Process Engineering
Division

GEA Wiegand
GmbH

GEA Evaporation Technologies

Evaporation Technology using Mechanical Vapour Recompression

Leading Technologies. Individual Solutions.

Contents

Thermal separation processes such as evaporation and distillation are energy intensive. In the course of their development, the aim of reducing energy costs first led to multiple-effect plants, then to thermal vapour recompression, and finally, to the use of mechanical vapour recompression systems.

In a conventional evaporator, the vapour stream produced is condensed, meaning that its energy content is lost to a large extent. In comparison, mechanical vapour recompression permits the continuous recycling of this energy stream by recompressing the vapour to a higher pressure and therefore, a higher energy content.

Mechanical vapour recompression reduces the consumption of primary energy and, consequently, the environmental load.

Main fields of application are currently the Food and Beverages industry (evaporation of milk, whey, sugar solutions), Chemical industry (evaporation of aqueous solutions), the Salt Works industry (evaporation of saline solutions) and Environmental Technology (concentration of waste water).

In each case, the decision on whether a vapour recompression system should be installed must be made on the basis of an efficiency study.

Mechanical Vapour Recompression and Evaporation

- **Importance**
- **Background**
- **Economic efficiency**

Plants for evaporation, distillation, evaporative crystallisation and evaporative drying are energy intensive. Operating costs of these plants are therefore primarily determined by the energy costs.

The reduction and optimisation of the specific energy consumption is therefore of prime importance.

There are three main techniques for minimising specific energy consumption, which can be applied either singly, or in combinations:

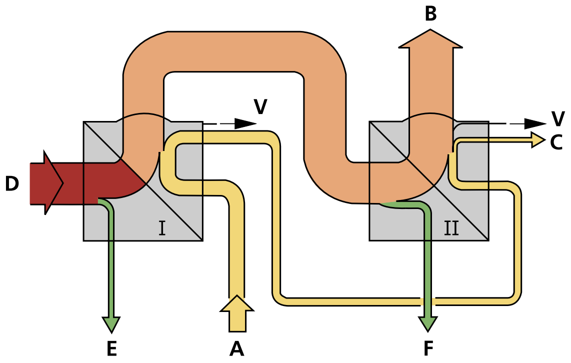
1. multiple effect arrangement
2. thermal vapour recompression
3. **mechanical vapour recompression**

1. Multiple Effect Arrangement

In a multiple effect evaporation plant, the vapour produced in the first effect by the live steam is not *lost* to the condenser, but is reutilized as the heating medium of the second effect. This effectively reduces the steam consumption by about 50%.

As this principle is repeated, further steam reductions follow.

The maximum heating temperature of the first effect, and the lowest boiling temperature achieved in the final effect creates a total temperature difference that is spread across the individual effects. As a result, the temperature difference per effect decreases as the number of effects increases. Their heating surfaces must consequently be larger in order to reach the specified evaporation rate. A first approximation shows that the heating surface to be used for all effects increases proportionally with the number of effects, and that in this way the investment costs considerably increase, whereas steam savings progressively decrease.



A	*product to be evaporated*
B	*residual vapour*
C	*concentrate*
D	*motive steam*
E	*heating steam condensate*
F	*vapour condensate*
V	*heat loss*

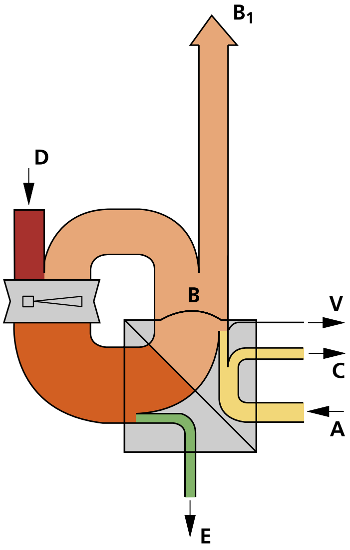
Heat flow diagram of a double-effect, directly heated evaporator

2. Thermal Vapour Recompression

During thermal vapour recompression, vapour from a boiling chamber is recompressed to the higher pressure of a heating chamber according to the heat pump principle; i.e. energy is added to the vapour. The saturated steam temperature corresponding to the heating chamber pressure is consequently higher, enabling the vapour to be reused for heating.

For this purpose, steam jet vapour recompressors are used. They operate according to the jet pump principle. They have no moving parts, ensuring a simple and effective design that provides the highest possible operational reliability.
The use of a thermal vapour recompressor has the same steam / energy saving effect as an additional evaporation effect.

A certain quantity of live steam, the so-called *motive steam*, is required for the operation of a thermal vapour recompressor. This motive steam quantity must be transferred to the next effect or to the condenser as surplus residual vapour. The surplus energy contained in the residual vapour approximately corresponds to the amount of energy supplied in the motive steam.



A	*product to be evaporated*
B	*vapour*
B_1	*residual vapour*
C	*concentrate*
D	*motive steam*
E	*heating steam condensate*
V	*heat loss*

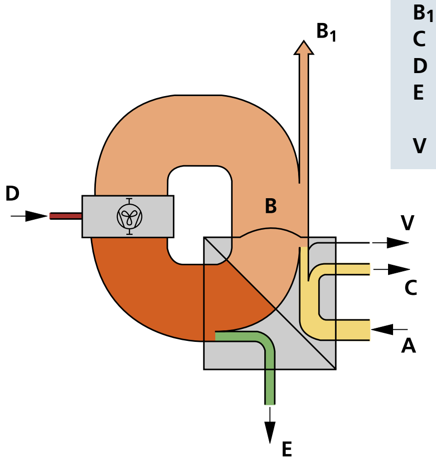
Heat flow diagram of an evaporator heated by thermal vapour recompression

3. Mechanical Vapour Recompression

During mechanical vapour recompression, the vapour of an evaporator is recompressed to a higher pressure by means of a mechanically driven compressor. The recompressor therefore also operates as heat pump, adding energy to the vapour.

Contrary to the compression heat pump with circulating process liquid (i.e. a closed system, refrigeration cycle) the vapour recompressor can be considered as a special case of the compression heat pump because it operates as an open system.
After compression of the vapour and subsequent condensation of the heating steam, the condensate leaves the cycle. The heating steam (hot side) is separated from the vapour (cold side) by the heat exchange surface of the evaporator.
The comparison between the open compression heat pump and the closed compression heat pump shows that the evaporator surface in the open system basically replaces the function of the expansion valve of the process liquid in the closed system.

By using a comparably small amount of energy, i.e. the mechanical energy of the compressor impeller in the case of the compression heat pump, energy is added to the process heat and continuously recycled. In this case, primary steam is not required as the heating medium.



A	*product to be evaporated*
B	*vapour*
B_1	*residual vapour*
C	*concentrate*
D	*electrical energy*
E	*heating steam condensate*
V	*heat loss*

Heat flow diagram of an evaporator heated by mechanical vapour recompression

The condensation heat to be dissipated in multiple effect and thermal vapour recompression systems, is still significantly high. In a multiple-effect plant, with *n* number of effects, the condensation heat is approximately 1 / n of the primary energy input. Furthermore, a steam jet compressor will only recompress part of the vapour stream, and the energy of the motive steam must be dissipated as residual heat through the cooling water. However, the use of the open compression heat pump principle can significantly reduce, and even eliminate, the amount of heat to be dissipated through the condenser.

A small amount of additional energy or condensation of excess vapour may be required to achieve the final heat balance, thereby allowing constant pressure ratios and stable operating conditions.

Reasons for using mechanical vapour recompression

- low specific energy consumption
- gentle evaporation of the product due to low temperature differences
- short residence times of the product, as a single-effect system is most often used
- high availability of the plants due to the simplicity of the process
- excellent partial load behaviour
- low specific operating costs

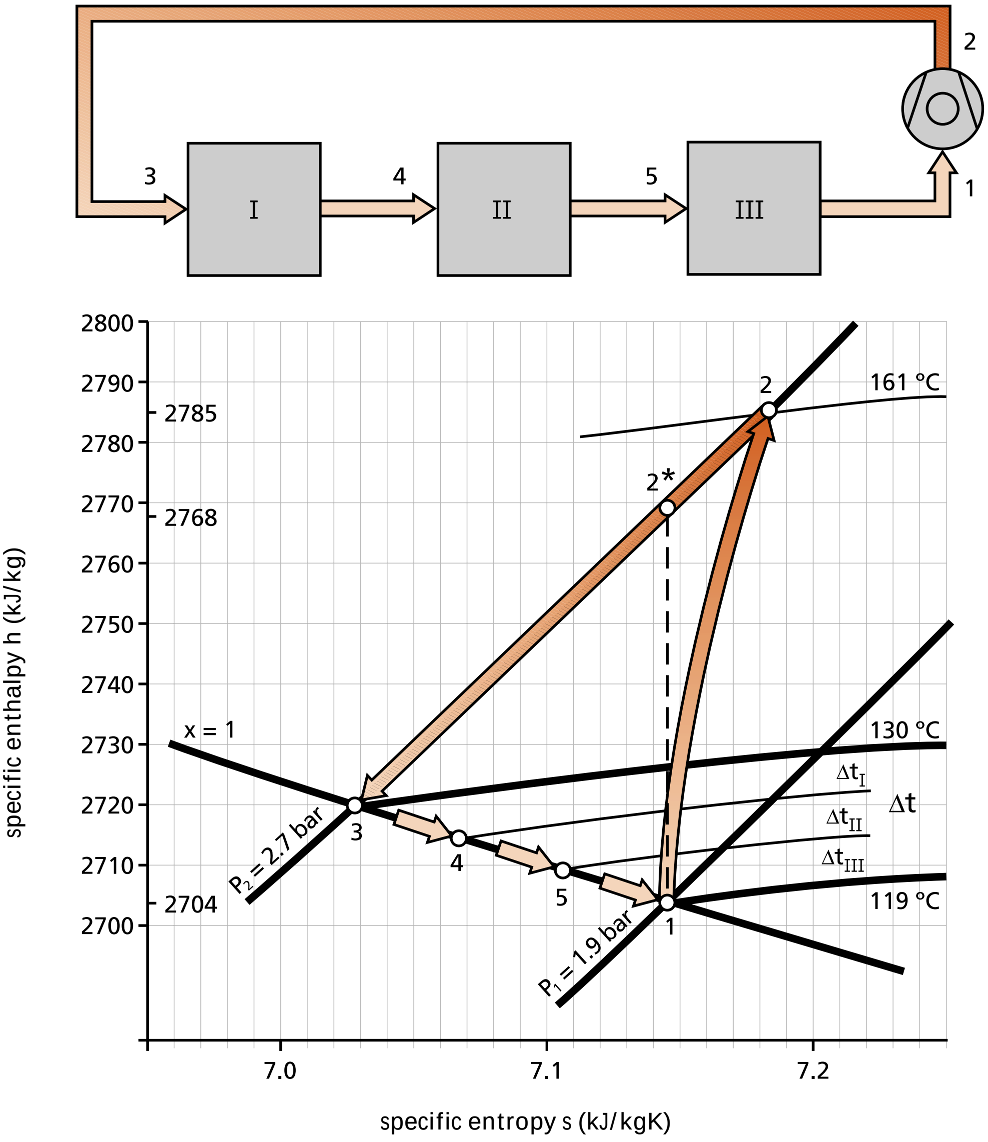
Principles of Mechanical Vapour Recompression

Single-stage centrifugal compressors and high pressure fans are generally used for mechanical vapour recompression systems for cost reasons. The explanations below are therefore limited to these designs.

Centrifugal compressors are volumetrically governed machines; i.e. the volumetric flow rate remains almost constant, regardless of the suction pressure. Mass flow does however change in proportion to the absolute suction pressure.

The compression cycle of the single-stage centrifugal compressor is depicted in the h,s diagram. The power that is required by the single-stage centrifugal compressor is

$$N = \dot{m} \cdot \Delta h_s / \eta_s$$



Change of state of water vapour in the Mollier h,s diagram in the case of single-stage compression

In the example: compression of saturated water vapour from the evaporator effect from suction state
$p_1 = 1.9$ bar and $t_1 = 119$ °C to
$p_2 = 2.7$ bar and $t_2 = 161$°C
(compression ratio $\Pi = 1.4$).

The compression cycle follows the polytropic curve 1 – 2, with specific enthalpy of the vapour increasing by the amount of Δh_p. For the specific enthalpy h_2 of the vapour, the value to be obtained by definition from the equation for the internal (isentropic) efficiency of the compressor

$$\eta_s \approx \frac{\Delta h_s}{\Delta h_p} = \frac{h_2{}^* - h_1}{h_2 - h_1} \approx 0.8$$

is $h_2 = 2785$ kJ/kg ($\eta_s \approx 0.8$ applies to single-stage centrifugal compressors in the case of water vapour). $t_2 = 161$ °C relative to h_2 and p_2. This vapour can now be used for heating of evaporator effect I. It first loses its superheat and is cooled to the saturation temperature t_3 (130 °C) of p_2 (2.7 bar). At this temperature, it passes to the calandria of the evaporator effect.

Based on $\Delta h_p \cong \Delta h_s / \eta_s$
$N = \dot{m} \cdot \Delta h_p / 3600$ (kW)

$\dot{m}$	the drawn-in vapour stream in kg/hr
Δh_p	the specific polytropic (effective) compression work in kJ/kg
Δh_s	the specific isentropic compression work in kJ/kg
η_s	the isentropic (internal) efficiency of the compressor

The specific polytropic compression work Δh_p depends *inter alia*, on the polytropic exponent κ and the molar mass M of the drawn-in gas, as well as the suction temperature and the required pressure increase. For the actual coupling power of the prime mover (electric motor, gas engine, turbine etc.) a further allowance for mechanical losses is taken into account.

Single-stage centrifugal compressors with impellers made of standard materials are capable of achieving a water vapour pressure increase by a factor of 1.8 , or, if higher-quality materials such as titanium are used, by a factor of up to 2.5.

The final pressure p_2 is then 1.8 , or max. 2.5 , times the suction pressure p_1, which corresponds to an absolute increase in saturated steam temperature of about 12-18 K, up to a max. 30 K depending on the suction pressure.

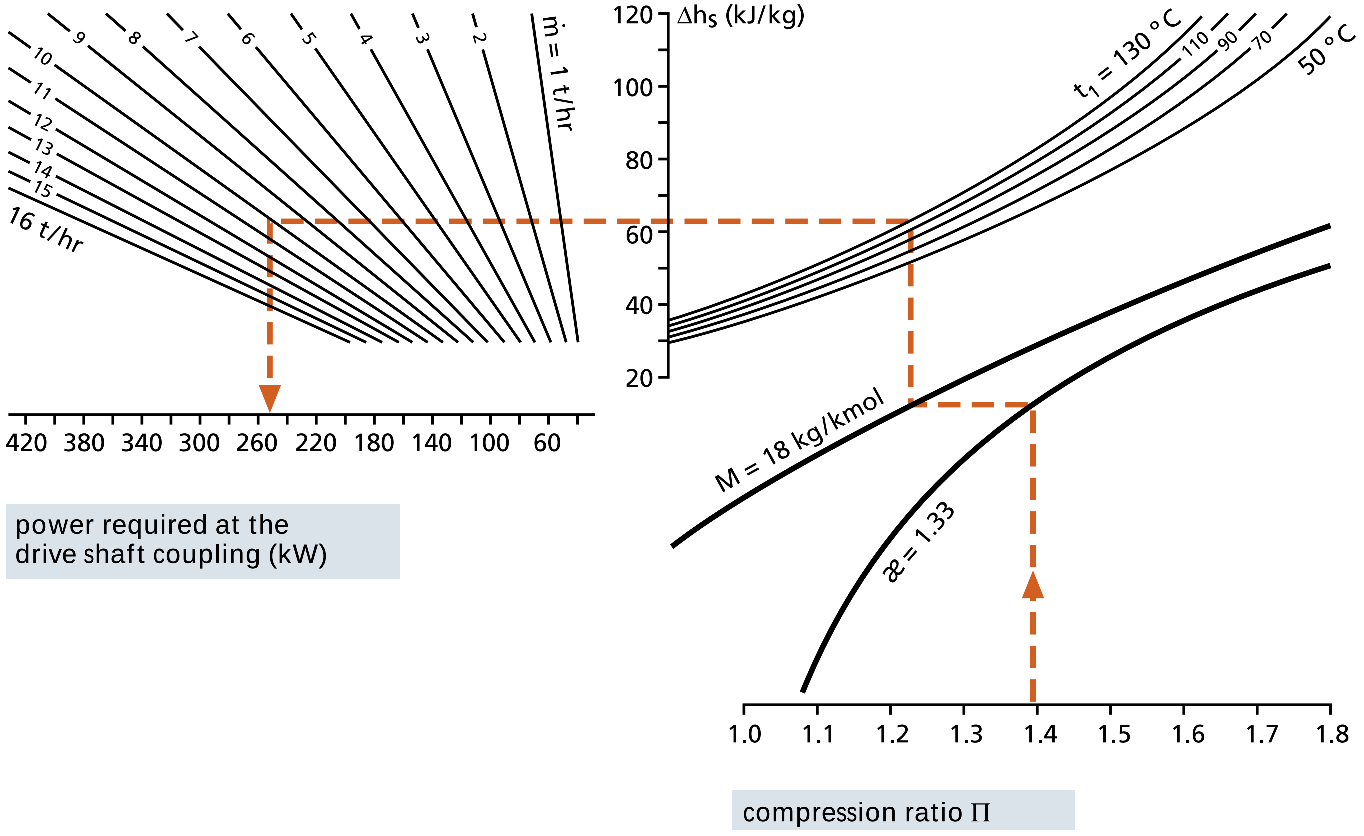
In evaporation technology, it is common practice to designate pressures by the corresponding water boiling temperatures. In this way, the temperature differences available can be directly indicated.

Example:

suction pressure $p_1 = 1$ bar corresponds to 100 °C
final pressure $p_2 = 1.7$ bar corresponds to 115.2 °C

$$\text{pressure ratio } \Pi = \frac{p_2}{p_1} = 1.7$$

saturated steam temperature increase: 15.2 K



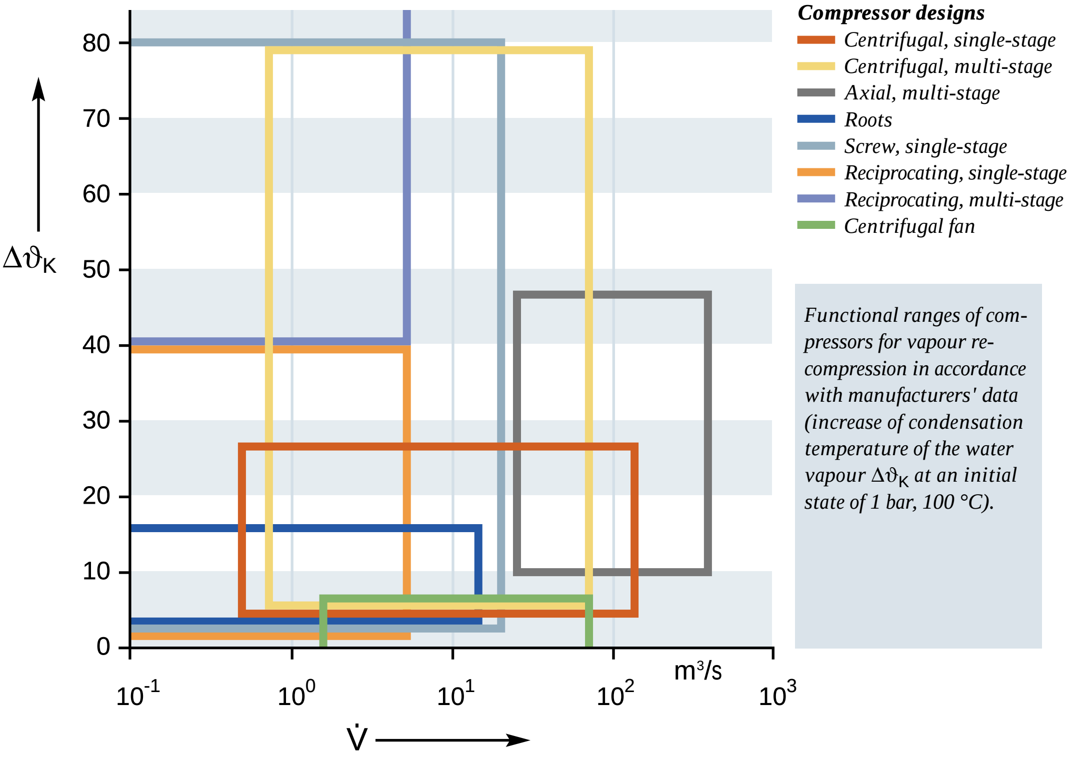
Determination of the coupling power (kW) for the prime mover.
Nomogram for isentropic compression work Δh_s of the single-stage centrifugal compressor for saturated steam (molar mass M = 18 kg/kmol, polytropic exponent $\kappa = 1.33$) in relation to the compression ratio Π and the suction temperature.

Mechanical Vapour Recompressors – Design and Functional Ranges

Machines for the compression of gases operate in accordance with positive displacement or dynamic principles.

With positive displacement machines, moving machine parts separate the suction chamber and the pressure chamber, and the gas pressure is increased as the volume of the operating chamber decreases. In the case of a reciprocating compressor, this is done by the movement of the piston within the cylinder.

In dynamically operating machines, the gas is supplied with energy by the impeller blades rotating at high circumferential speeds. The gas is first accelerated and is then decelerated through a diffuser situated downstream from the impeller. In this way, the high velocity is converted into pressure energy. Depending on the direction in which the fluid passes through the impeller, the relevant machine is either called an axial-flow, mixed-flow or centrifugal compressor.



Functional ranges of compressors for vapour recompression in accordance with manufacturers' data (increase of condensation temperature of the water vapour $\Delta\vartheta_K$ at an initial state of 1 bar, 100 °C).

The type of compressor best suited depends on the operating conditions relevant to the application. Key parameters are the required pressure rise and the flow rate of the vapour to be compressed.

Π is the pressure ratio of final pressure p_2 to suction pressure p_1 and is defined as compression ratio.

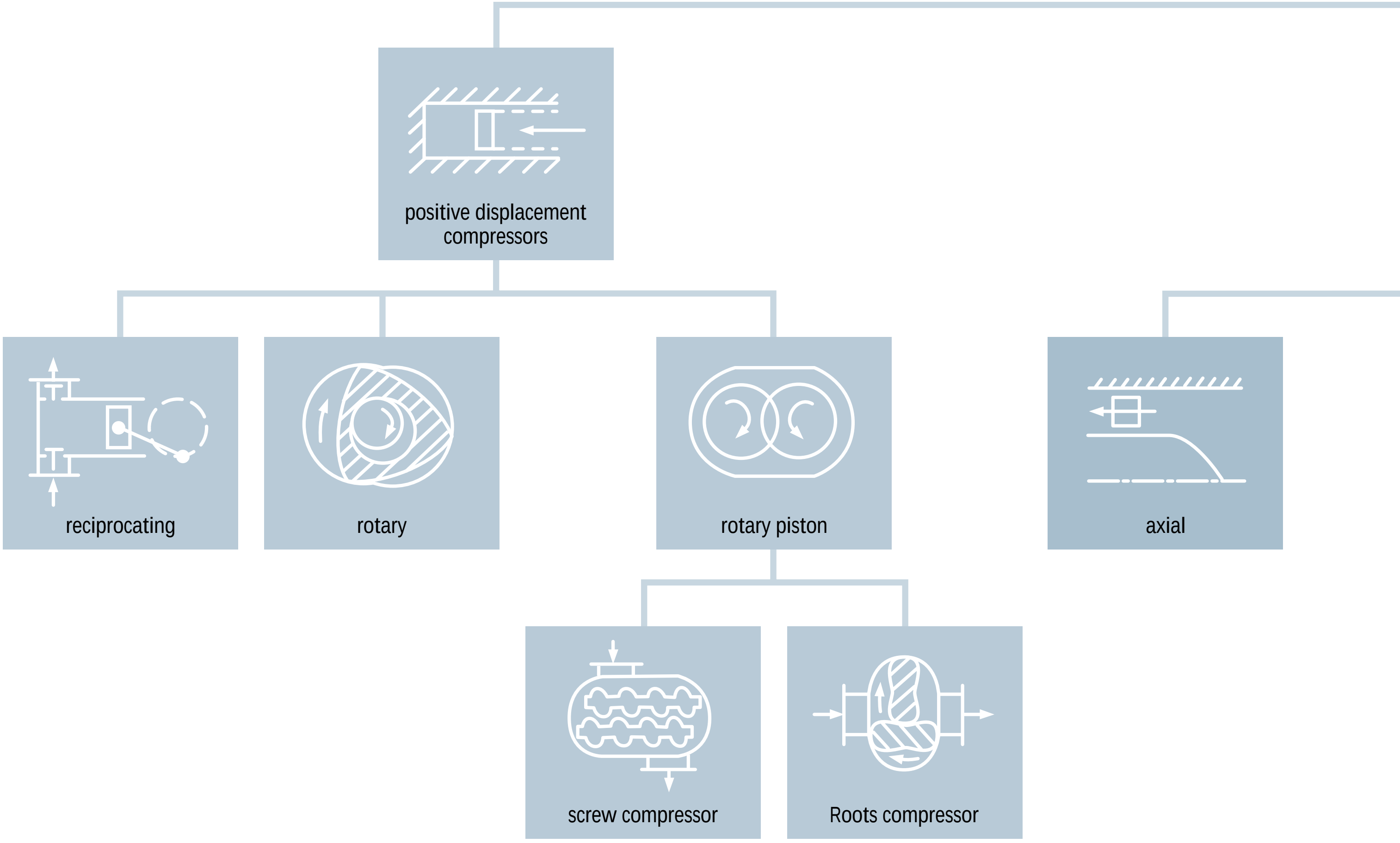
As evaporation plants are frequently operated in the vacuum range at medium heating surface loads and with small temperature differences, centrifugal recompressors are often used.

These are mainly:

- high pressure centrifugal fans
- single-stage centrifugal compressors

These machines are capable of a wide range of flow rates (e.g. 3,000 to 500,000 m^3/hr), at pressure ratios of 1.1 to 2.5.

Mechanically driven compressors



Reciprocating compressors operate in similar fashion to the principle of the internal combustion engine. The crankshaft moves the piston in a straight oscillating motion via the connecting rod and the piston rod running on the crosshead. The gas in the working chambers above and below the piston is displaced through valves actuated by the gas pressure. In order to avoid thermal stress at the sealing faces, the cylinder shell and the gland pocket may be heated with steam.

$\dot{V}_{min} = 0.01\ m^3/s$
$\dot{V}_{max} = 6\ m^3/s$

Rotary compressors are of little importance for the compression of water vapour. They are frequently used for the compression of cooling agents.

The working elements of the **screw compressor** are the primary rotor and the secondary rotor. Compartments are formed by the rotors between their intermeshing screw profiles and the casing. As the rotors turn, the compartments become progressively smaller. *The installed pressure ratio* Π_{in} is determined by the position of the outlet port and rotor dimensions.

$\dot{V}_{min} = 0.06\ m^3/s$
$\dot{V}_{max} = 22\ m^3/s$

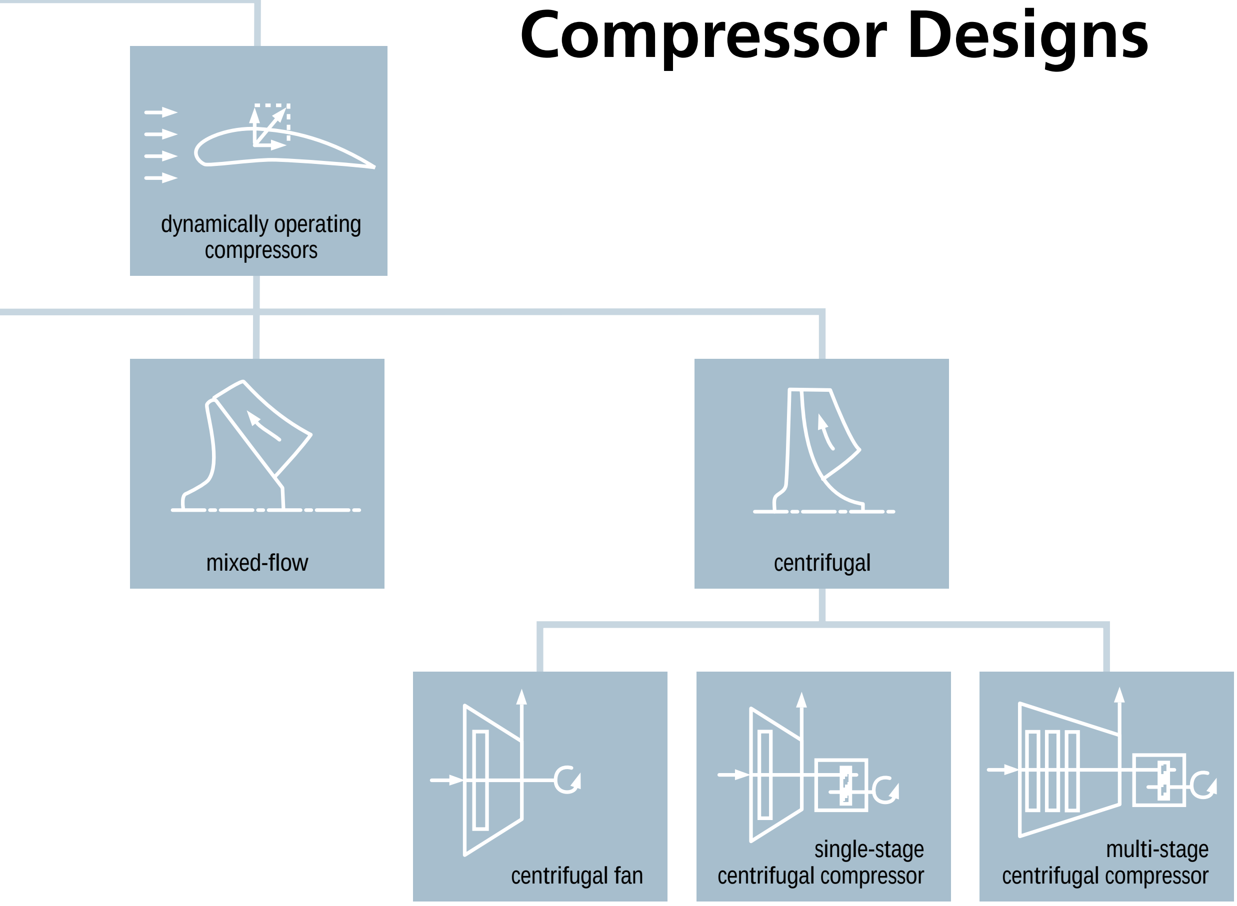
The two symmetrical, figure of eight shaped rotary lobes and the blower casing of the **Roots compressor** form compartments. As the lobes turn, the gas flows into these compartments and is transferred from the suction side to the pressure side. There is no internal compression in the rotating blades. The gas is compressed in the compartment on the pressure side by the positive displacement principle. A small gap remains between the lobes during rotation, and they do not actually touch.

$\dot{V}_{min} = 0.05\ m^3/s$
$\dot{V}_{max} = 25\ m^3/s$

Axial compressors are used for very large volumetric flow rates. They are nearly always designed as multi-stage systems. In a single axial stage, only a fraction of the pressure increase of a single centrifugal stage can be achieved. The efficiency of multi-stage axial compressors is however higher than that of multi-stage centrifugal compressors. Compared with the centrifugal compressor, a much smaller sized axial compressor can be used for the same type of compression work.

$\dot{V}_{min} = 25\ m^3/s$
$\dot{V}_{max} = 400\ m^3/s$

Operating Principles and Compressor Designs



Mixed-flow compressors are of little importance for the compression of water vapour.

Centrifugal fans can be used for low pressure ratios of up to Π=1.25. In the same way as a centrifugal compressor, the gas enters the eye of the impeller along its axis, exiting radially by means of centrifugal forces. The fan impeller and housing is of welded plate construction with reinforcing rib stiffeners as required. Gearboxes are generally not required, as the drive system gives the required speed of the impeller.

$\dot{V}_{min}$ = 1 m³/s
$\dot{V}_{max}$ = 140 m³/s

Single-stage, centrifugal compressors

The main feature of this type of compressor is the overhung impeller and the compact arrangement of compressor and gearbox. Motor, gearbox and compressor are mostly mounted on a common base-frame. Cast materials are used for the compressor casing. The impellers, which are highly stressed by the high tip speeds of > 400 m/s, are made of high-quality materials such as chrome-nickel steels or titanium alloys.

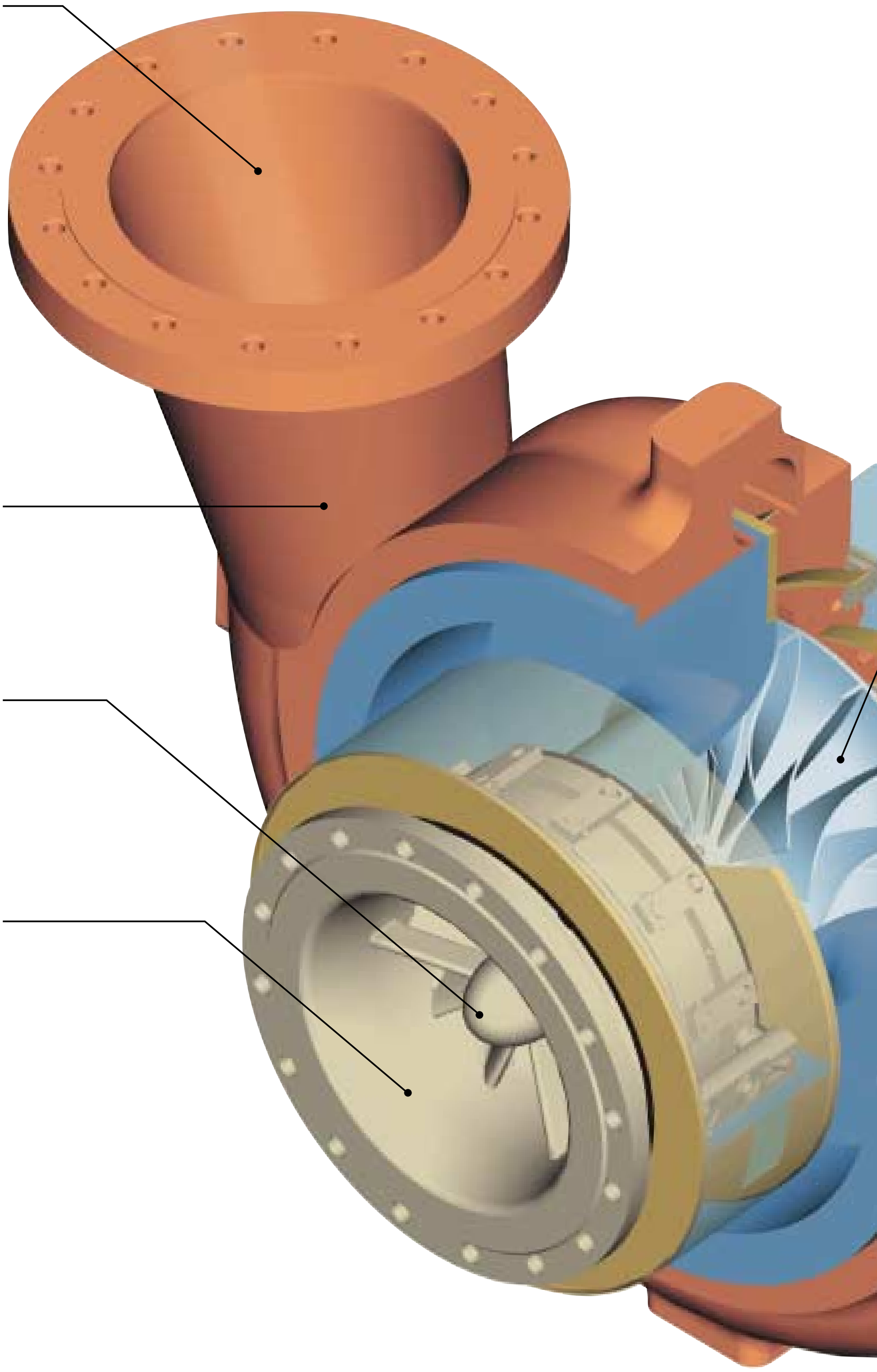
Π_{max} = 2.5
$\dot{V}_{min}$ = 0.5 m³/s
$\dot{V}_{max}$ = 150 m³/s

Multi-stage, centrifugal compressors

This type of compressor is used for large volumetric flow rates and high saturated steam temperature increases. The multi-stage centrifugal compressor is formed by the arrangement of several stages on a single shaft. After leaving one stage, the gas flows through a diffuser and interstage channel before entering the next impeller stage. The impeller shaft runs on bearings in the casing and is driven by a separate helical gear. For increasing the efficiency and for avoiding unacceptably high temperatures in the casing, water can be injected into the interstage channels.
In order to reach pressure ratios exceeding Π = 10, single-stage machines can also be connected in series. If the impellers were driven from a central drive with several pinions, the relevant unit would be called a two-, three-, or four-impeller compressor.

Π_{max} = 10 (in a casing)
$\dot{V}_{min}$ = 0.8 m³/s
$\dot{V}_{max}$ = 70 m³/s

Design Details of the Single-stage, Centrifugal Compressor

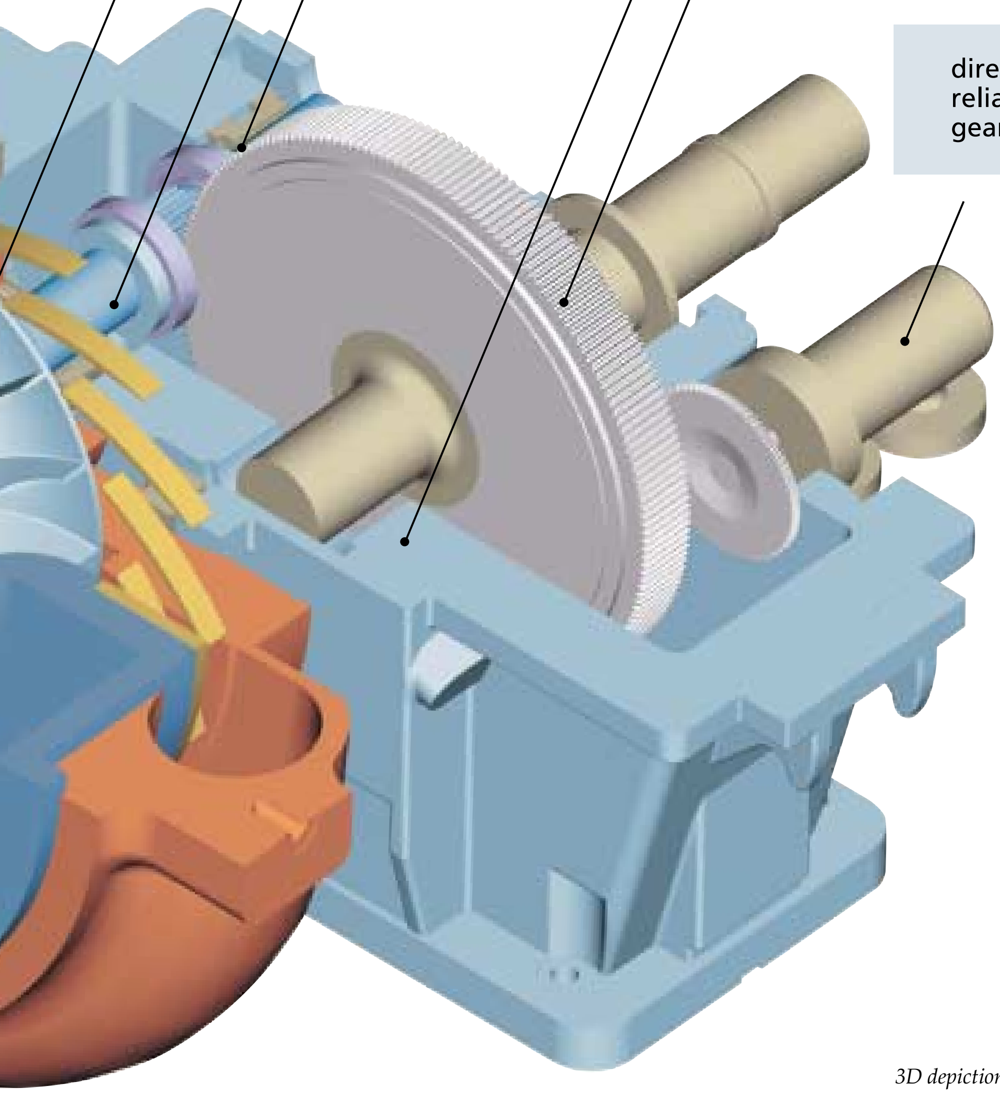


gas outlet

one-piece spiral casing permits final pressures of up to 60 bar

inlet guide vane for continuous adjustment achieves maximum partial load efficiencies

gas inlet



impeller of semi-open design permits maximum pressure ratio per stage

pinion shafts of highest quality ensure safe operation

maintenance-free pinion shaft bearings with optimum dampening properties

gearbox housing of solid, compact design

helical gear wheel

directly-driven main oil pump giving reliable lubrication to bearings and gears

3D depiction of a centrifugal compressor (illustration: Atlas Copco)

The impeller

The impeller is of the overhung design at the free end of the shaft (pinion shaft in the case of a compressor, main shaft in the case of a fan).

Depending on the compressor design, semi-open or closed impellers are used.

The blade geometry might be

- radial, or
- backward curved.

Radial bladed impellers are capable of achieving higher pressures due to the higher tip speeds that can be achieved as a result of their greater strength. Impellers with backward curved blades have lower permissible tip speeds, but their working range is wider and more stable.

For lower pressure increases, i.e. relatively low tip speeds, closed impellers are used due to their steep characteristic curve.

The impeller can be precision milled or of welded design. Frequently duplex steel of material EN 1.4462 is used. This material is corrosion resistant and has the required strength. Other CrNi steels and special materials such as titanium are also used.

Spiral casing

After leaving the impeller the accelerated gas stream flows into the spiral casing and tube diffuser. During this process, the high kinetic energy is converted into static pressure by deceleration of the flow.

While centrifugal compressor casings are mostly made of CrNi steel castings, fan casings are normally of welded design. To minimise corrosion, CrNiMo steel, typically material EN 1.4571, is used for the fan.

Casing thickness and external reinforcing is sized in such a way that the permissible deformation, which is of particular importance during vacuum operation, is not exceeded.

Gearbox

The helical gears of modern compressors are integrated within the compressor. For this reason, a coupling between gear and compressor shaft is not required. Thrust collars are situated on the high-speed pinion shaft. These thrust collars transmit the residual axial thrust to the low-speed main shaft (wheel shaft).

Centrifugal fans, which run at low speeds compared to compressors, do not require a gearbox. The impeller shaft is directly connected to the motor shaft by means of a coupling.

Bearing and lubrication system

The bearings of centrifugal compressors must ensure stable, vibration-free running conditions due to the high speeds, of up to 20,000 rpm, that can be encountered by a pinion shaft.

Radial tilting-pad bearings are therefore used for the high-speed pinion shaft. The wheel shaft of the gear runs on multi-faced, hydrodynamic journal bearings. The thrust bearing is designed as a combination radial/axial unit to contain the remaining axial thrust.

The bearings are lubricated with pressurised oil. For this purpose, a standardized lubrication system consisting of an oil tank, main oil pump, auxiliary oil pump, oil filter and oil cooler is installed.

Centrifugal fans are frequently equipped with less expensive roller bearings. For characteristic speed values (mean bearing diameter x speed) of up to 600,000 mm/min, simple forced oil lubrication is sufficient. At higher values, the same kind of lubrication system as for centrifugal compressors is used.

For fans running at characteristic speed values of more than 800,000 mm/min, hydrodynamic journal bearings are used.

Compressor Drives

Different types of prime mover can be used for driving vapour recompressors.

In each case, the drive is selected on the basis of its efficiency and the type of drive power available.

Electric motors are commonly used as drives. They offer considerable advantages due to the standardization of sizes and types of protection, their low power / weight, power / volume, price / performance ratios and minimum maintenance requirements.

Prime movers

Electric motors

Three-phase asynchronous motors

Three-phase asynchronous motors operate, according to the number of pairs of poles, at synchronous speeds of 3000, 1500, 1000 or 750 rpm (50 Hz, idle speed) or, if frequency converters are used, at variable speeds. Two types of motor are available: low voltage, and high voltage motors. Low voltage motors generally operate at capacities of up to 630 kW or 1,250 kW for supply voltages of 400 V or 690 V respectively. High voltage motors and converters can be used for capacities of up to approx. 6,000 kW. The efficiency of asynchronous motors is constant over a wide load range.

Direct current motors

Variable speed, direct-current motors are recommended for frequent partial load operation at high efficiency. They put a much lighter load on the operating current system during start-up than three-phase asynchronous motors. Their disadvantages are their higher prices and maintenance requirements. Compared to frequency controlled asynchronous motors, the direct current motor has lost some of its importance.

Gas engines

Gas engines are used if insufficient amounts of electrical energy are available. Good efficiencies, of up to 90 %, are reached if the waste heat from the cooling water and exhaust gas can be used, for example, for preheating purposes. The purchase price of a gas engine for waste heat recovery is considerably higher than that of a comparable electric motor. Its maintenance costs, which are several times higher than those of electric motors, are also a disadvantage.

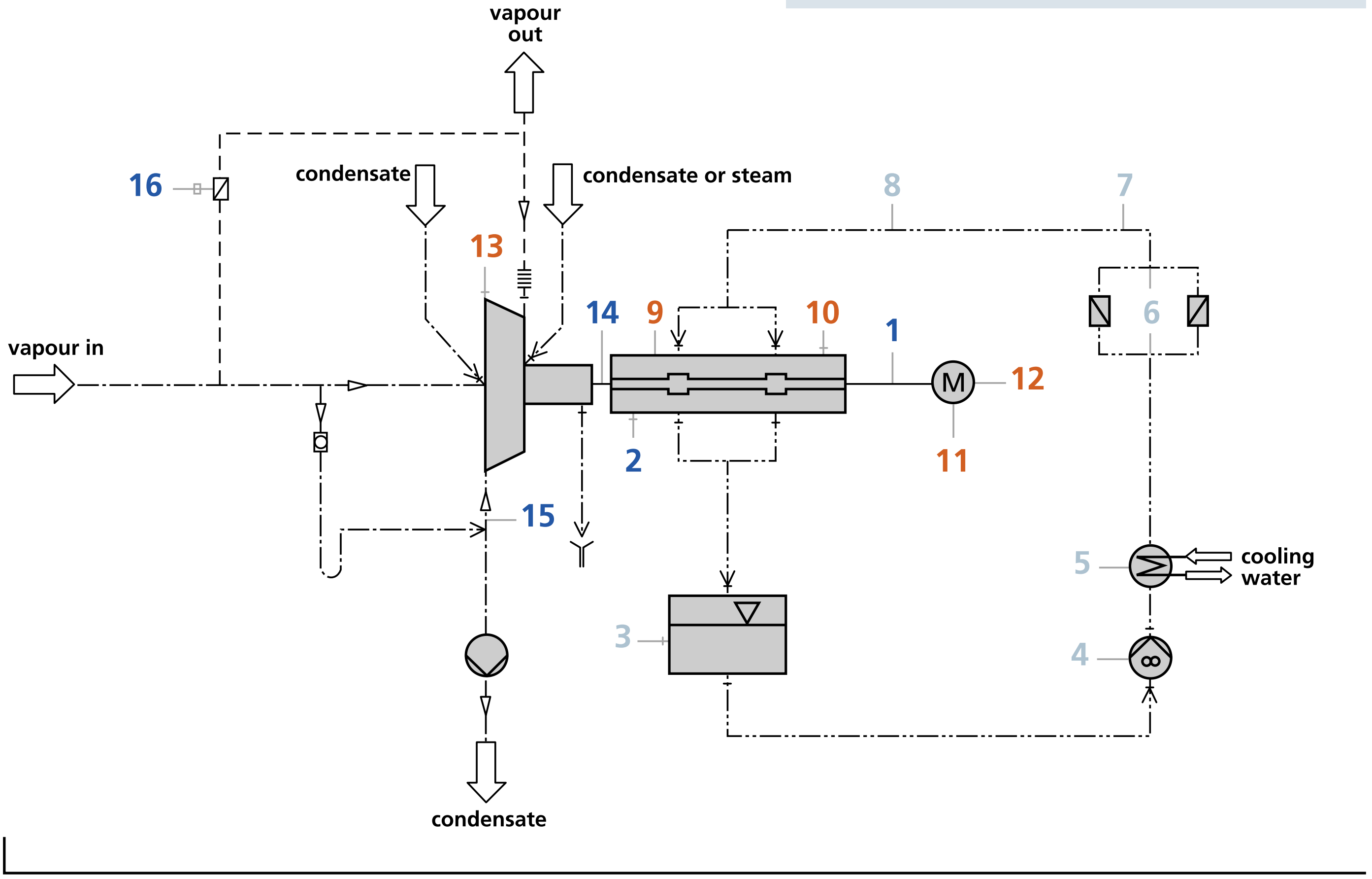
Steam turbines

The use of a variable-speed steam turbine as the prime mover of a compressor is sensible if the exhaust steam can be recovered. In this case, the relatively poor efficiency of a single-stage steam turbine, which may be used for price reasons, is of secondary importance.

Monitoring and Safety Equipment

A number of monitoring and safety systems are required to detect irregularities in compressor operation, to provide early warnings of wear and to prevent mechanical damage to the plant.

These are shown in detail in the example of a centrifugal fan:



1 Impeller speed

The speed is continuously measured by a revolution counter. The fan requires overspeed protection especially in the case of frequency converter operation. An alarm is given shortly before the maximum permissible speed is reached. When the maximum speed has been reached, the motor is automatically shut down.

2 Vibration monitoring

The vibration monitoring system monitors the dynamic behaviour of the rotating assembly. For this purpose, sensors are installed in the proximity of the bearings.

The vibration amplitude is determined by various factors, e.g. by:

- the relevant speed
- state of the bearings
- state of the impeller (incrustation / deposits)
- frequent changes in load required by the process

An alarm is given when the maximum permissible vibration is reached. Exceeding the maximum limit leads to an emergency stop of the system.

3 Oil tank levels

The oil level in the lubricating oil reservoir is measured. An alarm is given when this falls to the minimum level.

4 Oil pump

The operation of the oil pump is monitored. Pump failure leads to an emergency stop of the fan. During normal fan shutdown, the oil pump remains in operation at least until the complete standstill of the rotating assembly.
For safety reasons, centrifugal compressors are equipped with an auxiliary oil pump in addition to the directly connected main oil pump.

5 Oil cooler

A heat exchanger, supplied with cooling water, is installed in the oil circulation line for oil cooling. A temperature control loop keeps the oil temperature constant.

6/7 Oil filter differential pressure

The oil filter pressure difference (6) is measured and an alarm is given when the limit is exceeded. The pressure in the oil system (7) triggers the emergency stop of the fan when this value falls below the minimum pressure.

8 Oil flow

In addition to oil pressure monitoring, the oil flow can also be monitored and used as a shutdown condition in special cases.

9/10 Shaft bearing temperatures

The fan shaft runs on two bearings in a single bearing housing. The temperatures of intact bearings are considerably lower than the maximum permissible values. When elevated temperatures are reached, first an alarm is given. The system is immediately stopped when t_{max} is reached in order to avoid damage to shaft and impeller.

11 Motor winding temperatures

The driving motor requires protection against overheating. For this purpose, driving motors are equipped with temperature sensors in order to measure winding temperatures at different places. Excessive temperatures lead to motor shut down.

12 Motor bearing temperatures

For larger motor powers, e.g. > 100 kW, it is advisable to measure and monitor the motor bearing temperatures.

13 Fan/compressor casing temperature

Due to the compression work, the compressor casing itself is also heated by the pumped medium. An excessive casing temperature might arise if:

- the suction pressure and, consequently, the density of the pumped medium is excessive $(p_{operation} > p_{design})$
- the compressor is operated without pumped medium
- the compressor operates in circulation mode (bypass valve of centrifugal compressor is open)

The temperature of the casing is recorded and monitored. Excessive casing temperatures first lead to an alarm and then to an emergency stop. Continuous condensate injection at the fan impeller inlet and, consequently, saturation of the vapour, limits excessive casing temperatures.

14 Shaft axial position indicator

In order to prevent major damage by gradual wear of the axial/thrust bearing, it is advisable in some cases to monitor the axial position of the shaft. If a limit is reached the compressor is automatically stopped.

15 Condensate drain

The casings of fans and especially of centrifugal compressors must be drained thoroughly in order to avoid damage to the impellers.
A condensate level monitoring system, which also triggers the emergency stop, is installed at the lowest point of the casing.

16 Surge protection for centrifugal compressors

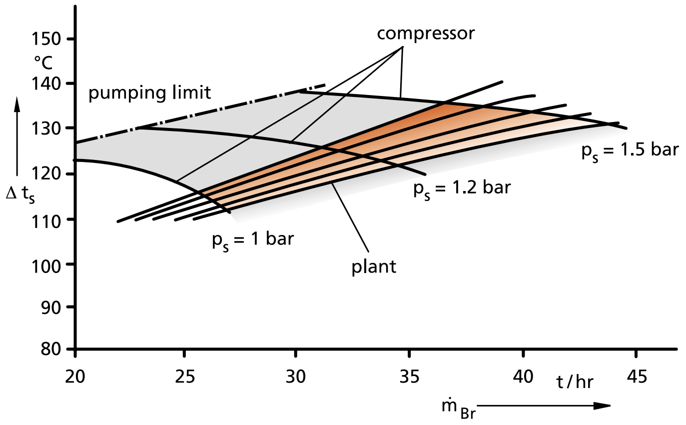
If the flow rate falls below the minimum value, e.g. during partial load operation, and thus below the stability limit of the compressor, the pumping direction of the vapour is momentarily reversed from the pressure side to the suction side. This surging leads to vibrations that may severely damage the machine.
For this reason, the machine is equipped with a surge limit safety system. If the flow rate falls below the safe pumping limit, the controller opens a bypass valve between the pressure line and the suction line in order to maintain an adequate flow rate.

Compressor Controls

Evaporation plants heated by mechanical vapour recompressors generally operate steadily to a limited extent, i.e. parameters such as mass flow rate, pressure and temperature fluctuate over time. Variations in evaporation rate (i.e. partial load operation) over a wide range are often desired. Therefore, different heat rates must be transferred. These changes in plant capacity are achieved by changing the temperature or pressure profiles.

The compressor design must take into account these variations in plant performance against the design duty. The operating behaviour of the plant is depicted in the so-called plant characteristics or performance curves. It shows the relationship between the necessary saturated steam temperature increase and the drawn-in vapour mass flow. The operating behaviour of the plant should be determined by tests to a large extent, or should at least be estimated.

The evaporator and compressor characteristic curves must correspond to each other for optimum operation of the vapour recompressor plant.



Specific changes in the flow conditions on the suction or pressure side of the compressor, for instance the suction pressure, allow the control concept to be varied. A variety of control concepts based on different performance criteria are available.

The following methods are preferred:

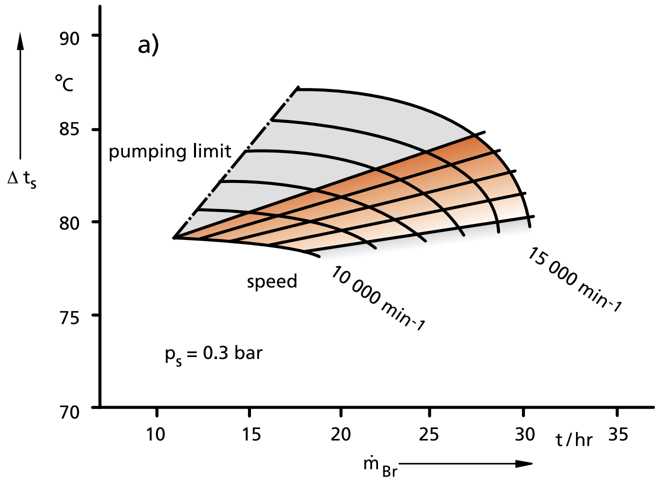
Single-stage centrifugal compressor

a) Speed control

Controlling the impeller speed and, consequently, the circumferential speed can influence the volumetric flow rate and the compression ratio. For speed control, a three-phase asynchronous motor equipped with a frequency converter is most commonly used. Especially for steep characteristic curves, i.e. for large changes in pressure against small changes in volume flow, control by continuous speed adjustment is advantageous.

The advantages of frequency converter operation are:

- Depending on the design, the motor can be operated at 20 to 60 % above its nominal speed so that a step-up gear is not required in most cases.
- A start-up coupling is not required.
- By limiting the starting current, the supply mains is not overloaded during start-up.
- Favourable partial load efficiencies are achieved.



b) Vane control

The vane control principle allows changes to the flow characteristic of the impeller. For this purpose, inlet guide vanes are installed in the suction nozzle of the compressor. The inlet guide vanes are adjusted from the outside by means of a drive. Whereas the compressor speed remains constant, the efficiency and performance of the impeller is changed.

Vane control is advantageous for characteristic plant curves based on considerable pressure changes in relation to the displaced volume. This results in a large control range and good partial load efficiencies.

c) Diffuser control

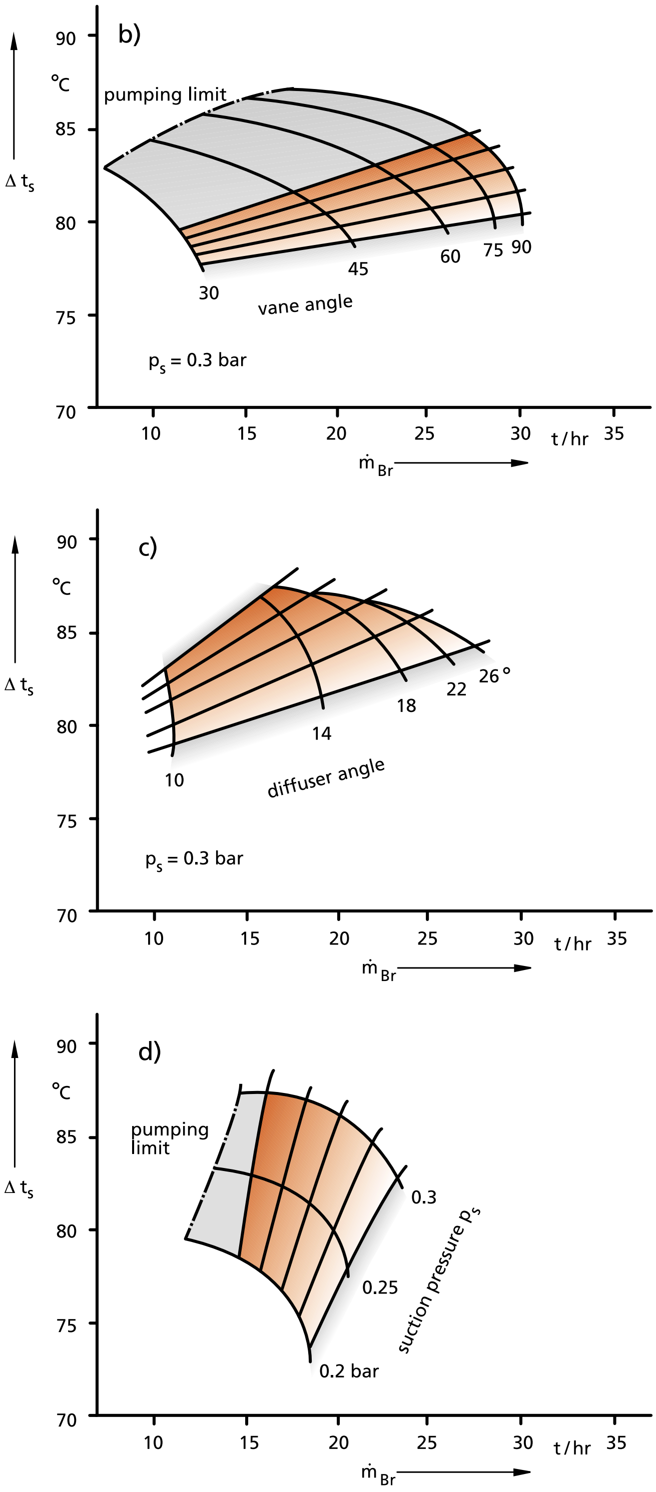
Adjustable vanes in the diffuser ensure a large change in the mass flow rate with low efficiency decrease and flat characteristic plant curves.

Diffuser control is used if the necessary temperature profile in the evaporator must remain approximately constant.

d) Inlet pressure control

If the process can take place at different temperatures, and the plant is not thermodynamically connected with other plants, the simple concept of controlling the pressure at the inlet by adjustments to the process parameters can be used. This control system ensures maximum variations in the mass flow rate, by changing the steam density at the evaporator separator, within the lower and upper process temperature limits. In many cases, a sufficiently large control range of the plant can be achieved in this way without special mechanical changes being necessary. Inlet pressure control can also be combined with one of the mechanical types of control, thus providing a particularly large control range.

Another possibility is the control of centrifugal compressors by throttling the suction line, thereby changing the mass flow rate. This type of control results in unfavourable partial load efficiencies.



Characteristic curves of a plant with fouling on the heating surface and for a single-stage, centrifugal compressor at different inlet pressures p_s

$\dot{m}_{Br}$ - *vapour mass flow rate*

Δt_s - *saturated steam temperature increase*

p_s - *pressure at inlet*

fouling on heating surface

1-effect falling film evaporation plant for the concentration of industrial waste water from tank cleaning

evaporation rate:
4.5 t/hr
final concentration:
40 % TS
compressor coupling power:
74 kW
drive:
frequency controlled electric motor

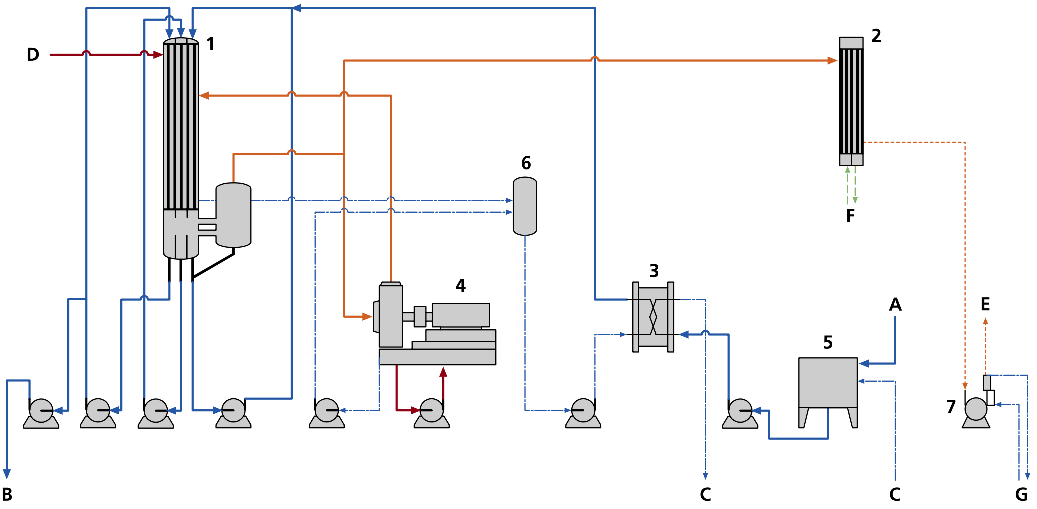
1-effect falling film evaporation plant with wrap-around separator and downstream high concentrator for various types of dairy and whey products

evaporation rate:
approx. 40 t/hr depending on the product
compressor coupling power:
390 kW

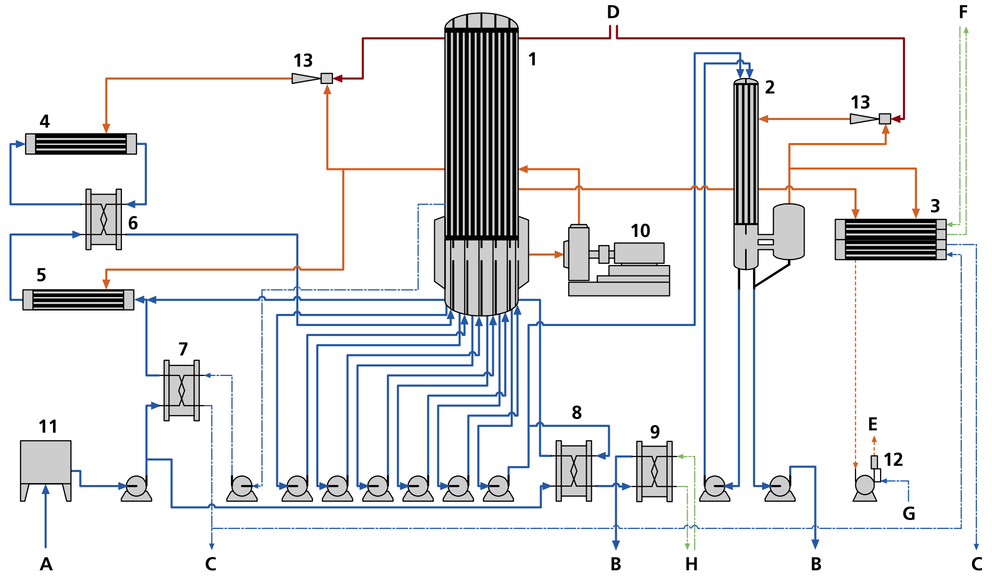
This plant concept is used for evaporation rates of 3 to approx. 55 t/hr water evaporation.

Evaporation Plants with Centrifugal Fans

1	*falling film evaporator*
2	*condenser*
3	*plate heat exchanger pre-heater*
4	*vapour recompressor (centrifugal fan)*
5	*feed tank*
6	*condensate collecting tank*
7	*vacuum pump*
A	*product*
B	*concentrate*
C	*condensate*
D	*live steam*
E	*deaeration*
F	*cooling water*
G	*service water*



1	*falling film evaporator*
2	*high concentrator*
3	*condenser*
4, 5	*pre-heaters*
6-9	*plate heat exchangers*
10	*vapour recompressor (centrifugal fan)*
11	*feed tank*
12	*vacuum pump*
13	*steam jet vapour recompressor*
A	*product*
B	*concentrate*
C	*condensate*
D	*live steam*
E	*deaeration*
F	*cooling water*
G	*service water*
H	*chilled water*



3-effect falling film evaporation plant consisting of 2 pre-evaporator effects heated by mechanical vapour recompressor and a finisher, heated by thermal vapour recompressor

evaporation rate:
50 t/hr
concentration range:
30 - 48 % TS
steam consumption:
15.5 t/hr of 38 - 11 bar (g) turbine
3.3 t/hr of 11 bar (g) steam jet
vapour recompressor
compressor coupling power:
730 kW

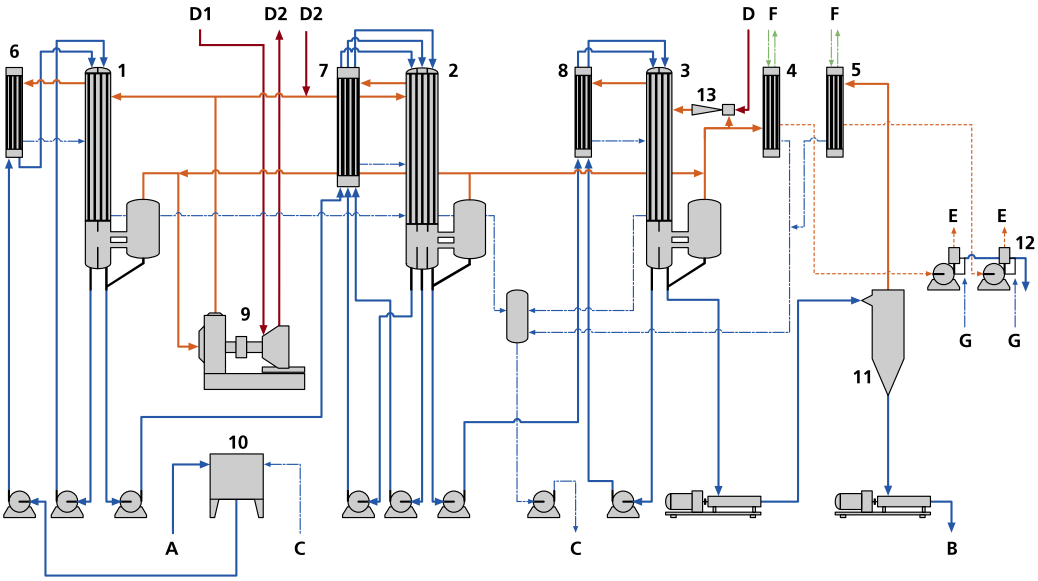
The centrifugal fan for vapour recompression is driven by a steam turbine.

1-effect falling film evaporation plant for wheat starch waste water. The plant can be operated as 1-effect system or as 2-effect system

evaporation rate:
approx. 17 / 33 t/hr
concentration range:
9 - 15 % TS
compressor coupling power:
1-effect operation: 230 kW
2-effect operation: 420 kW

1, 2	*falling film pre-evaporators*
3	*high concentrator*
4, 5	*condensers*
6-8	*pre-heaters*
9	*vapour recompressor (centrifugal fan)*
10	*feed tank*
11	*flash cooler*
12	*vacuum pumps*
13	*steam jet vapour recompressor*

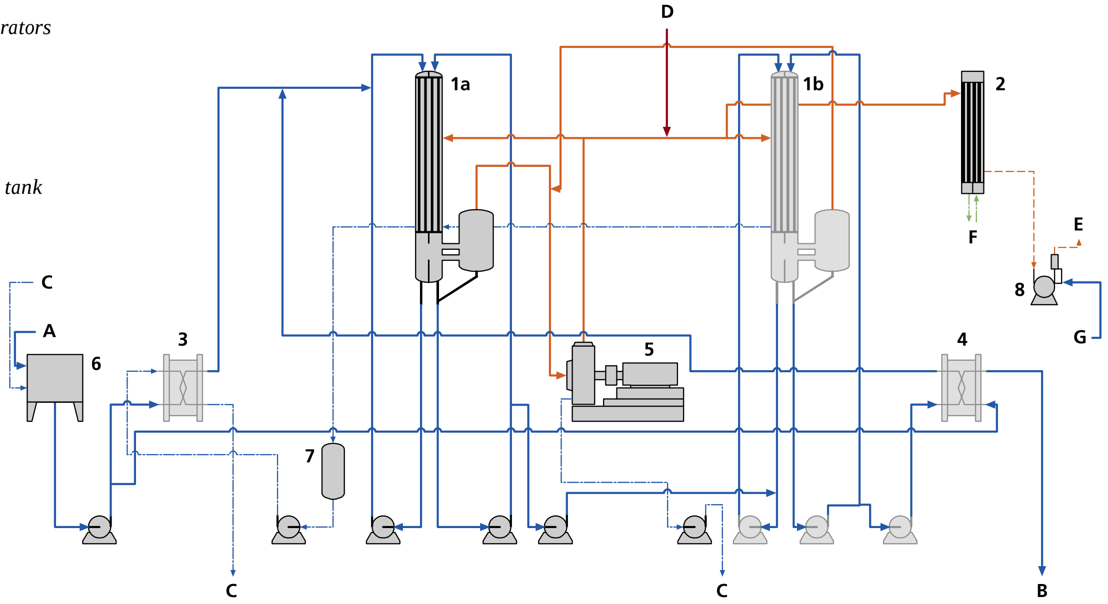
A	*product*
B	*concentrate*
C	*condensate*
D	*live steam*
D1	*high pressure steam*
D2	*low pressure steam*
E	*deaeration*
F	*cooling water*
G	*service water*



1a, 1b	*falling film pre-evaporators*
2	*condenser*
3, 4	*plate heat exchangers*
5	*vapour recompressor (centrifugal fan)*
6	*feed tank*
7	*condensate collecting tank*
8	*vacuum pump*

A	*product*
B	*pre-concentrate*
C	*condensate*
D	*live steam*
E	*deaeration*
F	*cooling water*
G	*service water*

The light grey equipment shows the planned plant expansion



Falling film evaporation plant for different glucose solutions, consisting of a 2-effect falling film pre-evaporator heated by mechanical vapour recompressor and a 2-effect falling film finisher in counter-flow arrangement, equipped with thermal vapour recompressor and flash cooler

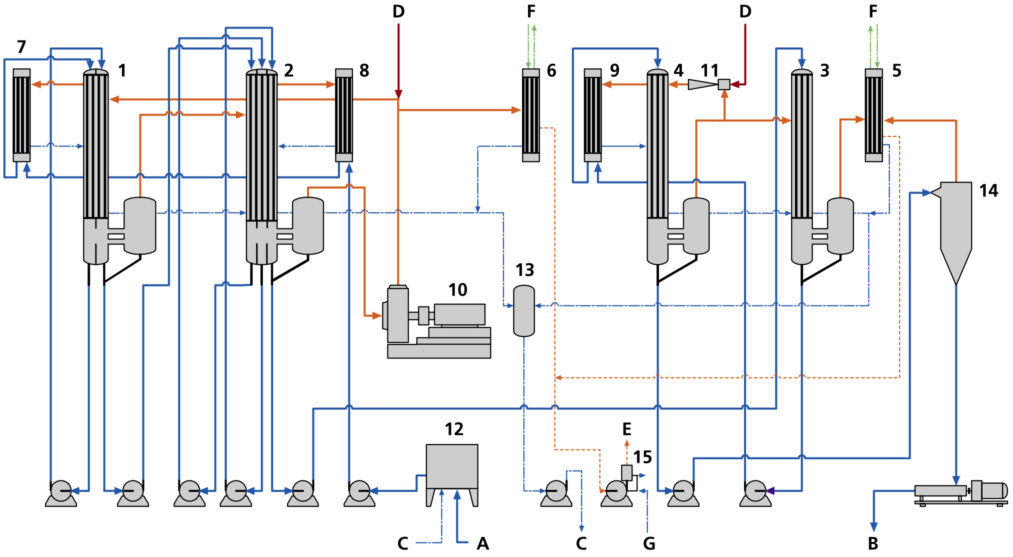
evaporation rate:
19 t/hr
concentration range:
32 - 83 % TS
steam consumption:
850 kg
compressor coupling power:
325 kW

3-effect falling film, forced-circulation evaporation plant consisting of 2 parallel evaporator effects for the pre-concentration of caprolactam water

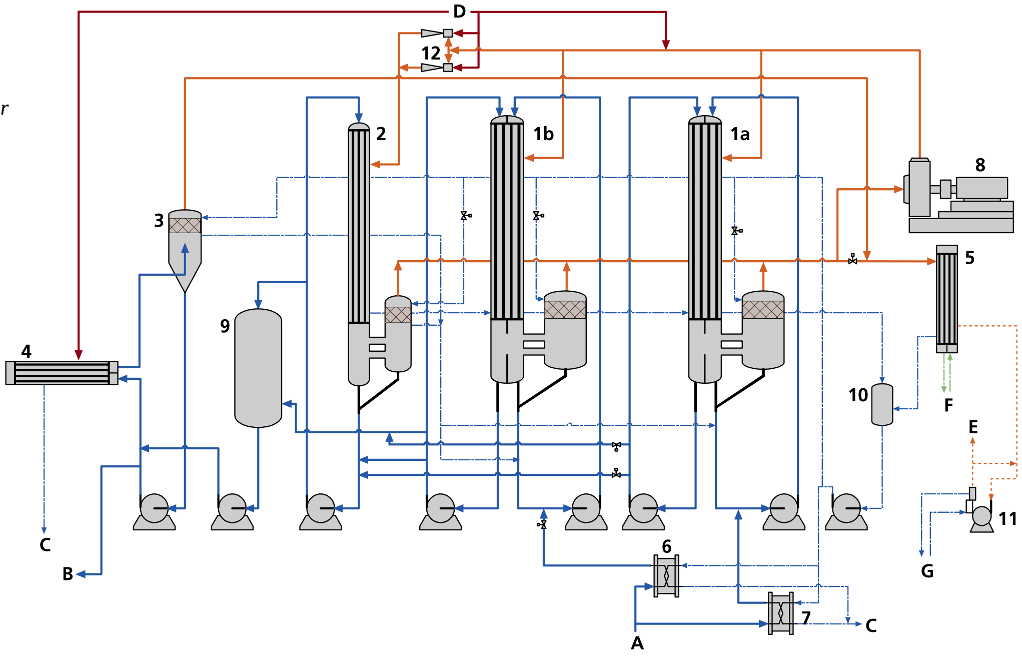
evaporation rate:
14 t/hr
concentration range:
7 - 95 % TS
steam consumption:
900 kg
compressor coupling power:
250 kW

Evaporation Plants with Centrifugal Compressors

1, 2	*falling film pre-evaporators*
3, 4	*falling film finishers*
5, 6	*condensers*
7-9	*pre-heaters*
10	*vapour recompressor (centrifugal compressor)*
11	*thermal vapour recompressor*
12	*feed tank*
13	*condensate collecting tank*
14	*flash cooler*
15	*vacuum pump*
A	*product*
B	*concentrate*
C	*condensate*
D	*live steam*
E	*deaeration*
F	*cooling water*
G	*service water*



1a	*falling film evaporator*
1b	*falling film evaporator*
2	*falling film evaporator*
3, 4	*forced circulation evaporator*
5	*condenser*
6, 7	*plate heat exchangers*
8	*vapour recompressor (centrifugal compressor)*
9	*buffer tank*
10	*condensate collecting tank*
11	*vacuum pump*
12	*steam jet vapour recompressor*
A	*product*
B	*concentrate*
C	*condensate*
D	*live steam*
E	*deaeration*
F	*cooling water*
G	*service water*



Our Range of Products in Summary

Evaporation plants

for the concentration of all types of liquid food, organic and inorganic solutions, waste water and other types of liquid products by means of thermal or mechanical vapour recompressors, single-effect or multi-effect systems, with additional equipment for heating, cooling, degassing, crystallization, rectification etc.

Membrane filtration

for the concentration of liquid food, process water, organic and inorganic solutions and waste water; for the separation of impurities for upgrading and valuable material recovery; based on technology and references by **GEA Filtration**, Hudson/USA.

Distillation/rectification plants

for the separation of multi-component mixtures, e.g. for the recovery of organic solvents, the recovery, purification and dehydration of bioalcohol of different qualities etc.

Lines for the production of alcohol

from the treatment of raw material, fermentation, distillation to stillage concentration/drying

Plants for crystallization

of special products as well as waste water containing salts

Product studies, engineering

for plants included in our range of products

GEA Wiegand

Process Engineering
Division

TÜV CERT
DIN ISO 9001

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