The Three Keys to Perfect Hydronic Control
Heating Degree Days by Census Region

Cooling Degree Days by Census Region

Source: Energy Information Administration, Annual Energy Review, Table 8.9. (June 2008)
Source: Energy Information Administration, Annual Energy Review 2008, Table 1.10
All HVAC installations should reach 2 fundamental objectives:

1. To deliver the specified comfort level

2. To reach the first objective, while using a minimum quantity of energy
Hydronic control

In theory, current technology involving state-of-the-art BMS systems makes it possible to achieve these 2 objectives.

In practice, even the most sophisticated control systems lead often to reduced comfort at increased operation costs.

In many cases, problems and therefore solutions are to be found on the hydronic side.
Symptoms related to typical hydronic problems

- **Too hot** in some parts, **too cold** in other parts

- Start-up after a set-back is difficult in some rooms

- Installed power is not deliverable

- Room temperatures fluctuate

- Higher energy consumption than expected
The Three Key Conditions for Perfect Hydronic Control

Why and how to satisfy them in the simplest way
Overflows and underflows

Without hydronic balancing, the first circuits are in overflow creating underflows in other circuits. Control valves cannot solve this problem.
Overflows and underflows

To avoid complaints from building tenants

1st action:

Pumps are pushed to the maximum

- overflows increase
- underflows are reduced:

\[ q \sim \sqrt{\Delta p} \] (turbulent flow)

To compensate an underflow of 20% in a circuit (raising the flow by 25% from 0.80 \( q_c \) to \( q_c \))

The local available \( Dp \) must be increased by 56%!
Unbalanced plant usually means too high total flow

Increasing Dp circuit by 56%

Requires a brute-force increase of the pump head $H$ by 56%

Leading to a total flow increase of 25%

$20\%$ underflow

$H \times 1.56$

$q_{\text{tot}} \times 1.25$
Pumping costs

Increasing the pump head to reduce underflows is very energy consuming.

\[
Pumping \text{ costs} \approx C_0 + \frac{\text{Pump head} \times \text{Flow}}{\text{Pump efficiency}}
\]

With pumping costs representing up to:
(in annual energy consumption)

**Heating**
- 2-6%
- 100%

**Cooling**
- 15-20%
- 100%
To avoid complaints from building tenants

2nd action:
Supply water temperature is: increased in heating (decreased in cooling)

- unfavoured rooms start to be comfortable
- tenants from favoured rooms will react

Energy waste!
Increased CO₂ emission!
The cost of discomfort

The cost of 1 F too high room temperature over a year 4 to 7% *

The cost of 1 F too low room temperature over a year 6 to 10% *

(*) of the plant annual energy consumption
Overflows means that \textbf{water velocity is higher than expected} per design.

\[
v = \frac{1273q}{d_i^2}
\]
with \( q \) in l/s, \( d_i \) in mm

Too high water velocity leads to erosion in pipe elbows and heat exchangers.

Control valves of circuits in overflows work with very short open/close cycles. This limits dramatically their actuator life-time.
In unbalanced buildings, tenants claim for bad comfort conditions.

Maintenance technicians waste their time repetitively visiting these buildings trying to fight symptoms.

When a plant is unbalanced, the startup time is longer for the last circuits in underflow. Overflows do not result in higher power output of the terminal units.

When a plant is balanced, start-up is achieved simultaneously in all circuits.
Control valves not under control

In frequent cases, control valves do not control the flow in terminal units any longer. They are set fully open by the control system:

- At start-up after a set-back
- Because of sudden load variation
- Because of weird set-point at the thermostat
Room temperature control

Hotel in Paris
Dear passengers:
Due to Central Airconditioning, we are unable to control the Lounge temperature.
Please bear with us.

Thank you!

Beijing International Airport
The design flow must be available at all terminal units in design conditions.
Achieving hydronic condition nr 1

Adjusting the **design flows** in all terminal units in **design conditions**

- Design conditions are the "worst" plant operating conditions, under which maximum flow is required: control valves are all fully open.
- If design flows are adjusted under design conditions, they can be obtained in all other conditions.

This should be achieved while creating the **absolute minimum amount** of additional pressure drops.
To enable **systematic balancing** with **optimal in pressure drop** result, hydronic distribution pipings must be decomposed into **hydronic modules**.
To enable **systematic balancing** with **optimal in pressure drop** result, hydronic distribution piping must be decomposed into **hydronic modules**.
At any bifurcation between many and many units **turn in the direction of the main flow** and place a balancing valve on the low flow side.

For finding balancing valve locations resulting in the **lowest possible sizes**
Proportional method
- adapted from air system balancing methodologies
- not optimal in pressure drops

Compensated method (Pr. Robert Petitjean, see CIBSE Code W)
- designed for application with balancing valves
- optimal in pressure drops
- by-product: all excess Dp is concentrated in the main valve

TA Balance method (Pr. Robert Petitjean)
- fully computerized: automatic determination of the index valve (on the worst unit)
- optimal in pressure drops
- by-product: all excess Dp is concentrated in the main valve
Order for balancing modules

The structure of hydronic modules can be seen as a hierarchical tree.

Before a module can be balanced, the whole descent of this module must be balanced.

Balancing order:
VSP set-point optimisation

A. System unbalanced
Total flow higher than needed
Pump power: 17.2 HP (100%)

B. System balanced
Total flow adjusted – Excess Dp in main valve
Pump power: 13.7 HP (80%)

C. System balanced
Main valve re-opened; VSP set-point reduced
Pump power: 9.8 HP (57%)

System curves

Pumping costs ≈ \( C_0 + \frac{\text{Pump head} \times \text{Flow}}{\text{Pump efficiency}} \)

Too high H

Optimum H

A
B
C

Too high H

Optimum H

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Pump power: 9.8 HP (57%)
Savings are real

Pfizer pharmaceutical production unit

- Installed cooling capacity of 1535 ton (refrig.) 5.4 MW (3 chillers in cascade)
- Total design flow: 3400 gpm = 773 m³/h
- Problem: production alarms!
- 80 balancing valves from ½” to 8”

- Audit of plant with TA Select based on a first measurement campaign (presettings calculated, viscosity corrections checked)
- Full balancing performed using TA-Balance on one TA-CBI
Savings are real

Before balancing

Industrial plant
1535 ton (refrig.) cooling
3914 gpm
112 ft pump head

3400 gpm (-13%)
90 ft pump head (-20%)
Pumping power reduction:
52 HP = 39 kW
Savings: 25,300 USD/year
17,200 €/year

After balancing
"When you can measure what you are speaking about and express it in numbers you know something about it; but when you cannot measure it, when you cannot express it in numbers, your knowledge is of a meager and unsatisfactory kind."

Lord Kelvin, 1883
Diagnostic is a key point

Through balancing, many hydronic problems may be detected:
- Filters or valves clogged
- Terminal units or exchangers wrongly mounted
- Pipe damaged or not connected as expected
- Shut-off valves partially shut
- Check valves or pumps installed back-to-front

Balancing exposes these flaws while they can still be cheaply repaired.

Diagnostic is one of the main use of balancing valves.
Key nr 2

\[ q = q_c \]

\[ \Delta p V \approx k \]

\[ q_p > q_s \]
Differential pressure variations

% of heating season below this load

Thermal plant load [%]

% of cooling season below this load

Thermal plant load [%]

St. Louis

Heating

57%

50 % load

Cooling

58%

Power

Flow

Dp piping

At constant supply water temperature

20 % flow

4% press. drop

Pressure drops are reduced to 4% of their design value.

\[ \Delta P \propto q^2 \]
Differential pressure variations

At low flow, the pump head applies itself almost entirely on the control valves.

VSP does not allow to compensate for all local Dp variations in the plant.

0.96*21 ft + 0.96*7 ft ≈ 26.9 ft in excess in the valve at half-load

5 ft in the valve

7 ft in the circuit

H: 33 ft

Max. flow (design conditions)

ΔP_{Pipe}: 21 @ design flow

Low flow (half-load)

H:

Pump head
Room temp. control by action on a terminal unit

- **Sensor** \( k1 \)
- **Controller** \( k2 \)
  \[ \Delta x = U - x \]
- **Actuator** \( k3 \)
  - **Signal** 0 - 10 volts
  - **Lift** 0-100%
- **Valve** \( k4 \)
  - **Flow** 0-100%
- **Terminal** \( k5 \)
  - **Power output** 0-100%
- **Room** \( x \)

**Flow**

- **Power output Lift Signal**

\[ x = \text{controlled value} \]
Control loop

Sensor \( k_1 \) → Controller \( k_2 \) → Actuator \( k_3 \) → Valve \( k_4 \) → Terminal \( k_5 \) → Room

Set value \( U \)

\[ \Delta x = \frac{U - x}{k_2} \]

Signal: 0 - 10 volts

Lift: 0-100%

\( x \) = controlled value

Power output %

Flow in %

Flow in % + Lift h in % = Power output %

Terminal unit characteristic

Control valve characteristic
The authority formulation how much the differential pressure builds up on the control orifice of a control valve when it is closing.

Its value indicates how effectively the control valve can reduce the flow while it is closing.
The lower the authority, the larger the $\Delta p$ variations on the control valve, the larger distortion of the valve characteristic.

Control valve with Equal-percentage characteristic (EQM)
2-way control valve authority (variable flow)

In a variable flow distribution, the authority of a control valve is variable.

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

**Constant** as soon as the valve \( \text{Cv} \) is chosen (\( \Delta p_V \)).

**Variable**, depends on flows in the piping, thus also on the opening of all the other control valves.
Variable authority of 2-way control valves

Authority in design conditions:
\[ \beta \approx \frac{5}{(5+7)} = 0.42 \]

Authority at half-load:
\[ \beta = \frac{5}{(5+7+0.96 \times 21)} = 0.15 \]

Low flow (half-load)

- Pump head: 33 ft
- \( \Delta P_{\text{Pipe}} \): 21 ft at design conditions

- 0.96 * 21 ft + 0.96 * 7 ft \( \approx 26.9 \) ft in excess in the valve at half-load
- 5 ft in the valve
- 7 ft in the circuit
Control valve oversizing

Control valves are commercially available with Cv values increasing according to the Reynard series:

\[ \text{Cv:} \ldots \ldots 2.0 \quad 3.0 \quad 4.0 \quad 5.0 \quad 10 \quad 20 \quad 30 \quad \ldots \ldots \]

For a water flow of 20 gpm, the commercially available control valves create a design \( \Delta pV \) of:

<table>
<thead>
<tr>
<th>Cv:</th>
<th>5.0</th>
<th>10</th>
<th>20</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \Delta pV ) [ft]</td>
<td>37.3</td>
<td>9.3</td>
<td>2.3</td>
</tr>
<tr>
<td>( \Delta pV ) [psi]</td>
<td>16.1</td>
<td>4.0</td>
<td>1.0</td>
</tr>
</tbody>
</table>

**Conclusion:**
Control valves are generally **oversized**.
Rangeability

**Rangeability** \((R_a) = \text{ratio between } Cv \text{ max and the } Cv \text{ corresponding to the min. controllable opening of the valve}**

Normal rangeability: 25-30 for Equal-percentage valves

Min controllable flow is 4% of the max flow (for the same Dp)

Min controllable heat output is around 10%

More generally, the minimum controllable flow is given by:

\[
q_{\text{min}} = \frac{q_{\text{max}}}{R_a \sqrt{\beta}}
\]

where \(q_{\text{max}} = 100% \text{ if the valve is not oversized} \)
Effect of $\Delta p$ variations on controlled heat output

$\Delta p$ variations distort the characteristic of the control valve
$\Rightarrow$ the nonlinear characteristic of the terminal unit is no longer compensated

Rangeability
area of the control valve

ON/OFF control
Cavitation: What is it and why did that happen?

- When liquid passes through the valve, increases the speed and thus the dynamic pressure.
- The static pressure drops to the same extent as the total pressure is constant.
- Decreases the static pressure below the vapor pressure creates steam bubbles.
- When the liquid has passed through the small opening in the valve seat decreases the speed and the static pressure rises again.
- Vapor bubbles collapse (implode)
- The strong “pressure waves” causing the short time erosion damage.
How do sounds and how to avoid?

- Cavitation causes vibrations in the valve wears down cone and valve seat.
- Cavitation normally cause noise in the valve as if it is “sand” to twirl in the valve. Noise can also sound like creaking
- Risk of cavitation increases by:
  - Low static pressure
  - Large pressure drop across a valve
  - High fluid temperature
  - Poor valve design

**RULE OF THUMB** to avoid cavitation:
Static pressure at valve inlet > 2 × Δp
Closing of control valves

According to its design, each valve has a required actuation close-off force or torque that depends on:

- Tension of the return spring, if any,
- Friction with o-rings and seals,
- Differential pressure applied on the plug.

Each control valve/actuator combination has a certain close-off differential pressure.
Hydronic condition nr 2

The differential pressure across control valves must not vary too much.
Control valve authority

To achieve good control it’s recommended to fulfill two rules on authority:
1. Size the control valve with a Cv with $\beta_{\text{design}} \geq 0.5$
2. Ensure that $\beta_{\text{min}} \geq 0.25$

Rule no 1:
$\Delta p_V \geq \Delta p_C + \Delta p_{\text{pipe}} + \Delta p_{\text{STAD}}$

or
$\Delta p_V \geq 0.5 \times \Delta H$

$\beta_{\text{design}} \geq 0.5$

Rule no 2:
$\Delta p_V \geq (\Delta p_{\text{piping}} + \Delta p_C)/3$

or
$\Delta p_V \geq 0.25 \times H$

$\beta_{\text{min}} \geq 0.25$
**Example:**

**Rule no 1:**
For obtaining a design authority of 0.5:
\[ \Delta p \text{ in control valve must be } \geq 0.5 \ \Delta H \]

Since \( \Delta p \) circuit = 7 ft,
\( \Delta p \) in control valve must be \( \geq 7 \) ft

**Final pump head** = 40 + 7 = **47 ft**
\( \beta_{\text{design}} = 7/14 = 0.5 \) but
\( \beta_{\text{min}} = 7/47 = 0.15 \)

**Rule no 2:**
For obtaining a minimum authority of 0.25:
\[ \Delta p \text{ in control valve must be } \geq 0.25 \ \Delta H \]

Since \( \Delta p \) piping + circuit = 33 + 7 = 40 ft,
\( \Delta p \) in control valve must be \( \geq 13.3 \) ft (40/3)

**Final pump head** = 40 + 13.3 = **53.3 ft**
\( \beta_{\text{design}} = 13.3/20.3 = 0.66 \) and
\( \beta_{\text{min}} = 13.3/53.3 = 0.25 \)

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**IDEA**
Ensure design authority of at least 0.5 and minimum on 0.25 in all control valves in the worst conditions.

\[ \beta_{\text{design}} = \frac{\Delta P_{\text{Control valve fully open and design flow}}}{\Delta H} \]
\[ \beta_{\text{min}} = \frac{\Delta P_{\text{Control valve fully open and design flow}}}{H} \]
Improved control with reduced pumping energy

Control valve sizing with Dp control:
For obtaining a design authority of 0.5 and min of 0.25:

\[ \Delta p \text{ in control valve must be } \geq 0.5 \Delta H \text{ and } \geq 0.25 \text{ of stabilized } \Delta p \]

Since \( \Delta p \text{ piping} + \Delta p \text{ circuit} = 7 \text{ ft} \),
\( \Delta p \text{ in control valve must be } \geq 7 \text{ ft} \)

Final stabilized \( \Delta p = 7 + 7 + 2 = 16 \text{ ft} \)

\[ \beta_{\text{design}} = \frac{7}{14} = 0.50 \text{ and} \]
\[ \beta_{\text{min}} = \frac{7}{16} = 0.44 \]

**Final pump head** = 31 + min \( \Delta p \) of DpC (2 ft) + 2 + 7 + 7 = **49 ft**
Differential pressure control - How does it work?

Measuring valve (Flow *measuring*)

Differential pressure control - How does it work?

Dp controller

Stabilisation

$\Delta H = 60$

$\Delta PL = 30$

P1

P2

P3

on
Differential pressure control - How does it work?

Measuring valve
(Flow measuring)

Differential pressure
Stabilisation

$\Delta H\ 70$

$\Delta P L\ 31$

P1

P2

P3

Dp controller
Differential pressure control - How does it work?

Measuring valve
(Flow measuring)

Dp controller

Differential pressure
Stabilisation

ΔH 80

ΔPL 32

P1

P2

P3

on
Differential pressure control - How does it work?

Measuring valve
*(Flow measuring)*

Differential pressure control - How does it work?

Stabilisation
Differential pressure control - How does it work?

Measuring valve
(Flow measuring)

Differential pressure
Stabilisation

P1

ΔH 100

ΔPL 34

P2

P3

Dp controller

on
Differential pressure control - How does it work?

Measuring valve
(Flow *measuring*)

Differential pressure
Stabilisation

Dp controller

$\Delta H\ 90$

$\Delta PL\ 33$

P1

P2

P3

on
Differential pressure control - How does it work?

Measuring valve
(Flow *measuring*)

Differential pressure
Stabilisation

Dp controller

\[ \Delta H \]
80

\[ \Delta P_L \]
32

\( P_1 \)

\( P_2 \)

\( P_3 \)

on
Differential pressure control - How does it work?

Measuring valve (Flow measuring)

Dp controller

Differential pressure Stabilisation

\[ \Delta H \]

\[ \Delta P_L \]

P1

P2

P3

on
Differential pressure control - How does it work?

Measuring valve
(Flow *measuring*)

Differential pressure control - How does it work?

Measuring valve
(Flow *measuring*)

Differential pressure
Stabilisation
Depending on project structure, Dp control will be applied:

On risers,

On branches,

On control valves.
Bigger plant with different Dp control configurations
Find the best Dp control solution…
Find the best Dp control solution…

First, decompose the plant into modules
Find the best Dp control solution... (1)

Dp control on each control valve

Parameters:
- On-off or modulating control
- Dp in pipes; length of branches
- Material cost
Find the best Dp control solution… (2)

Dp control on branches

Parameters:
- On-off or modulating control
- Dp in pipes; length of branches
- Material cost
Find the best Dp control solution... (3)

Dp control on main pipes

Parameters:
On-off or modulating control
Dp in pipes; length of branches
Material cost
VSP Pumps versus authority
Variable flow systems reduce by nature pumping costs at partial load.

Variable speed pumps reduce further pumping costs by avoiding an increase of the pump head when the flow diminishes.
Variable speed pump (VSP) control

Goal:

- Maximize pumping energy savings
- while keeping a 100% operational cooling/heating system

- Which control mode to select for the VSP?
- If a remote Dp sensor is used for the VSP, where to install it?
VSP – constant head or proportional head?

**Pump head [ft] (constant)**

- $H_0 = H_d$

**Flow [%]**

Differential pressure increase mainly on last circuits at partial load

**Pump head [ft] (proportional)**

- $H_0 = \frac{1}{2} H_d$

**Flow [%]**

Differential pressure decrease on most circuits at partial load, leads to possible underflows in these circuits!
VSP – integrated sensor or remote sensor at the end? (constant head)

\[ H_0 = H_d - \Delta p \]

Differential pressure increase mainly on last circuits at partial load

Differential pressure decrease mainly on first circuits at partial load, leads to possible underflows in these circuits!
If a VSP is used, is there a need for stabilizing differential pressure with differential pressure control valves?

Depending on the VSP control mode and the location of a circuit in a plant,
- How does the authority of control valves evolve at various loads?
- Under which conditions do we get the minimum authority?

Sample plant:

Assumption:
All circuits have simultaneous decrease of their load
Sample plant at design conditions
Variable speed pump – Constant head

\[ \Delta p_{RA} = 80 \text{ ft} \]

\[ \Delta p_{RJ} = 40 \text{ ft} \]

\[ H_d = 100 \text{ ft} \]

\[ H_0 = H_d \]

Min. authority (slightly improved) at min. flow in the plant
Variable speed pump – Proportional head

- Min. authority (improved) at interm. cond. depending on circuit location, at the cost of underflows in first circuits.

- $H_0 = \frac{1}{2} H_d$

- $\Delta p_{RA} = 80$ ft

- $\Delta p_{RJ} = 40$ ft

- $H_d = 100$ ft

- $H_0 = \frac{1}{2} H_d$

- $0.6$, $0.5$, $0.2$, $0.17$, $0.46$, $0.5$

- Authority
Variable speed pump – Remote sensor at cst head

\[ H_0 = H_d - \Delta p_{RK} \]

Min. authority improved (or not) at min. flow, depending on circuit location and pipe Dp
Variable speed pump – Remote sensor at cst head

\[ H_0 = H_d - \Delta p_{RA} \]

\[ \Delta p_{RA} = 80 \text{ ft} \]

\[ \Delta p_{RJ} = 40 \text{ ft} \]

\[ \Delta p_{RK} = 20 \text{ ft} \]

Authority becomes meaningless because design flow can no longer be obtained at part load.
VSP and controllability

- VSP are meant to **maximize pumping energy savings** at varying load in a variable flow system.

- VSP do act where they are installed, on the **total flow** going through them, based on the Dp sensed at **one** location by a Dp sensor.

- VSP **do not guarantee** by themselves a **stable and accurate room temperature control** throughout the system, whatever the control mode and Dp sensor location that is selected.

- **Differential pressure controllers** are required to protect control valves from large differential pressure variations experienced at varying load on the system…

  … And they lead, thanks to that, to very interesting energy savings.
Remote VSP Dp sensor with Dp controllers

Full load (design conditions)

- Pump head: 30 ft
- $\Delta p_{BV}$
- $\Delta p_{DpC}$
- $\Delta p_{Stabilized}$

Half-load (20% of the flow)

- Pump head: 13.7 ft
- $\Delta p_{BV}$
- $\Delta p_{DpC}$
- $\Delta p_{Stabilized}$

... nearby a further 55% reduction on pump head at half-load
Remote VSP Dp sensor with Dp controllers

Rules for optimum pumping energy saving and controllability:

1. Perform dynamic balancing with Dp controllers on each branch or on each unit with PIBC
2. Install the VSP Dp sensor on the index branch/circuit
3. Adjust the set-point of the VSP to the largest required available differential pressure amongst stabilized branches/circuits

VSP Dp sensor set-point:
+ 10 ft in DpL
+ 1 ft for BV
+ 1 ft (min Dp of DpC)
= 12 ft

Index detection:
+ 10 ft (distribution pipe)
+ 1 ft (riser pipe)
+ 1 ft for BV
+ 1 ft (min Dp of DpC)
+ 9 ft in DpL
= 22 ft
Optimum energy saving and controllability

- Good authority ensured to control valves with Dp controller on each branch or on each unit with PIBCV
- Enhanced pumping energy saving with VSP Dp sensor on index branch
- Complete Dp calculation of the distribution is warmly recommended
Key nr 3

\[ q = q_c \]

\[ \Delta p V = k \]

\[ q_p > q_s \]
Flow in production units are usually relatively well set, because of:
• published manufacturers' limits
• warranty

Flow on distribution side is generally well above design value, because of:
• safety factors
• "what can the most can the least" approach at all stages

In 90% of installations, flow in distribution is over 150% of the design value.
Source: Investigation by Costic (French Research and Training Centre in HVAC), published in CFP Journal April-May 2002.
Compatibility between flows

- Because the distribution pump is oversized, the distribution takes more flow than the production can provide.
- There is a mixing point in A between return water and supply water.
- The supply water temperature is higher than expected per design.
For a 43 F – 54 F – 75 F temperature regime

Decreased supply temperature effect -18% 

Overflow effect +10%
The problem of flow compatibility does not appear under all conditions:

- **At lower load**
everything seems to work fine.

- **At high load (when peak power is needed)**
incompatible flows limit the power which is transmitted from the production to the distribution. When $q_p < \Sigma q_s$, the flow reverses in the bypass and return water is mixed with supply water.
Compatibility between flows

**When the production side is designed as a loop:**

- Flow incompatibility starts by affecting the last circuits.
- The problem of flow compatibility appears only at high load.

\[ q_p > \Sigma q_s \]
Compatibility between flows – false solutions

- "Pushing" the distribution pump as a reaction to complaints makes the problem worse
- It increases the flow incompatibility and therefore the mixing
- Supply water temperature increases further in cooling (decreases further in heating)
Compatibility between flows – false solutions

- Decreasing (raising in heating) the set-point of the production unit can compensate for the incompatibility but at the cost of higher energy consumption.

Chiller manufacturers’ technical literature indicates extra energy use of approximately 3% per F that the chilled water supply temperature is lowered.
Compatibility between flows – false solutions

- Adding a production unit possibly solves compatibility issue but at the cost of an unnecessary production unit.
- It is not a good solution because the problem is not a problem of lack of installed capacity, it is a problem of too high flow in the distribution.
Compatibility between flows – the solution

Need to balance:

- Provide the design flow to all units.
- At the right supply water temperature.
- Do not take more flow than produced.

At each system interface, guarantee the flow compatibility thanks to balancing on the production side and on the distribution side.
The water flows must be compatible at system interfaces.
The design flow must be available at all terminal units in design conditions.

The differential pressure across control valves must not vary too much.

The water flows must be compatible at system interfaces.
The Three Keys

Three simple conditions to satisfy in order to make hydronic systems work as expected from design.

Production

Distribution

Terminal units
Thank you for your attention!

Questions?