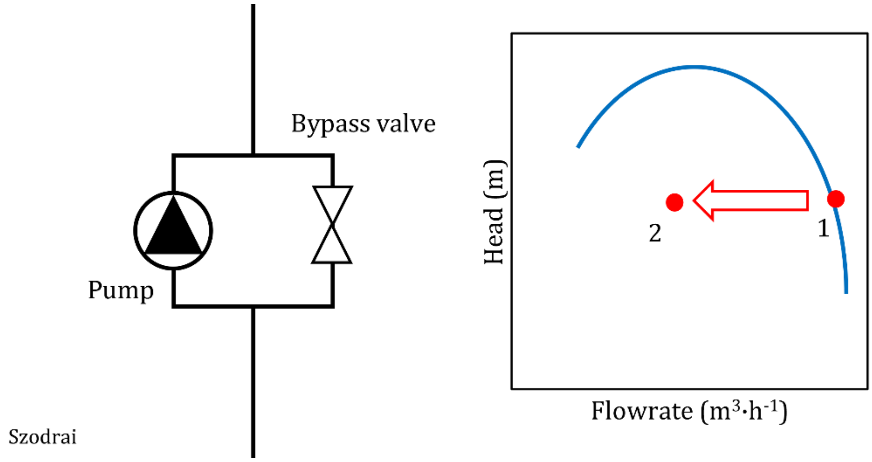

Figure 3.3. Bypass control (red dots are duty points)

3.3.Impeller diameter

Another way of adjusting the performance of a centrifugal pump is by modifying the impeller diameter in the pump. Usually, this means that the duty point is changed a new pump with a smaller or larger diameter is chosen.

3.4.RPM modulation

RPM modulation is done by modulating the engine's frequency. This one is the most efficient way of adjusting pump performance exposed to variable flow requirements. The minimal revolution reduction is 40%. This value belongs to 15Hz. At this point the cooling is not proper subsequently it will cause overheating. This can be useful when modifying the operating parameters of an existing pump.

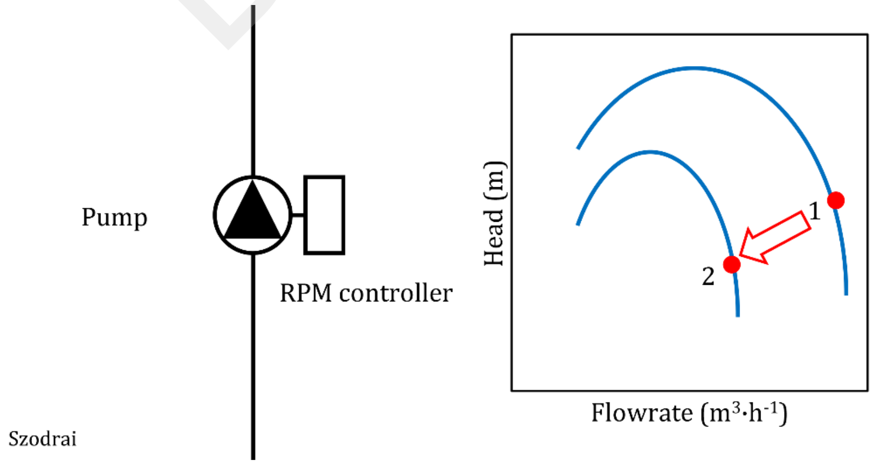

Figure 3.4. RPM control (red dots are duty points)

3.5.Affinity and similarity laws

The affinity laws are assuming that the pump efficiency remains constant if only the RPM is changed. Complete sets of pump curves are usually not available. The pump affinity laws allow us to determine the performance of the same or similar pump.

Similarity rules are used when a pump impeller parameter is scaled up or down and affinity rules used when the examined two different states are corresponding to one impeller geometry.

Similarity rules can be derived by using the head, flow rate, and power equations. The three equation parameters are divided into essential and non-essential parameters. The non-essential parameters are constant and cannot change if geometries are similar.

The volume flow rate can be expressed:

$$\dot{V} = D_2 \cdot \pi \cdot b_2 \cdot \psi_2 \cdot c_{2m} \tag{3.2}$$

where ψ_2 is the contraction factor and the width of the impeller (b_2) can be normalized with the dimeter.

$$b_2' = b_2 \cdot D_2^{-1} \tag{3.3}$$

The velocity at the meridian can be also expressed with diameters and revolution if the blade angle (β) and are known:

$$c_{2m} = (u_2 - c_{u2}) \cdot tan\, \beta_2 \tag{3.4}$$

Since the tangential velocity (u_2) is:

$$u_2 = D_2 \cdot \pi \cdot RPM \tag{3.5}$$

To determine c_{u2} blade transmission ratio is needed which is also a constant.

$$\xi = c_{u2} \cdot u_2^{-1} \tag{3.6}$$

Finally, the flow rate equation becomes:

$$\dot{V} = \pi^2 \cdot b_2' \cdot \psi_2 \cdot tan\, \beta_2 \cdot (1 - \xi) \cdot \ {D_2}^3 \cdot RPM \tag{3.7}$$

The pressure head applying the known equations becomes:

$$H = c_{u2} \cdot u_2 \cdot \mathrm{g}^{-1} = \xi \cdot u_2^2 \cdot \mathrm{g}^{-1} \tag{3.8}$$

$$H = \xi \cdot \pi^2 \cdot \ {D_2}^2 \cdot RPM^2 \cdot \mathrm{g}^{-1} \tag{3.9}$$

To determine the power Euler's turbomachine equation can be used:

$$P = \dot{V} \cdot \rho \cdot g \cdot H \tag{3.10}$$

With the multiplication of the flow rate and head, the power becomes:

$$P = \pi^4 \cdot b_2' \cdot \psi_2 \cdot tan\,\beta_2 \cdot (1 - \xi) \cdot \xi \cdot \rho \cdot \ D_2{}^5 \cdot RPM^3 \tag{3.11}$$

Similarity laws are attained by dividing two cases with each other where only outer diameter and RMP changes the efficiency remains constant.

$$\frac{V_1}{V_2} = \left(\frac{D_1}{D_2}\right)^3 \cdot \left(\frac{RPM_1}{RPM_2}\right) \tag{3.12}$$

$$\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2 \cdot \left(\frac{RPM_1}{RPM_2}\right)^2 \tag{3.13}$$

$$\frac{P_1}{P_2} = \frac{\rho_1}{\rho_2} \cdot \left(\frac{D_1}{D_2}\right)^5 \cdot \left(\frac{RPM_1}{RPM_2}\right)^3 \tag{3.14}$$

If the geometry is the same and the only RPM is changed according to the affinity law the efficiency is constant.

$$\frac{V_1}{V_2} = \left(\frac{RPM_1}{RPM_2}\right) \tag{3.15}$$

$$\frac{H_1}{H_2} = \left(\frac{RPM_1}{RPM_2}\right)^2 \tag{3.16}$$

$$\frac{P_1}{P_2} = \frac{\rho_1}{\rho_2} \cdot \left(\frac{RPM_1}{RPM_2}\right)^3 \tag{3.17}$$

3.6.Calculations

A pump is controlled with RPM modulation, calculate the missing parameters of the duty points (assuming the law of affinity can be applied).

Table 3.1 Pump parameters

	unit	1	2	3
V	$m^3 \cdot h^{-1}$	30		
H	m	4		
P	kW	3		
RPM	min^{-1}	500	600	700
η	%			

Calculate the missing shaft power, volume flow and head for 3 geometrically similar centrifugal pumps if parameters are given for the first 2nd 3rd pump diameters and RPM are known. (assuming law of similarity can be applied)

Table 3.2 Pump parameters

	unit	1	2	3
V	$m^3 \cdot h^{-1}$	30		
H	m	3		
P	kW	3		
RPM	min^{-1}	500	600	700
D	mm	120	150	200
η	%			

A flow must be reduced from V_1=300$m^3 \cdot h^{-1}$ to V_2=280$m^3 \cdot h^{-1}$. Pump parameters are given (D_2=300mm; b=30mm; β_2=45°; RPM=500min^{-1}). Depending on the method of performance adjustment (throttle, bypass, diameter, RPM) calculate the power consumption change.

Recommended literature for this section:

[1] Fluid Mechanics for Engineers in SI Units David A. Chin 2018

4. Hydraulics

Hydraulic system performance curve gives the pressure loss in a system at a given flow rate (see figure 4.1). The intersection of the hydraulic system performance curve and the pump performance curve gives the duty point. A hydraulic system includes all the pipes, fittings, and devices. The fluid must flow through to represent the pressure loss the fluid experiences. The use of fluid systems is widespread. They serve in a wide range of applications such as domestic, commercial, and agricultural services. They are applied in wastewater systems, industrial applications for food processing, chemical, petrochemical, and mechanical industries. In building engineering, pumps are used in heating and cooling systems. Fluid machines have two main purposes: Transfer fluid from one place to another place, these are called hydraulically open systems. Circulating fluid around a system is called hydraulically closed system. [1]

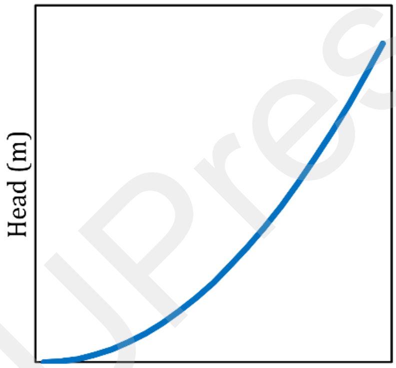

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Figure 4.1. Hydraulic loss curve of a closed system

4.1.Hydraulic losses valve and friction

To calculate how much pressure difference is in a hydraulic system, the Bernoulli equation can be used.

$$0.5 \cdot \rho_1 \cdot {c_1}^2 + \rho_1 \cdot g \cdot h_1 + p_1 = 0.5 \cdot \rho_2 \cdot {c_2}^2 + \rho_2 \cdot g \cdot h_2 + p_2 + \Delta p \qquad (4.1)$$

$$\Delta p = pressure\ loss = friction\ loss + head\ loss \qquad (4.2)$$

The friction loss pressure loss in pipes and tubes:

$$p_{firction} = p_{dynamic} \cdot \lambda \cdot L \cdot D^{-1} = 0.5 \cdot \rho \cdot c^2 \cdot \lambda \cdot L \cdot D^{-1} \qquad (4.3)$$

λ is the friction coefficient can be calculated or can be looked up from the Moody diagram. The Moody diagram represents the pipe friction coefficient in the function of the surface roughness and Reynolds Number. Reynolds number (Re) is a dimensionless number that gives a measure of the ratio of inertial forces to viscous forces. Flow in a pipe or a tube, the Reynolds number is defined as:

$$Re = inertial\ forces \cdot viscous\ forces^{-1} = c \cdot D_H \cdot \nu^{-1} \qquad (4.4)$$

where: D_H is the hydraulic diameter of the pipe (m), c is the mean velocity of the fluid ($m \cdot s^{-1}$), ν is the kinematic viscosity ($m^2 \cdot s^{-1}$). For shapes such as rectangular ducts, the hydraulic diameter defined as:

$$D_H = 4 \cdot A \cdot P^{-1} \qquad (4.5)$$

where: A is the cross-sectional area and P is the wetted perimeter. The wetted perimeter for a channel is the total perimeter of all channel walls that are in contact with the flow.

For turbulent flow, the friction coefficient depends on the Reynolds Number and the roughness of the pipe wall.

The head losses caused by fittings, elbows, valves, etc.

$$p_{head} = \sum \zeta \cdot p_{dynamic} \qquad (4.6)$$

where: the ζ is the head loss coefficient. It tells how much will change the static pressure at a given dynamic pressure. These coefficients are geometry dependent and vary also depending on the flow. This head loss coefficient can be measured or estimated by simulation.

4.2. Hydraulically open systems

There are hydraulically closed and hydraulically open systems and also airtight and not airtight systems however these are not the same. We talk about hydraulically open systems when the geodetic pressure influences the head. These systems have inlets and outlets, also it has level difference along the system’s length. Mass flow is equal at every point, pressure losses are adding up at every section.

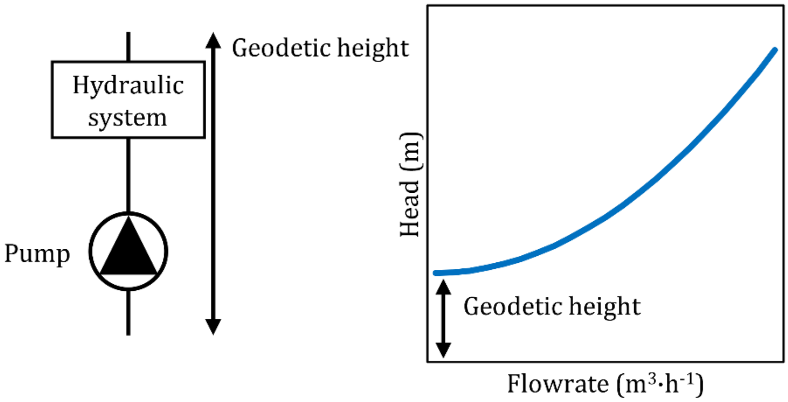

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Figure 4.2. Schematic of negative geodetic open system

For a negative geodetic system to produce flow pressure is required at least just as much height as the positive geodetic height. For open systems, these constructions are the most common. In some rare occurrences, positive geodetic systems do not even need a pump until a given flow rate.

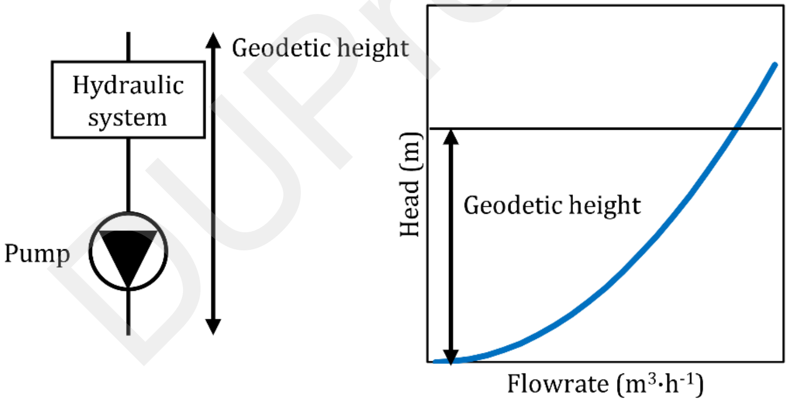

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Figure 4.3. Schematic of positive geodetic open system

4.3.Hydraulically closed systems

Typically, closed systems are systems, which transport heat energy in heating systems, air-conditioning systems, process cooling systems, etc. In hydraulically closed systems the geodetic pressure does not influence (on that significant level) the pressure difference between any two points. Closed systems that are higher than around 30m some geodetic pressure can appear. The pressure loss of these kinds of systems is always as large as the hydraulic loss in the system.

In general, closed systems are complex, and consists of numerous sections. In a closed system, the main circuit will have the largest pressure loss which is also the required head. The other sections that are not part of the main circuit and part of the closed system are the branches. In a system where a branch is placed see figure 4.4 the branch has less hydraulic resistance, thus more fluid could flow through it. In a larger system it means that in circuits where the hydraulic loss is large no flow can occur. That is why hydraulic balancing is needed. Hydraulic balancing is when the branches and the main circuit has the same pressure loss.

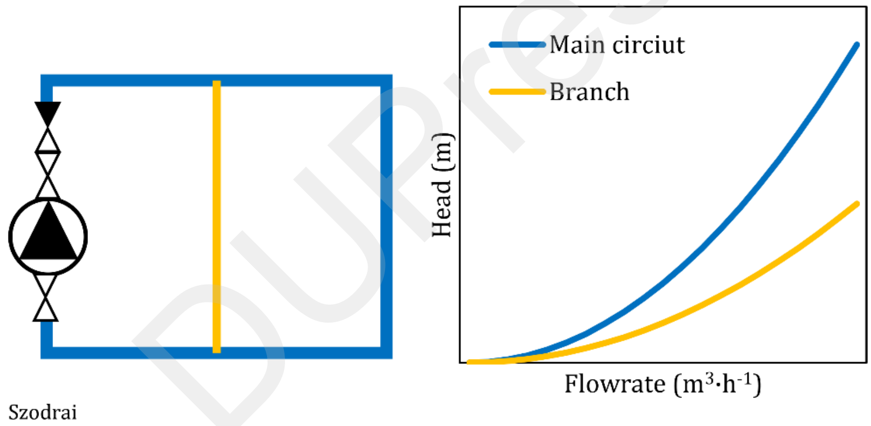

Figure 4.4. Schematic of a closed system

4.4.Siphon effect

A siphon is a pipe or tubing system that allows the transfer of fluid from an upper location to a lower one; the key feature of a siphon is that the fluid is moved upwards from its entry point before it turns down to its exit point. In fluid machinery, it is relevant because these systems only require the pump to start to flow, after the flow is self-sustainable.

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Figure 4.5. Schematic of a siphon

4.5.Calculations

Calculate the hydraulic resistance of the system if the flow rate of the pump is 0,1m^3·h^{-1}. Diameters, lengths, and loss coefficients are highlighted in figure 4.6. (Flow is laminar; D_1=32mm; D_2=16mm; L_1=30m; L_2=10m; ζ_1=1.5; ζ_2=0.5; ζ_3=0.8; ζ_4=2)

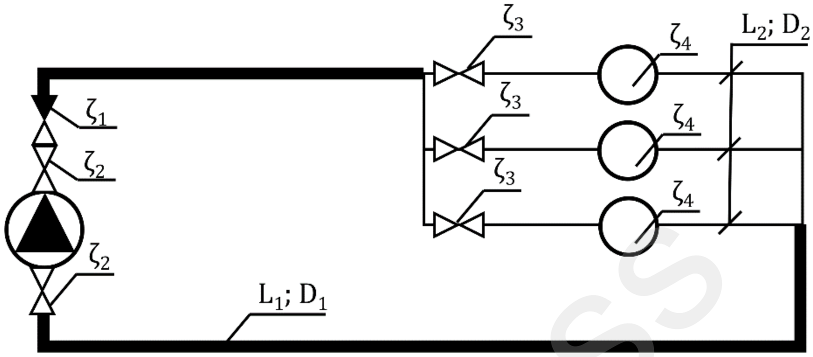

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Figure 4.6 Schematic of a closed system

Calculate the hydraulic resistance of the system if the flow rate of the pump is 0,2m^3·h^{-1}. Diameters, lengths, and loss coefficients are highlighted in figure 4.7. (Flow is laminar; D_1=32mm; D_2=28mm; H_1=4m; H_2=5m; L_1=10m; L_2=40m; ζ_1=1.5; ζ_2=0.5; ζ_3=0.5)

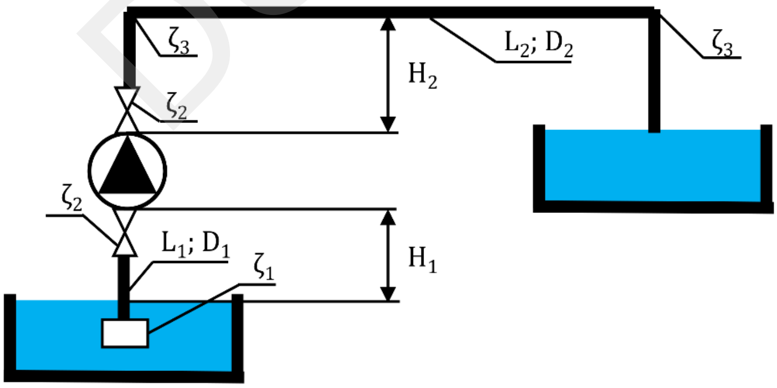

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Figure 4.7 Schematic of a closed system

5. Fluid machine utilization

5.1. Cordier diagram

The Cordier diagram is an empirical diagram based on measurements. Its main purpose is to help the design stage by highlighting what type of device would be the most efficient for a given case.

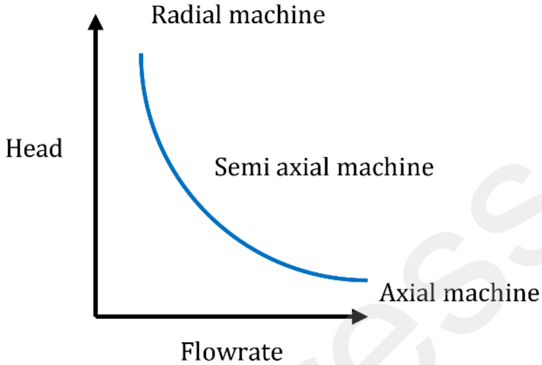

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Figure 5.1. Cordier diagram

As figure 5.1 shows the axis could be head and flow rate. Based on experience it can be concluded that for cases which require high flow rate and low head axial devices are preferred and in cases where low flow rate and the high head is required volumetric or radial devices are needed. The blue line in figure 5.1 represents the machines duty point, and the type corresponds to a value range.

These diagrams can be applied to any fluid machine. The values usually normalized on these diagrams thus for further details other diagrams or specific parameters are needed.

5.2.Fluid machine choosing

Every time we select a fluid machine the process can be divided into four major steps. Firstly, the purpose of the device must define, does it work or power machine and what is the working media.

Table 5.1. Fluid machine types

	Work idea	
Medium	Work machine	Power machines
Gas	Fan	Wind turbine
Liquid	Pump	Water turbine

Secondly, the system where it will be applied the head and flow must be determined. Measurements or hydraulic calculation can determine the head. which can be expressed by the following:

$$\rho_{device} \cdot g \cdot H_{device} = 0.5 \cdot \Delta\rho \cdot \Delta c^2 + \Delta\rho \cdot g \cdot \Delta H + \Delta p + \Delta p_{losses} \qquad (5.1)$$

The Δp represents the difference between the inlet and outlet of the fluid machine. The device subscript represents the values at the device. Volume flow has many options to be determined and these are:

given value of a device,
given value from a standard,
calculated value based on physics,
measured value.

Thirdly, with the help of a Cordier diagram a subtype of the device can be chosen. Fourthly, theoretical efficiency and the actual selection can be made. Axial and gas transferring devices are handled together with the centrifugal water pumps. Relevant differences were highlighted in the note.

5.1.Calculations

The pump impeller has a diameter of 300mm, a blade width of 20mm, and an exit blade angle of 40° and it rotates at 500min^{-1}. Under design conditions, water at 20°C flows through the pump at a flow rate of 60m^3·h^{-1}. Flow enters the impeller in a direction normal to the inflow surface. Estimate the required shaft power at the design flow rate.

In a heating system, the necessary heat power is 100kW. The supply water temperature is 80°C, the return temperature is 60°C. The pressure drop in the system is p=1bar. Calculate the flow rate and power of the pump.

Recommended literature for this section:

[1] Fluid Mechanics for Engineers in SI Units David A. Chin 2018

6. Positive displacement machines

Compared to turbomachines, which have rotating components, positive displacement machines operate by displacing fluids within a contained volume. Positive displacement pumps are characterized by their ability to generate high pressures at low flow rates, compared to turbomachines that generate moderate pressures at high flow rates. For example, a positive displacement pump might be capable of generating pressure on the order of 304bar with a flow rate of $21 m^3 \cdot h^{-1}$, whereas a turbomachines pump might be capable of generating pressure on the order of 5.1bar with a flow rate of $72000 m^3 \cdot h^{-1}$.

Positive displacement pumps can theoretically maintain the same flow independently of the discharge pressure (see figure 6.1). Therefore, they are constant flow machines.

It can guarantee a fixed flow because it always retains the same fixed amount of fluid and pushes it forward periodically. It should be noted that constant flow at all pressures is only a theoretical concept. As pressure rises there is a slight increase in leakage inside these pumps which reduces flow, however this difference is not relevant.

Positive displacement pumps should not be made to work with a closed valve on the discharge side. If a volumetric pump works against a closed valve it will continue working until something breaks (the pump or the line). For safety purposes, a relief valve should be used. Some pumps have internal relief valves that open when the safety pressure is overcome, but an external relief valve on the discharge line with a return line to the supply tank is highly recommended.

There are two main positive displacement pumps: reciprocating and rotary.

6.1. Reciprocating pumps

The three most used reciprocating pumps are piston, plunger, and diaphragm. They all work similarly with the piston, plunger or diaphragm moving forward and backwards. When it moves backwards it pulls water into the chamber (the check valves ensure this water comes from the suction side) and when it moves forward it pushes the water out through the discharge (again, the valves choose the flow direction).

One of the biggest problem with reciprocating pumps is that the flow is not steady like in centrifugal pumps. It goes from the maximum value, while the piston moves forward to zero while it moves backwards. In some applications this phenomenon can cause problems; by wasting energy (constantly accelerating/decelerating water) and creating inconvenient vibrations and "water hammers".

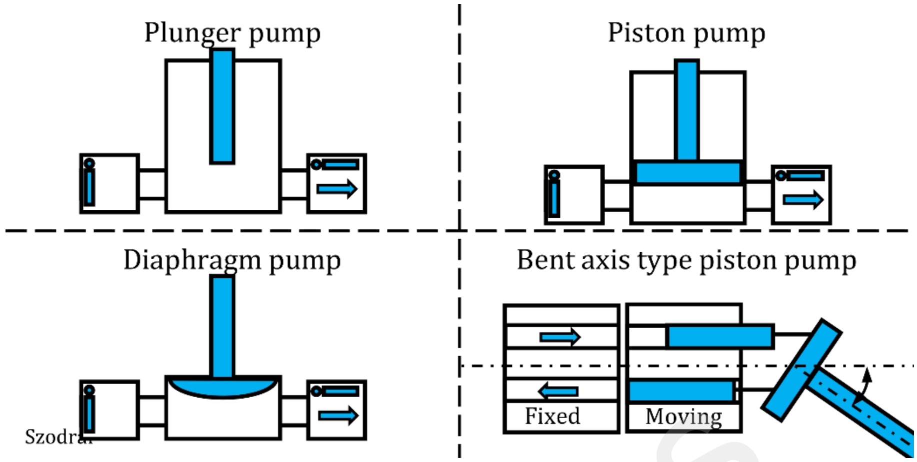

Figure 6.1. Reciprocating pumps

To increase the steadiness of the flow, more than one piston or plunger should be used. Pumps must work unsynchronized and parallel. Pumps with only one chamber are called simplex, others are duplex, triplex, or quad (two, three or four chambers respectively).

A piston can be a simple axle which called plunger pump (see figure 6.1), in this working chamber only a portion of it works. More effective when a piston is used since it increases its effective volume.

In multiplex (bent axis piston pumps see figure 6.1) pumps cylinder block contains several pistons arranged along a circle. The piston rods are connected to the drive shaft flange by a ball and socket joints. The pistons are forced in and out of their bores as the distance between the driveshaft flange and cylinder block changes. A universal joint connects the cylinder block to the drive shaft to provide alignment and positive drive. This type of pump can also be designed to have a variable displacement capability. The maximum swash plate angle is limited to 17.5° by construction.

For a simplex pump, the volume flow:

$$\dot{V}_{piston} = 0.25 \cdot \pi \cdot D_{piston}{}^{2} \cdot L \cdot n \cdot RPM \tag{6.1}$$

where L is the piston stroke, D_{piston} is piston diameter, n is the number of pistons, RPM is the periodicity of the piston. In case of a plunger pump the axle diameter must be used instead of the piston diameter.

For bent axis, piston pumps with an amount of chamber flow rate vary with the offset angle θ. There is no flow when the cylinder block centreline is parallel to the drive shaft centreline (offset angle is 0°). The total fluid flow per stroke can be given as:

$$\dot{V}_{bent\ axis\ piston} = 0.25 \cdot \pi \cdot {D_{piston}}^2 \cdot L \cdot n \cdot RPM \cdot tan\Theta \tag{6.2}$$

$$tan\Theta = L \cdot {D_{piston}}^{-1} \tag{6.3}$$

The diaphragm pump uses a flexible membrane called diaphragm (typically made of rubber, thermoplastic or Teflon) to displace volumes of fluid. The major advantage is that it does not require sealing like the piston or plunger pumps considering the diaphragm seals by itself. These pumps are mostly used to move toxic or hazardous fluids. There are pumps with two diaphragms back-to-back on the same rod so it can pump on both strokes.

Due to the complex geometry of the diaphragm, its volume is calculated with 3D modelling.

6.2.Rotary pumps

Rotary pumps always include rotational movement in the way they work. They have the advantage of being more efficient and creating fewer vibrations than reciprocating pumps. The worst drawback of these pumps is that they need very tight clearance between some moving parts which means they cannot move fast. If rotary pumps work too fast, they are very vulnerable to erosion which would reduce the pump's efficiency. There are a lot of rotary type pumps: gear (external or internal), lobe, vane, and screw.

The gear pump as the name implies works by trapping fluid between the teeth of the gear and w the rotation of the gear pushing the fluid forward. The gears are designed and positioned to not allow the fluid to go backwards. Gear pumps are widely used in car engine oil pumps and various hydraulic power packs. Internal gear pumps are exceptionally versatile (see figure 6.2). While they are often used on thin liquids such as solvents and fuel oil. They excel by efficiently pumping highly viscous. Gear pumps theoretical flow rate is:

$$\dot{V}_{gear} = 0.25 \cdot \pi \cdot (D_C - D_R)^2 \cdot b \cdot RPM \tag{6.4}$$

where D_C cam ring diameter, D_R rotor diameter, b vane width.

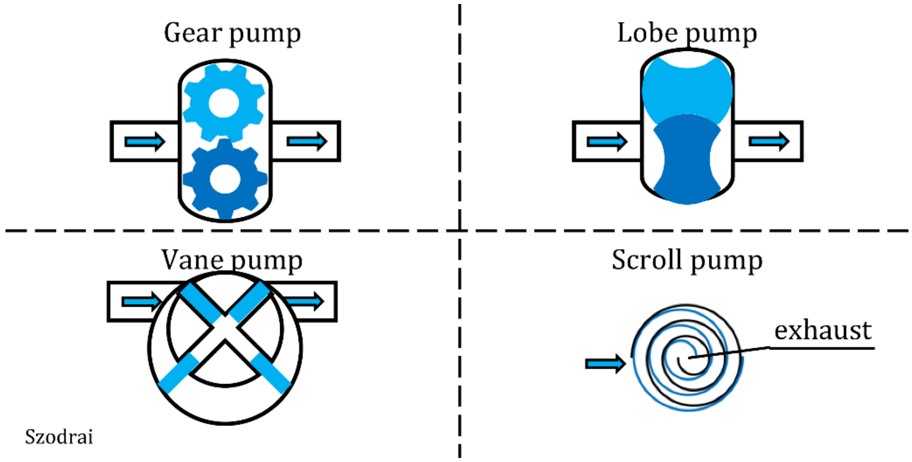

Figure 6.2. External and internal (respectively) gear pump

Lobe pumps (see figure 6.2) can pump solid seeds and small solid particles and viscous liquids. It is favoured due to sanitary qualities, high efficiency, reliability, and good corrosion resistance.

Vane rotary pumps (see figure 6.2) have vanes mounted to a rotor inside a cavity. As the rotor spins, the vanes move to accompany the cavity's wall; this movement is caused by either the springs or centrifugal force.

Flexible vane pumps can handle only the small solids but can create a good vacuum. Sliding vane pumps can run dry for short periods and can handle small amounts of vapour. The vane pumps are known for their dry priming, ease of maintenance, and good suction characteristics. The operating range of these pumps varies from -32°C to 260°C. Vane pumps theoretical flow can be defined if the cam ring D_C and rotor D_R diameters are known:

$$\begin{aligned}\dot{V}_{vane} &= 0.25 \cdot \pi \cdot (D_C - D_R)^2 \cdot b \cdot RPM \\ &= 0.25 \cdot \pi \cdot (D_C - D_R) \cdot (D_C + D_R) \cdot b \cdot RPM \\ &= 0.25 \cdot \pi \cdot 2 \cdot e_{max} \cdot (D_C + D_R) \cdot b \cdot RPM\end{aligned} \tag{6.5}$$

The maximum eccentricity of the rotor and the cam ring is the following:

$$e_{max} = 0.5 \cdot (D_C - D_R) \tag{6.6}$$

A scroll compressor, also known as scroll pump and scroll vacuum pump, uses two interleaved Archimedean spiral-shaped scrolls to pump or compress liquids or gases. One of the scrolls is fixed, while the other orbits eccentrically without rotating, thereby trapping, and compressing pockets of fluid between the scrolls. In heat pumps and refrigerators this is a preferred type of compressor. A flowrate of the lobe and scroll pump similarly to the diaphragm pump is difficult to define thus 3D modelling is needed.

Screw pumps have typically two or more screws although the first screw pump only had one screw. These screws spin in different directions trapping the fluid between the teeth and forcing it to move. They are suited for fuel-injection, oil burners, boosting, hydraulics, fuel, lubrication and so on. Screw pumps theoretical flow can be expressed if the pitch of the stator screw surface (m) is known.

$$\dot{V}_{screw} = 0.25 \cdot \pi \cdot {D_{screw}}^2 \cdot m \cdot RPM \tag{6.7}$$

There are 2 types of screw compressors: the oil-free type and the oil-supply type. The rotor shape is the same, but the position where they mesh is different: the oil-free type rotates without contact, the oil-supply type rotates with contact. One feature of the oil-free screw compressor is that it generates clean compressed air that does not contain oil. This type is widely used in industries where oil cannot be mixed with air and also as an air source for instrumentation in general industries. Also, in the oil-free screw compressor, the rotors rotate without contact.

6.3.Overall efficiencies

Volumetric pump losses occur due to the difference between the theoretical and actual flow rate differences, leakages, and drive losses.

Volumetric efficiency ($\eta_{volumetric}$) is the ratio of the actual flow rate of the pump and the theoretical flow rate of the pump. This is expressed as follows:

$$\eta_{volumetric} = \dot{V}_{actual} \cdot {\dot{V}_{theoretical}}^{-1} \tag{6.8}$$

It indicates the amount of leakage that takes place within the pump. This is due to manufacturing tolerances and flexing of the pump casing under-designed pressure operating conditions.

Mechanical efficiency ($\eta_{mecahnical}$) is the ratio of the pump output power, assuming no leakage, and the actual power delivered to the pump:

$$\eta_{mechanical} = P_{no\ leakages} \cdot {P_{actual}}^{-1} = M_{theoretical} \cdot {M_{actual}}^{-1} \tag{6.9}$$

Mechanical efficiency indicates the amount of energy losses that occur for reasons other than leakage. This includes friction in bearings and between mating parts. This includes energy losses due to fluid turbulence. Mechanical efficiencies are about 90%–95%. We also have the relation:

$$\eta_{mechanical} = p \cdot \dot{V}_{theoretical} \cdot (M_{actual} \cdot RPM)^{-1} \tag{6.10}$$

Where p is the pump discharge pressure, $V_{theoretical}$ is the theoretical flow rate of the pump, M_{actual} [N·m] is the actual torque delivered to the pump and RPM is the periodicity of the pump.

The theoretical torque required to operate the pump is the torque that would be required if there were no leakage.

$$M_{theoretical} = \eta_{mechanical} \cdot M_{actual} = p \cdot \dot{V}_{theoretical} \cdot RPM^{-1} \qquad (6.11)$$

Overall efficiency ($\eta_{overall}$) is defined as the ratio of actual power delivered by the pump to actual power delivered to the pump.

$$\begin{aligned} \eta_{overall} &= P_{actual,\ delivered\ by\ the\ pump} \cdot P_{actual,\ delivered\ to\ the\ pump}{}^{-1} \\ &= \eta_{mechanical} \cdot \eta_{Volumetric} \\ &= p \cdot \dot{V}_{actual} \cdot (T_{actual} \cdot RPM)^{-1} \end{aligned} \qquad (6.12)$$

$$\eta_{volumetric} = \dot{V}_{actual} \cdot \dot{V}_{theoretical}{}^{-1} \qquad (6.13)$$

$$\eta_{mechanical} = \eta_{overall} \cdot \eta_{volumetric}{}^{-1} \qquad (6.14)$$

6.4.Calculation

A vane pump has a rotor diameter of 60mm, a cam ring diameter of 90mm and a vane width of 50mm. What must be eccentricity for it to have a volumetric displacement of 115cm^3? Calculate the mechanical efficiency where the overall efficiency is 80% at 300min^{-1} and the actual flow is 120m^3·h^{-1}.

A gear pump has an outside diameter of 80mm, inside diameter of 60mm and a width of 25mm. If the actual pump flow is 1800min^{-1} and the flowrate is 6.3m^3·h^{-1}, what is the volumetric efficiency?

A pump has a displacement volume of 98.4cm^3. It delivers 5.5m^3·h^{-1} of oil at 1000min^{-1} and 70bar. If the prime mover input torque is 650Nm. What is the overall efficiency of the pump? What theoretical torque is required to operate the pump?

Recommended literature for this section:

[1] Fluid Mechanics for Engineers in SI Units David A. Chin, 2018

7. Wind and water Turbines

Turbines convert the kinetic energy of the fluid flow to rotating an axle thus generating energy. We would like to know what is the efficiency of a turbine and what depends on it. First, we examine open turbines. Open turbines mean that the device moving part is not placed in a tunnel, for instance, because wind turbines are open turbines.

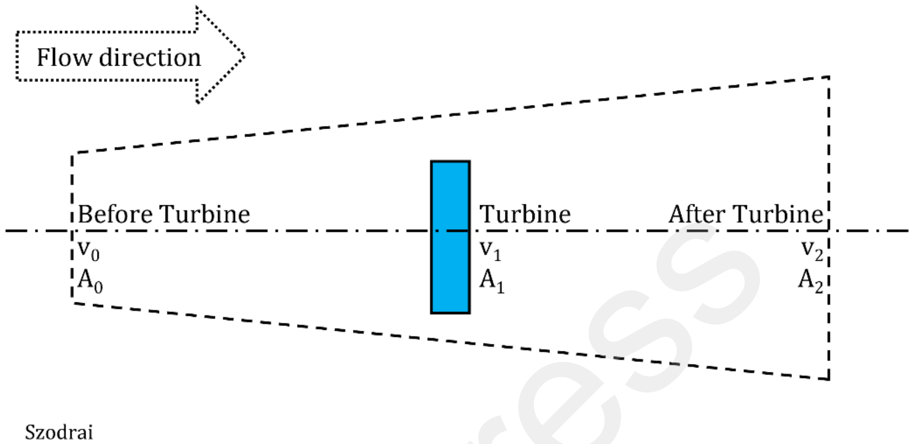

Figure 7.1. Concept of open turbines

The open turbines decrease the flow velocity while increases the examined cross-section area. (In closed turbines the cross-section is constant or defined.) This work idea is also called the "Lift machines" work idea.

Force on the blade is the inertia change:

$$F = \dot{m} \cdot v_0 - \dot{m} \cdot v_2 \tag{7.1}$$

Power of the turbine (P_T) is the inertia change at a given velocity, this velocity (v_1) is the velocity of the turbine blade.

$$P_T = F \cdot v_1 = \dot{m} \cdot (v_0 - v_2) \cdot v_1 \tag{7.2}$$

The difference of the kinetic power (P_W) tells how much power we have harnessed by the device.

$$P_W = 0.5 \cdot \dot{m} \cdot ({v_0}^2 - {v_2}^2) \tag{7.3}$$

We can assume that the power produced on the turbine is equal to the decreased kinetic power of the flow around it.

$$P_T = P_W \tag{7.4}$$

7.1.Power coefficient

Now our goal is to see, what the relationship is between the inlet and the blade velocity and how is it influenced by the power. We assume that the turbine slows down the v_0 flow velocity to v_2:

$$P_T = P_W \tag{7.5}$$

$$\dot{m} \cdot (v_0 - v_2) \cdot v_1 = 0.5 \cdot \dot{m} \cdot \left({v_0}^2 - {v_2}^2\right) \tag{7.6}$$

Inlet velocity can be expressed:

$$v_1 = 0.5 \cdot (v_0 + v_2) \tag{7.7}$$

Outlet velocity can be expressed:

$$v_2 = 2 \cdot v_1 - v_0 \tag{7.8}$$

We introduce the mass flow (due to the conservation of mass flow it is equal at every point), and we also want the equation to depend only by the inlet velocity.

$$\dot{m} = \rho \cdot A_1 \cdot v_1 \tag{7.9}$$

$$P_T = \rho \cdot A_1 \cdot v_1 \cdot (v_0 - v_2) \cdot v_1 \tag{7.10}$$

$$P_T = \rho \cdot A_1 \cdot {v_1}^2 \cdot (v_0 - v_2) \tag{7.11}$$

$$P_T = \rho \cdot A_1 \cdot {v_1}^2 \cdot \left(v_0 - (2 \cdot v_1 + v_0)\right) \tag{7.12}$$

For simplification purposes, we introduce the velocity induction factor "a" to simplify the equation. Which can be described as:

$$a = (v_0 - v_1) \cdot (v_0)^{-1} \tag{7.13}$$

$$a = (v_0 - v_2) \cdot (2 \cdot v_0)^{-1} \tag{7.14}$$

Inlet velocity can be expressed:

$$v_1 = v_0 \cdot (1 - a) \tag{7.15}$$

Outlet velocity can be expressed:

$$v_2 = v_0 - 2 \cdot v_0 \cdot a \tag{7.16}$$

The power harnessed by the turbine will be:

$$P_T = \rho \cdot A_1 \cdot {v_1}^2 \cdot \left(v_0 - (2 \cdot + v_0)\right) \tag{7.17}$$

$$P_T = 2 \cdot \rho \cdot A_1 \cdot {v_1}^2 \cdot (v_0 - v_1) \tag{7.18}$$

$$P_T = 2 \cdot \rho \cdot A_0 \cdot \left(v_0 \cdot (1 - a)\right)^2 \cdot \left(v_0 - v_0 \cdot (1 - a)\right) \tag{7.19}$$

$$P_T = 2 \cdot \rho \cdot A_0 \cdot {v_0}^3 \cdot (1 - a)^2 \cdot \left(1 - (1 - a)\right) \tag{7.20}$$

$$P_T = 2 \cdot \rho \cdot A_0 \cdot {v_0}^3 \cdot (1-a)^2 \cdot (1-1+a) \tag{7.21}$$

$$P_T = 2 \cdot \rho \cdot A_0 \cdot {v_0}^3 \cdot (1-a)^2 \cdot a \tag{7.22}$$

The wind kinetic power is:

$$P_0 = 0.5 \cdot \rho \cdot A_0 \cdot {v_0}^3 \tag{7.23}$$

The ratio of harnessed power and wind power is the power coefficient (Cp).

$$C_P = produced\ power \cdot kinetic\ power^{-1} = 4 \cdot a \cdot (1-a)^2 \tag{7.24}$$

This equation tells us how does the C_p changes in the function of the induction factor at open turbines.

7.2.Torque coefficient

For turbomachines it is important to know how much torque is required to spin the rotor or if it is a turbine, we would like to know how much torque we can get out from it. First, we calculate the load force by the flow.

$$F_{max} = 0.5 \cdot \rho \cdot A_0 \cdot v_0^2 \tag{7.25}$$

If we multiply the force with the rotor radius, we will get the maximum torque.

$$T_{max} = F_{max} \cdot R \tag{7.26}$$

$$T_{max} = 0.5 \cdot \rho \cdot A_0 \cdot v_0^2 \cdot R \tag{7.27}$$

The rate of the actual and maximum torque gives the torque coefficients.

$$T = T_{max} \cdot C_T \tag{7.28}$$

$$C_T = torque\ on\ the\ axis \cdot maximum\ torque\ on\ the\ axis^{-1} \tag{7.29}$$

The tip speed ratio is the best way to describe the work idea of an open turbine. This ratio is the rate of the blade velocity and fluid velocity. To put it in simple terms this coefficient tells us how fast will spin (RPM; min^{-1}) the turbine rotor at a given flow velocity.

$$\lambda = blade\ velocity \cdot fluid\ velocity^{-1} = R \cdot RPM \cdot {v_0}^{-1} = v_t \cdot {v_0}^{-1} \rightarrow R = \lambda \cdot v_0 \cdot RPM^{-1} \tag{7.30}$$

$$T_{max} = 0.5 \cdot \rho \cdot A_0 \cdot v_0^2 \cdot \lambda \cdot v_0 \cdot RPM^{-1} = P_0 \cdot \lambda \cdot RPM^{-1} \rightarrow P_0 = T_{max} \cdot RPM \cdot \lambda^{-1} \tag{7.31}$$

Power on the axis is the torque on the axis at a given RPM.

$$P_T = T \cdot RPM \tag{7.32}$$

$$C_P = P_T \cdot {P_0}^{-1} = T_{max} \cdot C_T \cdot RPM \cdot (T_{max} \cdot RPM \cdot \lambda^{-1})^{-1} = C_T \cdot \lambda^{-1} \tag{7.33}$$

Finally, we can see that the turbines torque and power coefficients have a connection which is the tip speed ratio.

$$C_T = C_P \cdot \lambda^{-1} \tag{7.34}$$

If we draw this function, we will be able to examine the idea and limit of open turbines. We can see in figure 7.2 that when the tip speed ratio is low (turbine spinning slowly) the torque is high. When the available torque is decreasing the RPM increases. In practice, if the λ=0-4 we are talking about drag machines that will provide large torque that can be used to power drives. If the λ=4-8 we are talking about lift machines that are excellent for electric power generation.

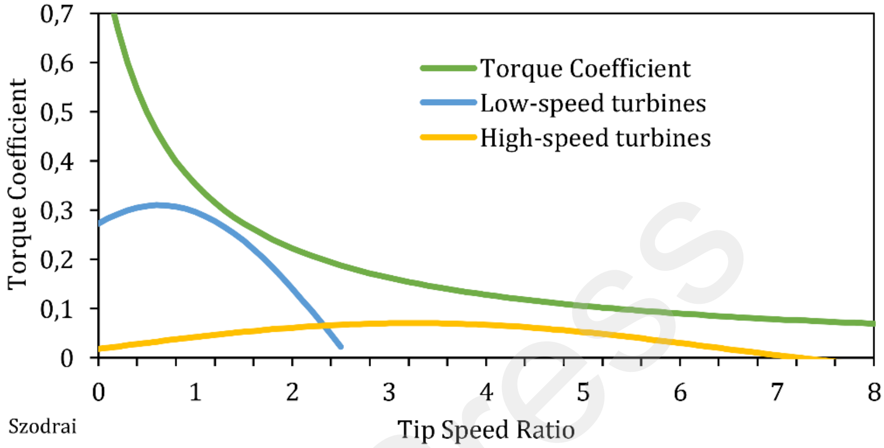

Figure 7.2. The function of the C_T and λ

7.3.Drag machines

Drag machines solely utilize drag when the kinetic energy of the flow is harnessed. This means that the blade velocity cannot be higher than the flow velocity. These work ideas are used to describe the impulse turbines or drag machines, where the blade and outlet velocity is the same.

$$v_1 = v_2 \tag{7.35}$$

Force on the blade can be expressed:

$$F_A = \rho \cdot A_1 \cdot (v_0 - v_1)^2 \tag{7.36}$$

In the optimal case the induction factor maximum is 1/3, thus:

$$v_t = 1/3 \cdot v_0 \tag{7.37}$$

$$C_P = 4/27 \cdot C_D \tag{7.38}$$

The limit of the drag machine performance is the drag coefficient which is a highly geometry dependent factor.

Impulse turbines, regardless of the phase of the fluid, operate at high torque. The rotating part of the device is a bucket that's main purpose is to generate as much drag as possible. The two well-known types of these turbines are the Savonius wind turbine and the Pelton water turbine. The Savonius wind turbine can generate relatively high torque even at low wind speed; however, its tips speed ratio is too low to produce sufficient amount of RPM to generate electricity.

On the other hand, the Pelton turbine can be the main element of a power plant, however, it requires a large amount of head pressure. What it means is that to build a Pelton turbine power plant a waterfall must be sacrificed. With a large amount of head pressure, the Pelton turbines require low flow rate, subsequently making this turbine excellent to store energy in the volume of the water.

7.4.Reaction turbines

Lifting machines do not only utilize drag, but the moving fluid also generates a lift force on the blade that leads to a higher blade velocity. Reaction turbines such as the three-bladed wind turbines or Darreius wind turbines are suitable to generate electricity. Their moving parts are wing-shaped and can achieve high RPM during its operation.

Reaction turbines are also suitable to generate electricity from the water flow. The reaction turbines require a large amount of flow and small head pressure. It must be mentioned that if the head is too low cavitation can occur. These turbines such as the Kaplan turbines can be used in rivers where only a small height difference occurs. In Kaplan turbine flow is entered through a spiral casing. Decreased area of the casing ensures that flow is entered to the central portion at uniform velocity throughout the perimeter. Water after crossing the guide vanes passes over the runner. Finally, it leaves through a draft tube.

Power demand may fluctuate over time. Controlling the water flow rate is the most efficient way to meet the required power demand. A governing mechanism, which controls the position of guide vanes is used to control the water flow rate. When power demand is high guide vanes are opened and when power demand is low guide vanes are closed. By varying flow conditions, relative velocity will change drastically. Kaplan turbine blades are adjustable. When the flow rate is high relative velocity of flow will be more axial, blades should pitch vertically. If the flow rate is low relative velocity of flow is more tangential, blades are pitched in a tangential direction.

Francis turbines are the most preferred hydraulic turbines. They are the most reliable hydroelectric power stations; they can work efficiently under a wide range of operating conditions. In runner, water enters radially and leaves axially. Francis turbine is not a pure reaction turbine; a portion of force comes from impulse action also.

7.5.Calculations

Calculate how much power does a three-bladed wind turbine generate when the average wind speed is $12m\cdot s^{-1}$ and the diameter of the turbine is 30m. It is assumed that the tip speed ratio of the turbine is 7. Also calculate the induction factor, the torque, and the RPM of the turbine.

Calculate how much energy can be generated with a Pelton turbine if the average effective height is 200m the flow rate is $1000m^3\cdot h^{-1}$. The water reservoir can only store $120000m^3$ of water.

Calculate the nominal power of a Kaplan turbine if it harnesses the kinetic energy from a 30m wide 5m meter deep water where a 5m jump occurs. The average velocity of the river is $3m\cdot s^{-1}$.

Recommended literature for this section:

[1] Fluid Mechanics for Engineers in SI Units David A. Chin, 2018

[2] Renewable energy resources John Twidell, Tony Wier Routledge, 2015

8. Thermal machines introduction

Thermal machines main purpose is to generate heat or harness the enthalpy of a fluid to generate useful power. When a thermal machine is mentioned usually, we think of a boiler that produces high-temperature water or a more advanced device like a heat pump that effectively increases the temperature level by harnessing a heat source. Also, thermal devices are used to categorize cooling devices such as heat pumps, cooling towers, dry coolers, absorption chillers etc. For these devices input energy is required to operate. These can be called as active devices, and heat exchangers, heatsinks can be called as passive. Thermal machines are used to harness only sensible heat, but nowadays the latent is also harnessed which makes these devices more effective.

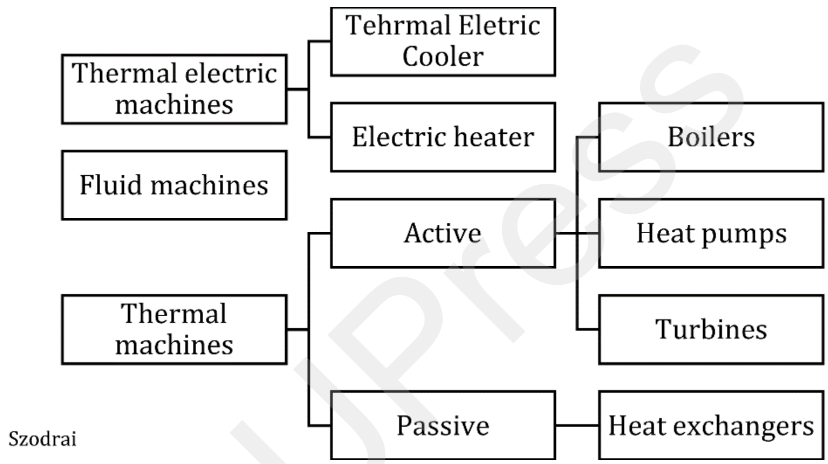

Figure 8.1. Thermal machine types

In the lecture note, the electric machines are not discussed, yet it is noted that electric heaters and thermal electric coolers are inefficient thermal machines.

Recommended literature for this section:

[3] Introduction to Thermal Systems Engineering: Thermodynamics, Fluid Mechanics, and Heat Transfer Michael J. Moran, 2002

9. Heat exchangers

Heat exchangers are passive devices that transfer energy between fluids at different temperatures by heat transfer modes such as heat conduction or convection. The main types can be seen in figure 9.1.

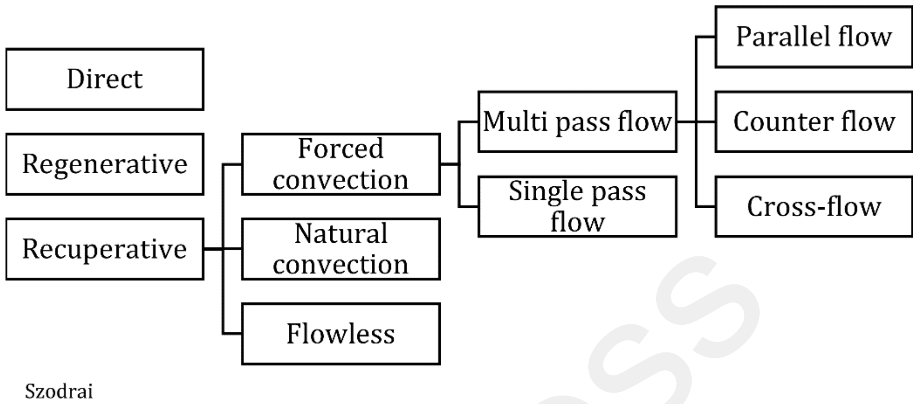

Figure 9.1. Types of heat exchangers

9.1. Direct and regenerative heat exchangers

One of the simplest type of heat exchanger is a T valve or a motoric mixing valve. (Side note: the regulating motor does need active power however heat generation does not.) When two-fluid contacts with different temperatures a new mixed temperature can be calculated in the rate of the fluid mass flows. In the direct heat exchangers, the pressure loss is low.

$$T_{mixed} = (T_A \cdot m_A + T_B \cdot m_B) \cdot (m_A + m_B)^{-1} \tag{9.1}$$

There are heat exchangers where the two-fluid medium could also change latent heat, these devices called regenerative heat exchangers. The ratios of the enthalpies can calculate the efficiency of these devices.

$$\eta_{heating} = \Delta h_{Fresh} \cdot \Delta h_{max}{}^{-1} = \left(h_{hot,outlet} - h_{hot,inlet}\right) \cdot \left(h_{cold,inlet} - h_{hot,inlet}\right)^{-1} \tag{9.2}$$

Regenerative heat exchangers are used in ventilation systems where the heat of the exhaust moist air is harnessed to reduce the heat demand. The idea is that it has a porous media that is rotating between the fresh and exhaust ducts which allows the latent heat transfer.

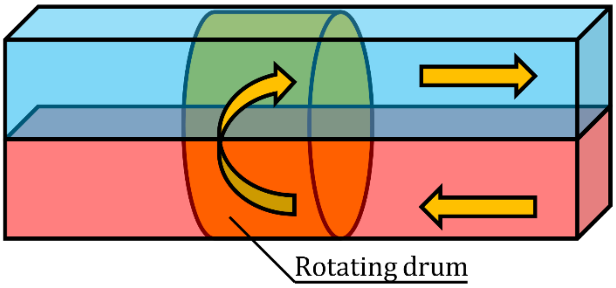

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Figure 9.2. Schematic of a direct and regenerative heat exchanger

9.2. Recuperative heat exchangers

Recuperative heat exchangers only allow sensible heat transfer, since the two different temperature fluids are separated from each other with a wall. The subcategories of this section are distinguished by flow is forced and if in what direction corresponding to another.

When a warm and cold steam pipe comes into contact the warmer gives its heat to the colder one, thus condensation occurs in the warmer pipe. Flow generates in these pipes due to condensation liquid drawing and new steam replaces it. These types are called flow-less heat exchangers since it does not have any fluid machine inside of the pipe system.

Natural convection could also generate flow if the temperature gradient is high enough. These types of heat exchangers are commonly used in domestic hot water buffer tanks or in solar collector systems.

If any of the side fluid machines used (forced flow is present) single- or multi-pass category can be used. Single-pass heat exchangers have two different types, depending on the latent heat generation: condenser or evaporator.

Usually, the multi refers to two-pass, where the flow direction of the two-fluid can be:

matching, parallel flow heat exchangers,
opposite counter flow heat exchangers,
and perpendicular, cross-flow heat exchangers.

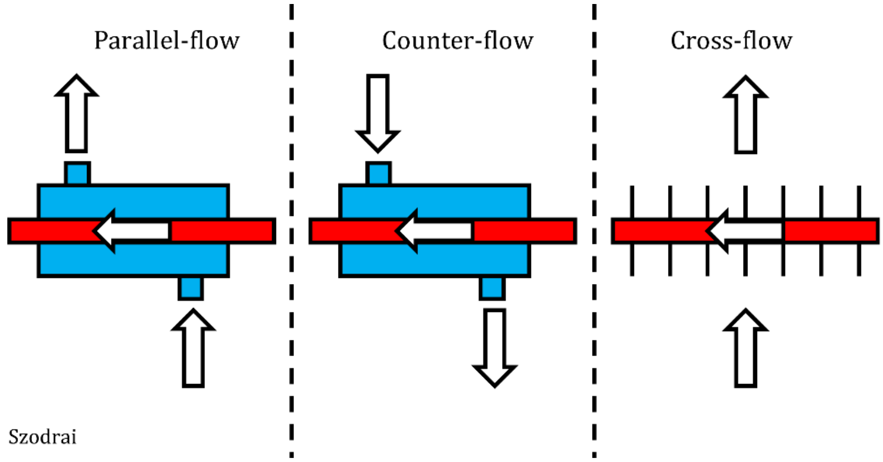

Figure 9.3 Recuperative heat exchanger types

9.2.1.Parallel flow heat exchangers

In theory a parallel flow heat exchanger infinitely large heat transfer surface would need to transfer the total amount of heat from the primer fluid to the secondary fluid. The temperature distribution of a heat sink can describe this phenome, where on one side there is heat gain and on the other side there is heat loss. The schematic of the heat exchanger and the temperatures can be seen in figure 9.4. For thermal machines, the source side is the primer side, and the other side of the heat exchanger is the secondary side.

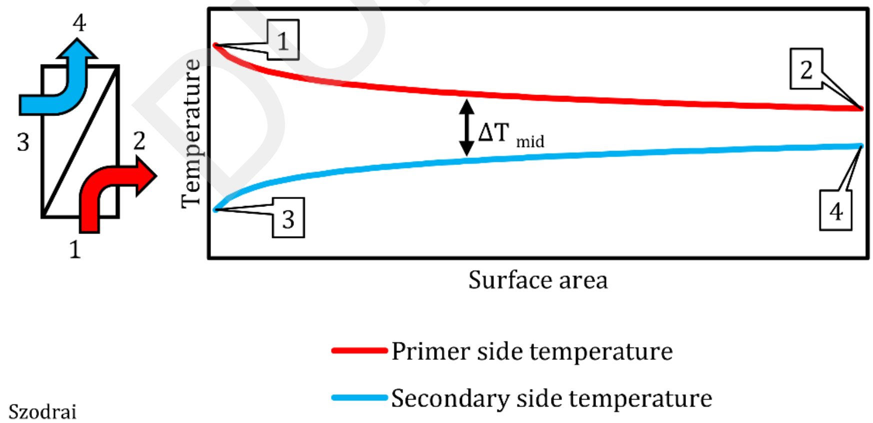

Figure 9.4. Parallel flow heat exchangers

9.2.2.Counterflow heat exchangers

In practice, counter flow heat exchangers are more preferred since it requires less heat transfer surface area to deliver the same amount of heat. Thanks to this arrangement it is possible to reach higher secondary outlet temperature than the primary outlet temperature.

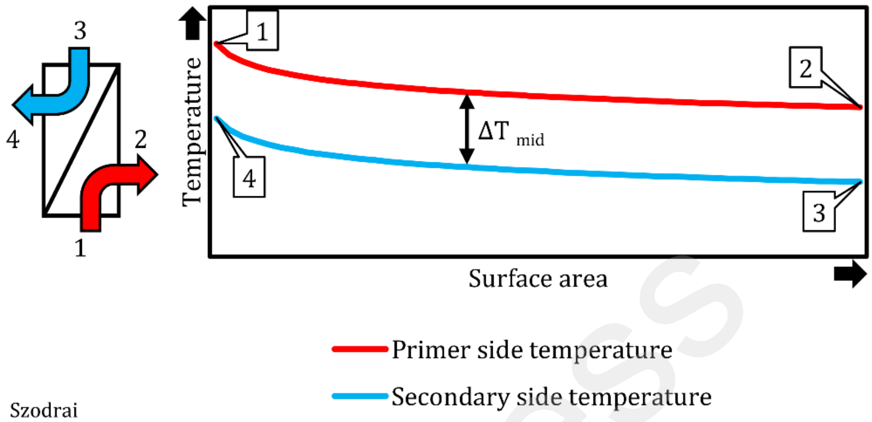

Figure 9.5. Counter flow heat exchangers

9.2.3.Cross-flow heat exchangers

Crossflow heat exchangers mostly used for ventilation systems where with the crossing of two air ducts a volume efficient heat exchange can be done. It is well known that the construction is not airtight thus it is limited to comfort applications. Another version of this device is how air condition units work. In those devices the water flows perpendicular to the airflow.

They are most efficient between the parallel and counter flow heat exchangers. To evaluate its performance, theoretical heat load-dependent temperature difference or temperature coefficient (Λ) is expressed by the following equation:

$$\Lambda = \Delta T_T \cdot \Delta T_M^{-1} \tag{9.3}$$

where a theoretical temperature difference is ΔT_T and measured temperature difference is ΔT_M on the secondary side. The theoretical temperature difference is the following:

$$\Delta T_T = \dot{Q}_{source} \cdot (c_s \cdot \dot{m}_s)^{-1} \tag{9.4}$$

where Q_{source} is the heat from the source (primer side), c_s and m_s is the specific heat and mass flow of secondary side fluid, respectively. Lower the temperature coefficient (Λ) is better. [4]

9.3.The logarithmic mean temperature difference

The heat balance for a heat exchanger can be described if the heat exchanger is perfectly insulated.

$$\Delta\dot{Q} = \underbrace{-c_p \cdot \dot{m}_p \cdot \Delta T_p}_{Pimer} = \underbrace{U \cdot LMTD \cdot \Delta A}_{Transfer} = \underbrace{c_s \cdot \dot{m}_s \cdot \Delta T_s}_{Secondary} \tag{9.5}$$

where c is the specific heat of the fluids, m is the mass flows, U is the separating plates heat transfer value and A is the surface area. LMTD is the difference between the average primer and the seconder plate temperatures. The LMTD is the logarithmic temperature difference which gives a good estimation between two surface temperatures when the surface temperature is not constant along the surface.

Now let us express the LMTD. Firstly, the heat transfer can be described by the following equation separately for the primer and secondary sides.

$$-c_p \cdot \dot{m}_p \cdot \Delta T_p = U \cdot LMTD \cdot \Delta A \tag{9.6}$$

$$c_s \cdot \dot{m}_s \cdot \Delta T_s = U \cdot LMTD \cdot \Delta A \tag{9.7}$$

Temperature difference change at both sides can change in the function of the surface area:

$$-dT_p = U \cdot LMTD \cdot dA \cdot \left(c_p \cdot \dot{m}_p\right)^{-1} \tag{9.8}$$

$$dT_s = U \cdot LMTD \cdot dA \cdot (c_s \cdot \dot{m}_s)^{-1} \tag{9.9}$$

The temperature difference between the primer and secondary fluid at a given magnitude of the surface area is:

$$d\left(T_p - T_s\right) = -U \cdot LMTD \cdot dA \cdot \left(\left(c_p \cdot \dot{m}_p\right)^{-1} + (c_s \cdot \dot{m}_s)^{-1}\right)^{-1} \tag{9.10}$$

For the sake of simplicity assume ΔT is T_p-T_s and $\left(\left(c_p \cdot \dot{m}_p\right)^{-1} + (c_s \cdot \dot{m}_s)^{-1}\right)^{-1}$ is B, thus our equation becomes:

$$d\Delta T = -U \cdot LMTD \cdot dA \cdot B \tag{9.11}$$

$$d\Delta T \cdot LMTD^{-1} = -U \cdot dA \cdot B \tag{9.12}$$

Integrating this expression between the initial temperature difference ΔT_0 and the corresponding to A surface area ΔT.

$$\int_{\Delta T_0}^{\Delta T} d\Delta T \cdot LMTD^{-1} = -U \cdot B \cdot \int_0^A dA \tag{9.13}$$

$$ln\left(\Delta T \cdot \Delta T_0{}^{-1}\right) = -U \cdot A \cdot B \tag{9.14}$$

And the following can be expressed:

$$\Delta T \cdot \varDelta T_0{}^{-1} = e^{-U \cdot A \cdot B} \tag{9.15}$$

$$\Delta T = \varDelta T_0 \cdot e^{-U \cdot A \cdot B} \tag{9.16}$$

$$B = -ln\left(\Delta T \cdot \varDelta T_0{}^{-1}\right) \cdot (U \cdot A)^{-1} \tag{9.17}$$

The general form of heat transfer for any A surface will be:

$$d\dot{Q} = U \cdot \varDelta T_0 \cdot e^{-U \cdot A \cdot B} \cdot dA \tag{9.18}$$

By integrating the function between A surface, the new form will be:

$$\Delta\dot{Q} = U \cdot \varDelta T_0 \cdot (-B \cdot U)^{-1} \cdot (e^{-U \cdot A \cdot B} - 1) = \varDelta T_0 \cdot B^{-1} \cdot (1 - e^{-U \cdot A \cdot B}) \tag{9.19}$$

Replacing parameters with the expressed equation, the heat transfer will be:

$$\begin{aligned} \Delta\dot{Q} &= \varDelta T_0 \cdot \left(-ln\left(\Delta T \cdot \varDelta T_0{}^{-1}\right) \cdot (U \cdot A)^{-1}\right)^{-1} \cdot \left(1 - \Delta T \cdot \varDelta T_0{}^{-1}\right) \\ &= U \cdot A \cdot \varDelta T_0 \cdot \left(ln\left(\Delta T \cdot \varDelta T_0{}^{-1}\right)\right)^{-1} \cdot \left(\Delta T \cdot \varDelta T_0{}^{-1} - 1\right) \end{aligned} \tag{9.20}$$

Dividing the heat transfer with the heat transfer coefficient and surface area the other side of the equation becomes the LMTD:

$$\Delta\dot{Q} \cdot (U \cdot A)^{-1} = (\Delta T - \varDelta T_0) \cdot \left(ln\left(\Delta T \cdot \varDelta T_0{}^{-1}\right)\right)^{-1} \tag{9.21}$$

$$LMTD = (\Delta T - \Delta T_0) \cdot ln\left(\Delta T \cdot \Delta T_0{}^{-1}\right)^{-1} \tag{9.22}$$

In engineering logarithmic mean temperature difference always gives a more precise value when the temperature changes along the surface.

9.4.Shell and tube heat exchangers

Shell and tube heat exchangers are the first type of heat exchangers that are used. The simple construction allows easy maintenance. This feature is crucial for steam systems or where solid deposition fouling can occur, which will lead to unwanted hydraulic loss increase.

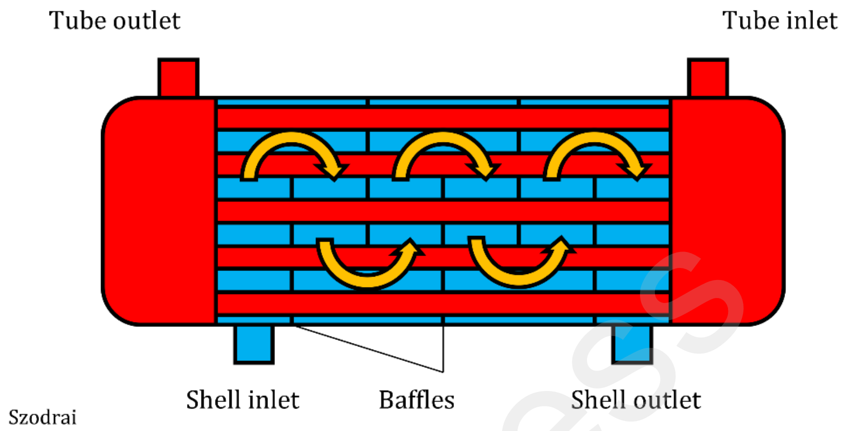

Figure 9.6. Shell and tube heat exchanger

This device has large volume requirements, thus in residential areas, it is not recommended to use. Due to complex flows its efficiency can only be determined by numerical simulation or measurements.

The working idea of this apparatus is to supply the primer heat in the tubes to transfer its heat to the secondary shell side. The shell side has a large volume and low hydraulic resistance. If a heat exchanger has a low-pressure loss on either side, it means that the fluid can easily flow through which means the temperature drop is small. To enhance its performance baffles are used, which extends the fluid route in the shell side.

9.5.Plate heat exchangers

A plate heat exchanger is a state-of-the-art device. It has high thermal efficiency and small volume. It is made from plates that are sealed in a way that on one side the primary and on the other side the secondary fluid flows. The hydraulic loss is usually high; however, this will be the largest loss in a hydraulic system.

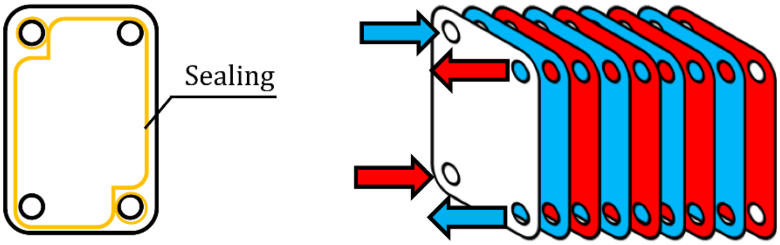

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Figure 9.7. Plate heat exchanger

9.6.Calculations

In a heat exchanger, the temperature difference is 120°C to 80°C on the primer side and 10°C to 60°C on the secondary side. Which one is better from the economic point of view Is it the cross-flow or the parallel flow when both features, the heat transfer and the heat quantity stay the same?

We are producing domestic hot water with a parallel flow heat exchanger that has a $2m^2$ surface area. On the primer side, the difference is 20°C at $72kg \cdot min^{-1}$ mass flow. How much domestic hot water can produce from 10°C cold water, if there is no loss to the environment. How much is the heat transfer value of the exchanger?

In wintertime to heat, we are using an air/fluid heat exchanger with a mass flow of $275kg \cdot h^{-1}$. The outside temperature is -15°C the indoor temperature is 20°C. The air specific heat is $1kJ \cdot kg^{-1} \cdot K^{-1}$ and the waters $4.18kJ \cdot kg^{-1} \cdot K^{-1}$. The fluid mass flow is $110kg \cdot h^{-1}$ with a 90°C temperature. Determine the heat transfer surface for parallel and counter flow if the heat transfer value is $15W \cdot m^{-2} \cdot K^{-1}$. Heat losses are disregarded.

Recommended literature for this section:

[3] Introduction to Thermal Systems Engineering: Thermodynamics, Fluid Mechanics, and Heat Transfer Michael J. Moran, 2002

[5] Heat exchanger design handbook Thulukkanam Kuppan, 2013

10. Heat pumps

A heat pump is a device that can transfer heat from a lower state to a higher state by using work. The heat source can change by the demand. One part of the heat pump cools and the other one heats. In that process sensible heat is used, as for heat pumps latent heat is applied. The ideal work principle is based on the reverse Carnot cycle: isothermal evaporation, isentropic compression, isothermal condensation, and isentropic expansion. However, to produce an isentropic expansion a turbine is needed instead of an expansion valve. This means instead of an isentropic expansion a well-controlled isenthalpic process will occur in a realistic cycle. Usually, the turbine is not preferred since the useful power that it could provide is small. A heat pump has five main parts: condenser, evaporator, compressor, expansion valve and the fluid that flows in the device.

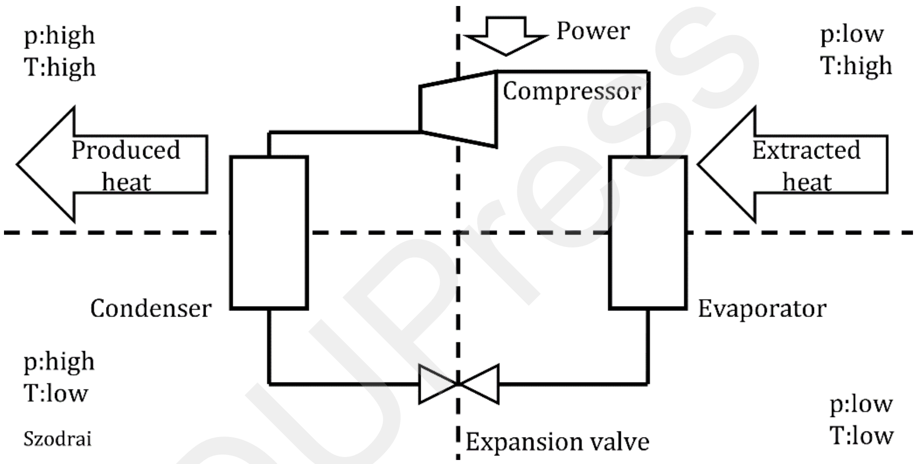

Figure 10.1.Schematic of a heat pump

10.1. Heat pump fluid

The fluid in the heat pump determines the efficiency of the pump. To maximize the latent heat transfer capability, evaporation pressure should be relatively high while the condensation pressure should be low. The reason is to minimize the pressure difference or in other words the efficiency of the compressor. If the pressure difference requirement is little and the latent heat capacity of the fluid is large the more efficient the cycle will be. Must be also noted that since the sensible heat does not take a major role in the process the specific heat capacity can be low. Applied fluids can be divided into three categories: f inert gases, toxic corrosive and able to explode, third and final category that can explode when contacted with air. The most notable fluids are nitrogen, carbon dioxide and hydrocarbons. Till 2030 a special blend called R410A can be used.

10.2. Condensator and evaporator

The condenser and evaporator are almost identical single flow heat exchangers. The only difference is the type of the phase when the change occurs in the device. As it was mentioned pressure level must be different, but in some multifunctioning heat pumps with few valves, the roles can be switched. Sizes only change when the designed heat flux is different.

As an example, let us look at an AC unit and assume we need cooling. The internal unit is an evaporator and the external is a condenser. When the two sizes are compared it can be noted that the external unit has a larger size. The reason for that it is easier to dissipate that heat and also that external unit could contain a compressor, expansion valve and a fan that could also enhance the heat flux. On the other hand, the internal unit must be silent, thus it only has a fan and a single flow heat exchanger.

One can see that if our goal is energy-efficiency than less the heat pump works the more efficient, we are. It means that the evaporator prefers low temperature while condenser prefers high temperature. Let us examine these single flow heat exchanger's temperature surface area diagrams.

On the evaporator in the primer side, fluid is cooled down by losing sensible amount of heat and converting it to latent heat at the secondary side, which is presented in figure 10.2. Ideally, the secondary side is isothermal, yet measurements show that a slight temperature difference can occur (it is also true for the condenser).

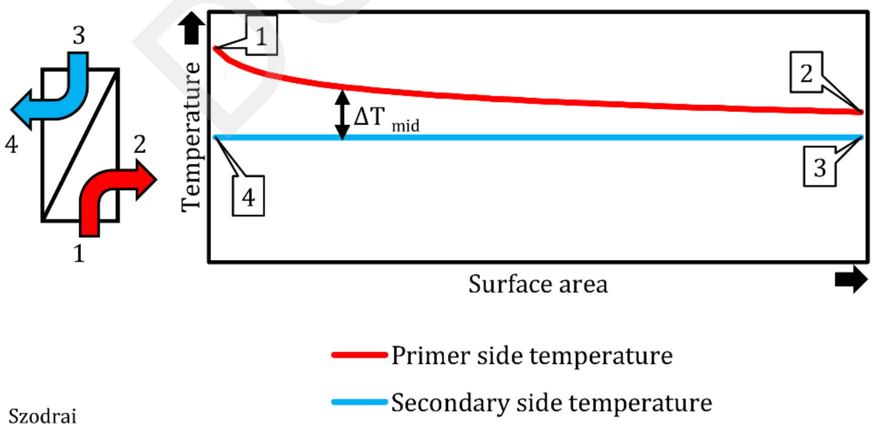

Figure 10.2. Evaporator schematic and temperature surface area diagram

In figure 10.3 we assume that the heat pump is in cooling mode and the primary side is the side of the heat pump (primary and secondary is always relative). In the condenser, the condensation occurs at a constant temperature while it transfers the heat to the secondary side.

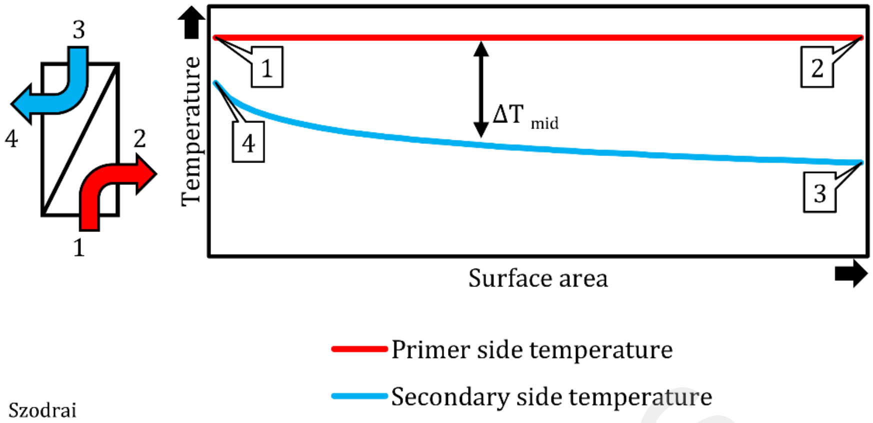

Figure 10.3. Condenser schematic and temperature surface area diagram

10.3. Compressor and expansion valve

The compressor ensures the pressure level for the condensation, while the expansion valve only lets a small amount of fluid in the low-pressure side to make sure evaporation. Since these parts does not have the isothermal restriction, therefore temperature change is relevant, yet this sensible change has a minor effect since the compressor is not able to supply heat load to the fluid. In a large-scale heat pump, the heat from compressors can be used with additional recovery parts. Usually, screw or scroll compressors are used in heat pumps. With the small flow rate and high head, it can easily increase the pressure, yet this efficiency leads to high-frequency turn on and off which means overheating. To decrease the amount of pressurizing cycles extra buffer tank is used in the high-pressure zone.

As for the expansion valve it is regulated by a measuring probe at a low-pressure zone. It can passively operate. To be a volume efficient system it is usually installed right after the condenser in the external unit for air conditioners.

10.4. Energy performance

The Coefficient of Performance (COP) is the ratio of the heating power output to the amount of power input of a heat pump. This value is highly sensible to the heat source temperature. From the heat source, it is required to provide high and constant heat flux throughout the year. The heat source of the heat pump is usually considered to be “free power” since it is extracted from (or dissipated to) the environment (e.g.: air, ponds, ground) or from a waste heat source.

COP is mostly used when the heat pump is to be tested in heating mode, while in cooling mode cooling subscript is given. COP is always the ratio of power values of the input power and the heat pump electric power demand. For yearly or seasonal evaluation Seasonal Performance Factor (SPF) is used which is the ratio of the time-integrated values of the COP (ratios of energies).

Selection of a heat pump always requires the following steps: firstly, heat demand has to be defined, secondly a device has to be chosen that can cover the demand and then in the third step, verification is needed that it can extract it for the heat source. If it is extracting more from the heat source than previously, it will lead to source cooling, thus its temperature will be decreasing. With the lowered source temperature larger pressure difference is needed thus COP will be higher. The extracted heat should smaller than the heat source regeneration ability.

10.5. Calculations

Calculate how much heat is extracted from the heat source. If the power output is 20kW and the COP is 4.5.

We have a heat pump that can work like a refrigerator. On the evaporator side the parameters are the following: throat size 16mm flow speed $0.55 m \cdot s^{-1}$ and the temperatures are 15°C, 10°C. On the condenser side, the parameters are the following: throat size 16mm flow speed $0.5 m \cdot s^{-1}$ and the temperatures are 37°C, 42°C. Calculate the COP of the device for cooling and heating.

Recommended literature for this section:

[2] Renewable energy resources John Twidell, Tony Wier Routledge, 2015

[3] Introduction to Thermal Systems Engineering: Thermodynamics, Fluid Mechanics, and Heat Transfer Michael J. Moran, 2002

11. Cooling

In general, cooling has two stages, cooling to and cooling from the ambient temperature. To cool down a warm object to ambient temperature natural convection or free cooling is needed. To cool down an object below ambient temperature a reversed working heat pump also known as chiller is needed. The chiller's most notable part is the condenser section. If the condenser section utilizes the latent heat and sensible heat it is called a cooling tower. If it only changes sensible heat with the environment it is called a dry cooler (see figure 11.1).

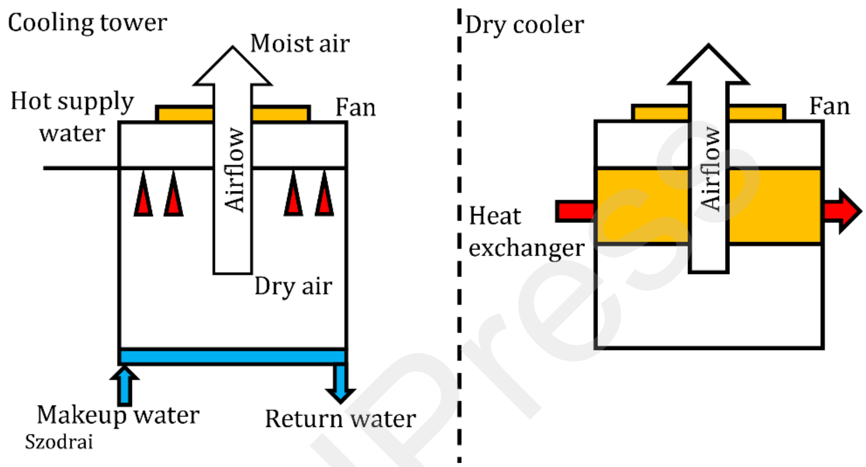

Figure 11.1. Schematic of cooling tower and dry cooler

11.1. Dry cooler

Dry coolers are also known as chillers. Its operation principles are the same as a reversed Carnot cycle. Usually, these are placed outside of the building, due to its high noise level and significant heat dissipation. Chillers main parts are the same as the heat pumps. The evaporator side connects to the refrigerator system as the cold energy source. Besides, to reduce cooling cycles frequency a cooling buffer tank is installed onto the device. The evaporator and the buffer tank can be combined when a shell and tube heat exchanger is used.

The condenser part consists of a cross-flow heat exchanger where one side has small diameter pipes where the phase change can occur and on the other side fins are placed to enhance the heat transfer to the air. A fan can enhance the heat transfer intensity. A dry cooler is widely used both in industrial and residential areas since it can operate on a high-temperature range.

11.2. Cooling tower

Cooling towers mostly used as condenser units of powerplants or larger buildings. The working idea of this device is that 32°C hot water flows to the tower and by using nozzles, it sprays the hot water into the airflow that is generated by a fan. In comparison to the dry cooler this case a direct heat exchange happens both sensible and latent. The inlet airflow creates moisture in the process, which means that the drier the ambient air is the more effective the device works. Since some amount of the hot water is taken away with the airflow makeup water is needed. In some cases, there can be water surplus, in that case additional drain outlet is needed. When the water is cooled down to around 27°C in the basin it is extracted at the return pipeline. The basin always has a free water surface which makes it too vulnerable to contamination and freezing. That is why the cooling tower circuit should be connected directly to the thermal machine and also for colder seasons dry coolers should be used.

For smooth operation, the water flow must be constant, and the airflow must be known. To ensure the flow rate the makeup water and airflow amount must be calculated. This amount depends on the ambient air's parameters. Ambient air parameters can be looked up from enthalpy-moisture (h-x) diagrams.

For calculating mass flow and energy, an equation can be written. The airflow has no losses in the process. On the contrary, the water flow has gains and losses. The gains are the inlet: makeup water flow and inlet air have some moisture content, while the losses occur in the outlet water and the moisture of the exhausting moist air.

The moisture content can be expressed with absolute humidity (x) which tells how much vapour (V) we have in the moist air.

Enthalpy values for dry air as was mentioned can be looked up in h-x diagram. The enthalpy of the water is 4.18 $kJ \cdot kg^{-1} \cdot$ (water temperature in °C). The specific heat of the moist air is calculated by first calculating the enthalpy of the moist air (MA) then dividing it by the temperature of the air (DA):

$$h_{MA} = c_{pDA} \cdot T + x \cdot (r + c_{p\,V} \cdot T) \tag{11.1}$$

$$c_{pMA} = h_{MA} \cdot T^{-1} \tag{11.2}$$

These equations assume that the contamination in the air does not modify the molar mass of the moist air by 2% and the specific heats are:

$c_{pDA}=1008.8 J \cdot kg^{-1} \cdot K^{-1}$ and $c_{pV}=1857.3 J \cdot kg^{-1} \cdot K^{-1}$ also the latent heat is $r=2.5 \cdot 10^{6} J \cdot kg^{-1}$.

Based on these assumptions the following expression can be made:

The makeup water amount is equal with the airflow moisture uptake.

$$\dot{m}_{makeup} = (x_{outlet} - x_{inlet}) \cdot \dot{m}_{air} \tag{11.3}$$

The airflow based on the derivation of the energy equation is:

$$\begin{aligned} \dot{m}_{air} = \dot{m}_{water} &\cdot \Delta h_{water} \\ &\cdot \left(\Delta h_{air} + x_{outlet\ vapour} \cdot h_{outlet\ vapour} - x_{inlet\ vapour} \right. \\ &\left. \cdot h_{inlet\ vapour} - \Delta x \cdot h_{makeup}\right)^{-1} \end{aligned} \tag{11.4}$$

11.3. Absorption machines

Absorption chillers do not have compressors in them and the fluid in the cycle is a solution of water with ammonia (R717) or lithium bromide (R718). Lithium bromide is more common because it is safer and non-toxic. In this section the work idea of lithium bromide machines is described. Lithium bromide can easily absorb water and can make a good mixture with it when the mixture of the heated water vapour leaves the concentrated solution. This chiller consists of four main parts: condenser, evaporator, expansion valve and thermochemical compressor. The thermochemical compressor has sub-cycle which also has four part: a generator an absorber a circulation pump and a regulating valve. The generator and the condenser work at high pressure, in some cases, these are even operating in one chamber. Similarly, the evaporator and the absorber can be freely linked together. In figure 11.2 a simple schematic is drawn of its work idea where the water vapour is noted with a dotted line and the liquid phase with solid lines.

The process starts at the generator where water vapour is produced by adding heat to the system. The added heat source ensures the evaporation process. This source is often linked with renewable energy sources since solar collectors can be an excellent source for this process or geothermal heat since usually it is less needed when cooling demand is high. In the generator along with the water vapour, a strong solution of lithium bromide is also created. To keep up the work cycle the generator and the absorber is connected. From the generator a strong solution is sprayed to the absorber and from the absorber a weak solution is pumped to the generator.

The weak solution is directly flowing to the condenser due to the added heat in the generator while the pressure in the condenser is high. In the condenser, the vapour is condensed and drained to the evaporator, while necessary heat is extracted from the cooling tower.

In the evaporator at low-pressure weak solution is sprayed in where heat is extracted. Heat is extracted from the cooling system, thus makes this part of our cold power source. Droplets that are not evaporated are collected in a basin and resprayed back into the chamber.

The fluid that is in a low-pressure chamber, heated by the cooling systems returns heat. As it was mentioned the strong solution is sprayed into the absorber and mixes with the weak solution from the evaporator. Both in the absorber and condenser, the condensation must ensure that the unwanted heat is removed, so it can be used by the cooling tower. Now the weak solution is ready to be injected into the generator. To increase the performance a counter flow heat exchanger can be used between the weak and strong solution lines.

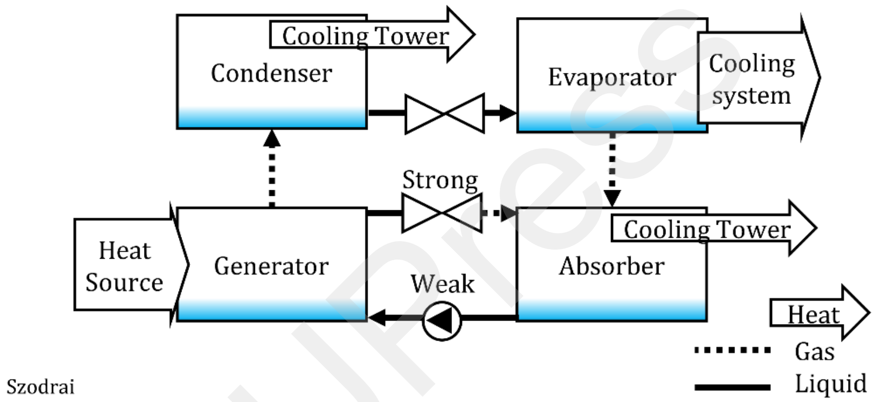

Figure 11.2. Schematic of an absorption chiller

11.4. Calculation

Water exiting the condenser of a power plant at 38°C enters the cooling tower with a mass flow rate of $4.5 \cdot 10^7 kg \cdot h^{-1}$. A stream of cooled water is returned to the condenser from a cooling tower with a temperature of 30°C at the same flow rate. Makeup water is added by a separate stream at 20°C. Atmospheric air enters the cooling tower at $40 kJ \cdot kg^{-1}$ and leaves at $80 kJ \cdot kg^{-1}$ enthalpy. Determine the mass flow rates of the dry air and the makeup water, in $kg \cdot h^{-1}$. The cooling tower operates at a steady state. Heat transfer with the surroundings and the fan power can be neglected, so can change in kinetic and potential energy. The pressure remains constant throughout at 1atm.

Recommended literature for this section:

[3] Introduction to Thermal Systems Engineering: Thermodynamics, Fluid Mechanics, and Heat Transfer Michael J. Moran, 2002

12. Combustion

Fuels are any materials that store potential energy in forms that can be practicably released and used as heat energy. This type of fuel source is favourable for thermal machines. Compared to heat pumps this source is highly independent therefore can be applied in any scale and location.

Fuels can have m states (gaseous, liquid, and solid). The ideal fuel has a gas phase. During its combustion, it requires a small amount of ignition energy meanwhile producing high amount of heat. The ignition energy is called endotherm energy and the heat that is produced is exotherm energy. For these kinds of endotherm reactions, every fuel and oxygen is needed. Based on the reaction duration it can be slow (e.g. rusting); medium (e.g. combustion) or quick: (e.g. explosion). The highest amount of energy and the most controllable reaction is combustion. Combustion is present at any boiler, gas turbine, or engine. At combustion-related problems, the tasks are to find how much fuel is needed to produce a given amount of heat, how much air is needed for complete combustion and how much will be the emission. For fuels mostly hydrocarbons are used, which contains carbon (C) and hydrogen (H). Complete combustion means that carbon and hydrogen combine with oxygen (O_2) to produce carbon dioxide (CO_2) and water (H_2O). Incomplete combustion means that part of the carbon is not completely oxidized and will produce soot or carbon monoxide (CO). Incomplete combustion fails to use the fuel efficiently.

Combustion fuel sources can be categorized into fossil fuels and biomass. The main difference is that fossil fuels are mined that can run out of capacity eventually and biomass can be produced in months or decades.

12.1. Gas fuels

Gas fuels, mostly including natural gas contains several hydrocarbon components which can be defined by volume fractions (r) is expressed in ($\%_V$) or ($m^3 \cdot m^{-3}$). The combustion of natural gases can be described by chemical equations where the chemical signs of the hydrocarbons are the following:

Table 12.1. Hydrocarbons

name	chemical symbol
methane	CH_4
ethane	C_2H_6
propane	C_3H_8
butane	C_4H_{10}
pentane	C_5H_{12}
hexane	C_6H_{14}

The chemical reaction in which hydrocarbons with oxygen produce an exothermic reaction and flue gas for 1m^3 of hydrocarbon is the following:

$$1 \cdot CH_4 + 2 \cdot O_2 = 1 \cdot CO_2 + 2 \cdot H_2O \tag{12.1}$$

$$1 \cdot C_2H_6 + 3.5 \cdot O_2 = 2 \cdot CO_2 + 3 \cdot H_2O \tag{12.2}$$

$$1 \cdot C_3H_8 + 5 \cdot O_2 = 3 \cdot CO_2 + 4 \cdot H_2O \tag{12.3}$$

$$1 \cdot C_4H_{10} + 6.5 \cdot O_2 = 4 \cdot CO_2 + 5 \cdot H_2O \tag{12.4}$$

$$1 \cdot C_5H_{12} + 8 \cdot O_2 = 5 \cdot CO_2 + 6 \cdot H_2O \tag{12.5}$$

$$1 \cdot C_6H_{14} + 9.5O_2 = 6 \cdot CO_2 + 7 \cdot H_2O \tag{12.6}$$

The number in front of the chemical sign shows how much volume is required or produced in the chemical reaction. In correspondence with the chemical signs some factors can be calculated such as the: O_2 coefficient denoted by f_1 ($m^3 \cdot m^{-3}$), CO_2 coefficient denoted by f_2 ($m^3 \cdot m^{-3}$), H_2O coefficient denoted by f_3 ($m^3 \cdot m^{-3}$). The f unit is in ($m^3 \cdot m^{-3}$) since it tells that for 1m^3 of fuel how much material is required or produced during the combustion. A well designed, adjusted and maintained gas flame produces only small amounts of carbon monoxide.

Incomplete combustion occurs: mixing of air and fuel; air supply to the flame; time of burn are insufficient and cooling of the flame temperature before combustion is complete.

12.1.1. Air demand

This specific quantity, which is called the minimum oxygen demand (O_{2min}) ($m^3 \cdot m^{-3}$) required for combustion of 1m^3 of combustible gas:

$$O_{2min} = \sum_{i=1}^{n} (r_i \cdot f_1) \tag{12.7}$$

The minimum air quantity (A_{min}) that can ensure the minimum oxygen requirement:

$$A_{min} = O_{2min} \cdot {O_{2\%v}}^{-1} \tag{12.8}$$

The $O_{2\%v}$ is referring to the O_2 volume fraction in the air. When fuel is injected into the combustion chamber it must mix with the air. The mixing is ensured by a swirler which is placed right after the injections. This mixing is imperfect, and the gas combustion demands need more air than in theory. The actual amount of air supply (A_{supply}) and the minimum rate of air demand can be determined with CO_2 measurements and their ratio is the excess air factor (λ).

$$\lambda = A_{supply} \cdot {A_{min}}^{-1} \tag{12.9}$$

12.1.2. Flue volume

From the hydrocarbon equations, it can be calculated that when 1m^3 of fuel is combusted how much m^3 of CO_2 and H_2O is produced, which is expressed by the following.

$$\sum CO_2 = \sum_{i=1}^{n} (r_i \cdot f_2) \tag{12.10}$$

$$\sum H_2O = \sum_{i=1}^{n} (r_i \cdot f_3) \tag{12.11}$$

For the volume flow calculation, different combustion models exist, based on the content of the flue gas.

The theoretical combustion model assumes that the combustion is complete (λ= 1) and pure oxygen is given for the combustion. During the combustion flue gas is produced that only contains CO_2 and H_2O.

Air-complete combustion model assumes that the combustion is complete (λ=1) and the oxygen is supplied by air. In the flue, gas nitrogen will appear additionally due to the presence of the air. Nitrogen oxides can be disregarded in combustions. The reason is that it only occurs in high temperatures and the amount is small even in those cases. Often its concentrations are limited yet rarely excessive.

Moist air-complete combustion flue model assumes that the combustion is complete. During the combustion water vapour is produced and additional air is added (λ> 1).

In the third most realistic model the volume flow of the flue gas is expressed with the following:

$$V_{flue} = \left(r_{N_2} + A_{supply} - O_{2,min} + \sum H_2O + \sum CO_2 \right) \cdot V_{fuel} \tag{12.12}$$

In an incomplete combustion model, unburned gases will also appear. In the resulting flue gas, the amount of unburnt fuel can be measured, statistically can be taken in an amount estimated using combustion modelling.

12.1.3. Dew point

During the combustion, the heat is extracted from the flue gas. During the extraction process, the water vapour could condensate on the exhaust pipe. If the exhaust pipe made from steel corrosion occurs which leads to leakage. To prevent it, two measurements can be taken. Non-steel (aluminium or plastic) pipe must apply, or the flue gas temperature must be above the condensation temperature. In some cases, if we wanted to extract the latent heat from the flue gas then our goal is to stay below the dew point. The condensation temperature or dew point is pressure-dependent, and the vapour partial pressure condensation temperature data can be looked up from a so-called Moliere diagram. The dew point can be expressed if the ambient pressure and vapour concentration is known in the flue gas.

$$p_{vapour} = \sum H_2O \cdot {V_{flue}}^{-1} \cdot p_{ambient} \tag{12.13}$$

12.1.4. Lower higher and Heating values

Higher heating value (HHV), ($MJ \cdot m^{-3}$) is determined by cooling all the components of combustion back to the initial pre-combustion temperature. After the combustion, the moisture is in liquid phase. HHV can be easily measured if the fuel is kept in a closed vessel during the measurement process.

Lower heating value (LHV), ($MJ \cdot m^{-3}$) is determined by subtracting the latent heat from the higher heating value. In many combustion-based machines, condensations must be avoided, thus this value gives the content of l the fuel. Now since latent heat is utilized the HHV is the energy potential for fuels. LHV is generated when moisture gets in is the gas phase after the combustion.

The heating value of a mixture is the volume fraction weighted average of heating values of the components.

$$HHV = \sum_{i=1}^{n} (r_i \cdot HHV_i) \tag{12.14}$$

$$LHV = \sum_{i=1}^{n} (r_i \cdot LHV_i) \tag{12.15}$$

For each component, HHV (and based on the known latent heat energy LHV) can be measured, and with stochiometric measurements, the volume fractions can be also determined. In any fuel type, the moisture always increases the difference between the HHV and LHV.

12.2. Liquid fuels

Liquid fuels in comparison with gas fuels have different applications. Liquid fuels are mostly used for vehicles and powerplants, while gas fuels have more residential applications. The liquid is mostly hydrocarbon mixtures operating on high pressure or liquid hydrocarbons that are sprayed into the combustion chamber.

For liquid fuels, the transfer is essential thus pour point must be defined since it will tell us the minimum temperature when fuel still flows. This is often needed for the kind of devices that operates at low temperature. Sometimes auxiliary preheating is needed for the ignition of liquid combusting device. Flashpoint is the temperature of the liquid where the evaporated fuel is ignited and a reaction occurs, however combustion has not started yet. In the meantime activation temperature will tell us what the lowest temperature is to start the combustion.

12.3. Solid fuels

Solid fuel combustion is also ideal when the fuel is mostly in dust form. Moreover, there are more differences: solid fuel volume is course and often porous, thus volume measurement is less practical. Solid fuel heat content is measured in ($MJ \cdot kg^{-3}$). Solid fuels have more components that cannot burn, thus making more solid depositions in the combustion chamber, known as ash. To keep up the solid combustion, ash must be removed frequently and burning cycles must be monitored.

Solid fuels can be divided into three main groups: conventional fuels (black and brown coal), biomass fuels (wood, pellets, briquettes), radioactive fuel (uranium).

Table 12.2. Solid fuels lover heating values [2]

Solid fuel type	Lower Heating Value ($MJ \cdot kg^{-1}$)
black coal	24
wood briquettes, pellets	18
brown coal	17
dried tree	15
fresh cut	6.8

Based on the LHV that can be seen in Table 12.2 the moisture content can be concluded. The black coal has the least amount of moisture in it. For solid fuels, this is the theoretical maximum or HHV. For biomass fuels, it is a lower value, only $18 MJ \cdot kg^{-1}$ and even this value can be theoretical since these are for forced dried solid fuels, which has a high moisture sorption affinity. The sorption affinity is relevant, and it can decrease the LHV. That is why these fuels (woodchip, briquettes, and pellets) must be kept in closed tanks. Pellet (6-12mm diameter cylindrical granules) and larger (100-155mm) briquettes have low moisture content and produce lower ash content than firewood. It is achieved by its composition: it is tuned to have a given heating value and low ash content.

In comparison a pellet of woodchips is a more inhomogeneous material with higher emission.

Freshly cut trees have $40\%_m$ moisture content. Without forced drying at least two years is needed to reduce its moisture to $20\%_m$. Still one favourable fact can be mentioned and that is that after drying tree when additional moisture added its sorption capabilities are worse.

Calculations are difficult to make for solid fuels since their material content is more complex and difficult to distinguish. Values could vary by location and time of extractions.

12.4. Calculation

With the given gas mixture, the gas mixture lover and higher healing value can be calculated. Calculate the flue’s condensation point if the air supply rate is 1.1.

Table 12.3. Components of a gas mixture

	r_i	LHV	HHV	f_1	f_2	f_3
	$m^3 \cdot m^{-3}$	$kJ \cdot m^{-3}$	$kJ \cdot m^{-3}$	$m^3 \cdot m^{-3}$	$m^3 \cdot m^{-3}$	$m^3 \cdot m^{-3}$
CH_4	0.9	35949.1	39886.8	2	1	2
C_2H_6	0.05	64615.7	70557.6	3.5	2	3
C_3H_8	0.05	93910.3	101980.4	5	3	4

Table 12.4. Dew point data

p_{vapour} (Pa)	T_{Dew} (°C)
15740	55
19917	60

Recommended literature for this section:

[3] Introduction to Thermal Systems Engineering: Thermodynamics, Fluid Mechanics, and Heat Transfer Michael J. Moran, 2002

13. Boilers

A boiler is a closed vessel or arrangement of enclosed tubes in which water is heated to supply steam to drive an engine or turbine or provide heat. It burns fuel to supply combustion to provide heat energy. Boilers can be classified by their temperature levels. Warm water boilers operate at maximum 110°C, at ambient or close to ambient pressure, in the system water flows. At higher temperatures, 130-150°C due to evaporation prevention higher pressures must be applied. When the pressure is low, and the heat load is high steam can be generated.

Usually, a boiler consists of a burner which responsible for the combustion, a burning chamber where the exothermic reaction could take place and dissipation can happen, a heat exchanger that extracts heat from the combustion chamber and an exhaust pipe where the flue gas can leave.

13.1. Burners

Gas burners purpose is to ensure the combustion and to provide oxygen and oxidizer to all of the components. Oxygen can be provided from the air around the boiler with a natural draft or with a fan. To ensure natural draft the combustion chamber must be open to the external fresh air source. A fan is needed when the combustion chamber is closed. The exhaust system has a large hydraulic resistance therefore natural draft is not enough, for large systems or when flue gas is cooled below dew point is compulsory. The fan produces unwanted noise load, which is why sound dampers are needed. The required oxygen for the flame formation is given by the combustion energy.

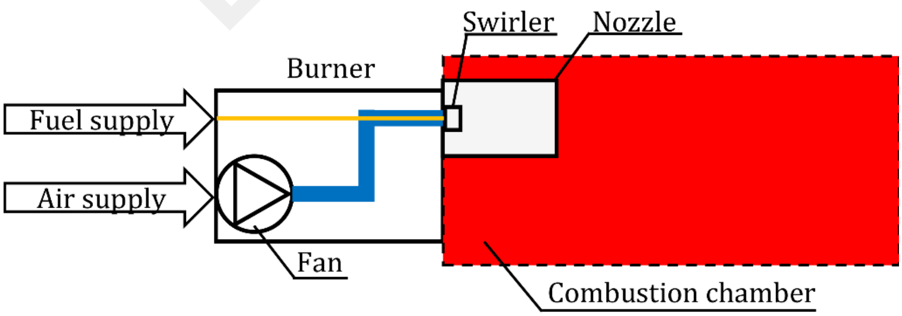

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Figure 13.1. Schematic of a burner

The oxidizer supply system is based on the fuel type. For gaseous or liquid fuels, the supply flow can be continuous, while for solid fuel it is different. Solid fuels usually ignited with a gas burner and when the combustion is started the burner turns off till the new combustion cycle starts.

The gas and air mixing location corresponding to the flame can be either pre-mixed, partially, and fully the mixing is done by a swirler that produces vortex at the burner nozzle.

13.2. Flame tube boilers

The volume ratio of the primary combustion chamber and the secondary waterside can distinguish boilers power level. If the secondary side is larger it is a flame tube boiler and if the combustion chamber side is larger it is a water tube boiler. Flame tube boilers as it can be seen on figure 13.2 are robust devices that usually provides few megawatts of heat. This could supply high-temperature water or steam. Heat extraction can be increased by putting passes above each other, whereas in a large water tank, flame tube does the heating.

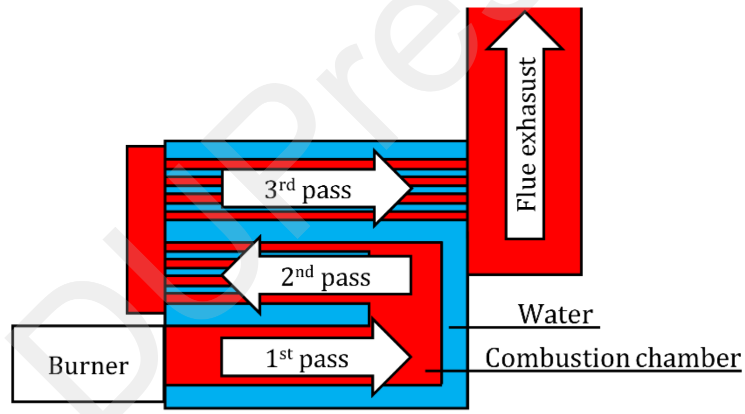

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Figure 13.2. Schematic of a flame tube boiler

The advantages of these boilers are the simple structure and easy to maintain its condition. The disadvantage is the outdated design, it is difficult to insulate because of the large volume, thus the radiative loss is high, and a separated building needed to place it. Also, with the newer energy-efficient technologies the temperature can be lower, and steam is rarely used.

13.3. Water-tube boilers

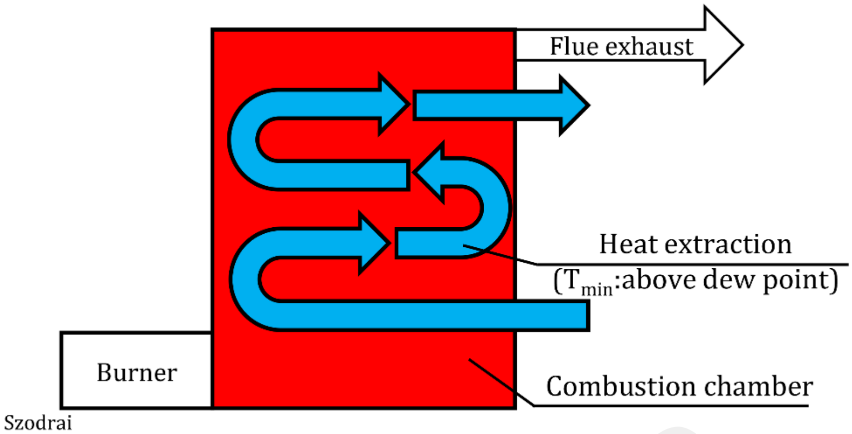


Figure 13.3. Schematic of a water-tube boiler

Water-tube boilers (see figure 13.3) represent an updated design; however, their performance level is lower than the flame tube boilers. The main principle is that in the combustion chamber a heat exchanger is placed, and by doing this the total volume of the boiler is reduced. Since the water volume is small, the desired temperature can be reached in a short time and easy to modulate its heat output.

13.4. Boiler efficiency

For any thermal system, the efficiency can be increased when the fluid temperature that is used for heat transfer is close to ambient temperature. Most of the boilers due to high stress and temperature are made from metals. If the temperature is lowered efficiency is increased, yet the risk of corrosion is raised. When boilers efficiency examined three categories can be mentioned: constant temperature, low temperature, and condensation boilers.

13.4.1. Constant-temperature boilers

Constant temperature boilers during their optimal operation produce high-temperature constant flow. These are excellent for high heat demand systems since this type was utilized first this category merges with the flame tube boilers. The loss distribution in a constant temperature boiler can be seen on figure 13.4. Its peak efficiency is around 77.5%. The highest loss occurs when it only utilizes the sensible heat from the flue gas since condensation is not allowed in the machine. This design does not allow the flue gas to cool down effectively thus high-temperature gas leaves which is an additional loss.

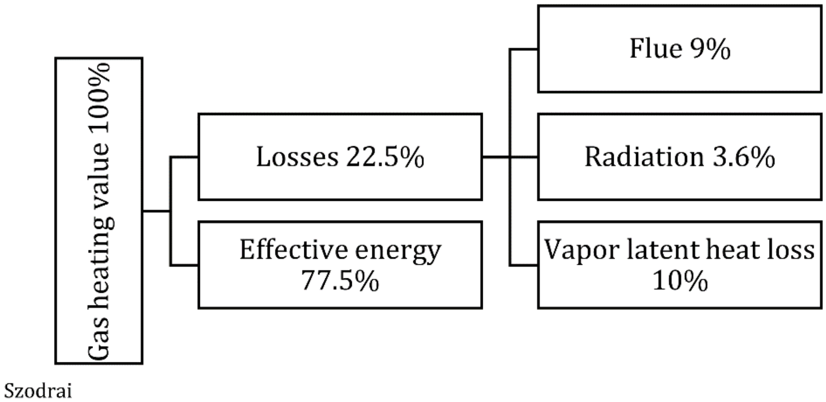


Figure 13.4. Loss distribution in a constant temperature boiler

The usual produced temperature range for these devices deviates from 150°C to 70°C. This cannot be lower since the technology that uses heat, cools down the fluid and if that fluid cools down the combustion chamber close to dew point it could damage the boiler.

13.4.2. Low-temperature boilers

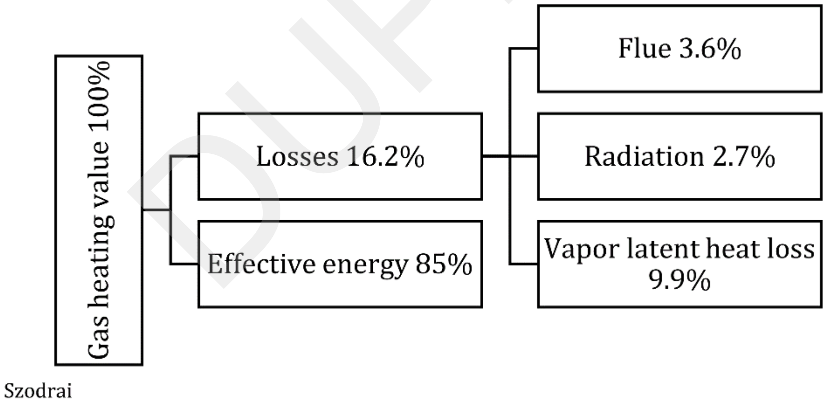


Figure 13.5. Loss distribution in a low-temperature boiler

Low-temperature boilers aim is to find the dew point and to warm up the fluid just a few degrees above it. With this idea, flue and radiation losses can be reduced significantly. This design is rarely used since it cannot be as simple as a constant temperature boiler and not as effective as a condensation boiler.

13.4.3. Condensation boilers

Condensation boilers are the most effective boilers, due to energy-efficiency regulations its application is compulsory. It must be noted that if the heat demand is constant or solid fuel is used then every boiler becomes a constant temperature boiler. A condensation boiler is used when heat needed in small cycles, e.g.: intermittent heating or domestic hot water demands. The produced warm fluid can go down to 30-40°C and the return temperature to the boiler even lower. As we can see in Figure 13.6 the losses are just 6% in the best case.

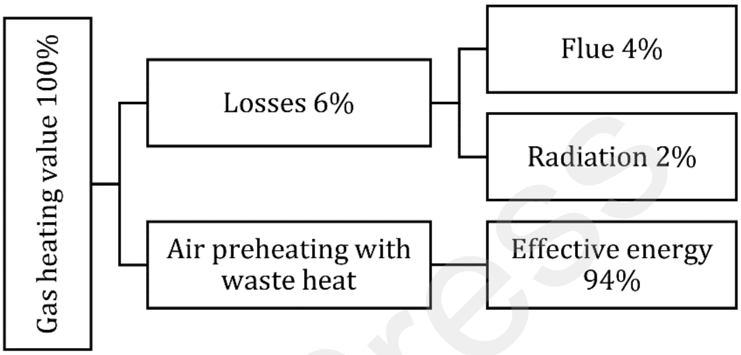


Figure 13.6. Loss distribution in a condensation boiler

Condensation can be achieved with two methods: firstly, an additional heat exchanger is applied after the boiler, thus corrosion due to condensation can only take place there. An additional heat exchanger (depicted in figure 13.7) raises the hydraulic resistance that why a fan is needed. In HVAC boilers they are called as turbo, which refers that it has a built-in fan. Secondly, a wet combustion chamber can be used where condensation is allowed, usually, this leads to a more expensive boiler yet significantly smaller one.

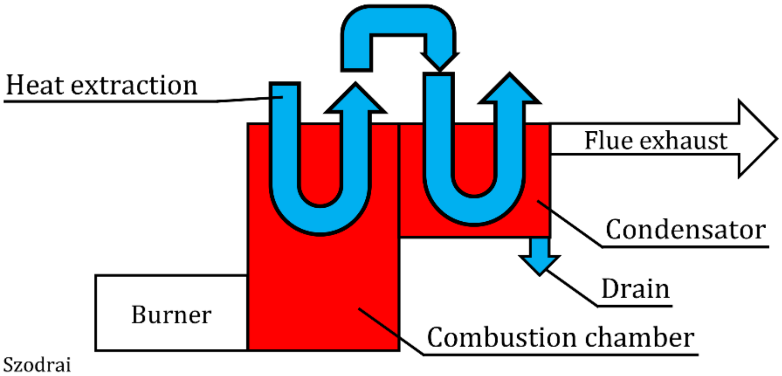


Figure 13.7. Condensation boiler

When 1m^3 of natural gas is combusted 1kg vapour is produced. This latent heat is about 11% of HHV of natural gas. It must be highlighted that the efficiency depends on the moisture content (a type of fuel, air change rate, air humidity and the rate of condensation).

To enhance a condensation boiler's cost-efficiency hybrid boiler can be used. This means that the condensation boiler is coupled with a heat pump. With an internal control mechanism, it decides which is more cost-efficient, the condensation boiler or the heat pump heat source.

13.4.4. Efficiency calculation

The efficiency calculation is the following for every boiler. Efficiency is the ratio of the useful heat ($\dot{Q}_{useful}$) and the input power ($\dot{Q}_{input}$). We are also known that if there would not be losses ($\dot{Q}_{loss}$) these two values were equal.

$$\eta = \dot{Q}_{useful} \cdot \dot{Q}_{input}^{\ -1} = \left(\dot{Q}_{input} - \dot{Q}_{loss}\right) \cdot \dot{Q}_{input}^{\ -1} \tag{13.1}$$

As we saw in figure 13.4-13.6 losses are from radiation, flue, and latent heat losses. Radiation losses can be calculated if the thermal transmittance (h), surface temperature ($T_{surface}$), air temperature (T_{air}) areas related to the boiler are known. It is called radiation loss since at the heat loss the radiative loss is more relevant compared to convective heat loss.

$$\dot{Q}_{radiation} = h \cdot A \cdot \left(T_{surface} - T_{air}\right) \tag{13.2}$$

Flue losses can be expressed if the specific heat, mass flow and temperatures are known.

$$\dot{Q}_{flue} = c_{flue} \cdot \dot{m}_{flue} \cdot \left(T_{flue} - T_{air}\right) \tag{13.3}$$

For radiation loss, the air temperature is the boiler rooms air temperature while at flue loss it is the ambient temperature. The exhaust flue temperature is measured directly after the boiler. Latent heat loss can be expressed with the specific condensation ($c_{condensation}$) heat and mass flow ($\dot{m}_{vapour}$).

$$\dot{Q}_{condensation} = c_{condensation} \cdot \dot{m}_{vapor} \tag{13.4}$$

The required heat can be calculated if the mass or volume flow rate of the boiler ($\dot{V}_{fuel}$) is known and the higher heating value of the fuel.

$$\dot{Q}_{input} = \dot{V}_{fuel} \cdot HHV \tag{13.5}$$

Efficiency is a dynamic value which can change at different loads. That is why for comparison figure 13.8 is made. Where the 3 types of boiler efficiency and load curve can be seen.

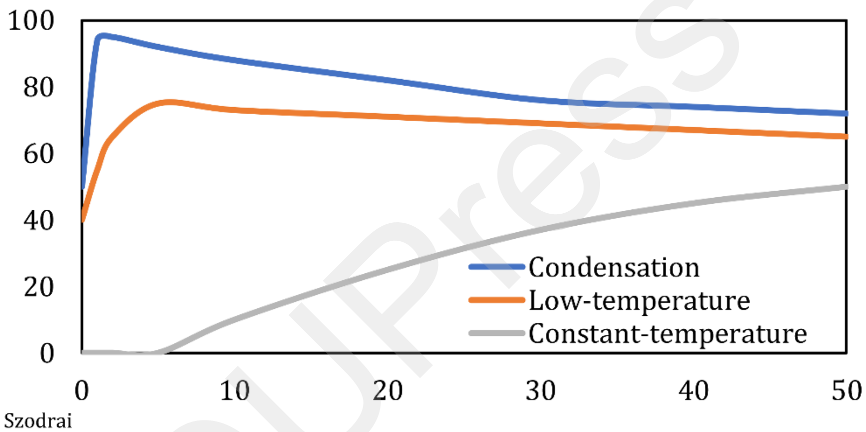


Figure 13.8. Boiler efficiencies

When small heating cycles are needed the condensation, boiler peaks with their efficiency. The low temperature follows it, yet it could not close its effectiveness since latent heat is not harnessed. It can be also seen that both the low temperature and condensation boiler's efficiency decreases when the heat demand becomes steady. On the other hand, constant temperature boiler cannot perform when the load is not steady, while it reaches a steady operation its efficiency increases regularly. Peak efficiency is good at the initial design however for energetic calculation, seasonal efficiency is more accurate. Seasonal efficiency is a time-averaged value of the efficiencies. These can be calculated with some coarse methods. During the operation period when a boiler is activated, it has some standby power uptake ($\dot{Q}_{standby}$) and when it works it uses its total capability ($\dot{Q}_{work}$). When the peak efficiencies, power ratios and duration ratios are known, seasonal efficiency can be calculated.

$$\eta_{seasonal} = \eta \cdot \left(\left(\tau_{standby} \cdot \tau_{work}^{-1} - 1 \right) \cdot \dot{Q}_{standby} \cdot \dot{Q}_{work}^{-1} + 1 \right)^{-1} \qquad (13.6)$$

DIN 4702 divides the heating season into 5 sections that have equal energy demands. Standardized percentages are 12.8%, 30.3%, 38.8%, 47.6% and 62.6%.

$$\eta_{seasonal} = 5 \cdot \left(\sum_{i=1}^{5} \frac{1}{\eta_{boiler,i}} \right)^{-1} \qquad (13.7)$$

13.5. Calculations

We are operating a 25kW atmospheric burner boiler with the given gas mixture. Determine the combustion and efficiency of the boiler. If the outside temperature is 0°C flue temperature is 140°C, the flue has $70 kg \cdot h^{-1}$ mass flow and $1.4 kJ \cdot kg \cdot K^{-1}$ specific heat. Device's gas consumption is $2.89 m^3 \cdot h^{-1}$.

With a 16kW boiler which yearly efficiency 87%. The heating season length is 190days and the heating hours are 2100h. Flue loss is 550W. Standby loss is 500W. How much is the natural gas demand?

Recommended literature for this section:

[3] Introduction to Thermal Systems Engineering: Thermodynamics, Fluid Mechanics, and Heat Transfer Michael J. Moran, 2002

14. Gas and steam turbines

Turbines can generate torque by harnessing dynamic pressure. Wind and water turbines produce power by slowing isothermal fluid flows. This flow is favoured, yet it can be only applied locally where conditions are given. On the other hand, gas turbines can be used freely from locations. Due to that gas turbines pressure difference and flow rate a high enthalpy expansion is ensured. This high enthalpy expansion can be from combustion which is the basis of open gas turbines. When a gas is heated its pressure increases and can supply a turbine with power. If the heating and cooling are solved a closed-circuit gas turbine can operate. On rare locations high enthalpy steam could leave the ground, thus this source can power turbines also, which can be the third category because the heat generation is independent of the turbine system.

Gas turbines work cycle is based on the Joule cycle which conations a compression, isobar heating and expansion, and a fourth step that makes a difference between the main two types of gas turbine. It is the cooling which can close the cycle. In a gas turbine a compressor, a heater and a turbine can be found with an optional heat extracting device.

Turbines are highly effective thermal machines that can convert heat energy to kinetic energy in multiple stages. The stage is the turbines smallest part, it both contains the accelerating and work producing part. The stage is made of fixed and rotating nozzles, stators, and rotors. Enthalpies change than occur either on stators or on rotors. The following can express the specific energy change:

$$h_2 - h_1 = 0.5 \cdot v_2^2 - 0.5 \cdot v_1^2 \qquad (14.1)$$

The r reaction rate is the rate of the enthalpy change on the rotor and the stage. At the impulse turbine, the r-value is low, in some cases it can be 0. The heat drop occurs at the stators on the rotors kinetic energy used to produce work. The gas accelerates up to a high level on the stators, so the outlet velocity is high. The high velocity is sensitive to the hydraulic losses, so the efficiency is highly dependent on the RPM. Short rotors with large diameters lower its losses.

For reaction turbines, the reaction rate is mostly 0.5. Rotors also use a huge amount of heat, unlike stators that only use a small amount, so the velocities are low. Hydraulic losses are low, and it will not depend on the rotation number by that much. Since the rotors use heat the pressure drop is significant, the partial losses will be also significant. With the greater partial losses, it needs longer rotors with small diameters that produce lower velocity.

14.1. Open gas turbines

In an open gas turbine, the Joule cycle is open, the exhaust fluid from the expansion is thrown out to the environment just as the inlet air sucked in for compression. This open cycle requires fresh air, which is delivered by the compressor This is more effective when the gas turbine moves. If the gas turbine is moving it could help to intake more air into the system and making it more effective It is the reason aeroplanes utilize this kind of propulsion. Based on the compressor or air intake subtypes can be made. Different subtypes of open gas turbines are optimized for specific flow rates where they can be used or where they excel. For low-speed Ma=0.2 (low intake flow rate) tubes and an additional propeller can be attached. These turboprop turbines are used for small aeroplanes such as Cessna type aeroplanes. Commercial aeroplanes travel closer to the speed of sound but in subsonic regime Ma=0.8. The airflow is much higher thus a densely made axial fan is applied before the compressor. These turbofan turbines not only provide air to the combustion but also uses it for propulsion. The ratio between the core and bypassing mass flow makes the bypass ratio. In commercial airlines high-bypass flow turbofans are used, however low bypass planes could use afterburners, which is an additional combustion chamber right after the turbine. This afterburner can increase the performance of the turbine and it can result in hypersonic speed Ma>1. Another rarely used turbine is the ramjet. This turbine only consists of a combustion chamber and a diffuser. It is optimized for hypersonic speed regime, and it cannot operate in the subsonic regime since solely the thrust from combustion is not enough.

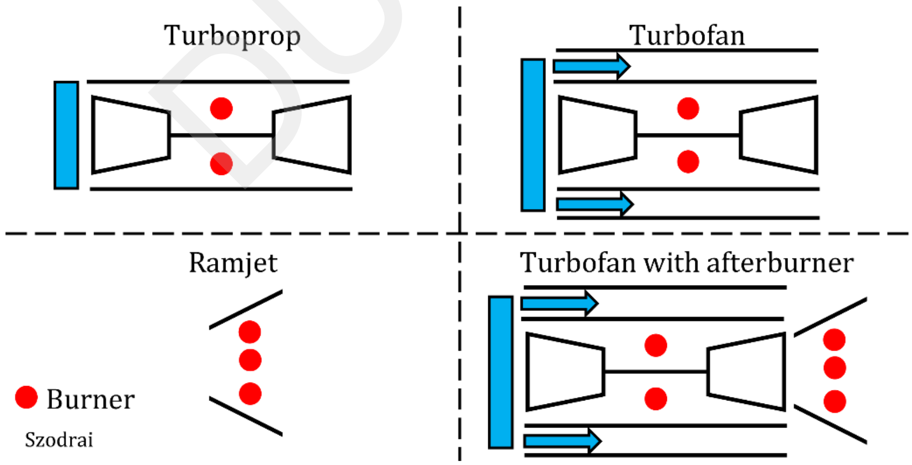


Figure 14.1. Schematic of a turbojet engines

As for the other parts of the turbine, the compressor and the turbine are similar in most cases. Compressors are axial type compressors where the pressure is maximized by several rows of axial blades. First stage usually has lower speeds and large diameters that keeps up the continuity while the velocity and pressure increases, in contrary the diameter of the blades decreases. The reverse process goes through the turbine. It starts with a small rotor and increases by every stage. As we have learnt it the axial velocity triangles, for optimization stationary rotors known as stators are used between each rotor. The turbine and the compressor are linked together. The ratio that tells how much power is transferred from the turbine (p_t) to the compressor (p_c) (values are specific $W \cdot kg^{-1}$) is the back-work ratio. Back-work ratio can be expressed with the following:

$$bwr = p_c \cdot p_T^{-1} \tag{14.2}$$

This reduces the effective propulsion, yet it also makes an effective drive for the compressor. To start a turbine usually a small motor is needed to spin both the turbine and the compressor since the inertia of the rotors is large. In open turbines, enthalpy is increased by combustion which is ensured by spraying liquid fuel into the combustion chamber. The combustion generates flue gas which cannot be recycled therefore open construction is needed.

14.2. Closed gas turbines

Closed cycle turbines have an additional heat exchanger, which purpose is to ensure the continuity of the cycle. In a closed construction, combustion could not occur in the main circuit. The fluid in the cycle can be air or any gas that has good thermal properties. Since high temperatures can occur the turbine has to be made of heat resistant materials. The system is closed, to work more effectively, hydraulic losses must be reduced. Heat added and removed by heat exchangers which means that in the heat exchanger the hydraulic resistance is high.

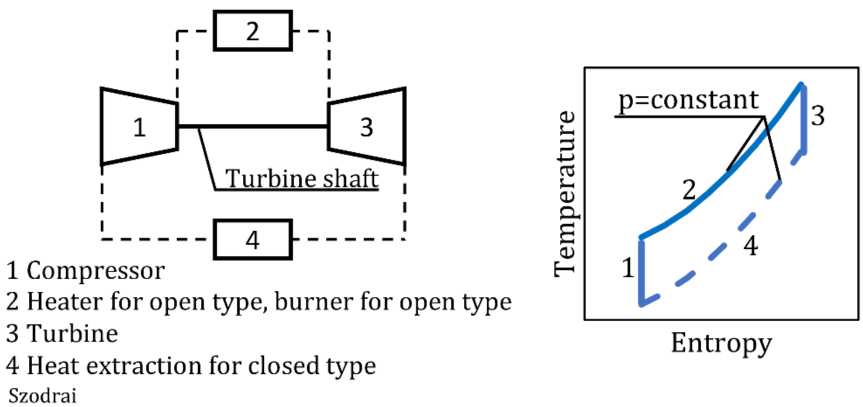


Figure 14.2. Open and closed cycles

Emission of this type is much lower if the added heat can be provided without combustion. Closed construction is used for power plants.

14.3. Steam turbines

Steam turbine refers to the fluid that goes into the turbine. Steam has large latent heat content, which is provided by a natural steam source (geothermal) or steam boilers, where overheating can increase the heat capacity of the steam. For steam turbines, it is essential to know how dry the steam is. From high enthalpy sources it is ensured that there are no liquid droplets in the steam. If a droplet would reach a high RPM turbine blade it would cause damage to it, so it must be avoided, the way it can be avoided by using a separator, which separates the gas and liquid phases of the wet steam. Between stages, condensation or separation is not possible, thus by design condensation cannot occur in a turbine. If more energy needed to be extracted, after the turbine another separator can be placed, and a lower pressure turbine can be added. These are called double flash steam turbines, when it only has one turbine it is called single flash.

14.4. Comparison

As a summary of the gas and steam turbines, the following table can be used. From Table 14.1 it can be concluded that steam turbines are mostly favoured impulse turbine construction while gas turbines are mostly excelling when reaction construction is used.

Table 14.1. Turbine comparison

	Gas turbine	Steam turbine
intake		
pressure	< 25bar	< 250bar
temperature	< 1200°C	< 550°C
outlet		
End pressure	> 1bar	> 0.02bar
End temperature	> 400°C	> 20°C
Heat drop	$500 kJ \cdot kg^{-1}$	$1500 kJ \cdot kg^{-1}$
Stages	4-8	20-40
Number of phases	1	2
Back-work ratio	40-80%	1-2%

14.5. Regenerative efficiency boost calculation

Regenerative efficiency boost happens when waste heat preheats the fluid before the actual heater. The preheater can be placed both on open and closed turbines as well. In figure 14.3 an open system regenerative boost can be seen. The exhaust is connected to a fresh air inlet. The waste heat is transferred right after the turbine. Point a shows how much does it cool the exhaust gas down and the b shows the preheated state. It means that instead of ambient temperature, T_b temperature where the heating has to start.

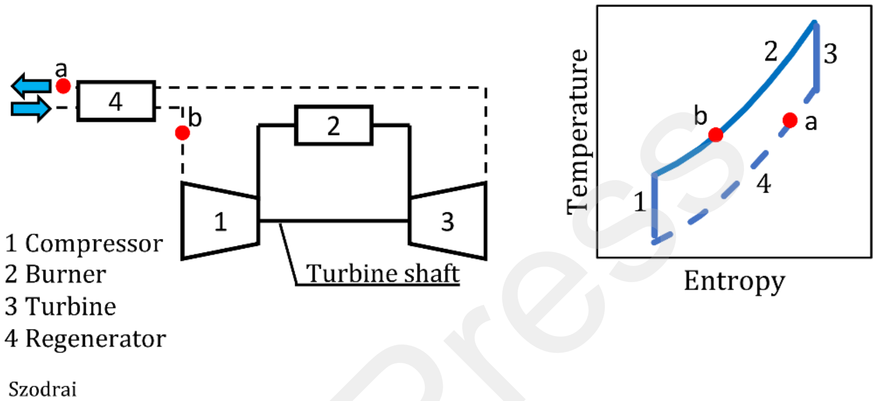
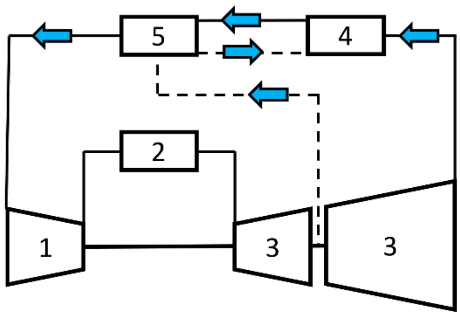


Figure 14.3. Regenerative efficiency boost schematic for an open turbine system

In figure 14.4 we can see a slightly different waste energy harnessing. In closed systems, the compressor (1) and the heater (2) could be merged into a steam generator, which is usually a flame tube boiler or a geothermal heat source. Between two stages of the turbine (3), low enthalpy steam is removed to a preheater (5) in the preheater the low enthalpy steam is condensed and drained to the condenser (4). In the condenser (4) heat is removed and drained back to the start of the cycle.



1 Compressor
2 Heater
3 Turbine
4 Condenser
5 Regenerator

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Figure 14.4. Regenerative efficiency boost schematic for a closed turbine system

For the efficiency calculation, specific values are used, which means energy values divided by the mass flow of the system. That is why lowercase values are used.

$$h_b = \eta_{regenerative} \cdot (h_4 - h_2) + h_2 \quad (14.3)$$

$$\eta_{total} = (p_t - p_c) \cdot q_{heating}^{-1} = \big((h_3 - h_4) - (h_2 - h_1)\big) \cdot (h_3 - h_2)^{-1} \quad (14.4)$$

$$\eta_{reg} = (p_t - p_c) \cdot q_{heating}^{-1} = \big((h_3 - h_4) - (h_2 - h_1)\big) \cdot (h_3 - h_b)^{-1} \quad (14.5)$$

14.6. Calculation

For a regenerator effectiveness of 80%, will determine the thermal efficiency. Also plot the thermal efficiency versus the regenerator effectiveness ranging from 0 to 80%. h_1=300kJ·kg^{-1}, h_2=580 kJ·kg^{-1}, h_3=1515kJ·kg^{-1}, h_4= 808kJ·kg^{-1}.

Determine the thermal efficiency for a closed steam system, if the system operates at 6bar, the maximum temperature of the system is 600°C and the temperature at the second stage of the turbine is 300°C.

Steam enters a turbine operating at a steady-state with a mass flow rate of 4600kg·h^{-1}. The turbine develops a power output of 1000kW. At the inlet, the pressure is 60bar, the temperature is 400°C, and the velocity is 10m·s^{-1}. At the exit, the pressure is 0.1bar, the quality is x=0.9, and the velocity is 50m·s^{-1}. Calculate the rate of heat transfer between the turbine and surroundings, in kW.

Recommended literature for this section:

[3] Introduction to Thermal Systems Engineering: Thermodynamics, Fluid Mechanics, and Heat Transfer Michael J. Moran, 2002

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