2014 NEBB Annual Conference
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Application of the Hydronic System Curve: Simple but Powerful Tool for Achieving System Performance
The ASHRAE Systems Handbook Describes

\[
\left( \frac{\text{New Flow}}{\text{Known Flow}} \right)^2 = \left( \frac{\text{New Head}}{\text{Known Head}} \right)
\]

\[
\left( \frac{Q_2}{Q_1} \right)^2 = \left( \frac{h_2}{h_1} \right)
\]

- Many answers to balancing issues can be given "depth" by using the System Curve concept as a simple analysis tool of the hydronic system.
• Use System "Flow Coefficient", and spreadsheets to get a reasonably close idea of how a system will perform.

Once the distribution system is designed, the pressure loss at design flow is calculated by the methods discussed in Chapter 35 of the 2001 ASHRAE Handbook—Fundamentals. The relationship between flow rate and pressure loss can be expressed by

\[ Q = C_v \sqrt{\Delta p} \]  

(15)

where

- \( Q \) = system flow rate, gpm
- \( \Delta p \) = pressure drop in system, psi
- \( C_v \) = system constant (sometimes called valve coefficient, discussed in Chapter 42)

Equation (15) may be modified as follows:

\[ Q = C_s \sqrt{\Delta h} \]  

(16)

where

- \( \Delta h \) = system head loss, ft of fluid \([\Delta h = \Delta p / \rho] \)
- \( C_s \) = system constant \([C_s = 0.67C_v]\) for water with density

Equations (15) and (16) are the system constant form of the Bernoulli pressure equation. If the flow rate and head loss are known
So Why Don't We Use This More Often?

- The "Elegance" and "Power" Inferred Are Often Not Used

![Graph showing system flow and pressure relationships](image)

1 PSI or 2.31 Ft. Head

\[ h \approx \text{Head} \]

\[ Q \approx \text{Flow} \]

\[ C_s \approx C_v \]

Fig. 15  Typical System Curve
So Is The Exponent Really 2?

$\left( \frac{Q_2}{Q_1} \right)^2 = \left( \frac{h_2}{h_1} \right)$

- Some folks like to debate the magnitude of the exponent used in System Curves, but there is enough "slop" in hydronic systems calculations to make a discussion on others irrelevant, or to quote "A difference, in order to be a difference must be a big enough difference to make a difference"!
6” Pipe

- In normal design range; close enough
- Fitting losses are very inaccurate
So why bring the subject up?

• There are two different methods
  • Darcy-Weisbach (considered more accurate)
  • Hazen-Williams

• There are inherent inaccuracies in all fluid calculations
  • Pumps are not flow meters
  • Fitting losses vary by fluid velocity (e.g. “K” factors)
  • Certain fittings can have non-logical results, like branch lines of “Tees”

• “Squares” have been the norm for many complimentary calculations

• The method we present is to help apply some fundamental logic and method of calculation... not absolute precision
System Curve Review

- The System Curve Relationship Predicts What The Head Loss Will Be In A Given Circuit Path When Flows Change Per the Relationship;

\[
\left(\frac{Q_2}{Q_1}\right)^2 = \frac{h_2}{h_1}
\]
We might approach this by looking at drawings....

- Could calculate path losses
- Presumption: Pump is selected for the design by calculation
Suggestion: Analysis Schematic

- Flat layout; all paths with respect to pump easily seen and labeled
- Add flow data for segments
- Use this example to illustrate concepts
Typical System Curve Application

Model 5011
Fi & Ci Series
November 1, 2010
1760 RPM

Curve no. 2093
Min. Imp. Dia. 8.25"
Size 6 X 5 X 11.0

HEAD IN FEET

FLOW IN GALLONS PER MINUTE

Curves based on clear water with specific gravity of 1.0
What if System Had One Circuit...

Supply → SOURCE

<table>
<thead>
<tr>
<th>FLOW (GPM)</th>
<th>HEAD (FEET)</th>
</tr>
</thead>
<tbody>
<tr>
<td>800 GPM</td>
<td>100'</td>
</tr>
</tbody>
</table>

Return

FLOW (GPM) vs HEAD (FEET) graph showing 800 GPM at 100'
Flow Coefficient ($C_V$) can also represent system

- Equivalent to valve flow equation

\[
Q = C_V \sqrt{\Delta P}
\]

\[
C_V = \left( \frac{Q}{\sqrt{\Delta P}} \right)
\]

\[
\Delta P = \left( \frac{Q}{C_V} \right)^2
\]

\[
C_V = \frac{800}{\sqrt{100/2.31}}
\]

\[
C_V = \frac{800}{\sqrt{43.3}}
\]

\[
C_V = 121.6
\]

---

April 4, 2014

2014 NEBB National Conference Tech Session; Making System Curves Work For You
What if System Had One Circuit...

SOURCE

800 GPM
100’

CV = 121.6

RETURN

121.6

2.31’

800 GPM
100’

0 100 200 300 400 500 600 700 800 900 1000

FLOW (GPM)

0 10 20 30 40 50 60 70 80 90 100

HEAD (FEET)
So What?

• If we know that the $C_v$ is 121.6, we can predict what the flow will be at conditions other than 100’ when the system remains unchanged (valves open)

• If differential pressure were reduced across branch to 4 PSI, then flow would be 243.2 GPM
What if system had these pressure drops...

\[
C_V = \left( \frac{800}{\sqrt{58.5/2.31}} \right) = 158.97
\]

\[
C_V = \left( \frac{800}{\sqrt{30/2.31}} \right) = 221.99
\]

\[
C_V = \left( \frac{800}{\sqrt{11.5/2.31}} \right) = 358.55
\]

\[
C_V = \left( \frac{800}{\sqrt{100/2.31}} \right) = 58.5
\]

\[
C_V = \left( \frac{800}{\sqrt{800/2.31}} \right) = C_V = 121.6
\]

\[
C_V = \left( \frac{800}{\sqrt{100/2.31}} \right) = 2.31
\]

\[
C_V = \left( \frac{800}{\sqrt{800/2.31}} \right) = 2.31
\]

\[
C_V = \left( \frac{800}{\sqrt{30/2.31}} \right) = 2.31
\]

\[
C_V = \left( \frac{800}{\sqrt{11.5/2.31}} \right) = 2.31
\]

\[
C_V = \left( \frac{800}{\sqrt{100/2.31}} \right) = 2.31
\]

AVERAGE

\[
\text{AVERAGE} = \frac{158.97 + 221.99 + 358.55}{3} = 246.5
\]

WEIGHTED AVERAGE

?
Graphically there’s an explanation....

```
HEAD (FEET)

FLOW (GPM)

2.31'

ΔP

100'

58.5'

30'

11.5'

“SYSTEM” CURVE

“PVF” CURVE

“COIL” CURVE

“VALVE” CURVE

ΔP

2.31'

0 100 200 300 400 500 600 700 800 900 1000

10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160

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April 4, 2014
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The Math....

System Head = PVF Head Loss + Coil Head Loss + Control Valve Head Loss

\[ \Delta h \approx \Delta P \]

\[ \Delta P_{\text{Syst}} = \Delta P_{\text{PVF}} + \Delta P_{\text{Coil}} + \Delta P_{\text{Valve}} \]

\[ \Delta P = \left( \frac{Q}{C_V} \right)^2 \]

\[ \left( \frac{Q}{C_V-\text{Syst}} \right)^2 = \left( \frac{Q}{C_V-\text{PVF}} \right)^2 + \left( \frac{Q}{C_V-\text{Coil}} \right)^2 + \left( \frac{Q}{C_V-\text{Valve}} \right)^2 \]
The Example....

\[
\left( \frac{Q}{C_{V\text{-SYST}}} \right)^2 = \left( \frac{Q}{C_{V\text{-PVF}}} \right)^2 + \left( \frac{Q}{C_{V\text{-COIL}}} \right)^2 + \left( \frac{Q}{C_{V\text{-VALVE}}} \right)^2
\]

\[
\left( \frac{800}{C_{V\text{-SYST}}} \right)^2 = \left( \frac{800}{C_{V\text{-PVF}}} \right)^2 + \left( \frac{800}{C_{V\text{-COIL}}} \right)^2 + \left( \frac{800}{C_{V\text{-VALVE}}} \right)^2
\]

\[
\frac{1}{(C_{V\text{-SYST}})^2} = \frac{1}{(C_{V\text{-PVF}})^2} + \frac{1}{(C_{V\text{-COIL}})^2} + \frac{1}{(C_{V\text{-VALVE}})^2}
\]

\[
\left( \frac{1}{C_{V\text{-SYST}}} \right)^2 = \left( \frac{1}{159} \right)^2 + \left( \frac{1}{222} \right)^2 + \left( \frac{1}{358.5} \right)^2
\]

- Flow is common to all components

\[
\frac{1}{(C_{V\text{-EQUIV}})^2} = \frac{1}{(C_{V\text{-1}})^2} + \frac{1}{(C_{V\text{-2}})^2} + \ldots + \frac{1}{(C_{V\text{-n}})^2}
\]

- For as many components in series in the flow path

**GENERAL FORM OF EQUATION**
The Example....

\[
\left( \frac{1}{C_{V-SYST}} \right)^2 = \left( \frac{1}{159} \right)^2 + \left( \frac{1}{222} \right)^2 + \left( \frac{1}{358.5} \right)^2
\]

\[
\frac{1}{\left( C_{V-SYST} \right)^2} = 3.957 \times 10^{-5} + 2.03 \times 10^{-5} + 7.78 \times 10^{-6} = 6.76407 \times 10^{-5}
\]

\[
C_{V-SYST} = \sqrt{\frac{1}{6.76407 \times 10^{-5}}} = 121.59
\]
Notes

• Every component with a known flow and head loss can have a flow coefficient implied
  • Pipe; Size, , Flow Friction Loss / Length, Length of Pipe
  • Coils, Fixed Position Valves, Strainers

• Non-variable devices have a constant flow coefficient
  • Composite device coefficient may be defined, grouping NV devices

• Variable devices (Flow Control Valves) have a unique flow coefficient for each position
  • Calculating system flow coefficient for each variable device position with the device coefficient shows the ability of the device to adjust flow (Valve Authority)
Two Variables

\[
\frac{C_{V-1} \times C_{V-2}}{\sqrt{(C_{V-2})^2 + (C_{V-1})^2}} = C_{V-SYST}
\]

- Concept: Control Valve Authority compares the variable flow coefficient of the valve and the ability to adjust system flow under the influence of the other system pressure losses.
Understanding is best worked out by examples

• Classic example: Valve Authority

• Equal Percentage valves roughly follow this relationship

\[ C_v = C_{VMAX} R^{(\ell-1)} \]

<table>
<thead>
<tr>
<th>Height</th>
<th>Range-ability</th>
<th>Cv Max</th>
<th>Cv Position</th>
</tr>
</thead>
<tbody>
<tr>
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<td>71.2</td>
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<td>50.6</td>
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<tr>
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<td>100</td>
<td>3.3</td>
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</table>
Valve Authority

• Concept: Control Valve Authority compares the variable flow coefficient of the valve and the ability to adjust system flow under the influence of the other system pressure losses.

\[ \beta = \frac{\Delta P_{\text{MIN}}}{\Delta P_{\text{MAX}}} \]

"Index Value"

• Our example: Control valve had pressure drop of 11.5’ Head, System has 100’ Head

\[ \beta = \frac{11.5}{100} = 11.5\% \]
### Our example: System components have 88.5’ (100-11.5) Head; $C_v=129.3$

<table>
<thead>
<tr>
<th>Height</th>
<th>Range-ability</th>
<th>Cv Max</th>
<th>Cv Position</th>
<th>Modified Cv</th>
<th>System Components</th>
<th>System Cv</th>
<th>Percent Valve</th>
<th>Percent System</th>
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<tr>
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<td>357.8</td>
<td>357.8</td>
<td>129.3</td>
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<td>100%</td>
</tr>
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<td>90%</td>
<td>30</td>
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<td>254.6</td>
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<td>33.1</td>
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<td>357.8</td>
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<td>4%</td>
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<td>357.8</td>
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<td>129.3</td>
<td>0.0</td>
<td>0%</td>
<td>0%</td>
</tr>
</tbody>
</table>

\[
C_{V-1} \times C_{V-2} = \sqrt{(C_{V-2})^2 + (C_{V-1})^2} = C_{V-SYST}
\]
Parallel Paths

- System 800 GPM @ 100' Head
- All circuits are proportionally balanced
  - All circuits have same head loss (100')
- Distribution losses as shown
  - Coil & Valve same as previous example
Parallel Paths

<table>
<thead>
<tr>
<th>PATH</th>
<th>FLOW</th>
<th>HEAD LOSS</th>
<th>FLOW COEFFICIENT</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>50</td>
<td>100</td>
<td>7.6</td>
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<td>2</td>
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<td>100</td>
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<td>3</td>
<td>200</td>
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<td>30.4</td>
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<tr>
<td>4</td>
<td>450</td>
<td>100</td>
<td>68.4</td>
</tr>
<tr>
<td>SYSTEM</td>
<td>800</td>
<td>100</td>
<td>121.6</td>
</tr>
</tbody>
</table>

- Is the answer obvious?
- System flow coefficient is the sum of path flow coefficients
If we drew System Curves...

- Parallel paths add flows on lines of constant head to create system curve
  - Conceptually can be used graphically in air systems where flow coefficient “doesn’t exist”
- What happens when the system isn’t “balanced”
  - What are the effects of balancing valves
  - Control Valves, VS Pumps, etc.
Example: Parallel Paths

• What Happens when Path 3 control valve closes?

<table>
<thead>
<tr>
<th>PATH</th>
<th>FLOW</th>
<th>HEAD LOSS</th>
<th>FLOW COEFFICIENT</th>
</tr>
</thead>
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<td>1</td>
<td>50</td>
<td>100</td>
<td>7.6</td>
</tr>
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<td>450</td>
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<td>68.4</td>
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<tr>
<td>SYSTEM</td>
<td>800</td>
<td>100</td>
<td>91.2</td>
</tr>
</tbody>
</table>
Typical System Curve Application
Example: What happens when not balanced?

- What Happens when Path 3 control valve closes?

- Organize layout into grid for spreadsheet
  - Letters for main pipes
  - Numbers for branch pipes
### Spreadsheet Analysis

<table>
<thead>
<tr>
<th>PATH 1</th>
<th>PUMP TRIM</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>COIL</th>
<th>VALVE</th>
<th>BV</th>
<th>E</th>
<th>F</th>
<th>G</th>
<th>H</th>
<th>PATH HEAD</th>
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<tr>
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</tr>
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</table>

$$\frac{1}{C_V} = X \Rightarrow \sqrt{\frac{1}{X}} = C_V$$

<table>
<thead>
<tr>
<th>PATH 2</th>
<th>PUMP TRIM</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
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<th>VALVE</th>
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<th>C</th>
<th>D</th>
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<th>VALVE</th>
<th>BV</th>
<th>E</th>
<th>F</th>
<th>G</th>
<th>H</th>
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April 4, 2014

2014 NEBB National Conference Tech Session; Making Systyem Curves Work For You

43
### Spreadsheet Analysis

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**SYSTEM CV**

126.5113
Draw a system curve to intersect the pump...
What happens if the large Path 4 flow rate was in Path 1?

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SYSTEM CV
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Location Matters

- Unbalanced, just locating the 450 GPM load close to the pump increases unbalanced flow from 830 GPM to 900 GPM (+12.5%)

- Is this surprising? Past anecdotal guidance has suggested not locating “big coils” close to the pump.
Careful calculation starts with the organized layout

Normally:
- Bigger
- Spreadout
- Mix pipe layouts
- Combine dissimilar devices
- etc.
Consider the flat hydronic schematic

- Flat Layout; No crossing lines, leads to easy spreadsheet calculation
- Does this really add a lot of time to the contract?
- Some systems really require doing this to understand what’s going on and to make adjustments.
Spreadsheets are the Designers & Balancers friend...

- Calculate system and path head losses
- Calculate system and path flow coefficients
  - $C_v$ makes initial analysis infinitely clearer
  - Makes balancing the diverse system understandable, possible
A comment on Calculation

• Designers have an ethical responsibility to calculate the hydronic system...
  • ASHRAE Standard 111 TAB requires designers to forward their calculations to the TAB agency

• Balancers don’t feel paid to calculate system hydraulics
  • It’s not that hard! It doesn’t need to be that precise!
  • Balancers are paid to provide a report and test data, part of which are data collected from submittals and design schedules...
  • A day or two in the office probably saves a week or more in the field feeling mystified
Example

- Four Floors
- 2 coils per floor
- 100 GPM per coil
- Direct Return distribution
- 75%/25% Diversity; 800 GPM Block, 600 GPM Design Flow
- Distribution selected at 600 GPM
- Proportional operation and balance
- Required Control Characteristic; Linear
  - VAV AHU with Outdoor Air Intake
• Flat layout; all paths wrt pump easily seen and labeled
• Add flow data for segments
• Path to Coil 1 has most friction loss for sizing pump
• Typically pump selected for 600 GPM @66 Ft. Hd.
• However, this selection is a “fake” number
  • If all valves open there is a potential greater capacity of 800 GPM
  • Pipes & friction losses based on 600 GPM
Issue; Do you have to go to the field and measure?

• No (sort of)
• Review application of System Flow Coefficient theory
### PATH 1

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**CALCULATED AS NON-BALANCED CONDITION**

Req To Balance

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**CALCULATED AS NON-BALANCED CONDITION**

Req To Balance

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**CALCULATED AS NON-BALANCED CONDITION**

Req To Balance
Review; Paths in parallel

\[ \text{SYSTEM} C_v = \text{PATH}_1 C_v + \text{PATH}_2 C_v + \ldots + \text{PATH}_8 C_v \]

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• Several things for the price of one!
  • System flow coefficient allows you to draw “system curve”
  • Doing the calculation tells you how much drop initially needs for balance (Preset Method)
  • Analyze balanced vs. unbalanced conditions
Establish “Un-Balanced” System Curve Via Flow Coefficient

1750 RPM

DESIGN FLOW
600 GPM @ 66.23 FT

UNBALANCED FLOW
844 GPM @ 60 FT

“SYSTEM CURVE”
600 GPM @ 66 FT
C₅ = 112@2.31 FT

25% DIVERSITY CV
165.6 GPM @ 2.31 FT
Typically balancers will want to adjust...

- No calcs, start at Path 8, work way out from pump
- Probably try to adjust to 100 GPM (design)
- Because of our pipe selections, they’ll get close, but...
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Start balancing from furthest point

- If calculations are done, pre-setting the valve head loss is simple, makes the flows all close to equal and proportional

- The catch... Flow is still greater than the diverse flow requirement of 600 GPM
  - Positive; If pump has flow capacity when valves all open they have almost design flow
  - Negative; Excess flow to chiller / boiler changes thermal performance
If design flow is important...

- **DESIGN FLOW**: 600 GPM @ 66.23 FT
- **UNBALANCED FLOW**: 844 GPM @ 60 FT
- **BALANCED FLOW**: 777 GPM @ 62.5 FT
- **25% DIVERSITY CV**: 149.4 GPM @ 2.31 FT
- **1750 RPM**: 1750 RPM
- **1351 RPM**: 1351 RPM
A note on the numbers...

• Basic design principles...
• Pump selection just happened to cover unbalanced and balanced flow rates... this doesn’t always happen!
• Not a lot of change in pump head...
  • Design 66’
  • Unbalanced 60’
• Big change in flow because of piping network
• Horsepower
  • Design: 600 @ 66’/73% → 13.8 Hp
  • Unbalanced: 844 @ 60’/81% → 15.8 Hp
  • Balanced System All Open: 777 @ 62.5’/80% → 15.3 Hp
  • Balanced System Diverse Flow: 600 @ 37.3’/80% → 7 Hp
Key to the diverse balance

• Do not primarily adjust Individual paths to the percentage of the diverse flow...
  • Must adjust to coil full design flow; Coil controller must have ability to get full flow for design heat transfer
  • A diverse pump selection is a “fake” selection point, not necessarily accounting for full design flow
  • The “extra” flow paths, or path capacity due to the control valve opening fully will move the system curve out on the pump curve, causing more flow

• How can systems be made to operate better, especially in variable speed
  • Decision analysis, More involved control algorithms/sequences
Control & Balancing Valve

• Standard Modulating Control Valves
  • Remember to account for control valve authority effects
  • Static Balance Valve provides proportionality and \textit{coordinated} flow limiting, however VS Pump control area can cause issues
  • Flow limiting valves do little to help in this application; Not Proportional
    • Under true “design” operation the pump doesn’t have enough capacity
    • Far circuits starve (proportionality)

• Pressure Independent Control Valves
  • Can limit maximum flow with “start up sequence” from the controller
  • Much more integrated control system sequence of operation...don’t expect your balancer or control programmer to figure it out for you
What do I mean by that?

• We established early on:
  • “Diversity” links to proportional control hence part load throttled flow
  • At any given time only a percentage of block design flow is required
  • Control valve has to close part way

• We have to “engineer” the performance for all conditions

• Start Up: All zones are below or above set point
  • Zone controllers are stupid: All valves open
  • All valves open they attempt to get full load design flow (demand greater than supply)
  • Pump can over-flow or give unacceptable performance; Water goes to path of least resistance

• Start Up Sequence: Control signal output is limited, corresponding to maximum diverse flow requirement
Field Example; 50% Block Load Flow

- Residential/Commercial
- Fan Coil
- Lot’s of risers
- Distribution piping at 50%
- August / Rain / Leaks
  - Mold
  - Want to dehumidify
- Couldn’t cool building
- Balancing
  - Tech tried preset, Designer rejected
- “Run 2 Pumps”
- Lawsuits
Pump(s)....

Unbalanced System Curve via Analysis

“ORDERED”

SPEC’D

Spec Flow & Head

Parallel Pump Curve

50% “Diverse Flow”
Field Example; 50% Block Load Flow

- 2 Pumps does not equal twice the flow
- However, if they had temporarily re-piped...
Pumped in series...
Is this an uncommon occurrence?

- Probably not
  - 3 semi-similar occurrences that same year

- What should solution have been?
  - Re-size certain pipes (Didn’t do)
    - Modified distribution system design (more major)
  - Modify pumps (Did do)

- Questionable design decisions
  - Aside from the obvious...
  - Fan Coil & 2 Position Valve...
Another example... sort of second hand

• PICV Valve application
• Water hammer on floors during commissioning & balancing process
• According to owner...
  • Valves set at factory for twice as much flow as design schedules
• Think though about similarities
  • Valve capacity twice that of pump capability to provide... sort of in the same realm as “diversity”
Working Example

- Four Floors
- 2 coils per floor
- 100 GPM per coil
- Direct Return distribution
- 75%/25% Diversity; 800 GPM Block, 600 GPM Design Flow
- Distribution selected at 600 GPM
- Proportional operation and balance
- Required Control Characteristic; Linear
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