Hydronic College

IMI Hydronic Engineering

Kontrolni krugovi za rashladne sisteme

Nastevski Gjorgjija

Engineering GREAT Solutions

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The hydronic imbalance problem

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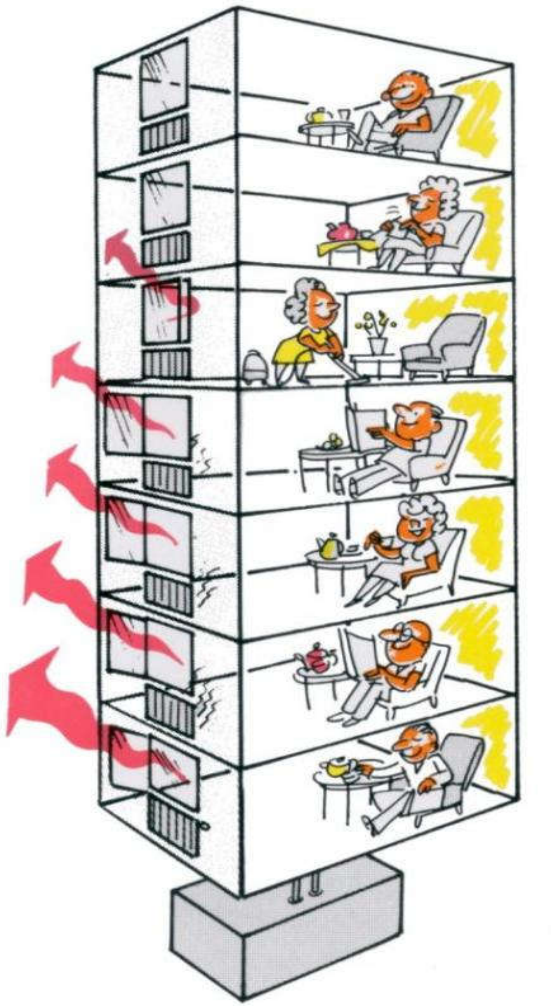

To avoid complaints from tenants

Pumps are:

- Oversized
- Pushed to maximum speed
- Replaced by more powerful pumps

  – Overflows are larger
  – Less underflows

  – Installation works globally in overflow
  – Pump head is vastly increased
  – Pumping cost is doubled to compensate a local underflow of 20%

Variable load in cooling

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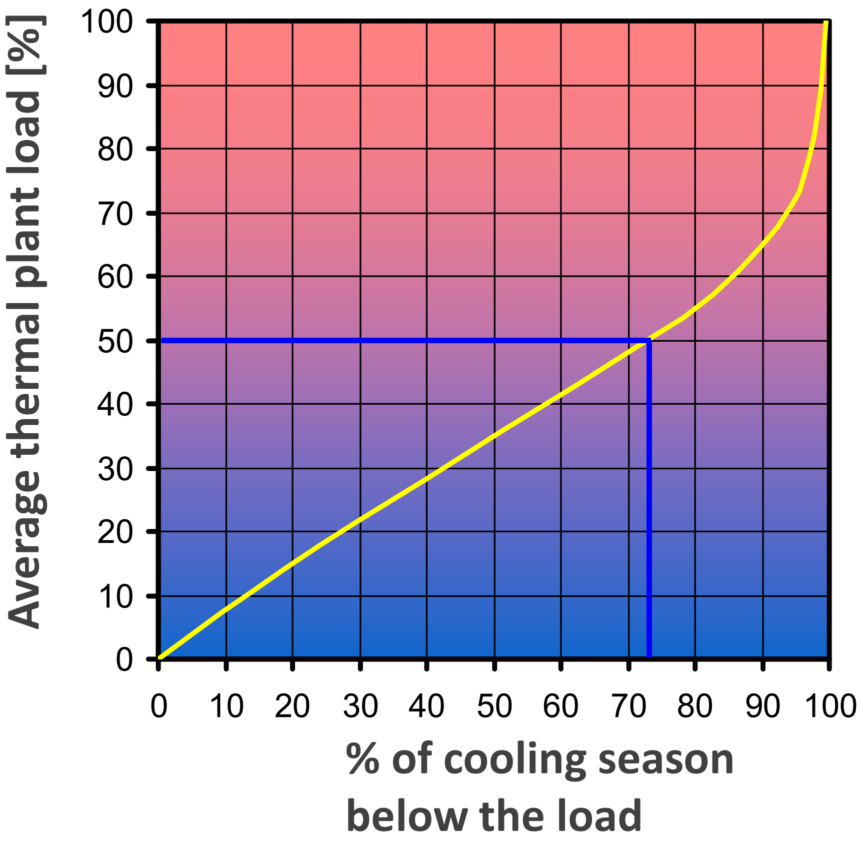

Paris

More than 72% of the cooling season the load is lower than 50%

Load variations are heavily influenced by:

- Sunshine effects (up to 750 W/m² for a West façade in July around 4pm at 50° North)
- Building occupancy (1 sitting person: ±110 W, computers …)

Constant or variable flow distributions

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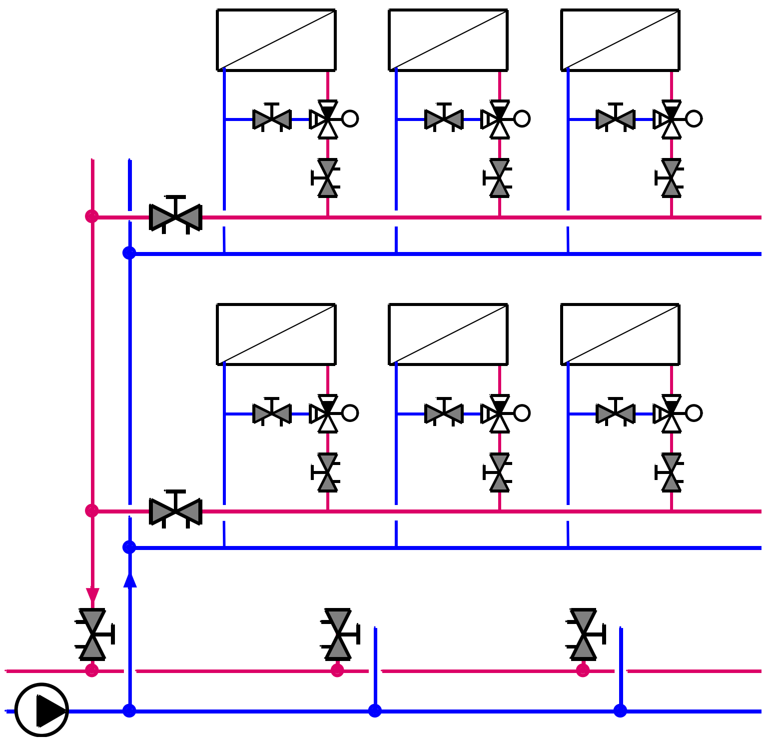
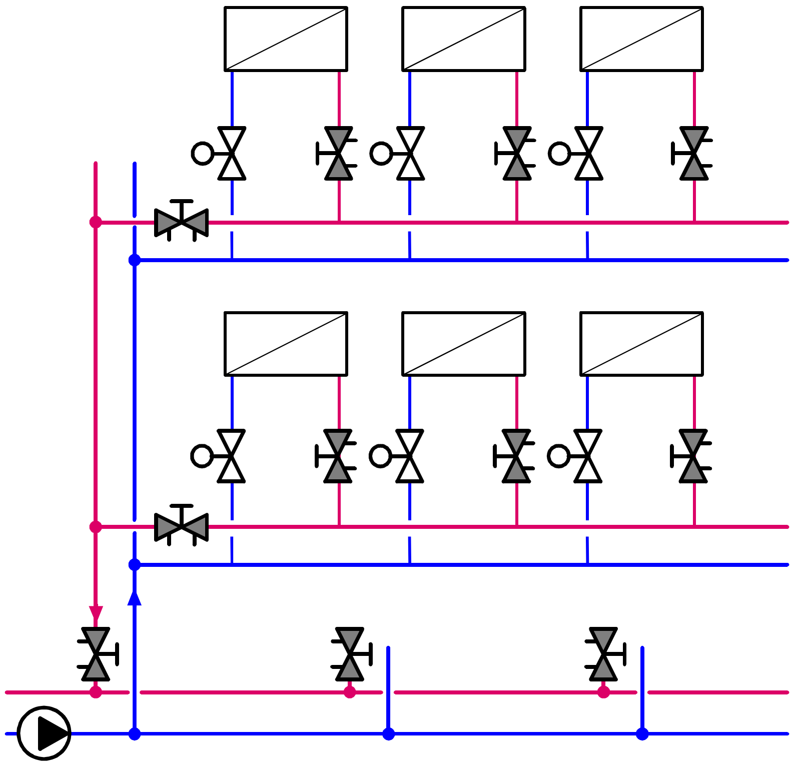
Variable flow is well suited to load variations

Constant flow



Variable flow



Variable flow – advantages and drawbacks

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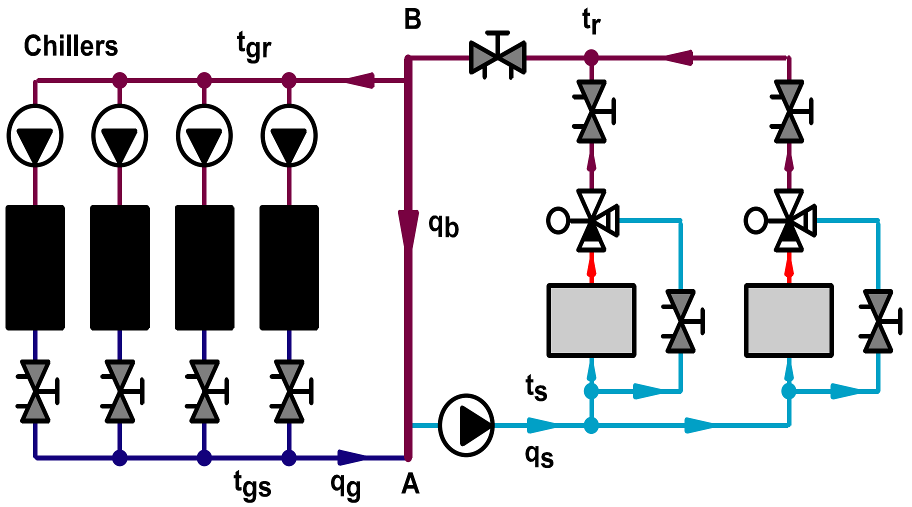
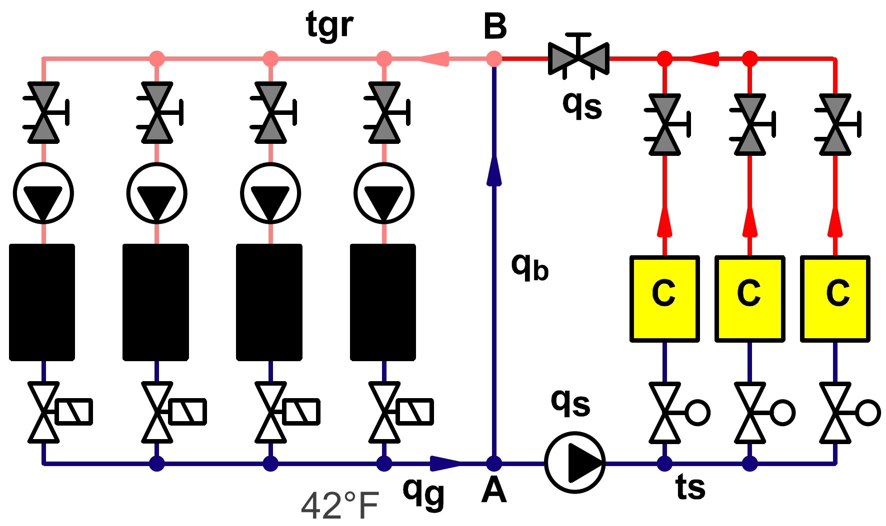
- Reduced pump energy consumption
- Compatibility between production and distribution flows
- Easy to work with a diversity factor
- Return temperature lower





- Variable authority of the ATC valves
- Need to ensure a minimal flow

Differential pressure variations

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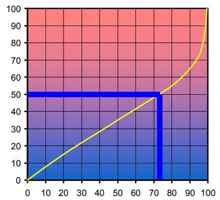
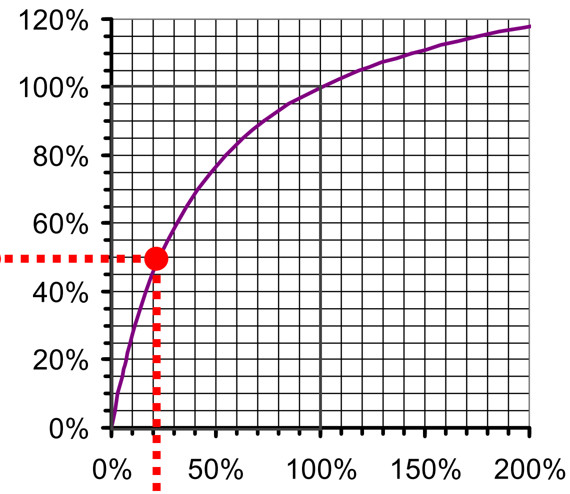
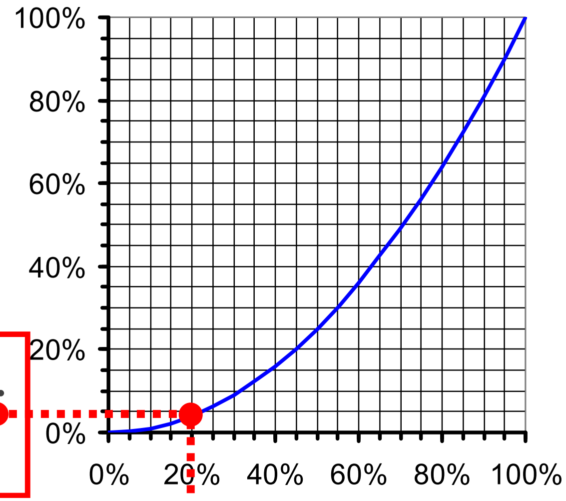


Half-load and below represents a considerable fraction of the cooling/heating season

Emission



Dp



$$\Delta P \propto q^2$$

50 % load

4% press. drop

At constant supply water temperature

20 % flow

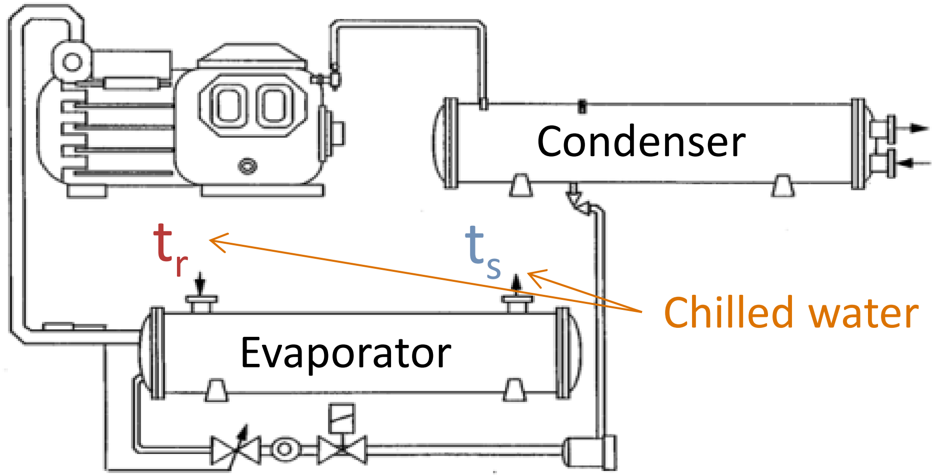
Pressure drops reduced to 4% of their initial value.

The pump head applies itself almost entirely on the 2-way automatic temperature control valves.

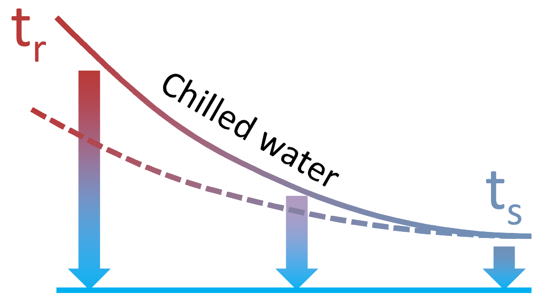
Chillers

- Energy Efficiency Ratio (EER) is used to indicate the chiller efficiency:

$$EER = \frac{cooling\ \ power\ [kW]}{absorbed\ \ electric\ power\ of\ \ unit\ [kW]}$$



- Heat transfer (and thus EER) is good when **Log Mean Temperature Difference** between water and refrigerant is kept high
  - Evaporator refrigerant temperature remains constant
  - Supply water temperature t_s is usually kept constant
  - Thus return water temperature t_r must be kept "high" to keep LMTD high

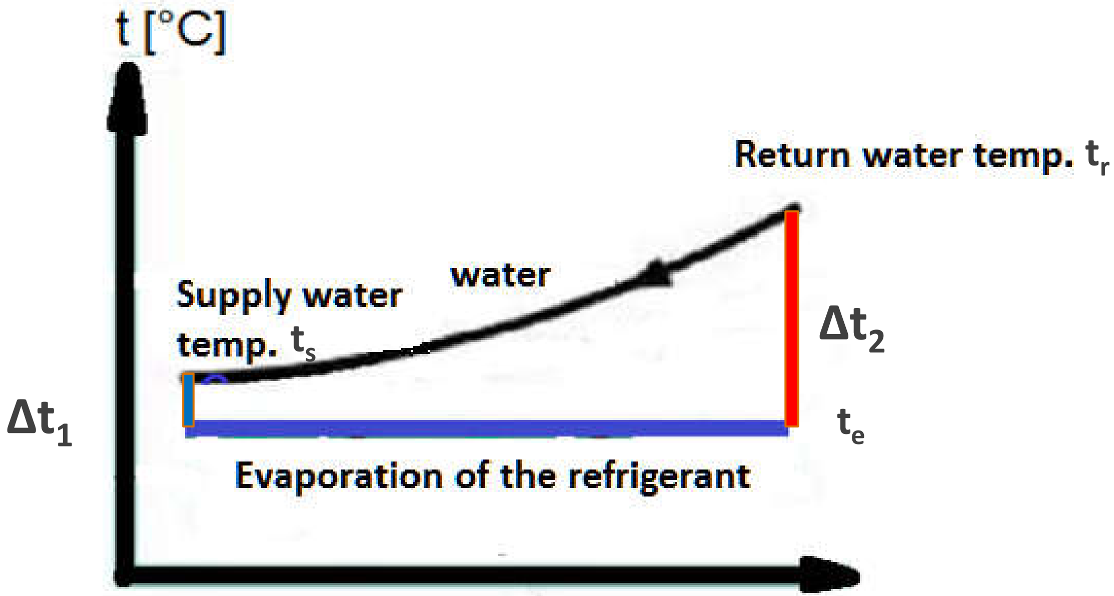


Refrigerant saturated suction temp.

- ***Keeping a high t_r (thus a high $\Delta t = t_s - t_r$) provides higher EER at partial load***

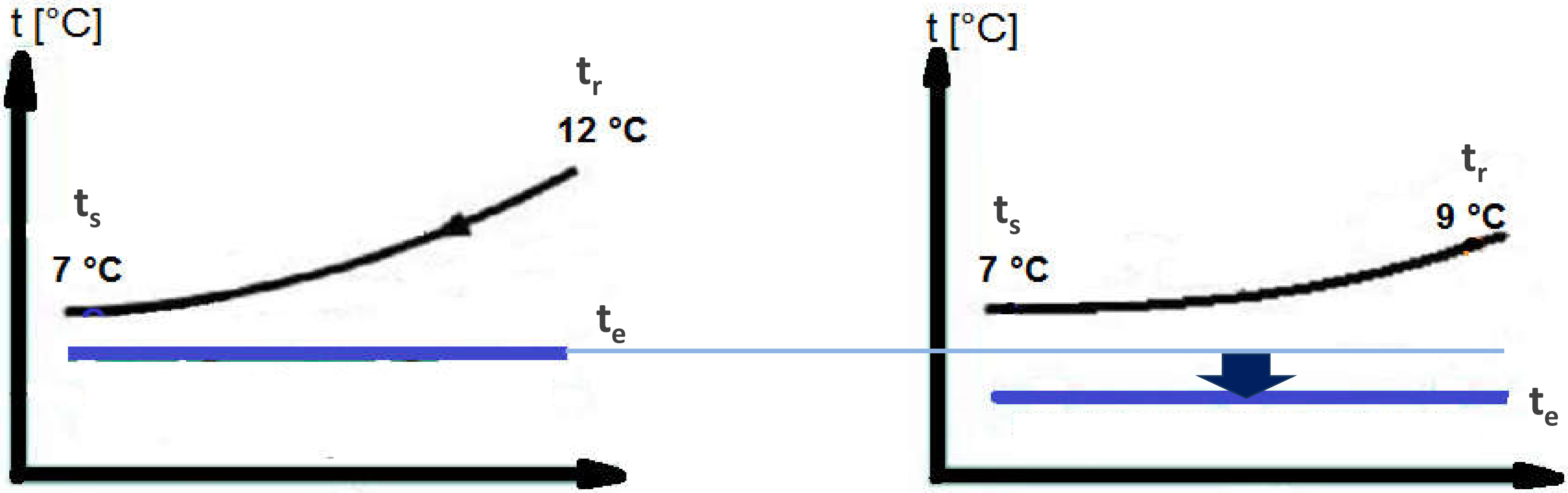
Constant supply water temperature (t_s) control !

Log Mean Temperature Difference



$$\Delta t_{\ln} = \frac{\Delta t_2 - \Delta t_1}{\ln \dfrac{\Delta t_2}{\Delta t_1}}$$

Chiller efficiency at the partial load

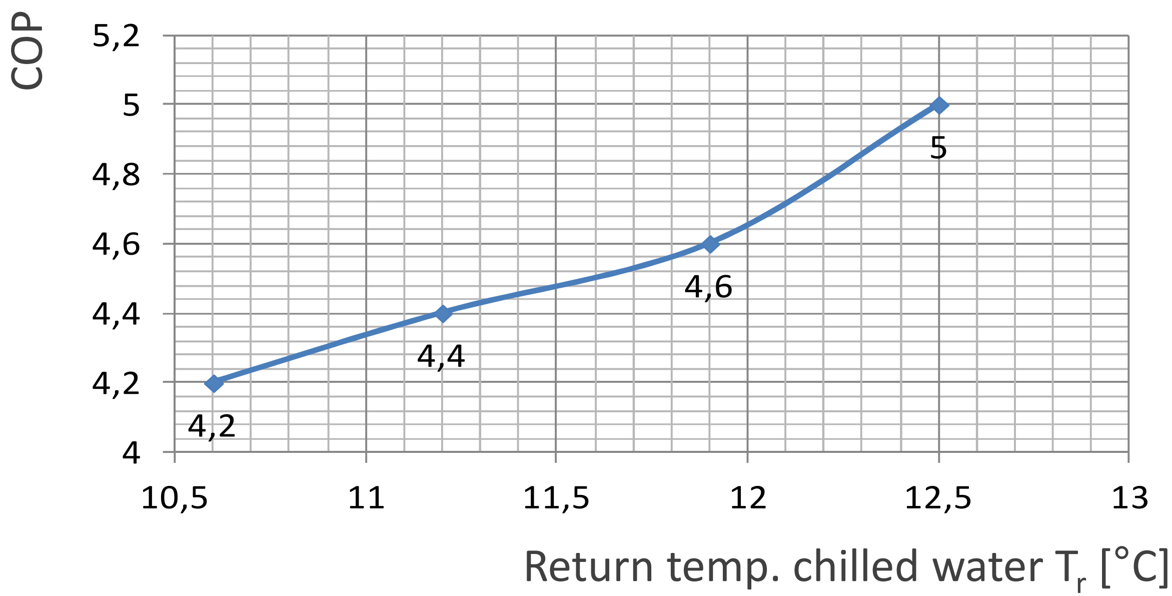


Design temperature regime: 7/12 °C

At smaller return water temperature (t_r) the LMTD is smaller, too. The smaller LMTD will be compensated by a lower evaporating temperature which causes a smaller EER coefficient!

Effect of a decrease of the return water temp. on COP

- **Example :**
  Chiller: 200 tons (703 kW)
  Water condenser temperatures: 29,5°/35°C
  Supply temperature of chilled water T_s : 7°C



- A **reduction of return temperature of chilled water** can lead to a **15% drop of the COP**

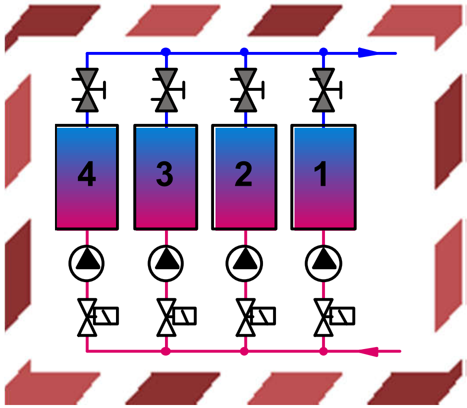
Cooling system circuits

1. Constant primary (chillers) – constant secondary (consumers network) flow

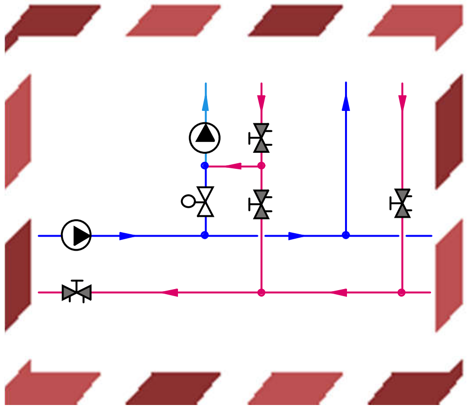
2. Constant primary (chillers) – variable secondary (consumers network) flow
(with pressure break tank)

3. Variable primary (chillers) – variable secondary (consumers network) flow, (VPF system)

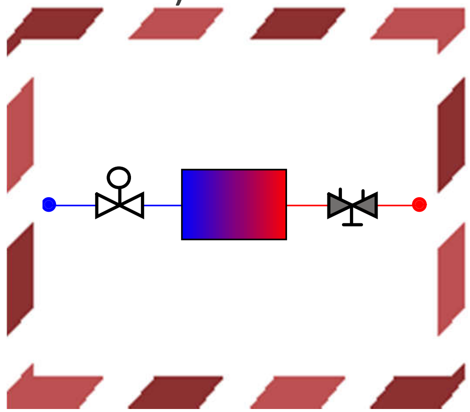
Primary (chillers)



Distribution



Secondary (consumers network)



Constant primary – constant secondary flow

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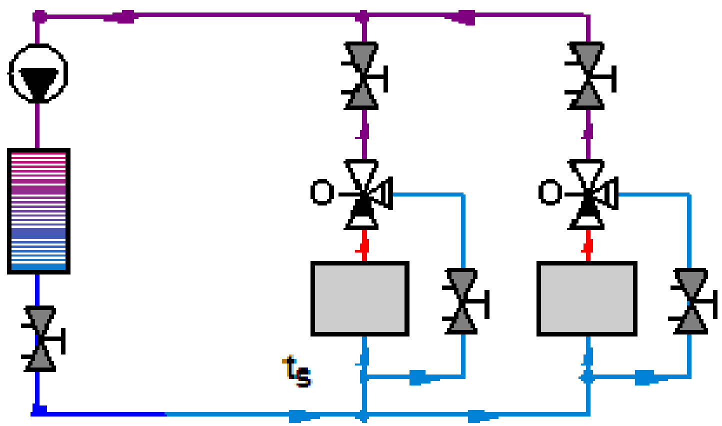
One chiller



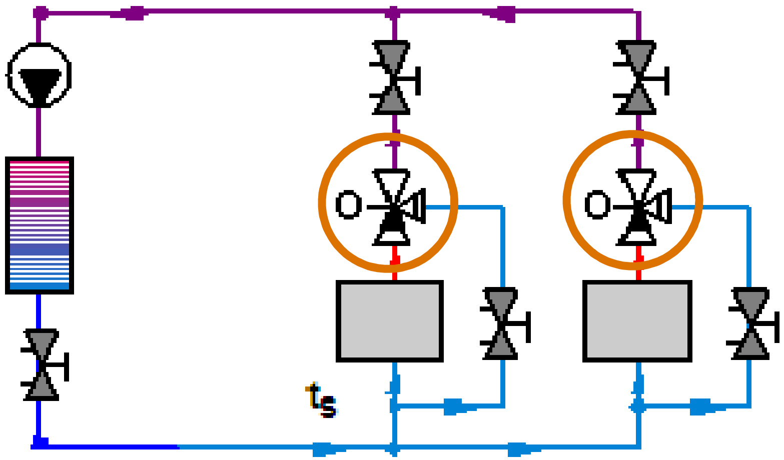
Properties

- **constant speed pump** on the primary side
- pump for **the total system pressure loss**
- **no pumping energy** saving
- **no diversity factor** using

Chiller efficiency:
Due to low return water temperature at the partial load, the EER coefficient decreases **(Low Temperature Syndrome).**

Used for small old and new type systems (until 40-50 kW)

One chiller



One chiller in the system with three way control valves

As the pressure drop through by pass of the three way control valve is smaller than the pressure loss of the consumers, you have to use:

- throttle valve in the by pass *or*
- three way valves with asymmetric k_{vs} value

Return water temperature

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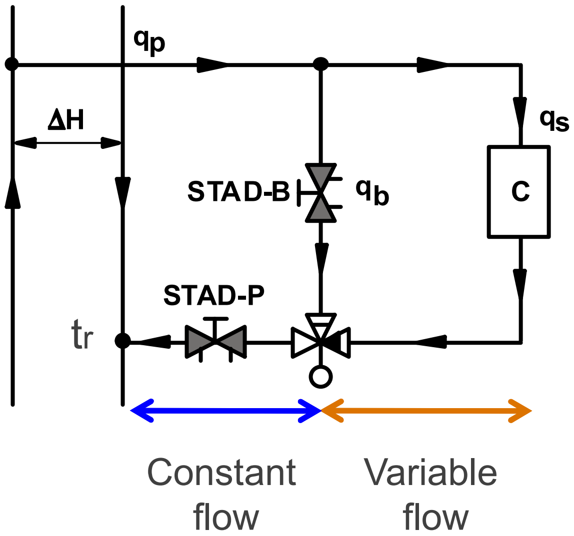
Three-way control valve



Low Temperature Syndrome.

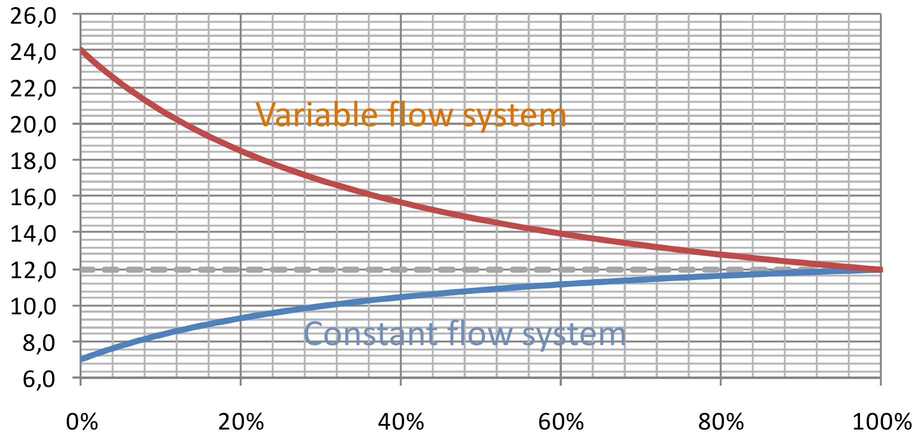
The return water temperature after the consumer increases, when the flow decreases

as the return water mixes with the supply water…

the return water temperature t_r will be decreased !

Cooling

Return water temperature t_r (°C)



Temperatures:
$t_s/t_r/t_i$ = 7/12/24°C

Flow in heat exchanger (%)

Several chillers in parallel connection

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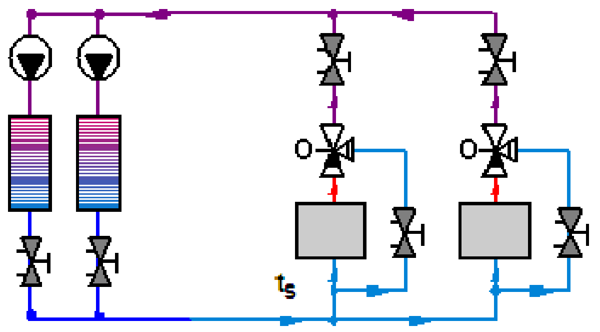
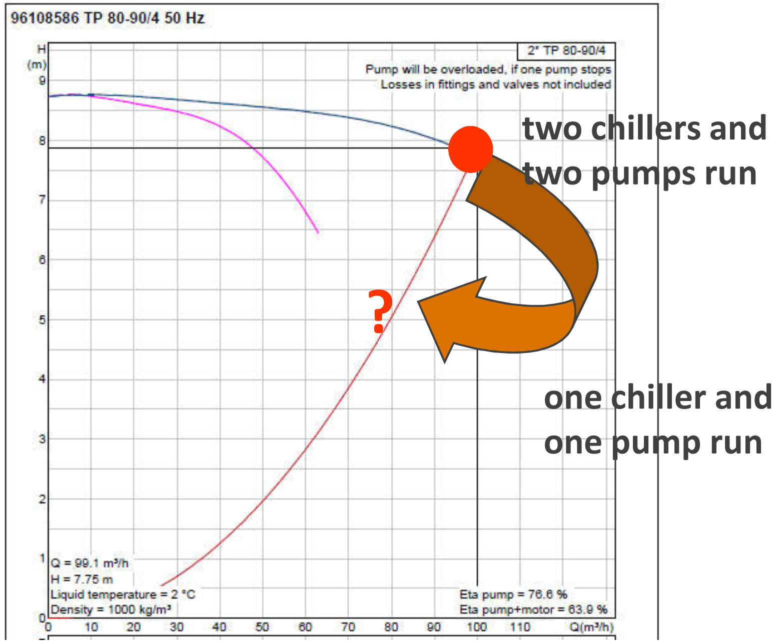




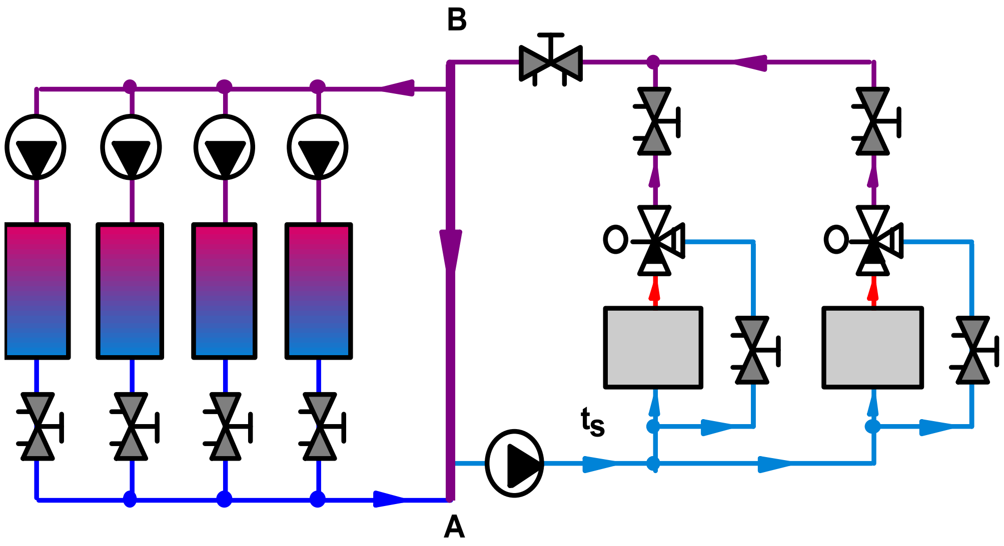
Properties

- **when only one chiller runs,** the work pointof the pump will be out of the range of the pump curve, when the consumers network pressure loss much larger than the chiller's pressure loss (*interactivity*)

- **the pump will be stoped or damaged**

Used for large and old type systems

Several chillers in parallel connection



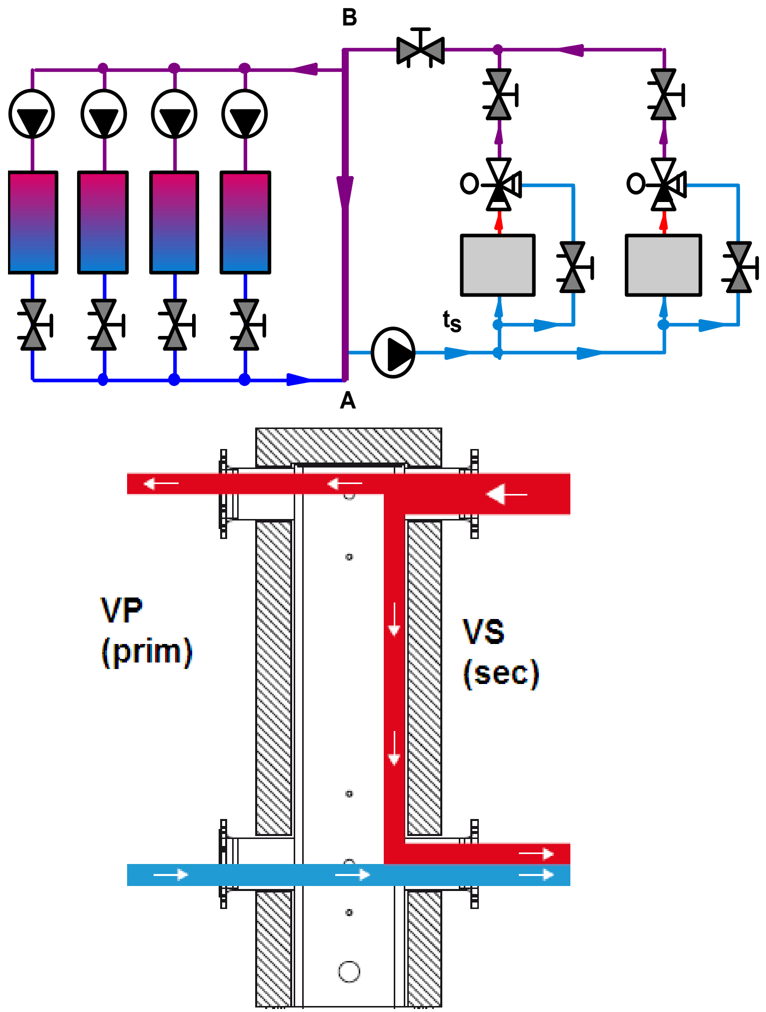
... with by pass

- to avoid the interactivity, you have to use by pass or pressure break tank

Chiller efficiency:
Due to the low return water temperature at the partial load the EER coefficient decreases **(low temperature syndrome).**

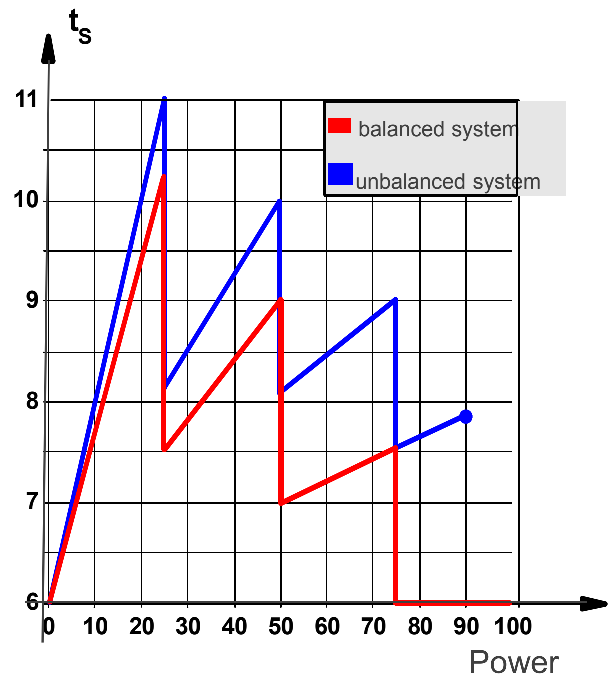
Used for large and old type systems

Several chillers in parallel connection



… supply water temperature

- **due to incompatiblity of primary and secondary** flows at partial load, you can achive the design t_s temperature, when all chillers run, only



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Constant primary – variable secondary flow

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Constant primary – variable secondary flow

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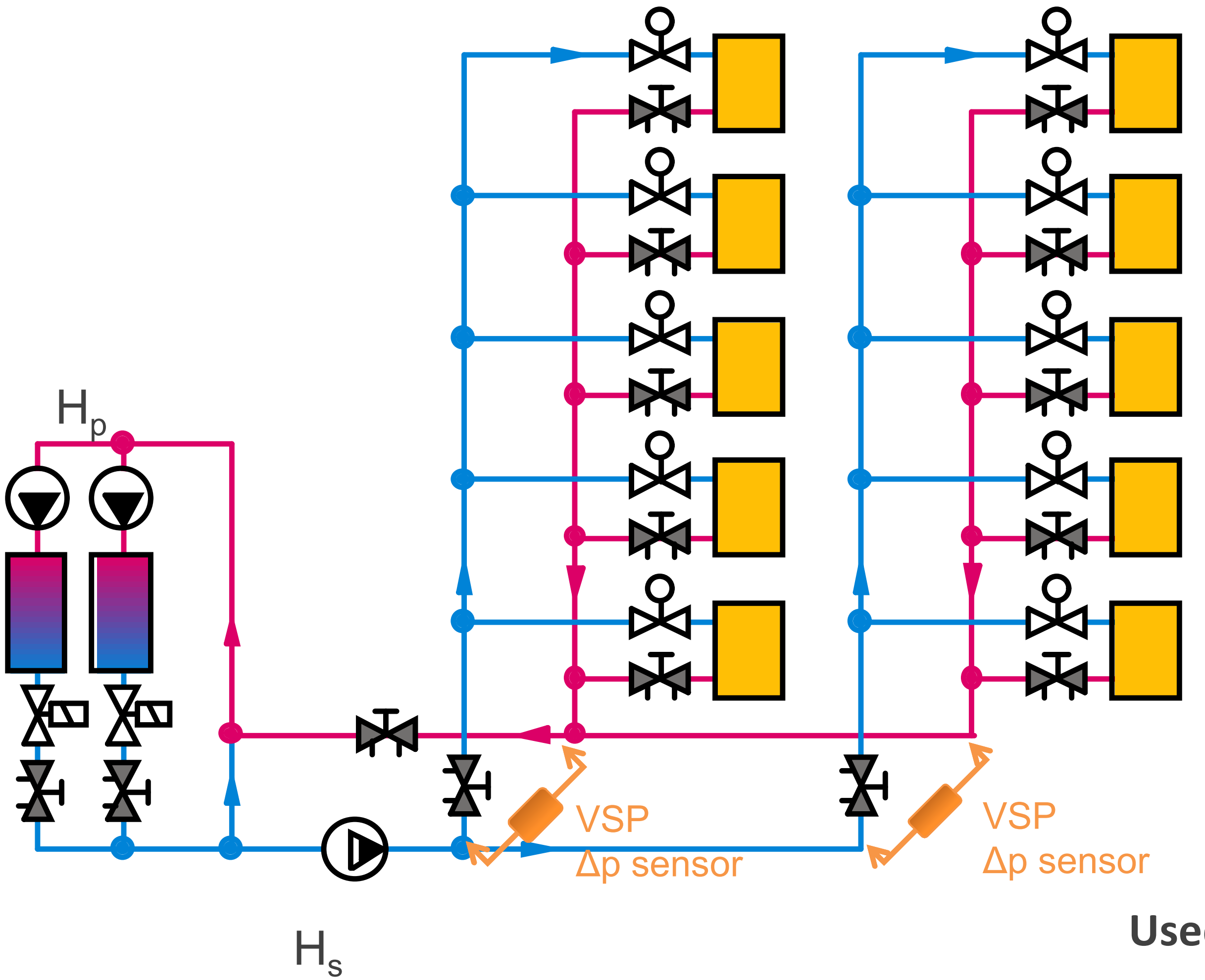


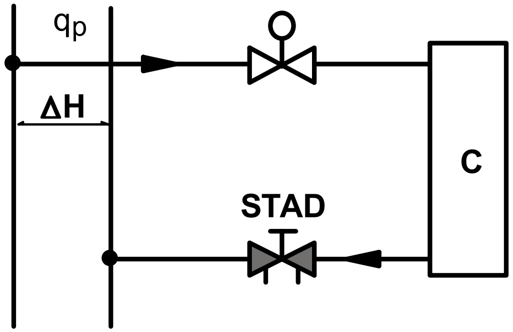
Properties

- constant flow at each chiller evaporator (variable flow by steps on primary side)
- constant speed pumps on primary side
- variable flow on the secondary side
- variable speed pumps on the secondary side:
  - allow pumping energy savings;
  - different options for locating the Δp sensor of the VSP

Used for small and large new type systems

Return water temperature

Two-way control valve (variable flow)



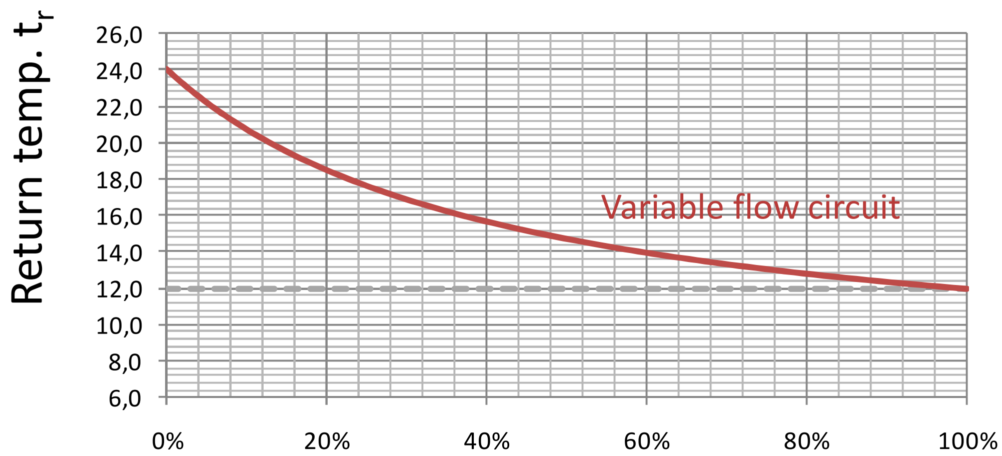
Modulating control

The Δt through a terminal unit increases when the flow reduces.

Thus the **return water temperature increases when the flow reduces**.

All benefits for chiller (?)

Cooling



Temperature regime:
ts/tr/ti = 7/12/24°C

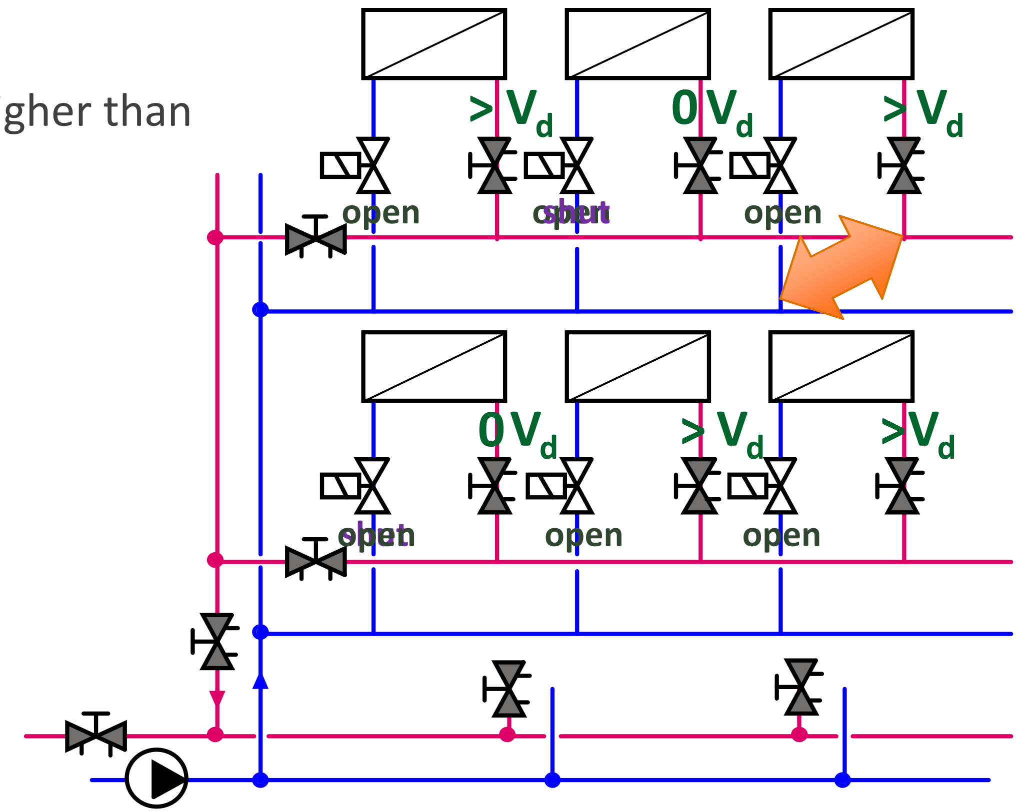
Flow through terminal unit

Interactivity

- When some CV are closed:
  - there is less total flow and Δp in piping
  - and thus more available Δp everywhere in the system
  - open valves receive a flow that is higher than design flow

- At partial load in the system, if a valve is open: $V_{actual} > V_{design}$

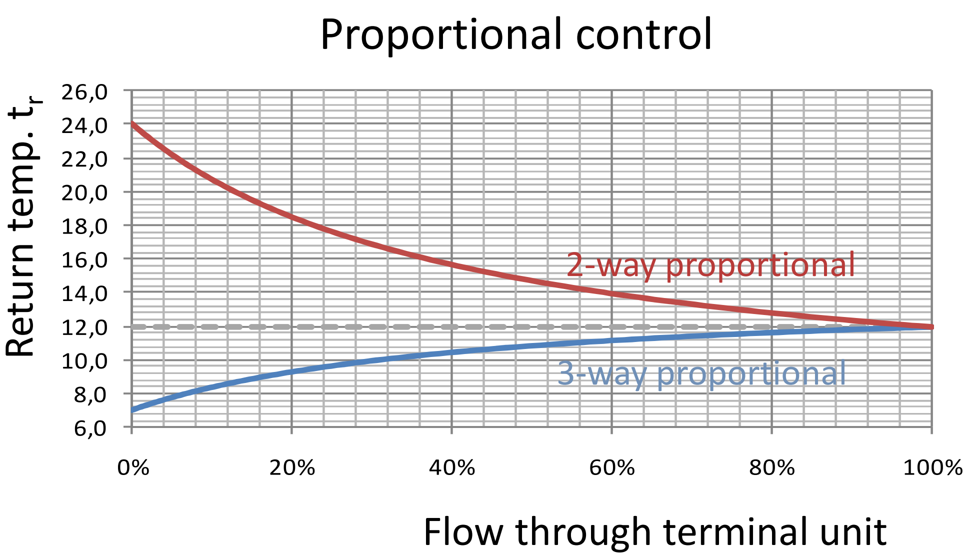
ON/OFF control

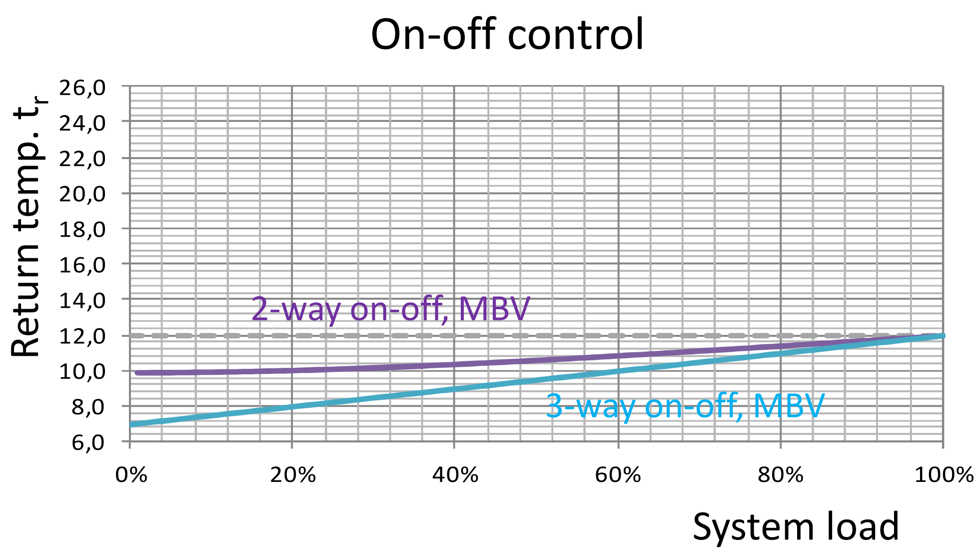


Return water temp. – proportional vs on-off control

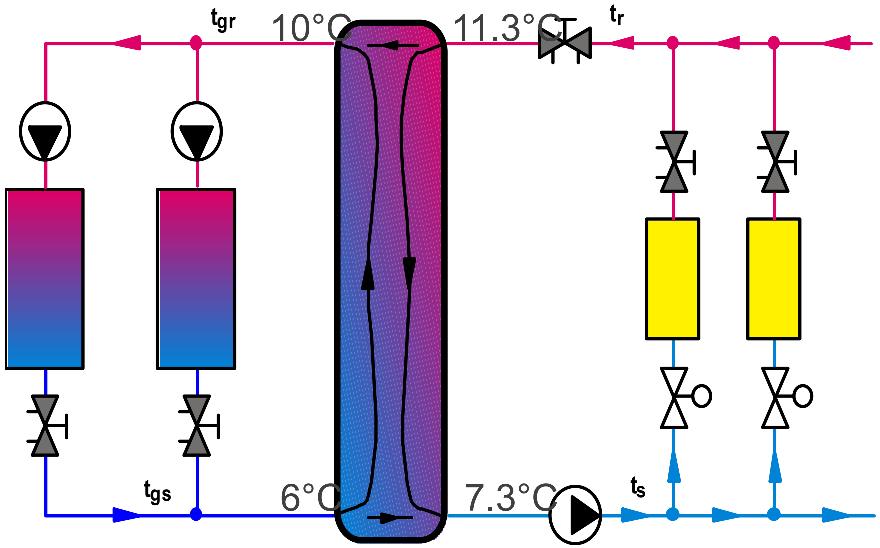
Cooling

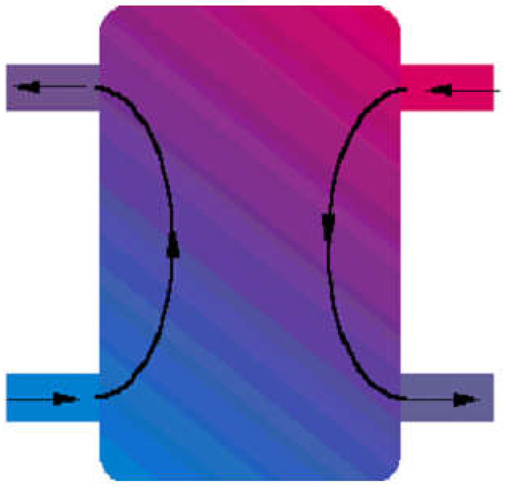
Temperature regime:
ts/tr/ti = 7/12/24°C





Pressure break tank



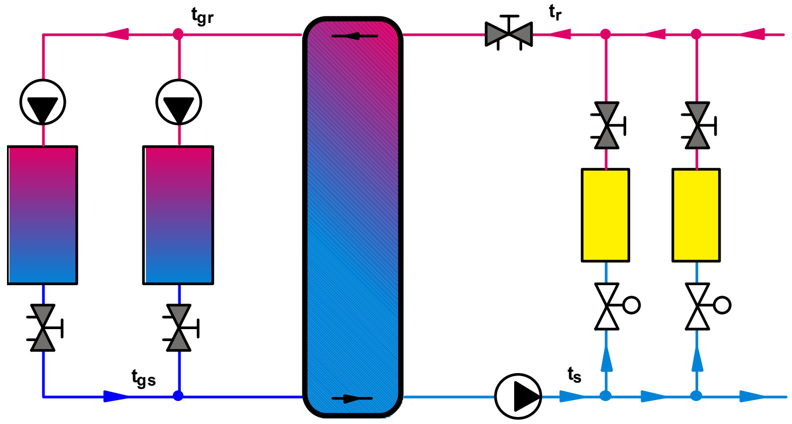


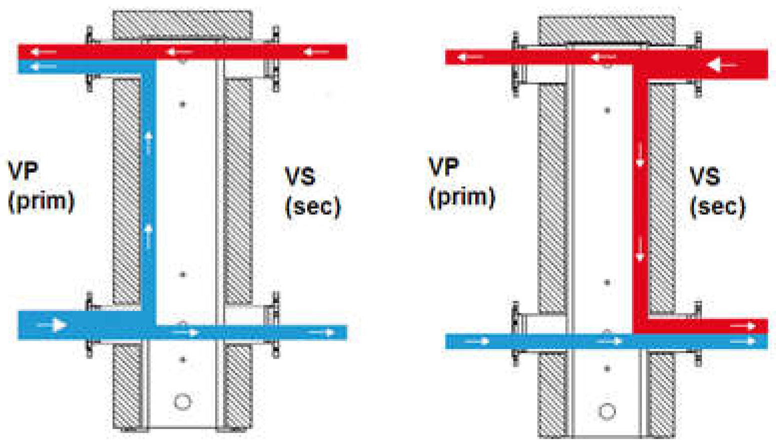
Oversized pressure break tank

-pressure break tank installed to avoid interactivity between the chillers

- when the tank diameter oversized, due to **bi-circulation**:
  - return temp. tgr decreases; chillers cannot deliver their full capacity.
  - supply temp. ts increases; terminal units cannot reach their full capacity.
- the chillers often start/stop

Flow compatibility in the pressure break tank





Flow compatibility at partial load

-to achive the compability of primary and secondary flows at partial load (to avoid the Low Temperature Syndrome), recommended to use more than one chiller on the primary side

Chiller efficiency:
If they can avoid the tgr tempreture's decrease (compatibility of flows!),
they can avoid the **Low Temperature Syndrome!**
EER coefficient will be closed to optimum at each load.

Variable primary– variable secondary flow

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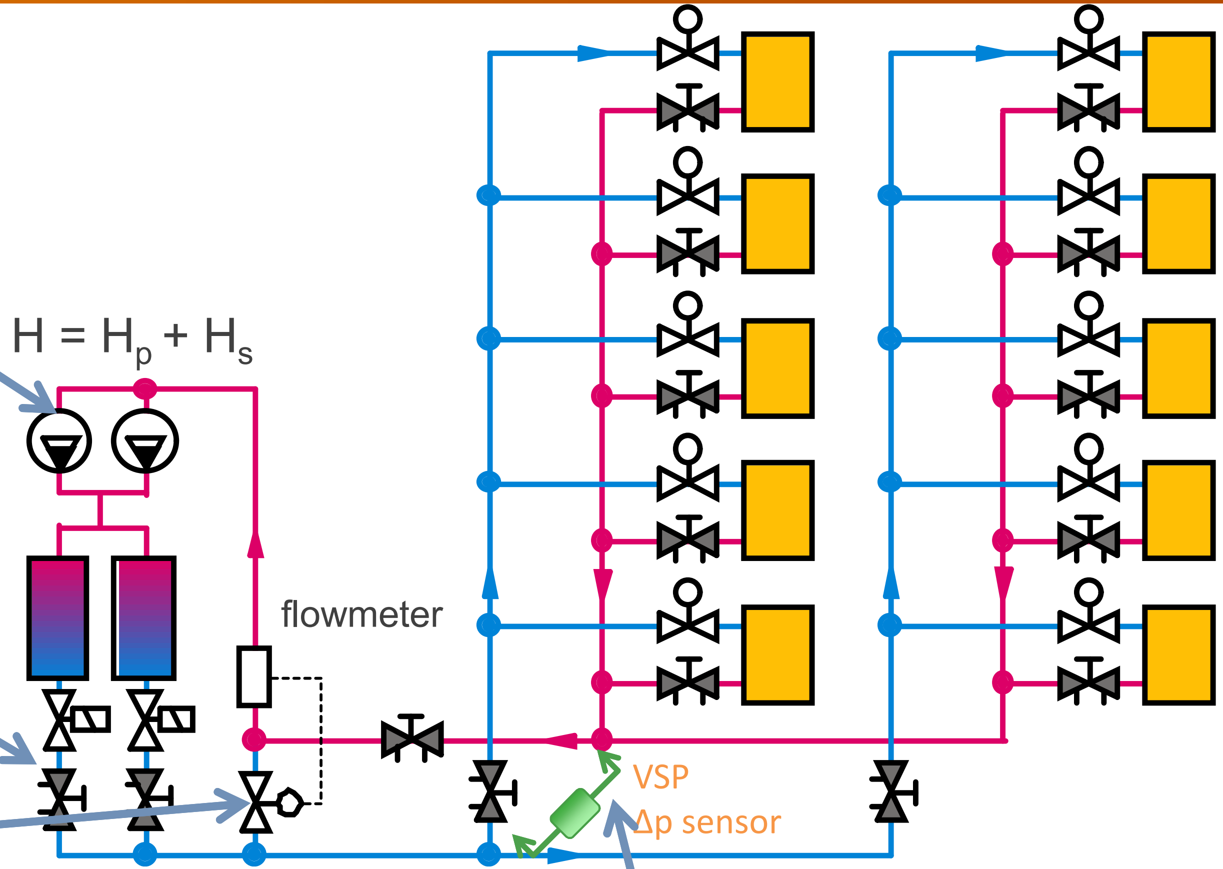
System

VSP primary pumps group circulate water throughout the entire system at variable flow

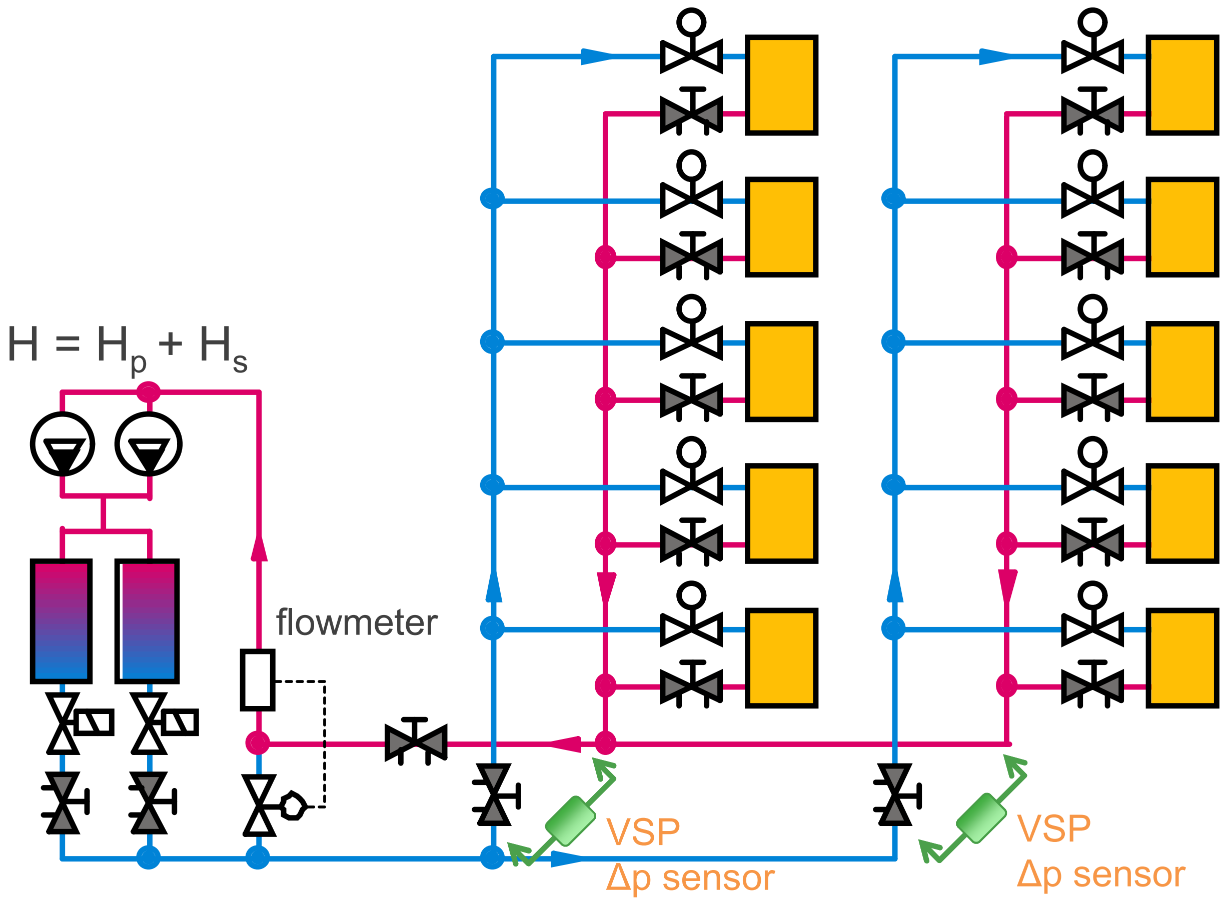
They have to use **manual balancing valves at chillers**

At low loads, the bypass valve opens to maintain the primary flow above the minimum evaporator flow limit



VSP primary pumps are operated to maintain a **constant Δp at a specific point in the system**

Properties



- variable flow in the distribution
- variable flow in chillers
- but, the bypass control valve maintains the chiller flow rates above the minimum limit of the evaporators
- variable speed pumps **group** on primary side:
    - sized for total plant head
    - headered for "flow-boosting" in evaporator
    - **allow pumping energy savings**
    - different options for locating the dp sensor of the VSP in the system

Hydronic control circuits

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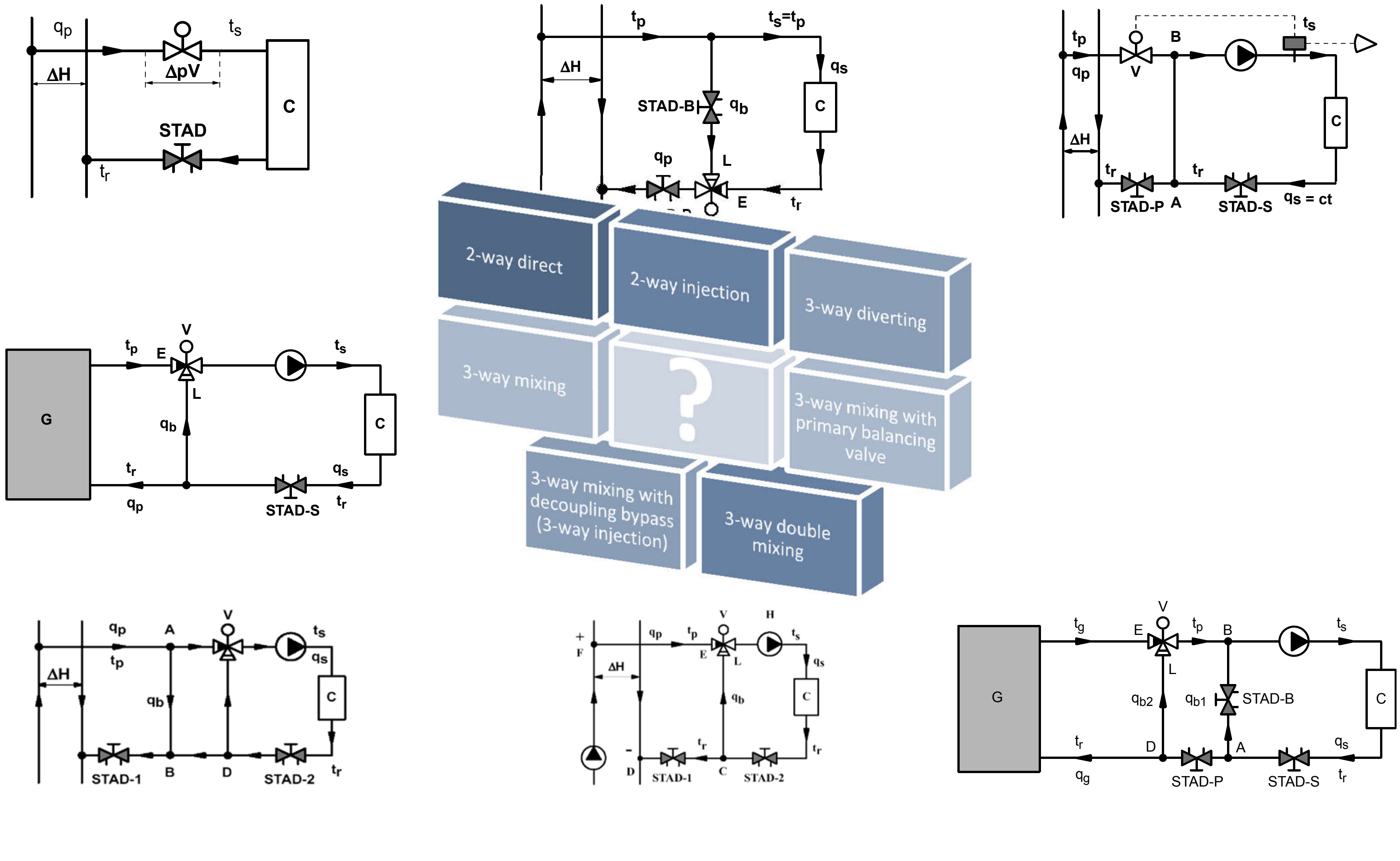
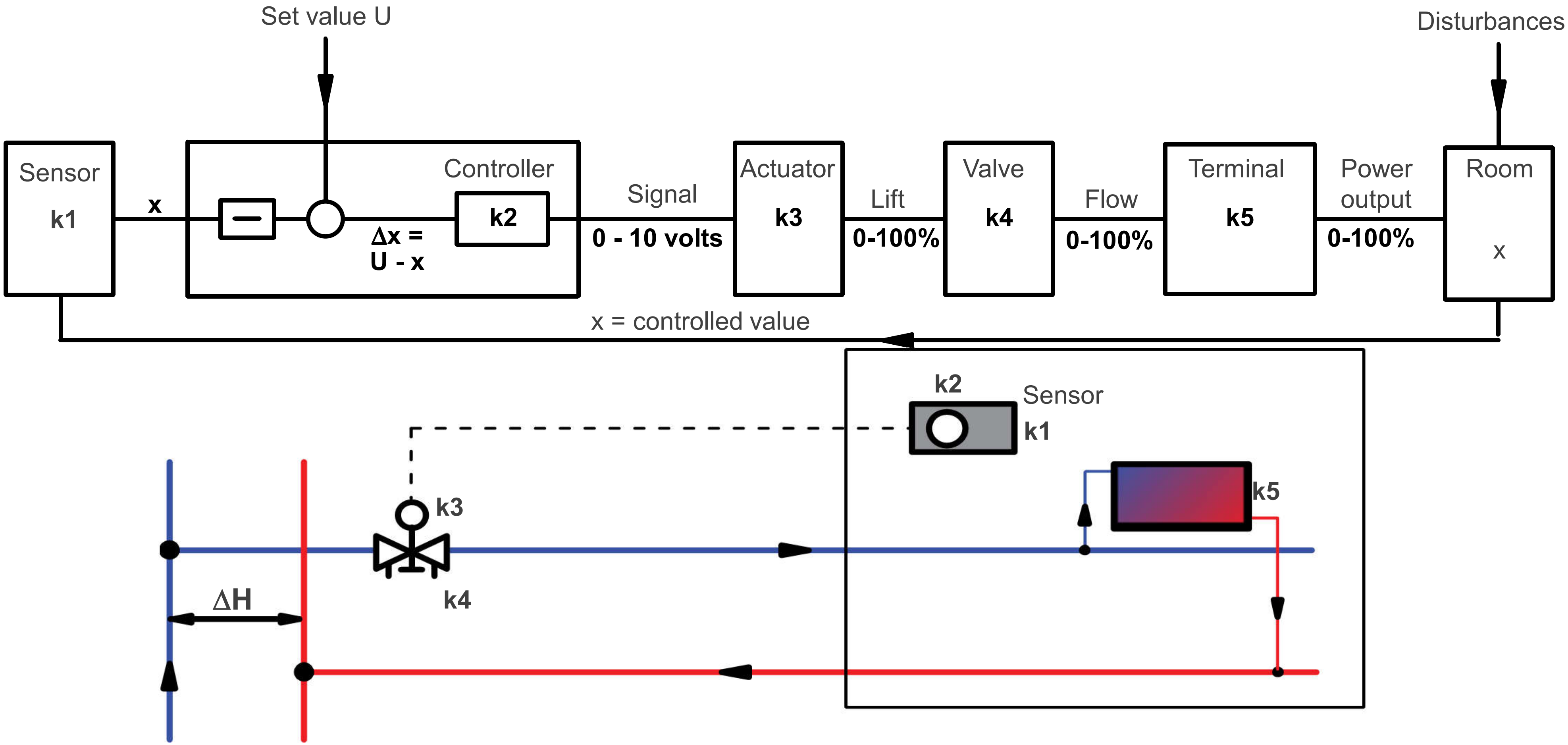


Room temp. control by action on a terminal unit



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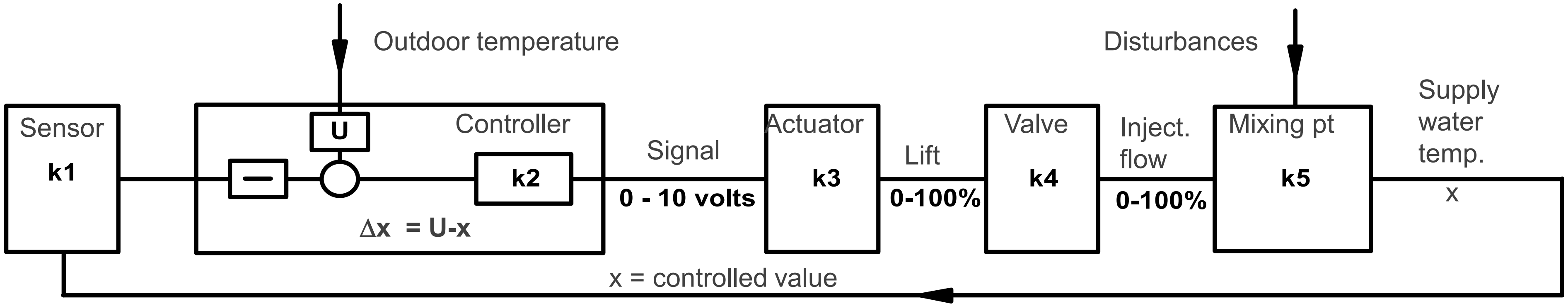
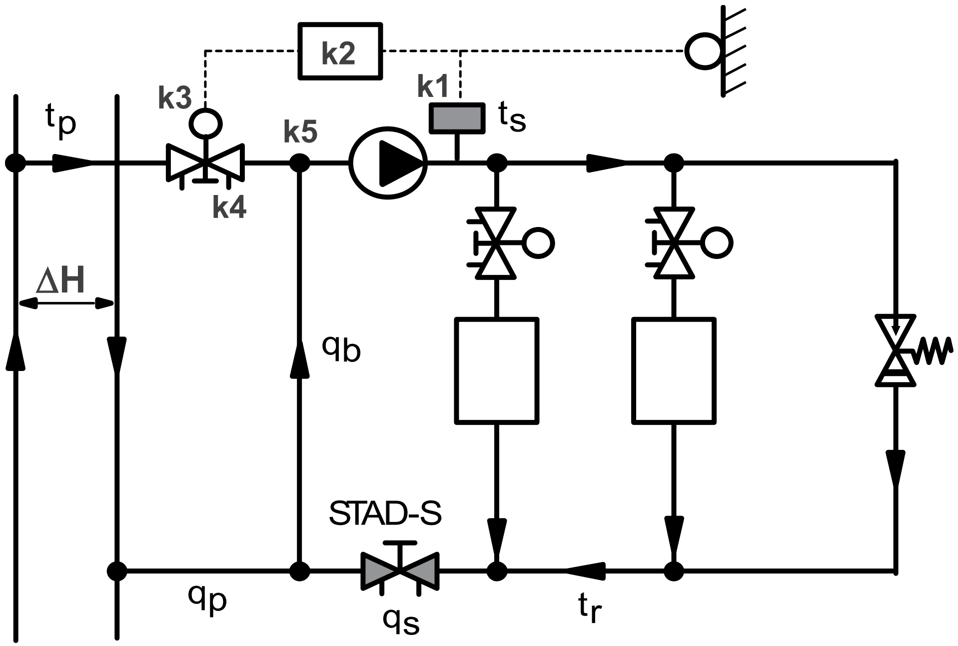
Supply water temp. control by action on an injection circuit





Global Circuit Characteristic

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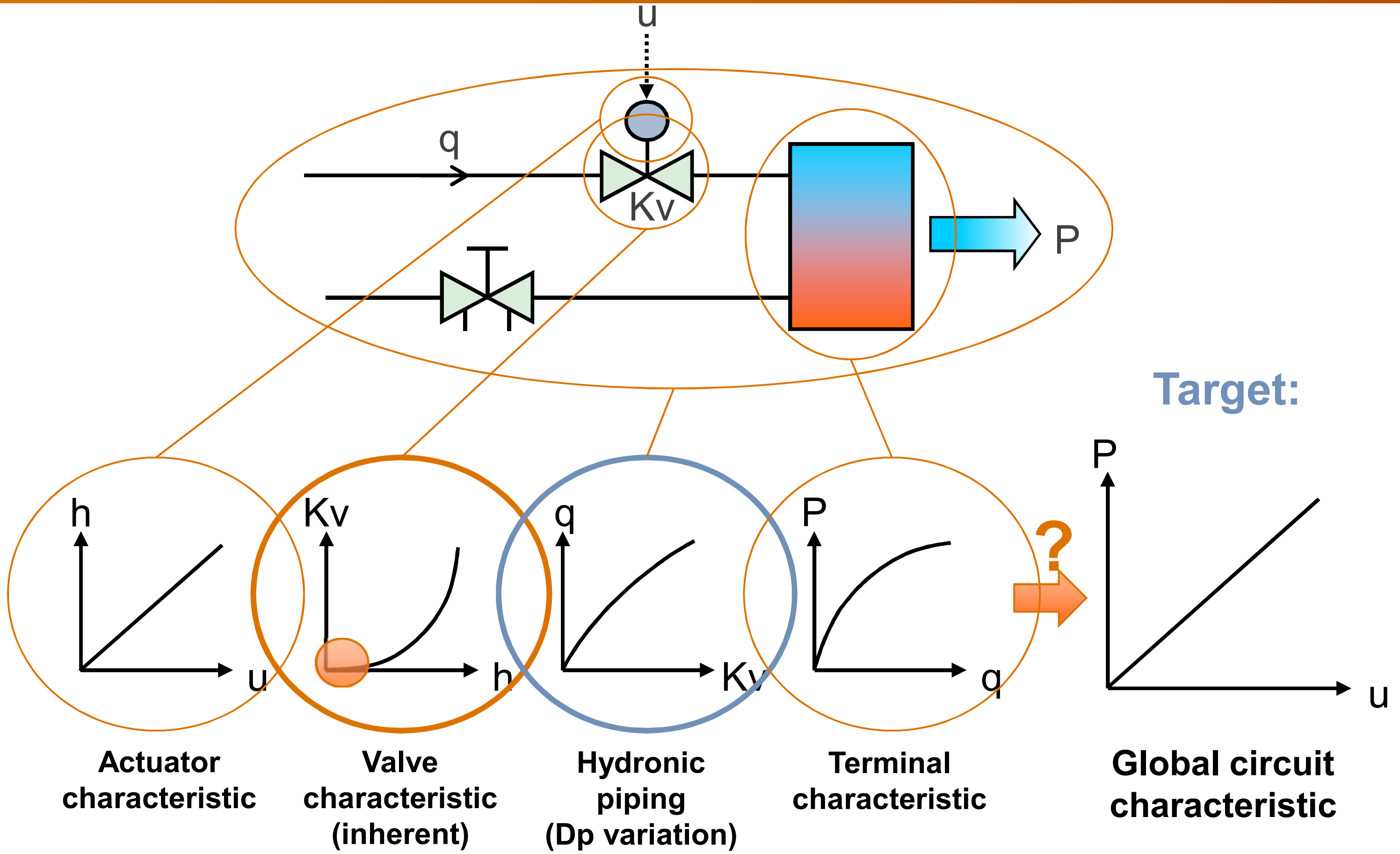
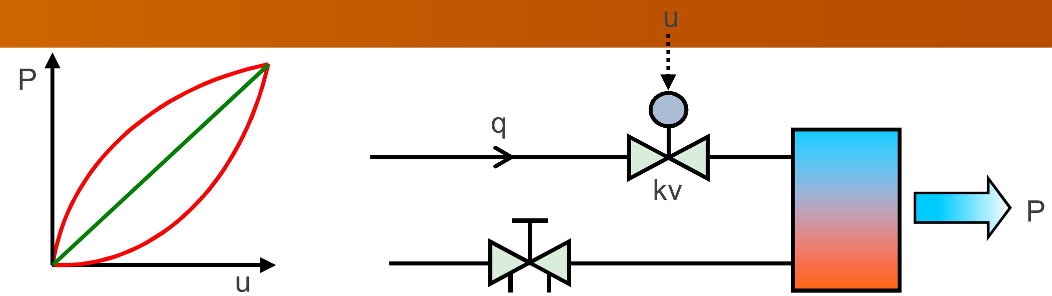
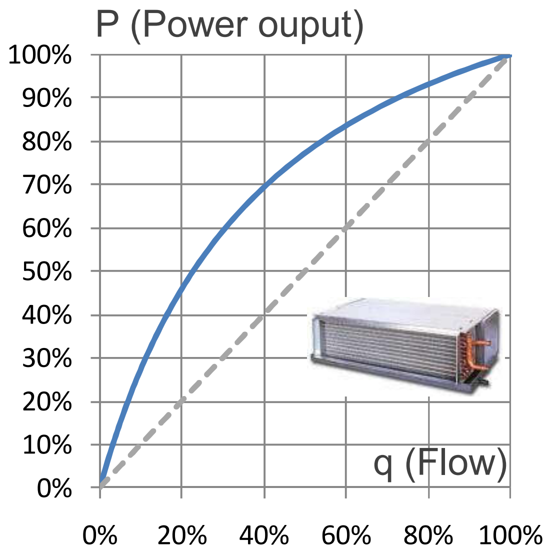


Proportional control
Terminal unit characteristic vs control valve character.

To obtain a global circuit characteristic that is as linear as possible, the **nonlinearity of the terminal unit characteristic** is **compensated by** an **equal percentage characteristic** of the control valve.

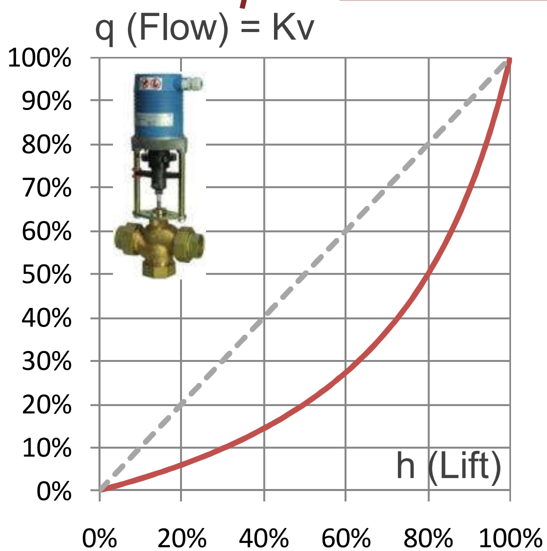


True only if Δp is constant since: $q = Kv \sqrt{(\Delta p)}$



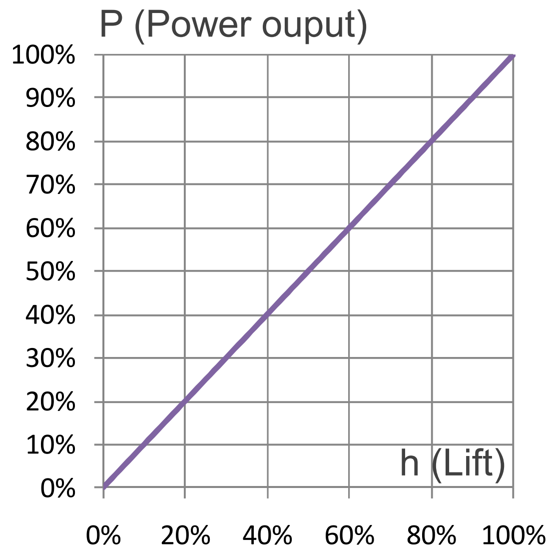
Terminal unit characteristic

+

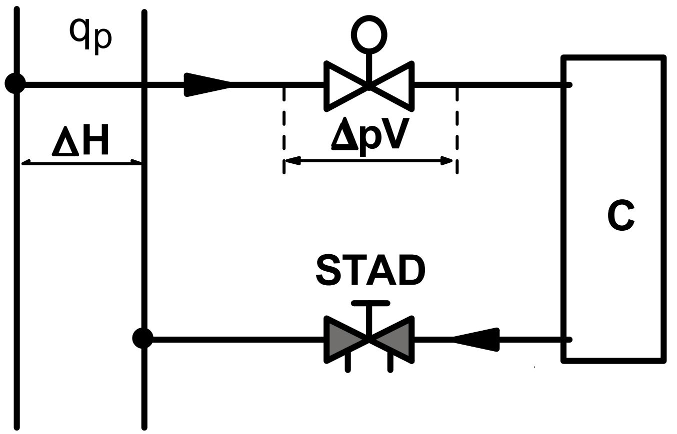


Control valve characteristic (Equal-percentage)

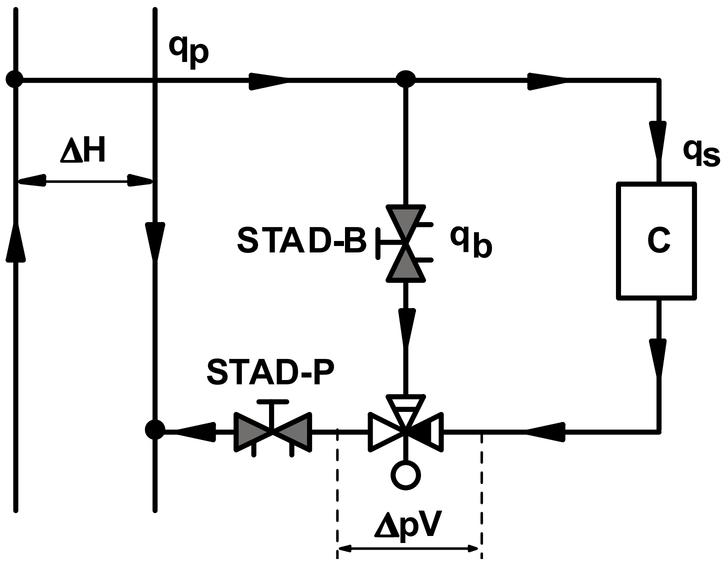
=



Control valve authority



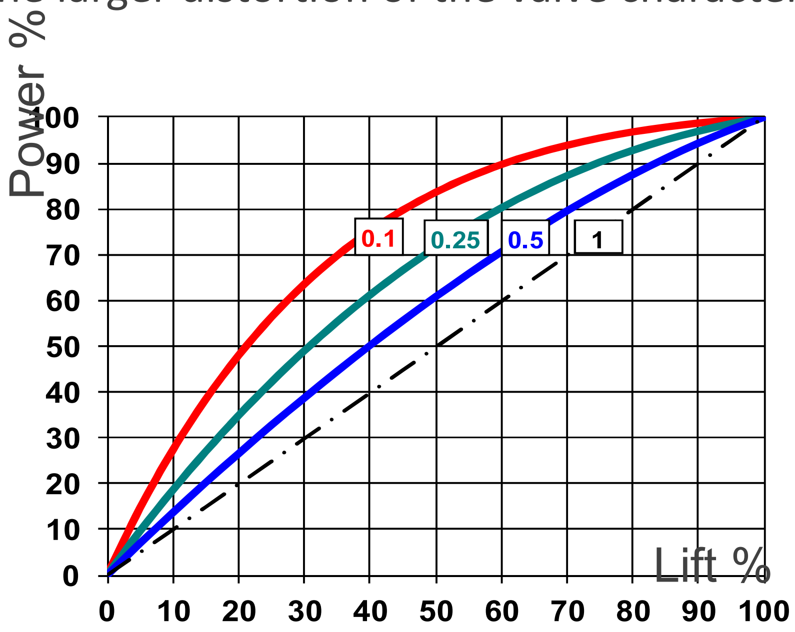
$$\beta = \frac{\Delta P_{\text{Control valve fully open and design flow}}}{\Delta P_{\text{Control valve fully shut}}}$$



The authority formulates how much the differential pressure builds up on the control orifice of a control valve when it is closing

Control valve authority guidelines

The lower the authority,
the larger the Dp variations on the control valve,
the larger distortion of the valve characteristic



Control valve with Equal-percentage characteristic

How to obtain a good (minimum) authority?

$$\beta = \frac{\Delta P_{\text{Control valve fully open and design flow}}}{\Delta P_{\text{Control valve fully shut}}}$$

Control valve sizing

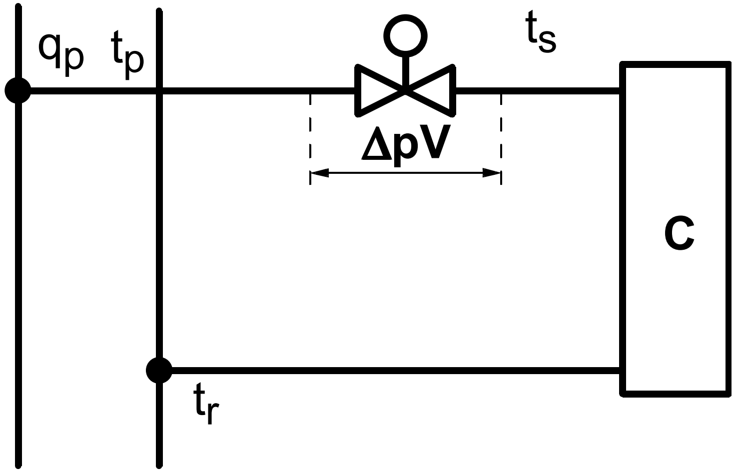
Control valve sizing for the exact Dp that gives a minimum authority of 0.25

Differential pressure control

Keep the differential pressure applied to control valves small enough

Differential pressure controllers
(stand-alone or integrated)

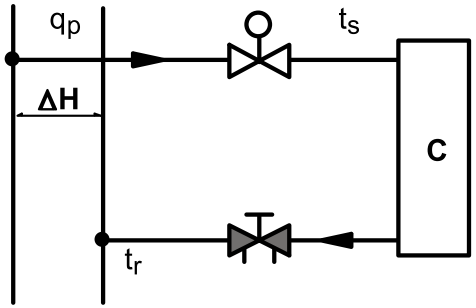
2-way direct circuit



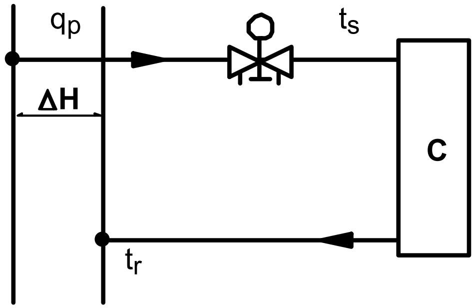
- **Primary network:** active or passive?
- **Primary/secondary flows:**
  variable or constant?
- **Flow directions:** ?
- **Temperatures tp , ts and tr:** ?
- **Control mode:** ?
- **Authority b:** minimum and design?

- **Balancing function/valve location:** ?
- **Balancing procedure:** ?

2-way direct circuit

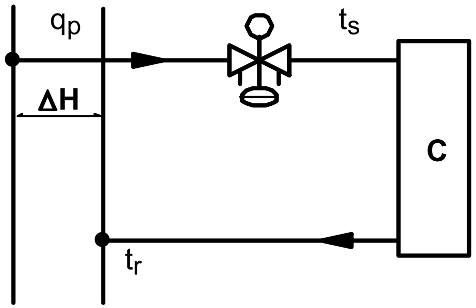


- **Active primary network**
- **Variable primary and secondary flows**
- **Temperatures:** ts = tp

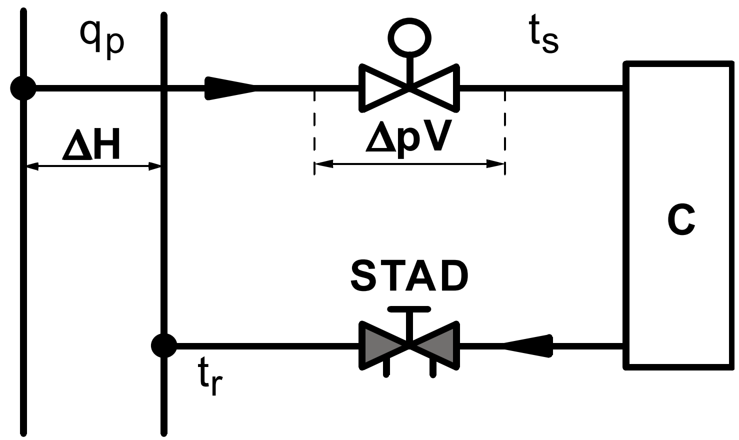


For proportional control, authority:

- min. authority:
  b_{min} = DpV/pump head
- design authority:
  b_d = DpV/DH



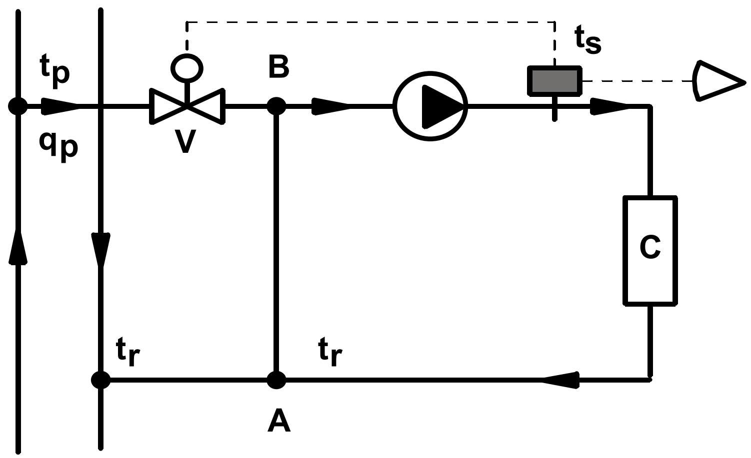
2-way direct circuit – Increasing authority



How to increase authority (if necessary)?

1. Decrease the available diff. pressure ΔH (smaller pressure losses of primary circuit, larger pipes, readjustment of STAD, …) ☹
2. Select control valve with smaller Kvs-value (flow will be smaller than expected or we have to increase ΔH) ☹
3. Choose other Kvs-value than from Renard series
   ➔ production of special valve with special Kvs-value (only for very special cases) ☹
   ➔ or… TA-FUS1ON-C ☺
4. Use a Dp controller
   to stabilize the Dp on the branch (STAP) ☺
   or use a PIBCV (TA-FUS1ON-P) ☺

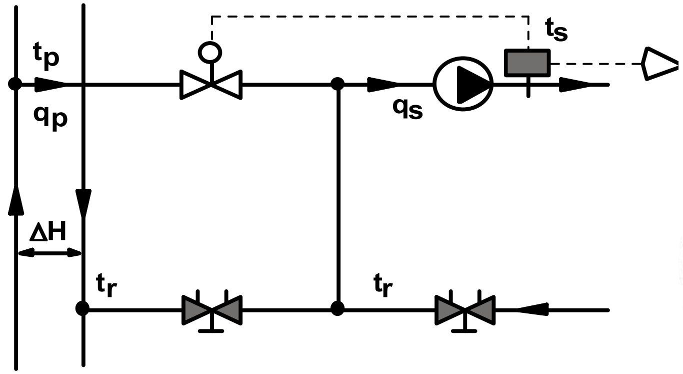
2-way injection circuit

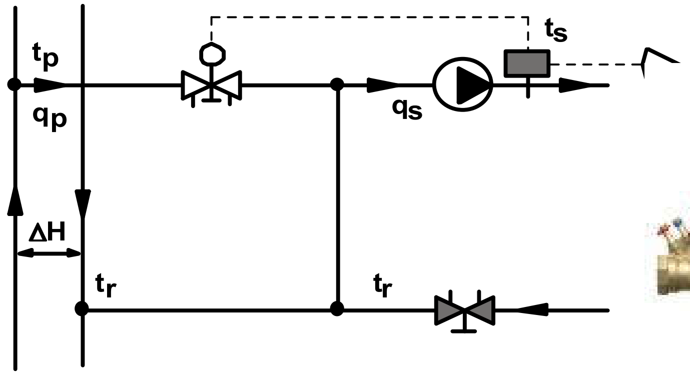


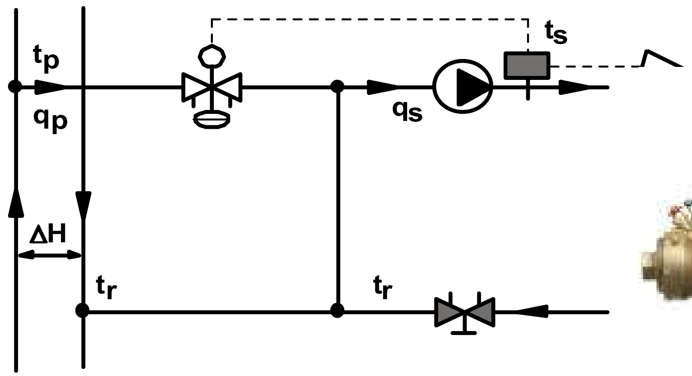
- **Primary network:** active or passive?
- **Primary/secondary flows:**
  variable or constant?
- **Flow directions:** ?
- **Temperatures tp , ts and tr:** ?
- **Control mode:** ?
- **Authority b:** minimum and design?

- **Balancing function/valve location:** ?
- **Balancing procedure:** ?

2-way injection circuit







- **Active primary and secondary networks**
- **Variable primary flow**
- **Constant secondary flow**
- **Temperatures:**
  - ts < tp (heating)
  - ts > tp (cooling)
  - ts can be = tp only in design cond.

For proportional control, authority:

- min. authority:
  b_{min} = DpV/pump head
- design authority:
  b_d = DpV/DH

Primary circuit is not influenced by terminal unit pressure drop

Valve characteristic for 2-way injection circuit

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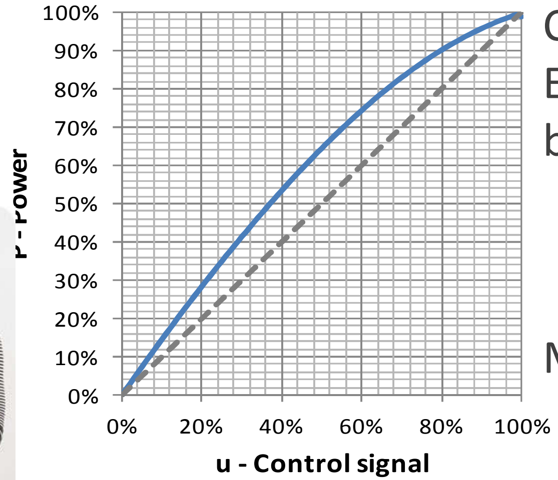
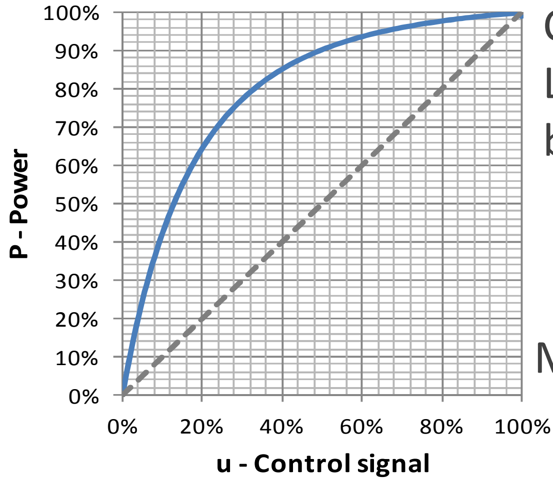
- Temp. regime: 15/18/24°C



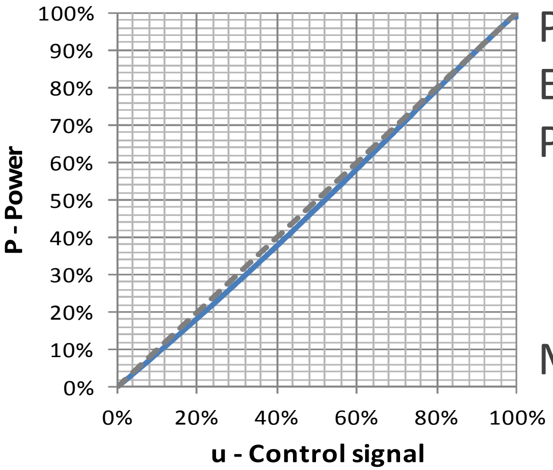
CV/BCV
EQM
b = 0.25

Max. gain 1.49



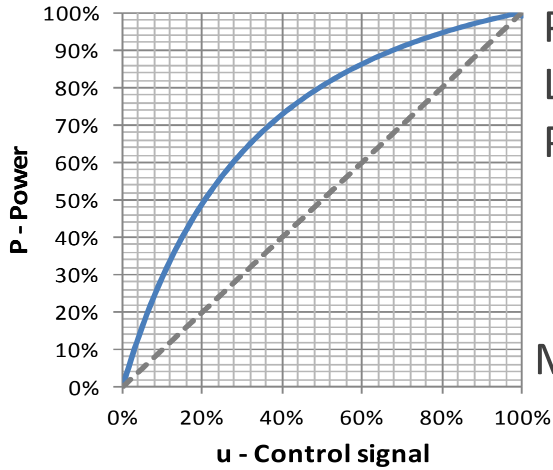
CV/BCV
Linear
b = 0.25

Max. gain 5.77



PIBCV
EQM
P-band 40%

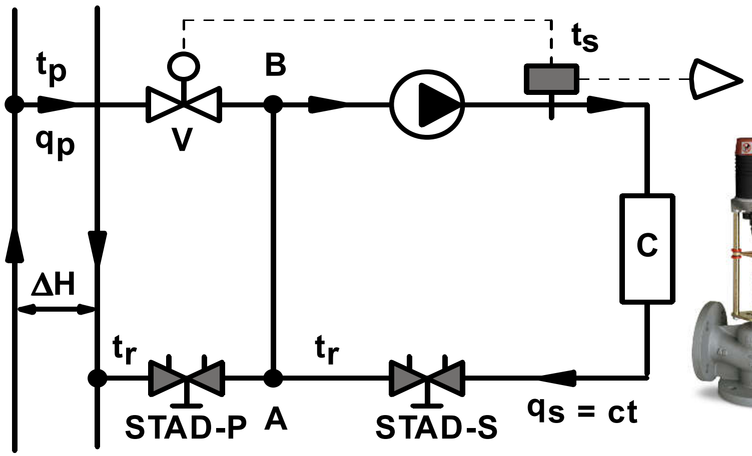
Max. gain 1.06



PIBCV
Linear
P-band 50%

Max. gain 3.58

2-way injection circuit – Sizing example



Example:

- Term. unit design flow: 72 m^3/h
- Supply / return temp.: 15/18 °C
- Chilled water supply t°: 6°C
- Primary pump head: 60 kPa
- Avail. diff. pressure ΔH: 45 kPa
- Circuit pressure drop Δp_C: 65 kPa

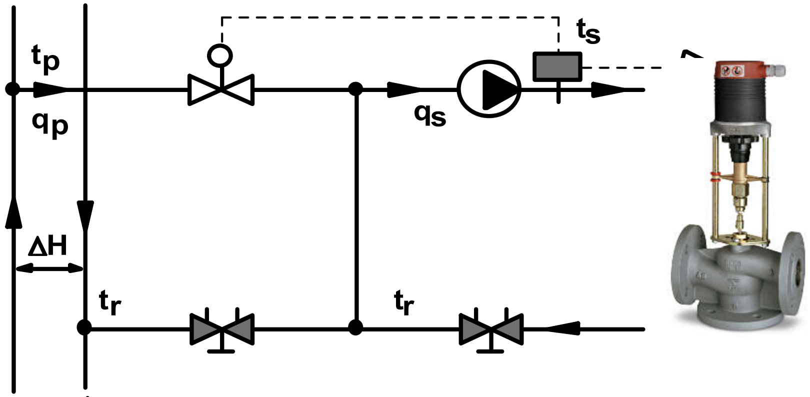
Energy and flow conservation:

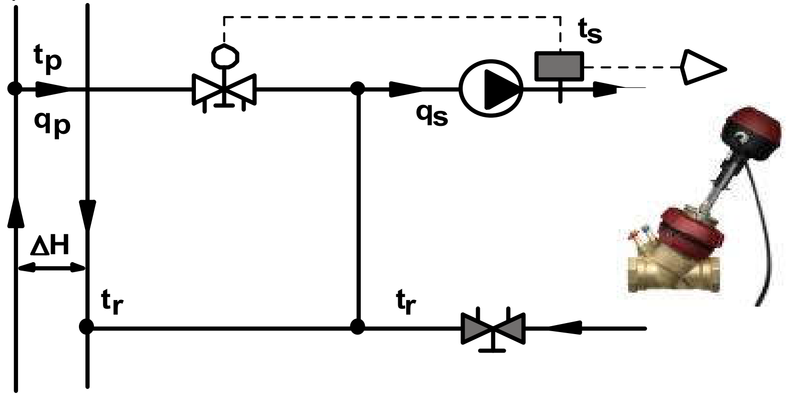
$q_p.(t_p-t_r) = q_s\ .(t_s-t_r)$

Primary flow:

- q_p = 72 . (15-18) / (6-18) = 18 m^3/h

2-way injection circuit – Control valve sizing





Flow: 18 m³/h

- **TA-FUS1ON-P DN 65;** set. 9.4

Control valve required Dp:

Largest between 0.25 . 60 kPa = 15 kPa

and

- Δp_{req} (CV)= 45-3 (STAD) = 42 kPa
- Δp_{req} (TA-FUS1ON)= 45 kPa

➔ If 15 kPa was not fulfilled, Dp control (Stand-alone or PIBCV) should be used

Valve sizing:

$$Kvs_{req} = 0.01 \cdot \frac{q}{\sqrt{\Delta p}} = 0.01 \cdot \frac{18000}{\sqrt{42}} = 27.75$$

- **CV216 RGA DN 50** Kvs=31.5
  ΔpV = 32.6 kPa

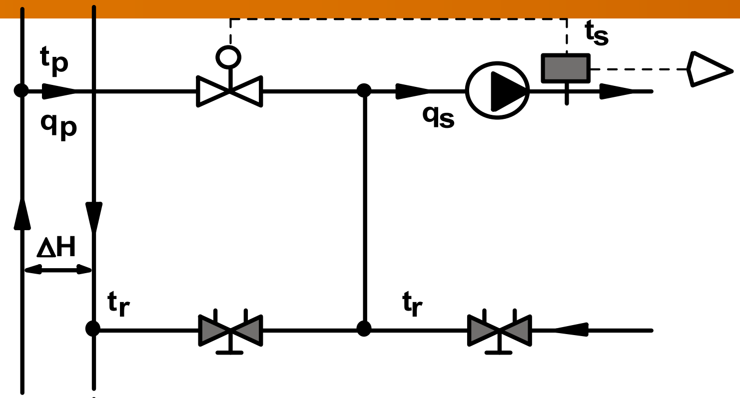
$$\Delta p_{STAD} = 45 - 32.6 = 12.4\ kPa$$

- **STAF 65-2**
  Setting 4.9

$$Kvs_{req} = 0.01 \cdot \frac{18000}{\sqrt{45}} = 26.81$$

- **TA-FUS1ON-C DN 50;** set. 7.95

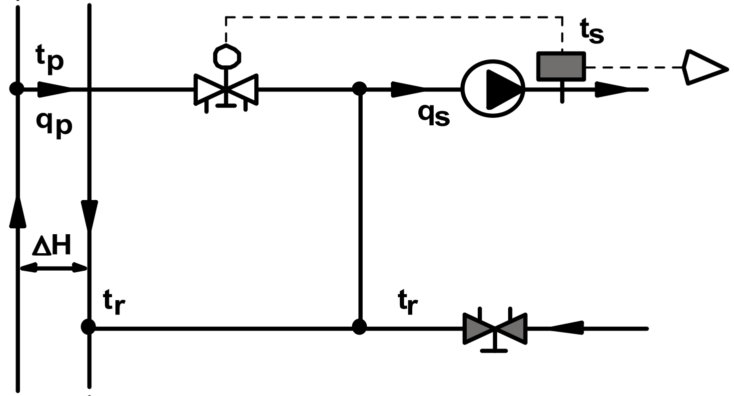
2-way injection circuit – Authority



- **Std CV**

  $b_{min} = \Delta pV/H_{pump} = 32.6/60 = 0.54$

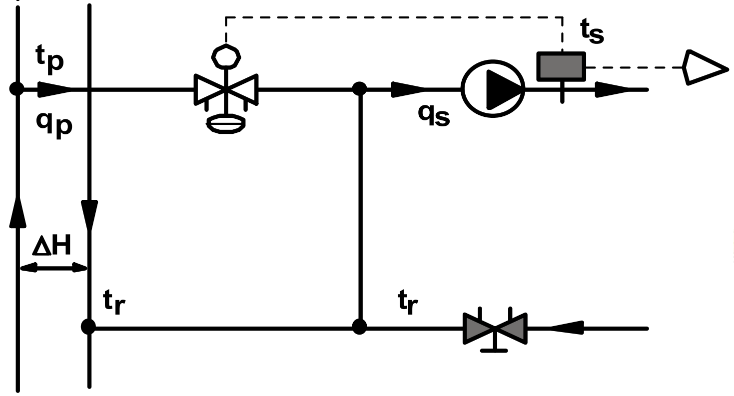
  $b_d = \Delta pV/\Delta H = 32.6/45 = 0.72$



- **TA-FUS1ON-C**

  $b_{min} = \Delta pV/H_{pump} = 45/60 = 0.75$

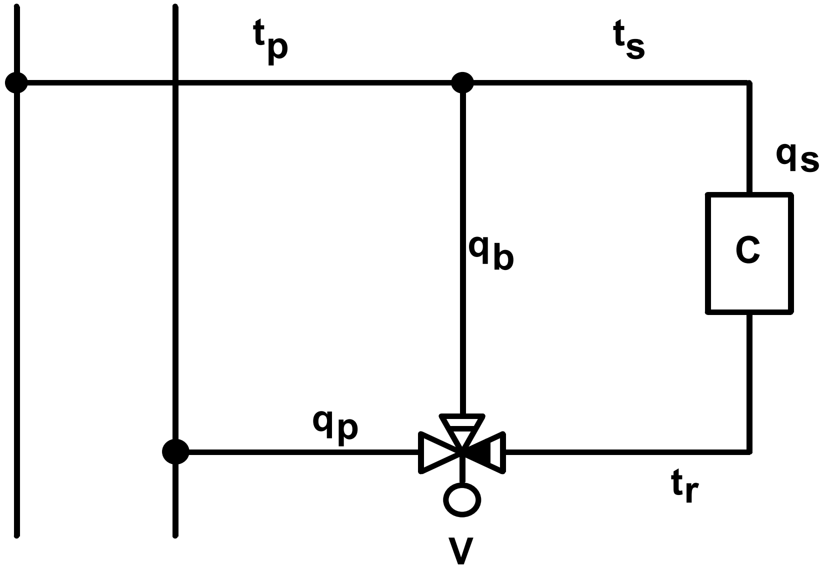
  $b_d = \Delta pV/\Delta H = 45^{-}/45 \approx 0.98$ (not 1!)



- **TA-FUS1ON-P**

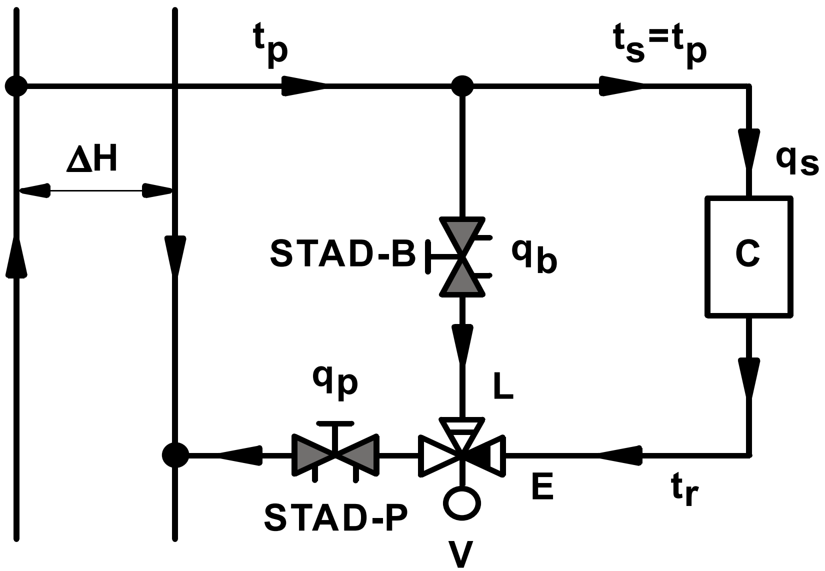
  $b_{min} > 0.7$ (P-band)

3-way diverting circuit



- **Primary network:** active or passive?
- **Primary/secondary flows:** variable or constant?
- **Temperatures tp , ts and tr:** ?
- **Authority b:** Formula?
- **Flow directions:** ?
- **Balancing valve location:** ?

3-way diverting circuit



- **Active primary network**
- **Primary flow:**
  - **Constant** if DH is constant (cst flow distribution)
  - **Variable** if DH is variable (variable flow distribution)
- **Variable secondary flow**
- **Temperatures:** ts = tp

- In **variable flow distributions,** min. authority:

$$\beta_{min} = \frac{\Delta p_V}{H_{pump}\left(1-\frac{\Delta p_{STAD-P}}{\Delta H}\right)}$$

design authority:

$b_d = Dp_V/(DH-Dp_{STAD-P})$

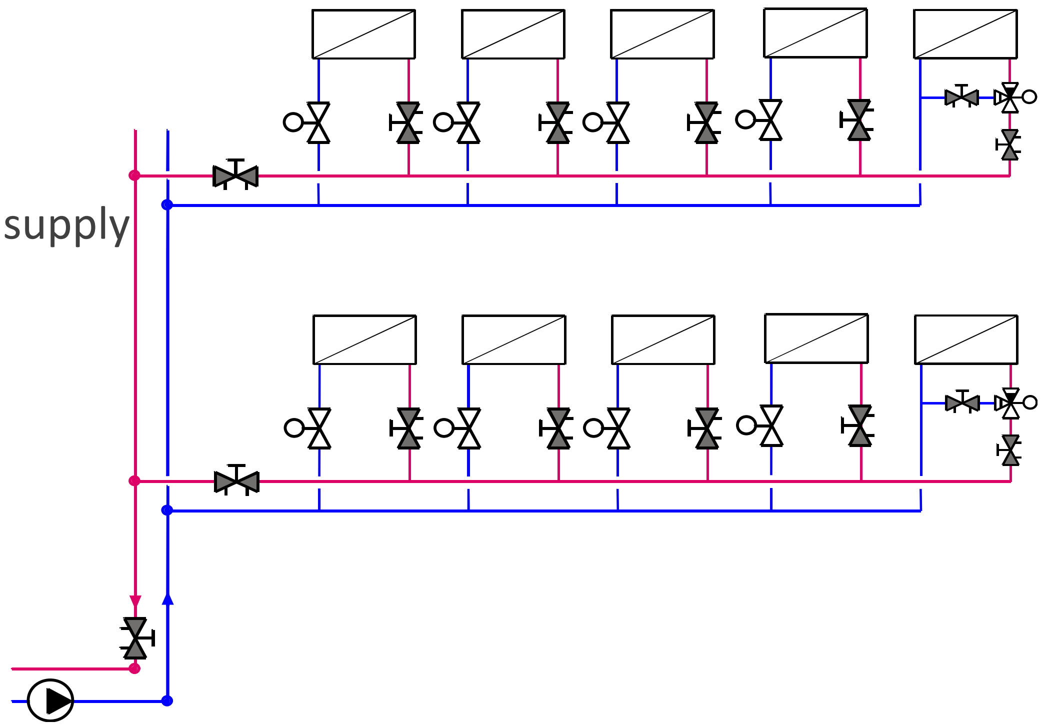
- **For proportional control, authority:**
  - In **cst flow distributions,** design & min. authority:
    $b = Dp_V/(Dp_V+Dp_C)$

3-way diverting circuit

Application in variable flow distribution

Obtain a minimum flow in each branch:

- To minimize dead time for control system
- To avoid too high heat losses/gains on the supply water
- For the pump



In this application:
Authority of the 3-way control valves to be calculated similarly as 2-way control valves since the flow in the distribution is variable and thus DH is variable too

3-way diverting circuit

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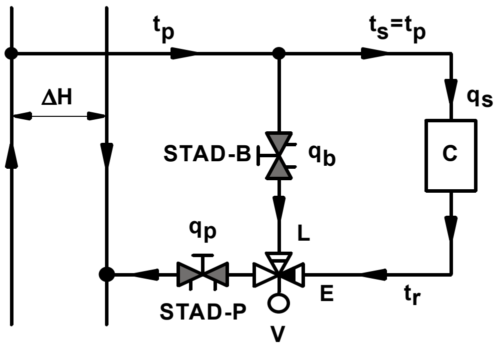

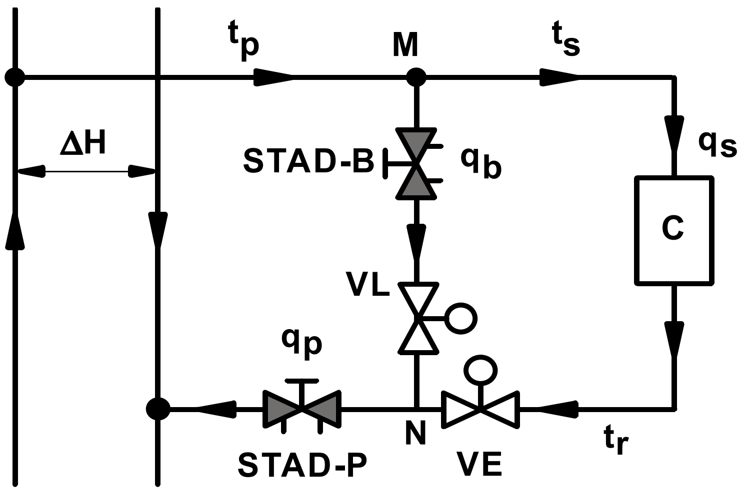

Authority (in constant flow distributions):

- The 3-way valve can be replaced by 2 identical two-way control valves working in opposition
- Valve VE represents the control port (to be examined for authority)

$$\beta = \frac{\Delta P_{\textit{Control valve} \text{ fully open and design flow}}}{\Delta P_{\textit{Control valve} \text{ fully shut}}}$$

Δp_V

$\Delta p_{MN} = \Delta p_V + \Delta p_C$

- Constant flow distribution → DH constant
  → Dp_{MN} constant
- Dp_{MN} applied on VE when VE is closed
- $Dp_{MN} = Dp_V + Dp_C$

$$\beta = \Delta p_V / (\Delta p_V + \Delta p_C)$$

3-way diverting circuit

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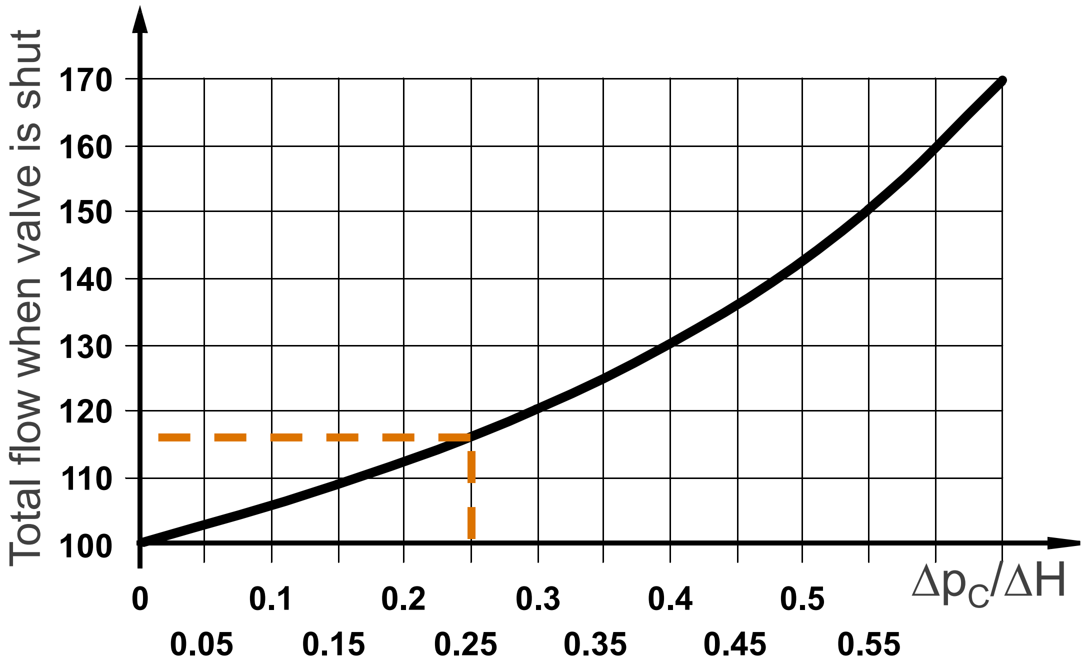
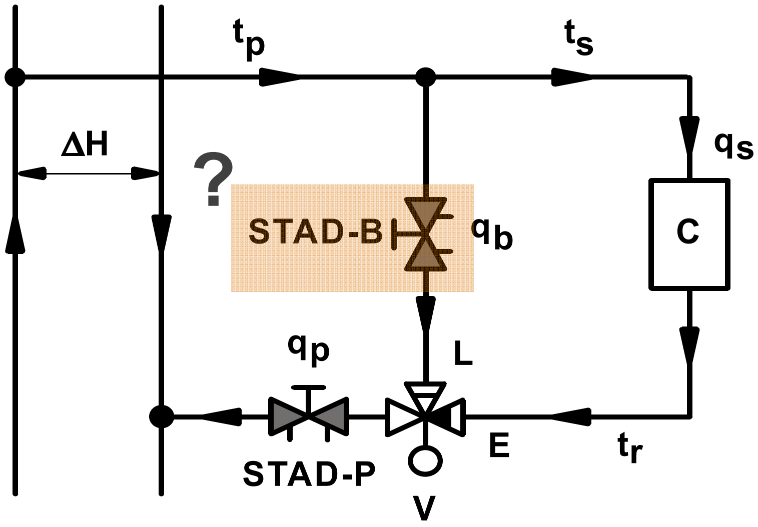
Is the balancing valve in the bypass required ?

YES, if:

$Dp_C > 0.25$ DH

(If the pressure drop in the coil exceeds 25% of the available differential pressure for the circuit)

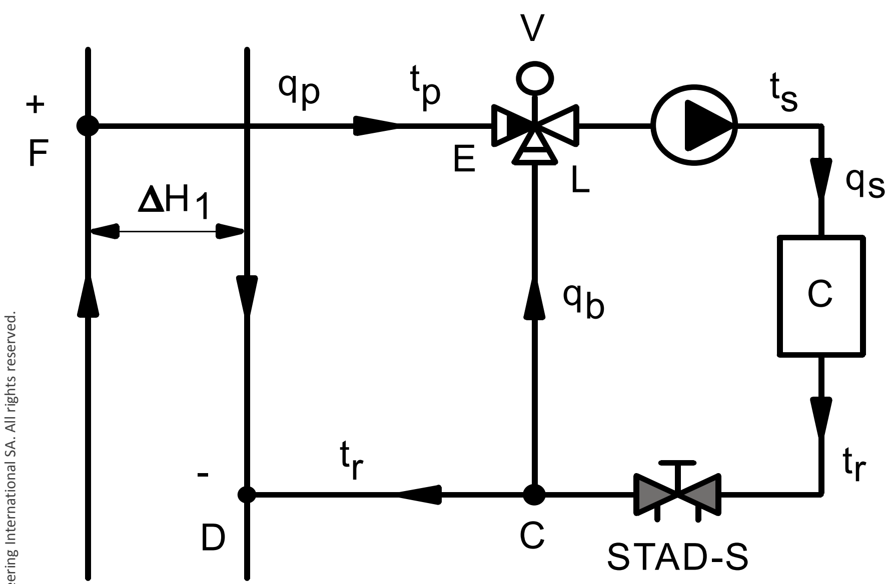




A 'full' balancing valve is actually not required in the bypass, see balancing procedure

3-way mixing circuit – applicability

The 3-way mixing circuit <u>cannot be used as is</u> with an active primary



With an active primary:

- DH_1 tends to increase q_p and to reduce q_b
- Mixing temperature t_s increases more rapidly than intended
- The flow can even be reverted in the bypass above a certain opening of the 3-way control valve

→There is no mixing any longer:
  - water at high temperature could be sent to floor heating
  - chilled water could be sent to cooling ceilings

Alternative to standard 3-way mixing circuit for use with an active primary

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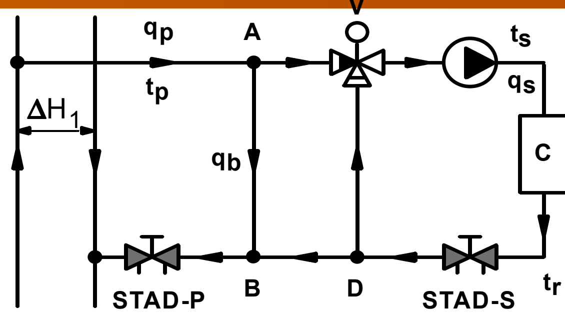
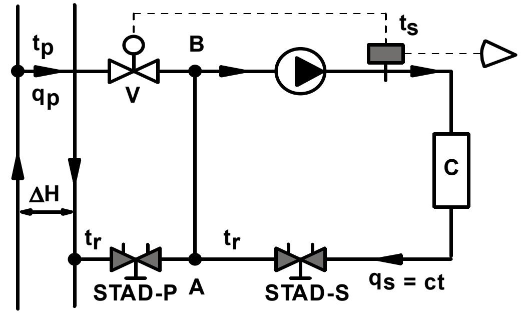
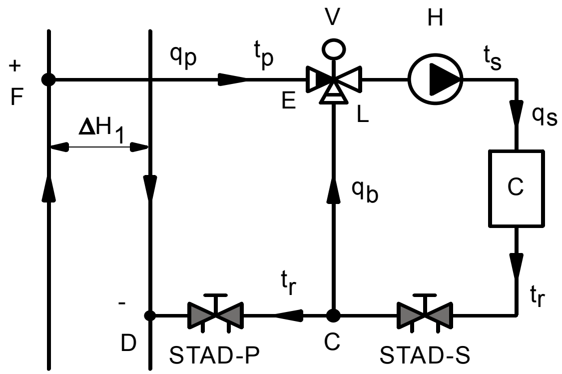
- If primary flow is constant or if it is acceptable to have some circuits at constant flow (for instance to generate a minimum flow)
  → **3-way mixing circuit with decoupling bypass**



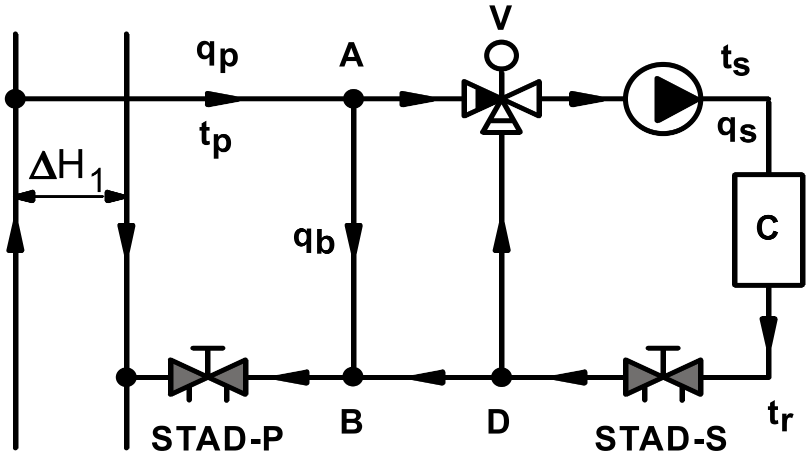
- If variable primary flow is a must and *redesign is possible*
  → **2-way injection circuit**



- If variable primary flow is a must but *minimal modifications* should be brought
  → **3-way mixing circuit with primary balancing valve**



3-way mixing circuit with decoupling bypass



- **Active primary network**
- **Constant primary flow**
- **Constant secondary flow**
- **Temperatures:** $t_s \leq t_p$ (heating);
  $t_s \geq t_p$ (cooling)
- **Authority:**
  - design & min. authority:
    $\beta = \Delta p_V/(\Delta p_V+\Delta p_{AB}) \approx 1$

When design $t_s = t_p$, the scheme as shown above applies and flow compatibility has to be ensured at bypass AB. Flow q_p is thus set slightly larger than flow q_s.

- **Pressure drop of 3-way control valve is covered by the secondary pump**

3-way mixing circuit with decoupling bypass

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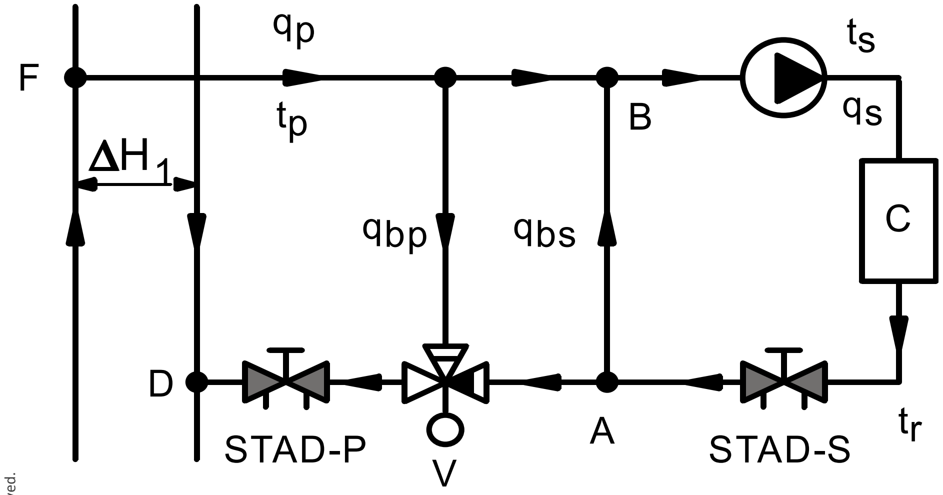
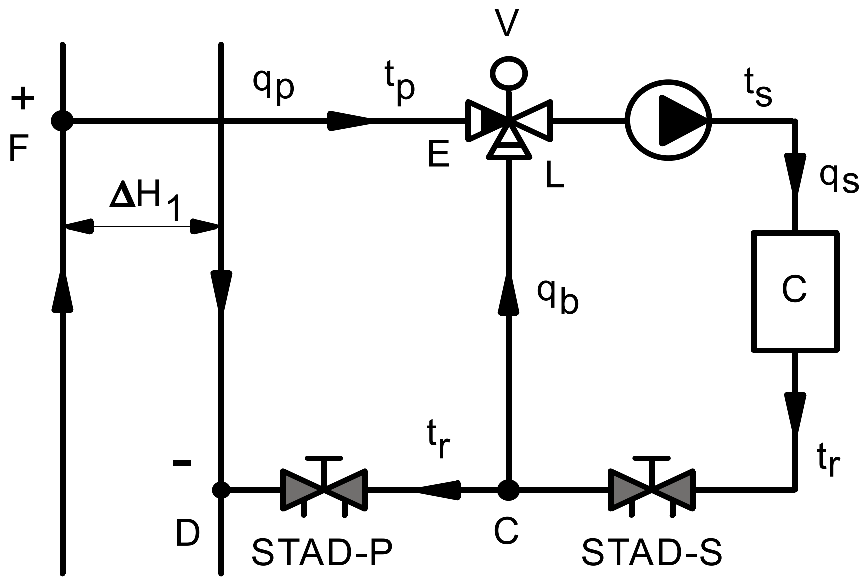
When design $t_s \neq t_p$, swapping the 3-way valve and the bypass is preferable
the flow q_s is then lower than q_p, thus allowing the use of a smaller control valve



- **Active primary network**
- **Constant primary flow**
- **Constant secondary flow**
- **Temperatures:** ts ≤ tp (heating); ts ≥ tp (cooling)
- **Authority:**
  - design & min. authority: $\beta = \Delta p_V/(\Delta p_V+\Delta p_{AB}) \approx 1$
- **Pressure drop of 3-way control valve is covered by the primary pump**

This scheme and the previous one are functionally equivalent

3-way mixing circuit with primary balancing valve



Note:
Balancing valve STAD-P does not compensate the primary ΔH_1 when the primary flow is small. This why the control valve authority is not improved by STAD-P.

- **Active primary network**
- **Variable primary flow**
- **Constant secondary flow**
- **Temperatures:** $t_s \leq t_p$ (heating); $t_s \geq t_p$ (cooling)

- **Authority:**
  - design & min. authority: $\beta = \Delta p_V/(\Delta p_V+\Delta H_1)$
- **Δp_V must be $\geq \Delta H_1$ to give $\beta = 0.5$**
  → This Δp_V must be compensated by secondary pump:
  - More expensive secondary pump
  - Pumping costs

TA-COMPACT-P

Fan-coil (floor-standing)

2 TA-COMPACT-P DN 15
with actuator EMO T

Short valve body…
always above condensing container

TA FCU valves

OVERVIEW	TA-COMPACT-P	TBV-CMP	TBV-C	TBV-CM
Characteristics	linear	EQM	linear	EQM
Pressure independent	yes	yes	no	no
On-Off control	EMO-T	EMO-T (use TA-COMPACT-P)	EMO-T	not recommended
Modulating control	not recommended	EMO-TM or MC15/24-C	not recommended	EMO-TM or MC15/24-C
3-point control	EMO-3 or MC15/24-C or MC15/230-C	EMO-3 or MC15/24-C or MC15/230-C	EMO-3 or MC15/24-C or MC15/230-C	EMO-3 or MC15/24-C or MC15/230-C

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TA FUS1ON Range

TA FUS1ON-C Range

DN32 DN40 DN50 DN65 DN80 DN100 DN125 DN150

TA FUS1ON-CP Range

Overview of the range

HVAC Control Valves

DN 15-150
Kvs 0.25-315
PN 6-16
0-120(130)°C

Industrial (& District Energy) Control Valves

DN 15-300
Kvs 0.16-1250
PN 16-40
0-180°C
For -30°C to 350°C, contact TA Hydronics

Motorized Butterfly Valves

DN 25-350
Kvs 36-13500
PN 6-16
-10-110°C

TAH Actuators

IMI Hydronic Engineering

EMO T
EMO TM

EMO, EMO EIB
EMOLON

TA-MC15
TA-MC15-C

TA-MC50-C

Used for TA-FUSION
Covered last year as 1st step

TA-MC55

TA-MC100 FSE/FSR

TA-MC100
TA-MC103

TA-MC160
TA-MC163

TA-MC250/500
TA-MC253/403
/503

TA-MC1000
TA-MC1003/1503

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Thank you for your attention!

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IMI PNEUMATEX

IMI TA

IMI HEIMEIER