Facility Dynamics Engineering

Introduction to the Controlled Systems

Ducts and Pipes (Supplemental)

Presented By:

David Sellers; Facility Dynamics Engineering

Senior Engineer

NAVFAC, San Diego

What We Will Cover in This Module

- Basic principles for flow in pipes and ducts
- How fans and pumps interact with ducts and pipes

The Square Law:
Common to both Air and Water Systems

- Roots in the Darcey - Weisbach equation

$$H_L = f\left(\frac{L}{D}\right)\left(\frac{V^2}{2g}\right)$$

The Square Law:
Common to both Air and Water Systems

- Roots in the Darcey - Weisbach equation
- Applies to fully developed turbulent flow

Nikuardse's Experiment

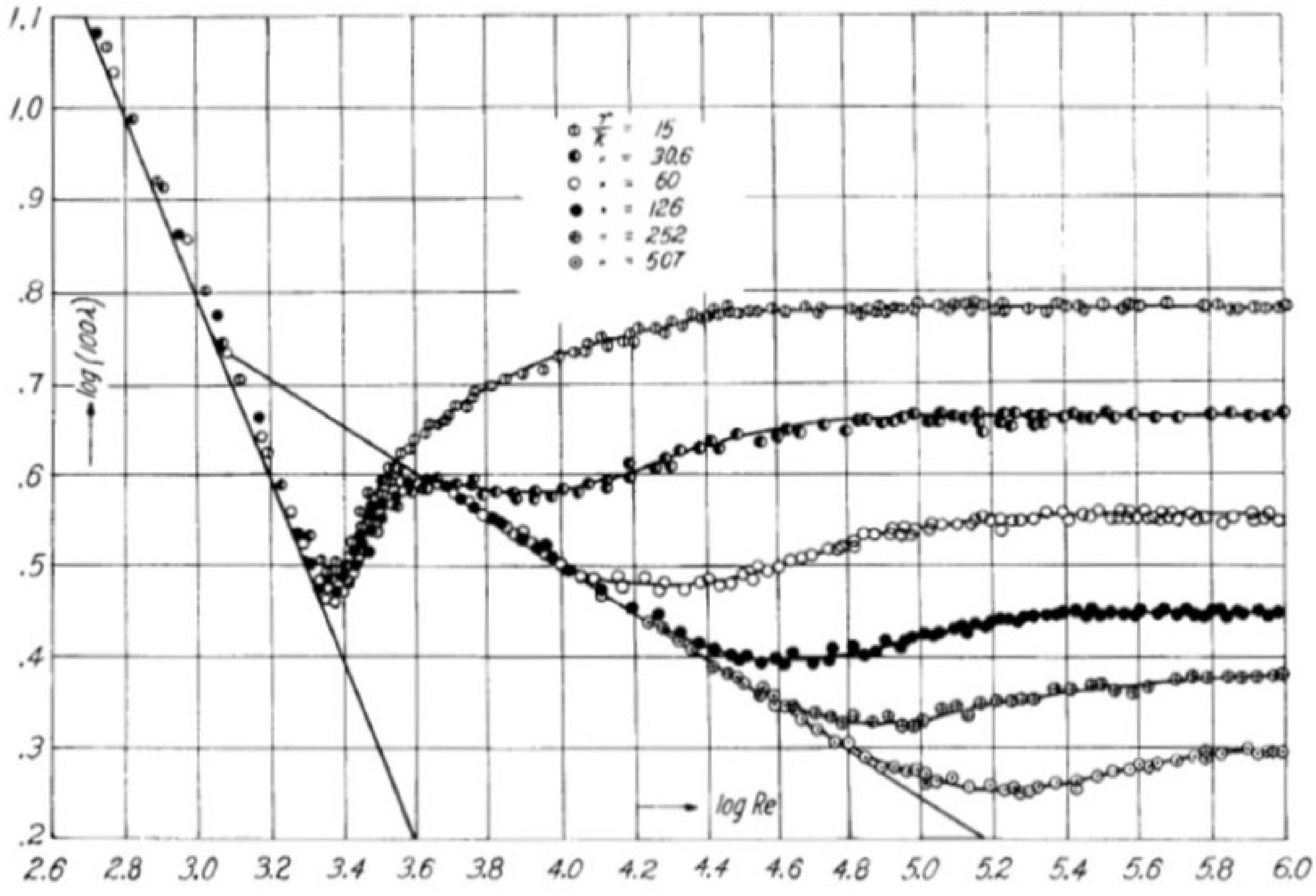
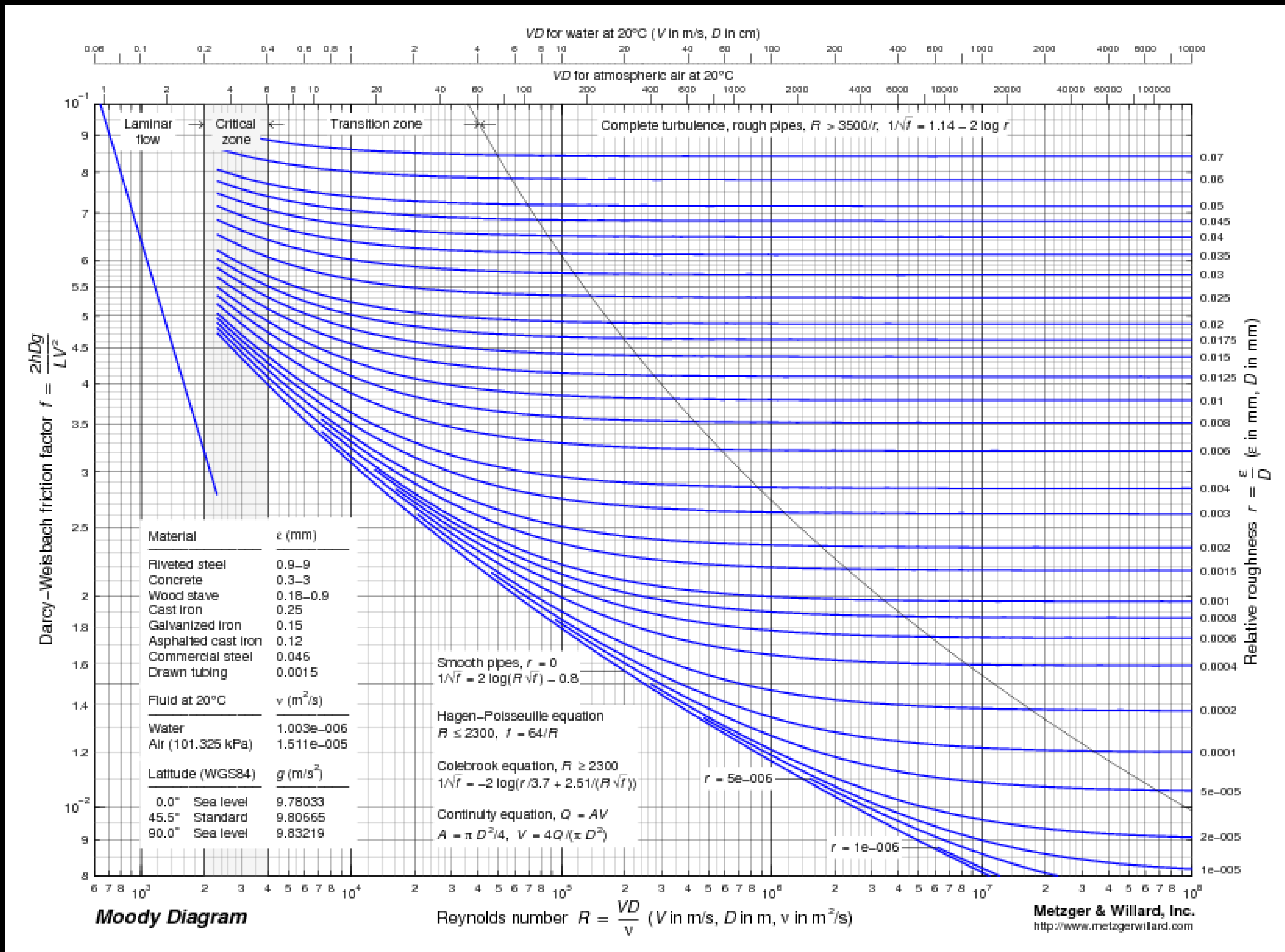


Figure 9.- Relation between log(100 λ) and log Re.

NACA TM 1292

51

Moody Diagram



Duct Friction Chart

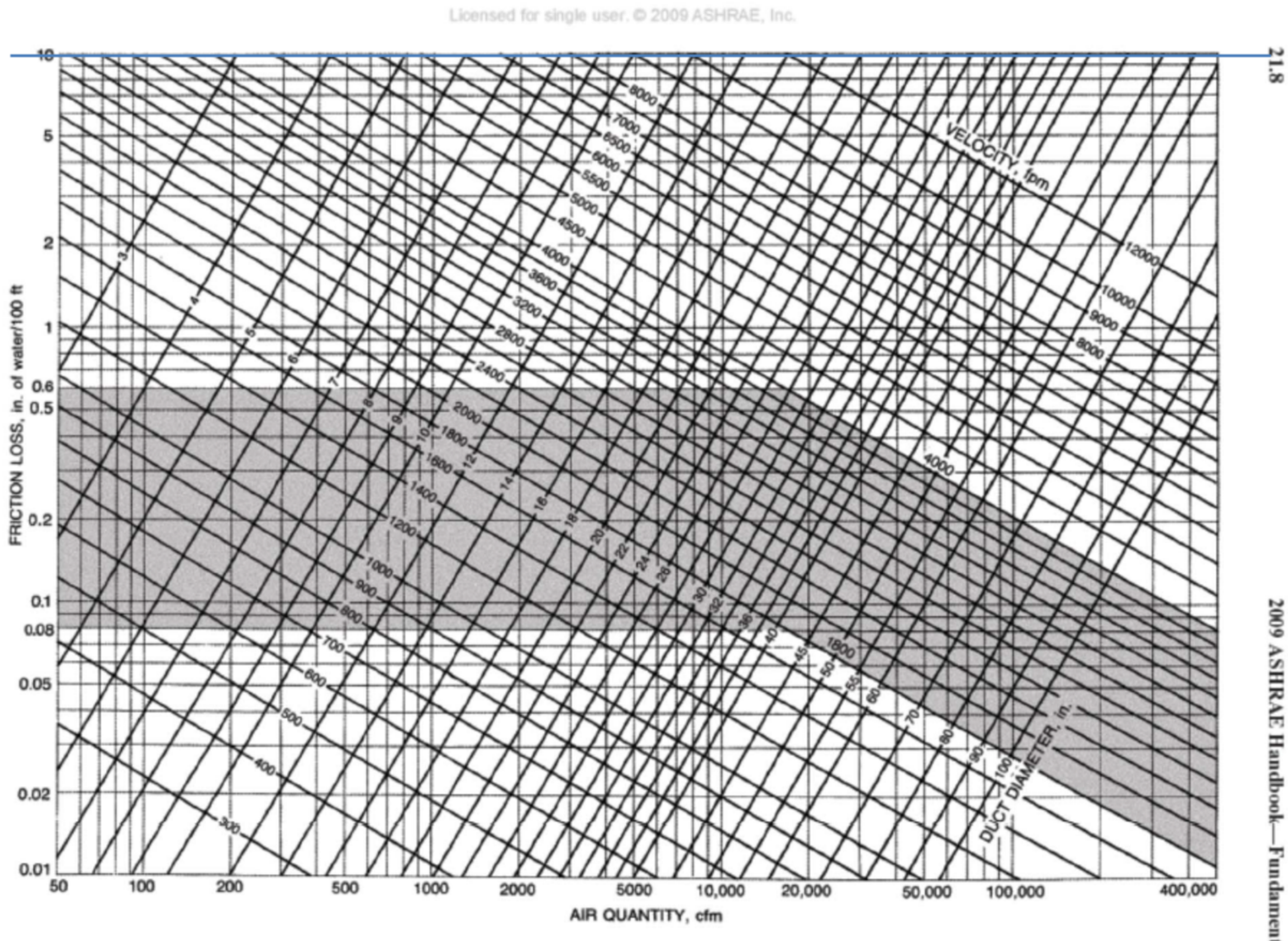
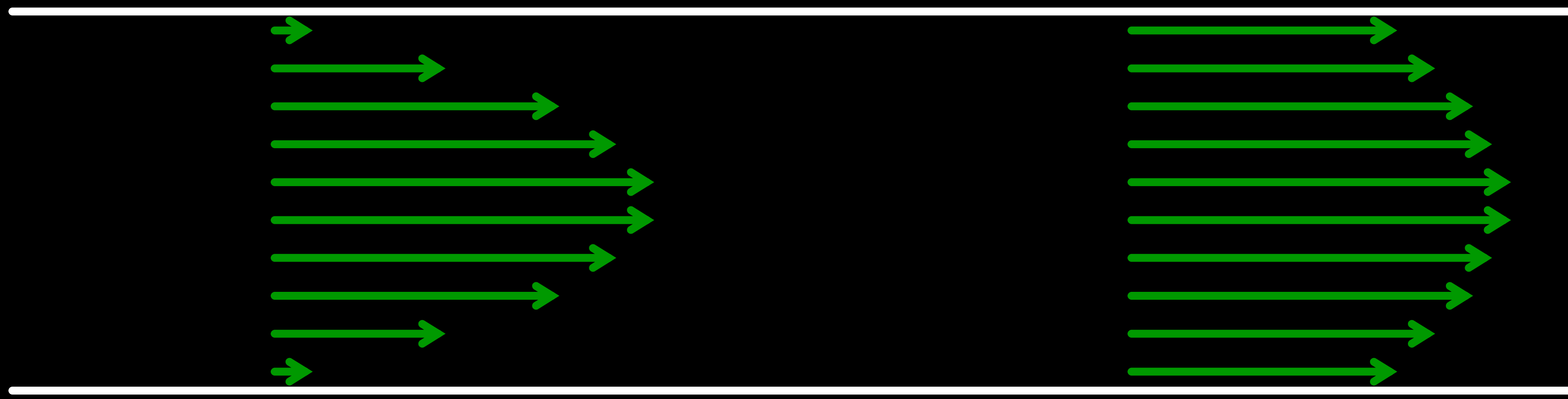


Fig. 9 Friction Chart for Round Duct ($\rho = 0.075\ lb_m/ft^3$ and $\varepsilon = 0.0003$ ft)

Turbulent vs. Laminar



Fully Developed Laminar Flow
Average Velocity

Fully Developed Turbulent Flow
Average Velocity

The vectors typically shown to represent the velocity profile in a pipe are usually an indication of the average velocity for a given fluid particle over time not the streamlines

- The actual motion of any given particle could be laminar (parallel lines) or turbulent (swirling lines)
- Laminar flow will tend to have a parabolic average velocity profile shape
- Turbulent flow will have a flatter, fuller velocity profile shape
- Fully developed flow exists when the average flow profile does not change as the fluid moves down the conduit

The Square Law:
Common to both Air and Water Systems

- Roots in the Darcey - Weisbach equation
- Applies to fully developed turbulent flow

$$Pressure_{New} = Pressure_{Old} \times \left(\frac{Flow_{New}}{Flow_{Old}} \right)^2$$

Where:

$Pressure_{New}$ = The pressure you want to know in consistent units

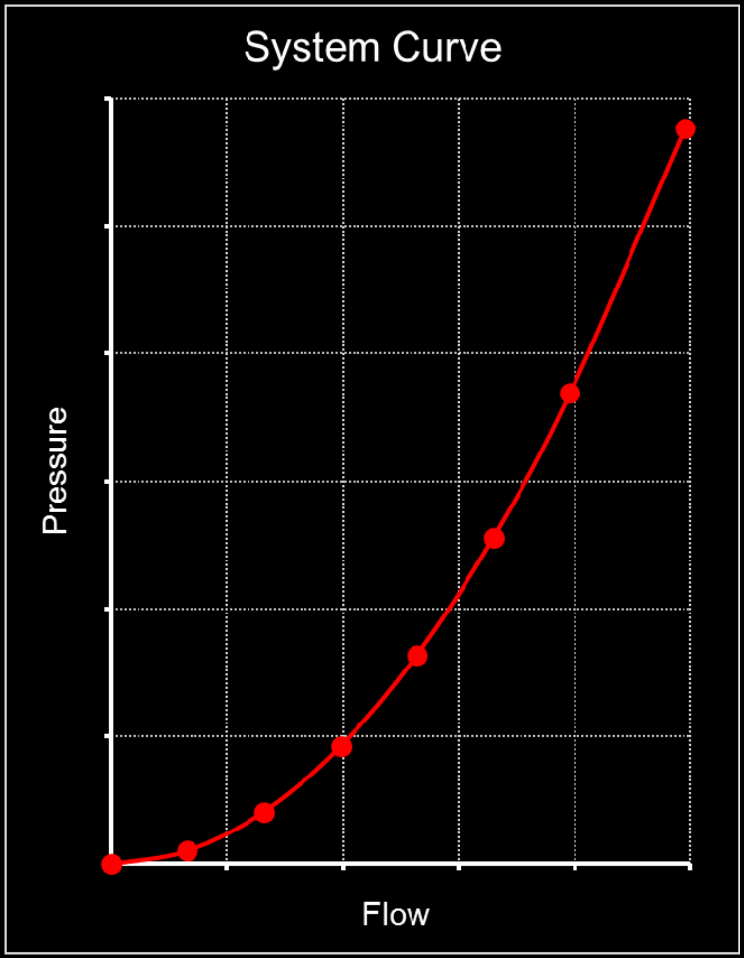
$Pressure_{Old}$ = The pressure you know in consistent units

$Flow_{New}$ = The pressure you want to know in consistent units

$Flow_{Old}$ = The pressure you want to know in consistent units

The Square Law:
Common to both Air and Water Systems

- Applies to a fixed system
  - If a damper moves in an air system, you generate a new system curve
  - If a valve moves in a water system you generate a new system curve
- There are places in our systems where the flow is not fully developed turbulent flow
  - ASHRAE research suggests the nominal exponent is 1.85 – 1.89
  - Using 2 is close enough for field work most of the time



Why This Matters if You are Working with Control Systems

Fan power is a function of flow and static pressure

$$bhp = \left(\frac{cfm \times static}{6{,}356 \times \eta_{fan_{static}}} \right)$$

Where:

bhp = Brake horse power into the fan drive shaft

cfm = Flow rate in cubic feet per minute

$static$ = Fan static pressure

$6{,}356$ = A units conversion constant

$\eta_{fan_{static}}$ = Fan static efficiency; .40 = .60 for small fans,

.68 - .78 for large fans

Divide by motor efficiency and multiply by .746 kW per horse power to get killoWatts

Why This Matters if You are Working with Control Systems

The Square Law says that if we reduce the flow rate in a given system, the static required to deliver the flow will be reduced exponentially

- Cut the flow by 50%, then you cut the static required to 25% of what it was (.5 x .5 or $.5^2$)

$$Pressure_{New} = Pressure_{Old} \times \left(\frac{Flow_{New}}{Flow_{Old}} \right)^2$$

Where:

$Pressure_{New}$ = The pressure you want to know in consistent units

$Pressure_{Old}$ = The pressure you know in consistent units

$Flow_{New}$ = The pressure you want to know in consistent units

$Flow_{Old}$ = The pressure you want to know in consistent units

Why This Matters if You are Working with Control Systems

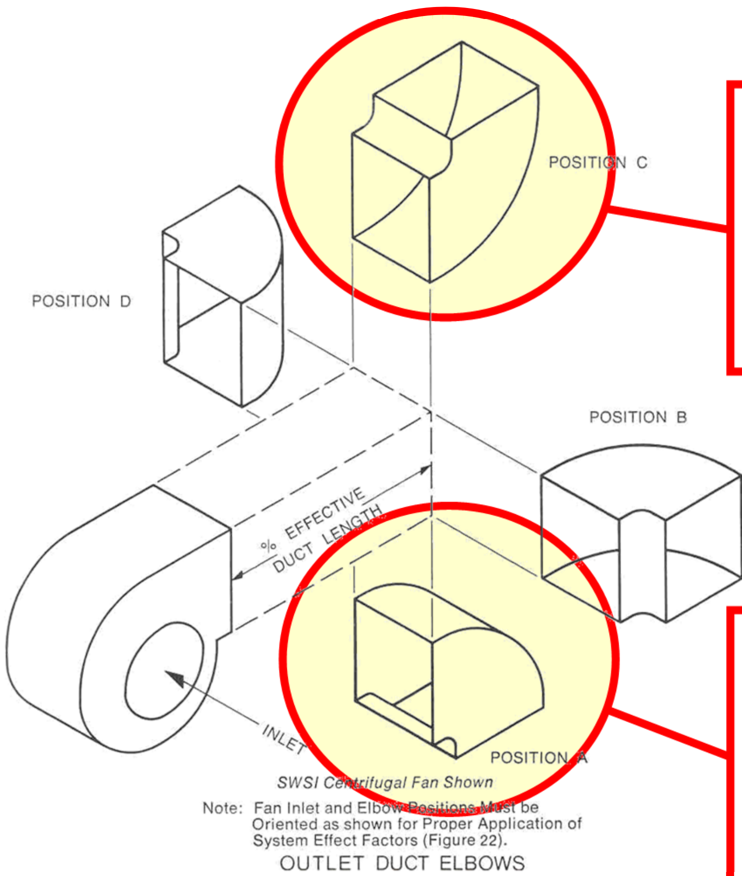
As a result, for a fixed system, fan power varies as the cube of the flow rate

$$bhp \propto Flow^3$$

- *Control systems can optimize both the flow and the static required by a system, thus control systems can optimize the fan power required by a system*
- *Control systems can also mess this up if we are not careful and as a result waste energy and resources*

System Effect:
Where the Duct Meets the Fan

Fitting placement and/or lack of a discharge duct has the same effect as adding static pressure

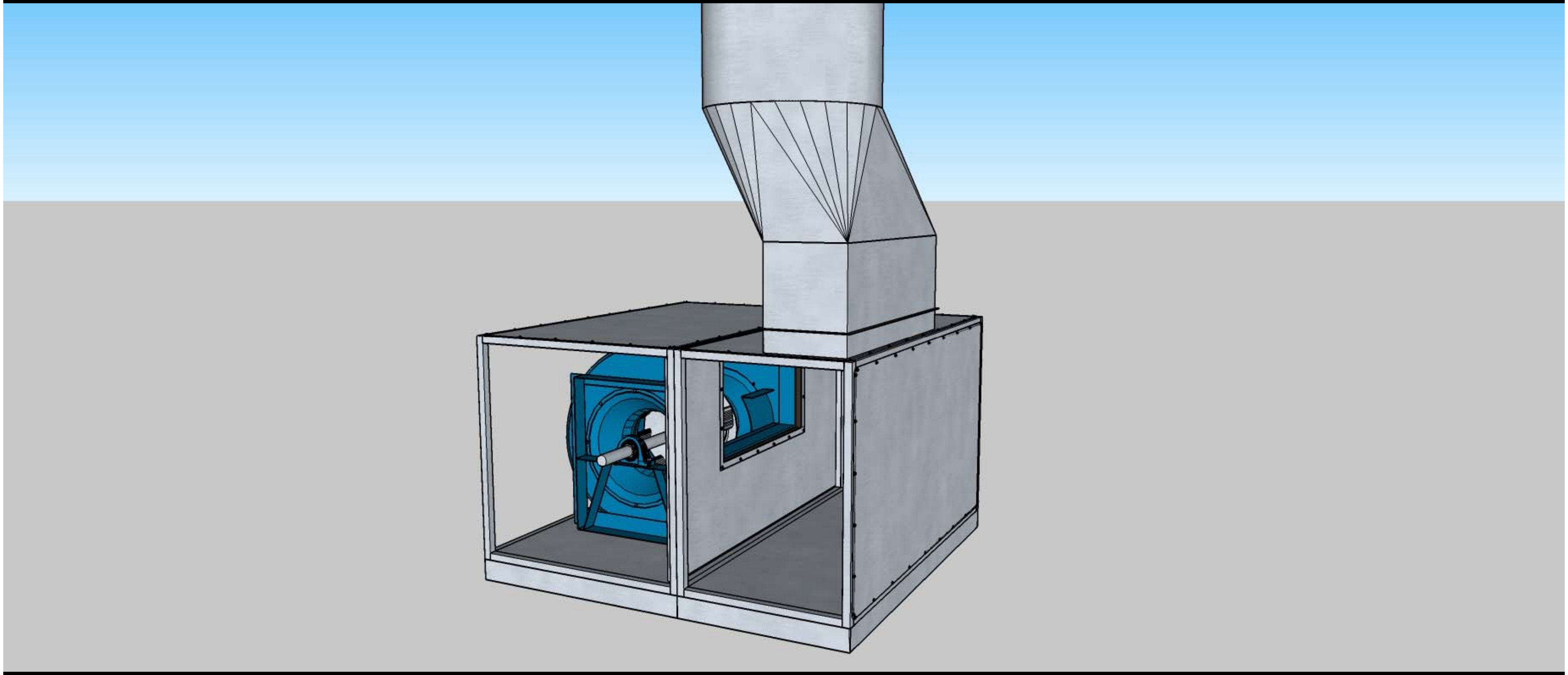


- .55 in.w.c. at 0% effective duct length
- .18 in.w.c. at 50% effective duct length

System effect assessed at 2,500 fpm and an outlet area to blast area ratio of 70%

- .20 in.w.c. at 0% effective duct length
- .05 in.w.c. at 50% effective duct length

System Effect:
Where the Duct Meets the Fan



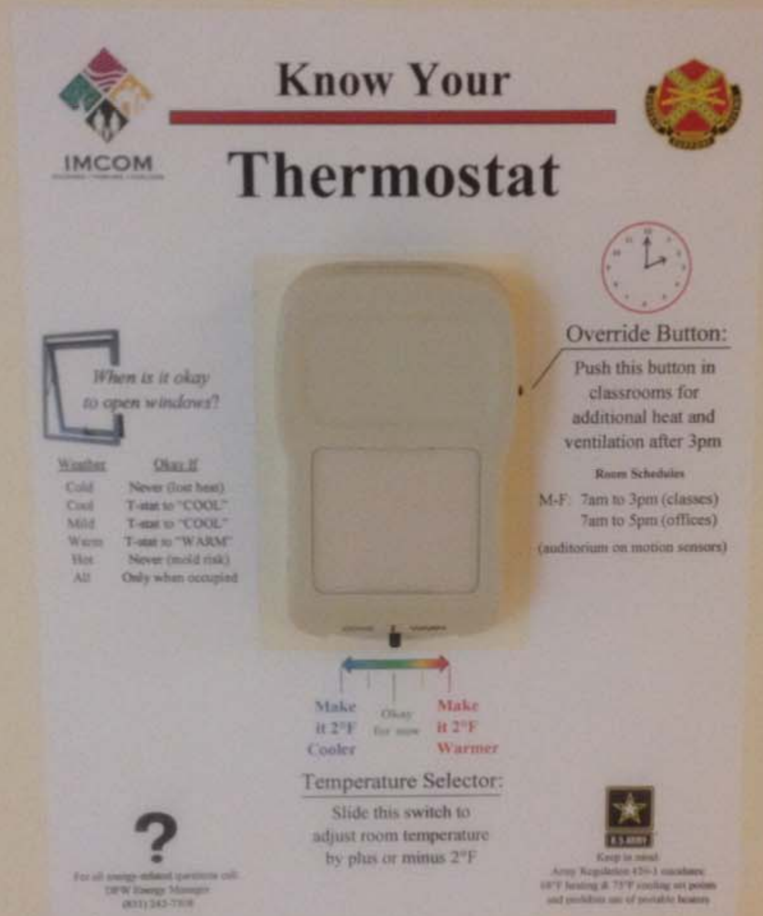
See *System Effect–Dealing with the Point Where the Fan Meets the Duct* for more in formation at www.Av8rDAS.Wordpress.com

The Control System *Can Not* Change Elbows or Correct System Effect Problems

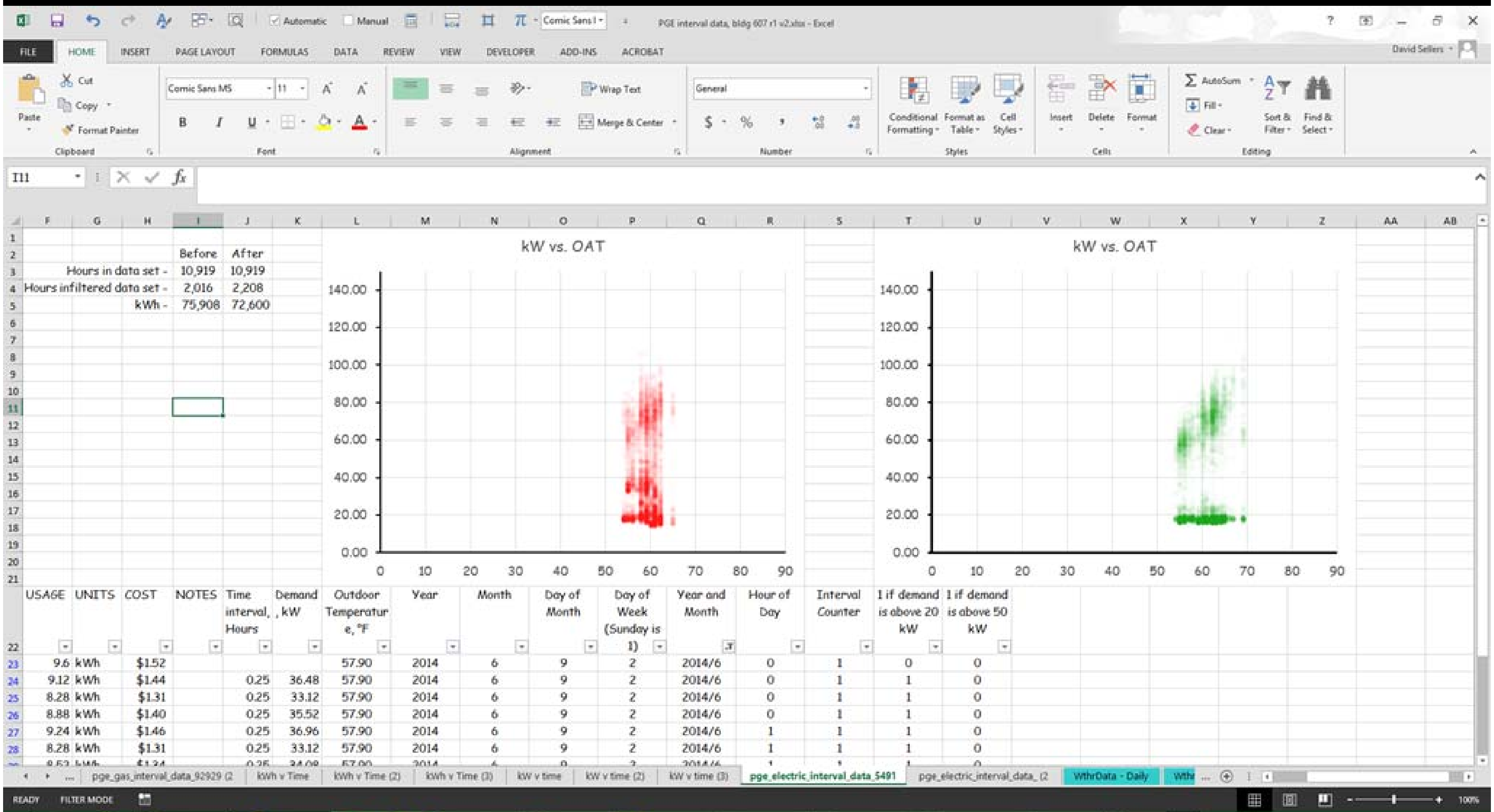
But it can optimize the system and minimize their impact

- For the system in the example, improvements to the control system logic:
  - Optimized operating hours by scheduling at the zone level
  - Optimized minimum flows
    - Reduced fan power
    - Reduced reheat

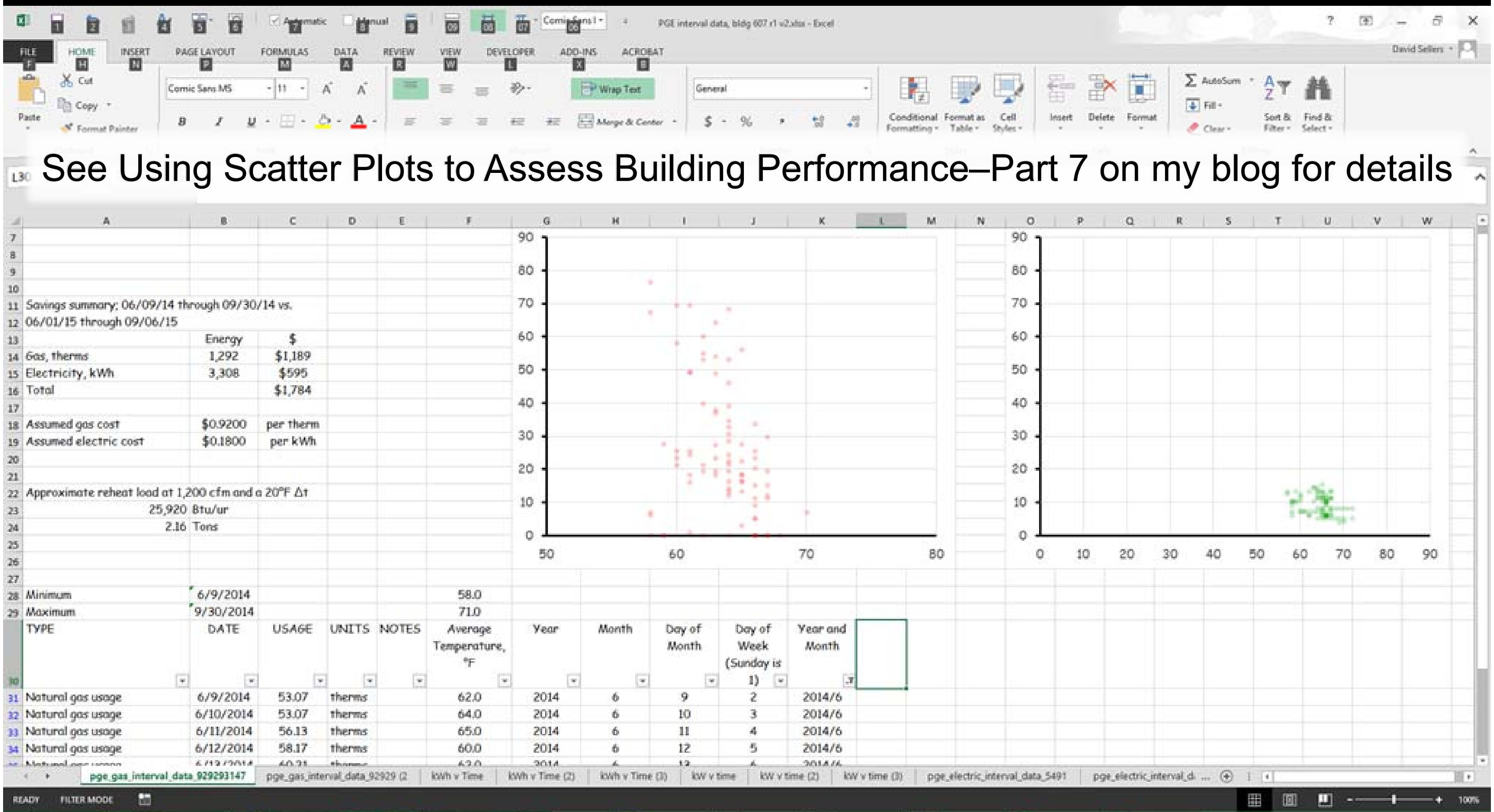
Training the Users; An Important Step



Scheduling and Training Alone had Significant Impact on Energy Consumption



Scheduling and Training Alone had Significant Impact on Energy Consumption



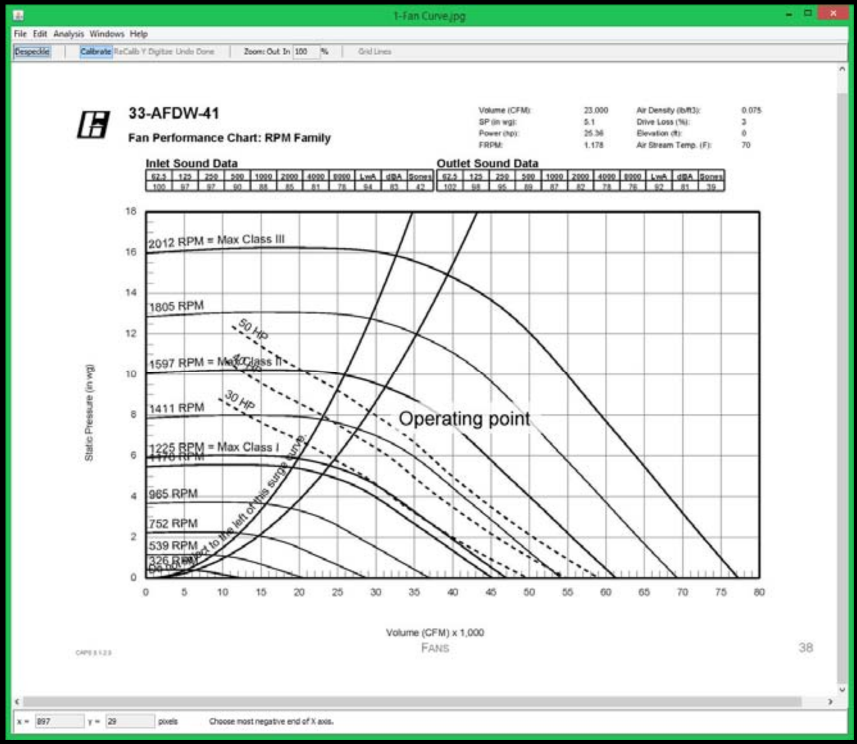
See Using Scatter Plots to Assess Building Performance–Part 7 on my blog for details

Real World Systems are More Complex

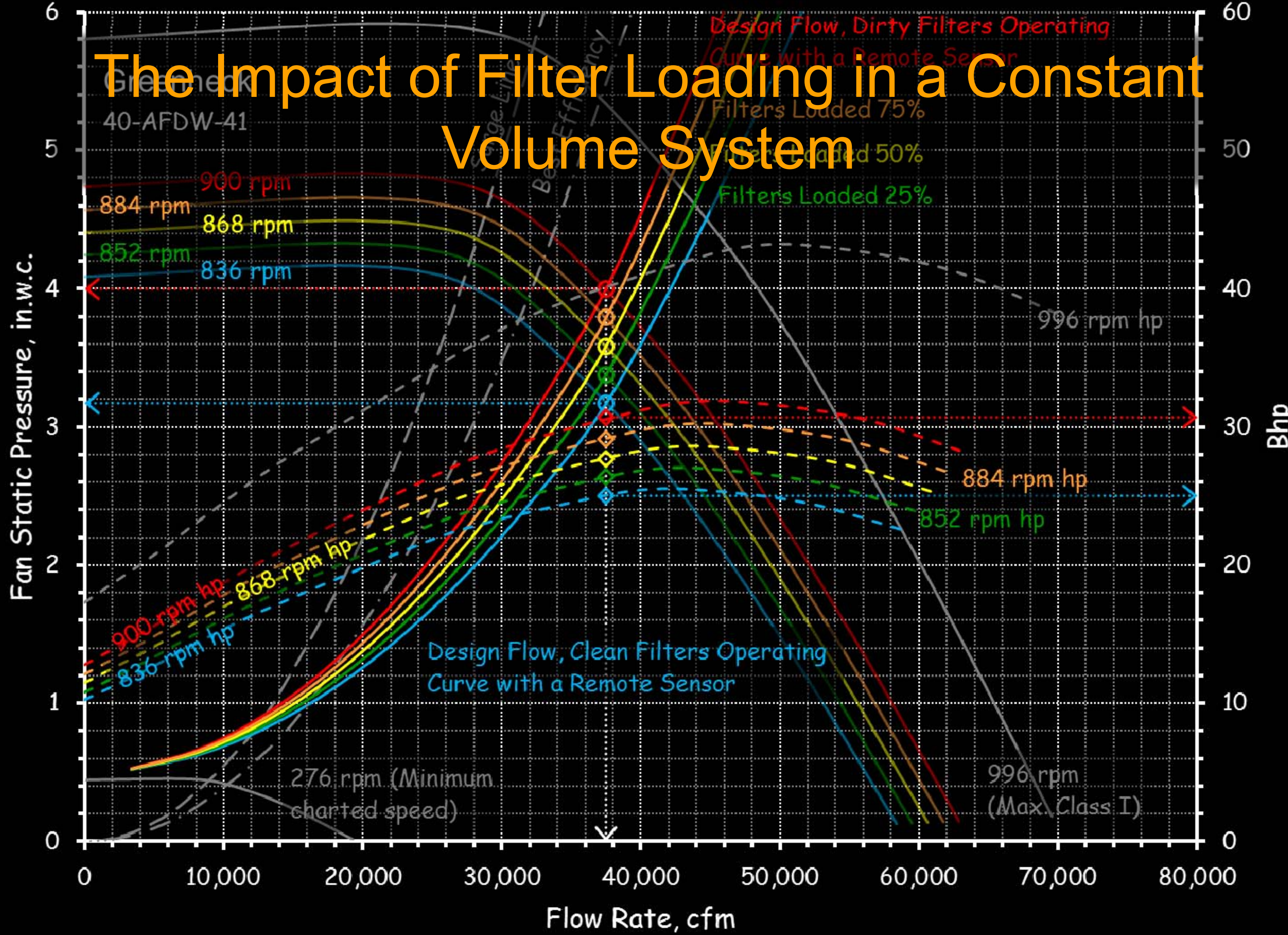
- Anything that moves or changes in the system creates a new system curve
- Constant volume systems are not constant volume
  - Filters load over time in air handling systems
  - Parallel pumps sharing common headers interact
- Variable air volume systems operate on a family of system curves by design

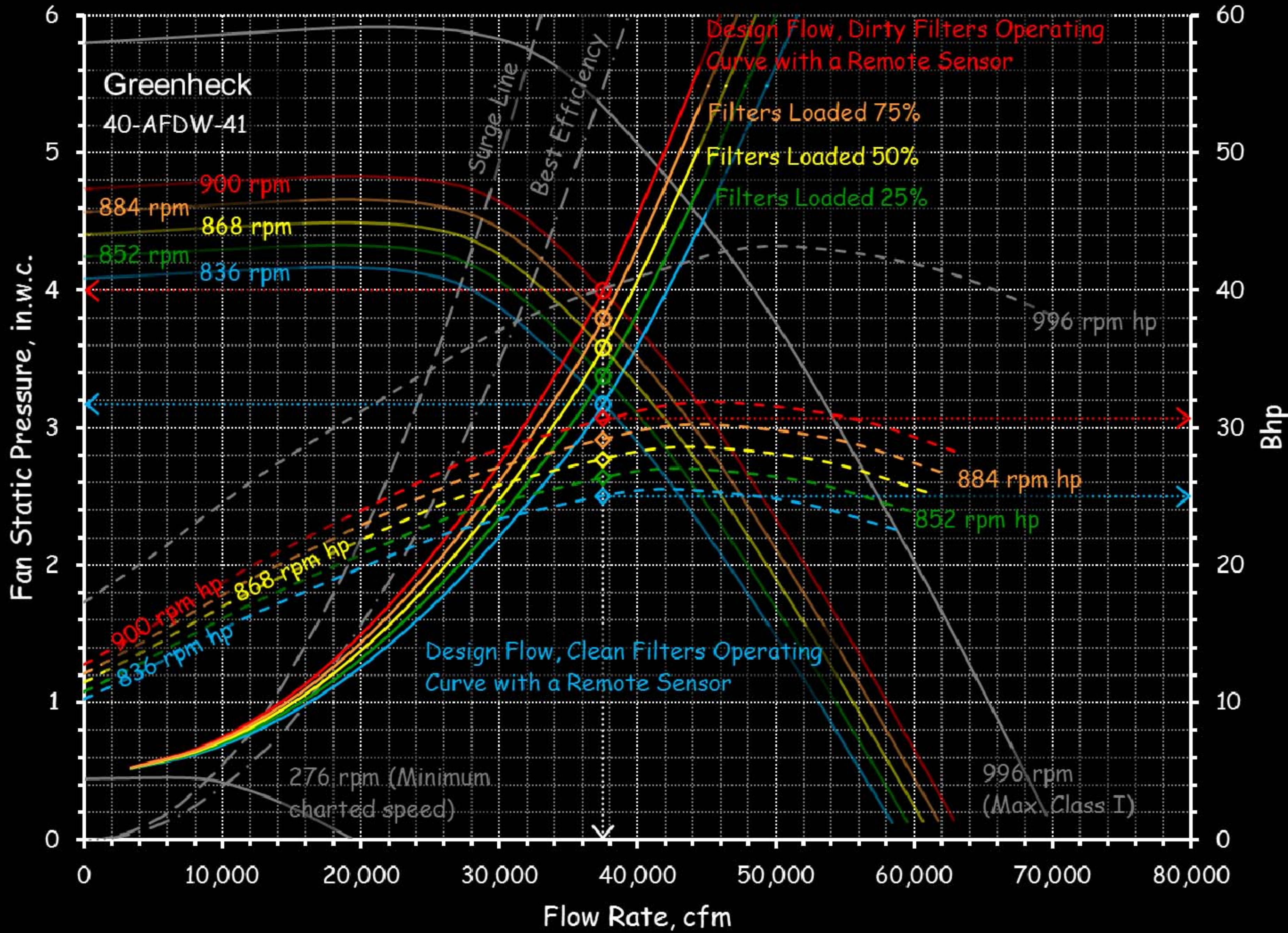
Plot Digitizer; A Free, Handy Resource

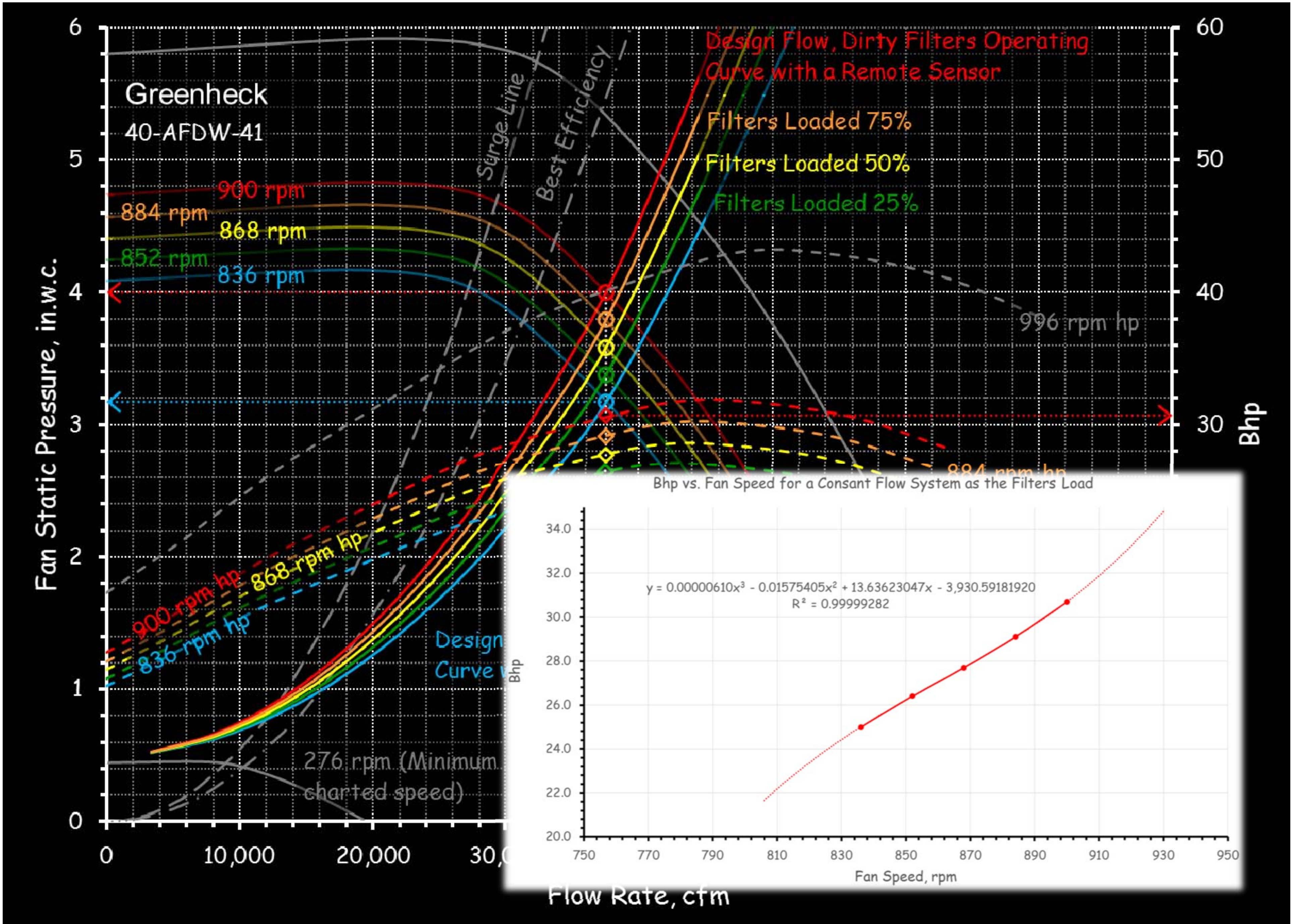
- Shareware
- Lets you digitize lines by clicking on them over an image
- http://tinyurl.com/PlotDigitizer



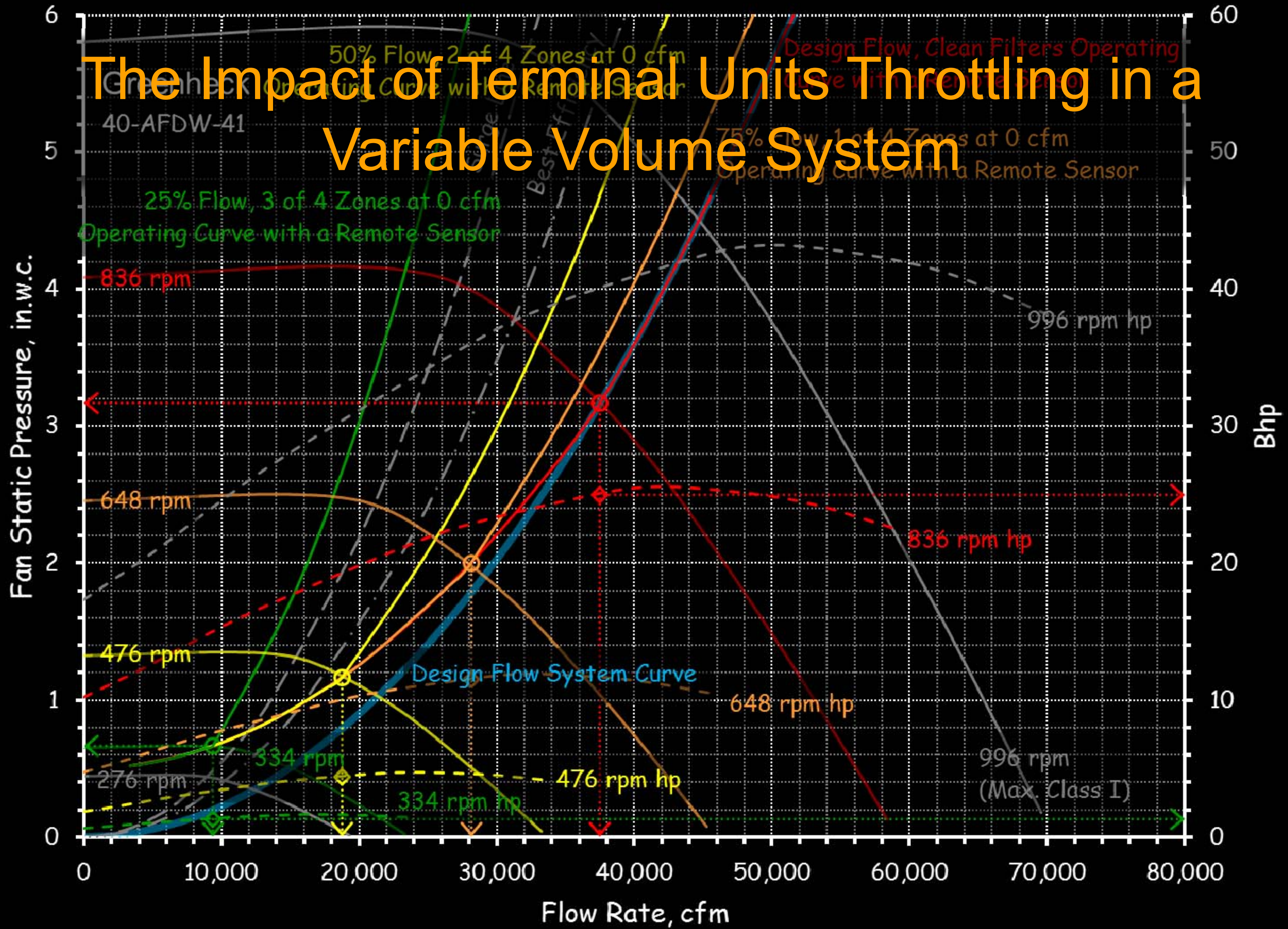
The Impact of Filter Loading in a Constant Volume System

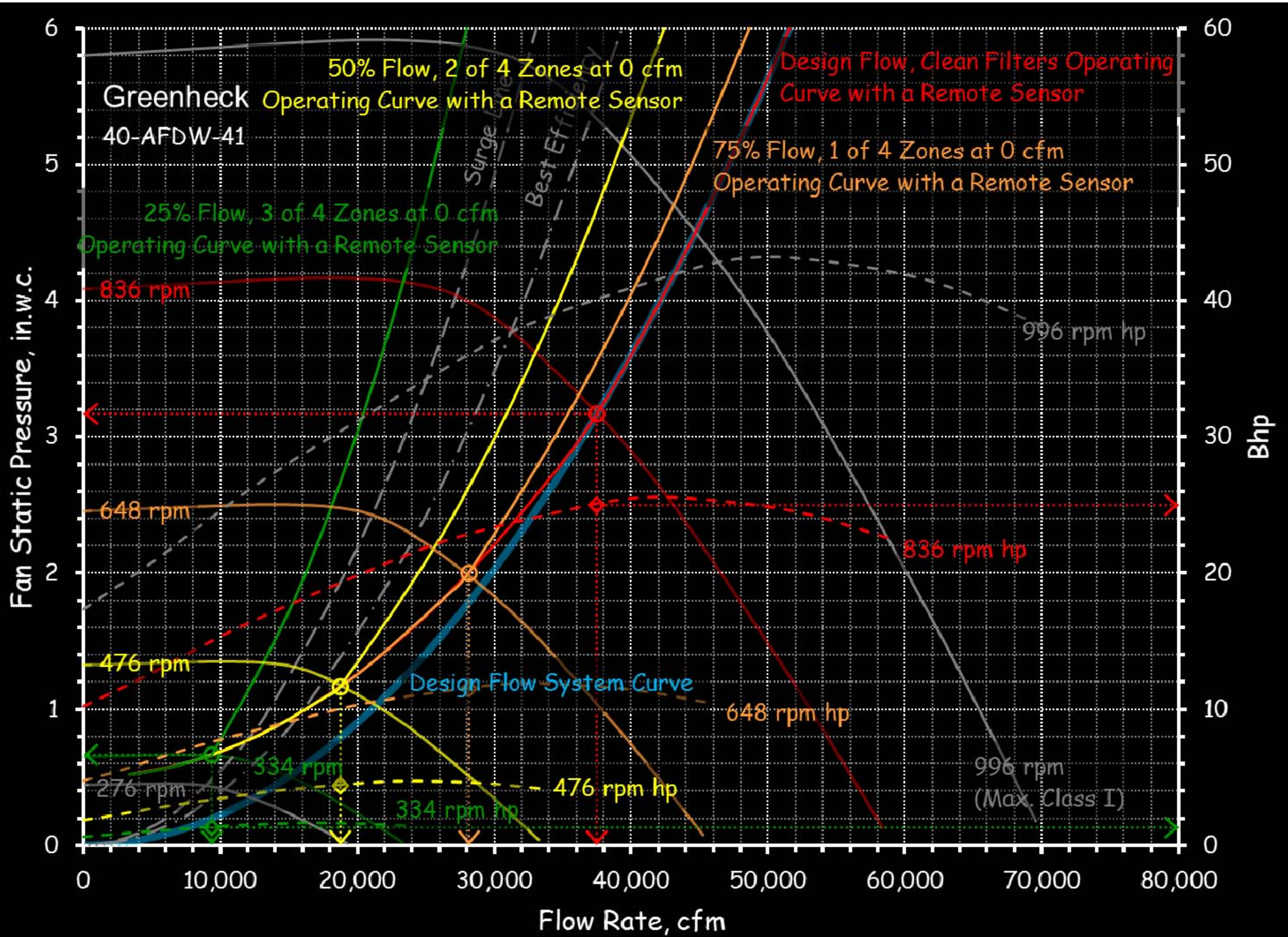


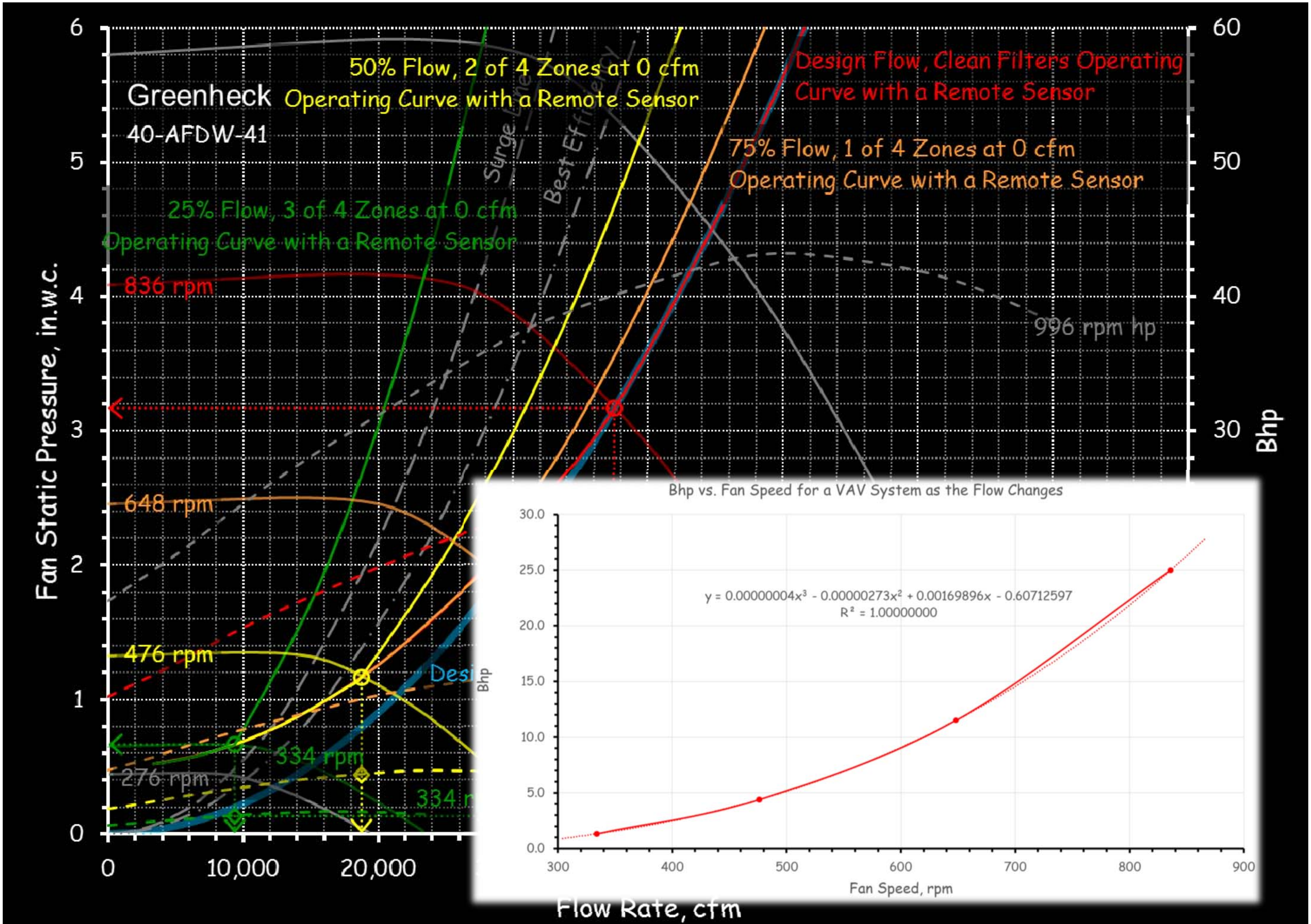

Greenheck
40-AFDW-41
Design Flow, Dirty Filters Operating Curve with a Remote Sensor
Filters Loaded 75%
Filters Loaded 50%
Filters Loaded 25%
Surge Line
Best Efficiency
900 rpm
884 rpm
868 rpm
852 rpm
836 rpm
996 rpm hp
884 rpm hp
852 rpm hp
900 rpm hp
868 rpm hp
836 rpm hp
Design Flow, Clean Filters Operating Curve with a Remote Sensor
276 rpm (Minimum charted speed)
996 rpm (Max. Class I)
Fan Static Pressure, in.w.c.
Bhp
Flow Rate, cfm
0
1
2
3
4
5
6
10
20
30
40
50
60
10,000
20,000
30,000
40,000
50,000
60,000
70,000
80,000


Greenheck
40-AFDW-41
Surge Line
Best Efficiency
Design Flow, Dirty Filters Operating Curve with a Remote Sensor
Filters Loaded 75%
Filters Loaded 50%
Filters Loaded 25%
900 rpm
884 rpm
868 rpm
852 rpm
836 rpm
996 rpm hp
900 rpm hp
868 rpm hp
836 rpm hp
884 rpm hp
276 rpm (Minimum charted speed)
Fan Static Pressure, in.w.c.
Bhp
Flow Rate, cfm
Bhp vs. Fan Speed for a Consant Flow System as the Filters Load
y = 0.00000610x³ - 0.01575405x² + 13.63623047x - 3,930.59181920
R² = 0.99999282
Fan Speed, rpm

The Impact of Terminal Units Throttling in a Variable Volume System




Greenheck
40-AFDW-41
50% Flow, 2 of 4 Zones at 0 cfm
Operating Curve with a Remote Sensor
Design Flow, Clean Filters Operating
Curve with a Remote Sensor
75% Flow, 1 of 4 Zones at 0 cfm
Operating Curve with a Remote Sensor
25% Flow, 3 of 4 Zones at 0 cfm
Operating Curve with a Remote Sensor
Surge Line
Best Efficiency
836 rpm
996 rpm hp
648 rpm
836 rpm hp
476 rpm
Design Flow System Curve
648 rpm hp
334 rpm
276 rpm
476 rpm hp
334 rpm hp
996 rpm
(Max. Class I)
Fan Static Pressure, in.w.c.
Bhp
Flow Rate, cfm
0
1
2
3
4
5
6
10
20
30
40
50
60
10,000
20,000
30,000
40,000
50,000
60,000
70,000
80,000


Greenheck
40-AFDW-41
50% Flow, 2 of 4 Zones at 0 cfm
Operating Curve with a Remote Sensor
Design Flow, Clean Filters Operating
Curve with a Remote Sensor
75% Flow, 1 of 4 Zones at 0 cfm
Operating Curve with a Remote Sensor
25% Flow, 3 of 4 Zones at 0 cfm
Operating Curve with a Remote Sensor
Surge Line
Best Efficiency
836 rpm
648 rpm
476 rpm
334 rpm
276 rpm
996 rpm hp
Fan Static Pressure, in.w.c.
Bhp
Flow Rate, cfm
Bhp vs. Fan Speed for a VAV System as the Flow Changes
y = 0.00000004x³ - 0.00000273x² + 0.00169896x - 0.60712597
R² = 1.00000000
Fan Speed, rpm

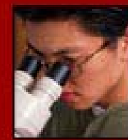
Similar Relationships Apply to Piping Systems

- Fitting fabrication issues and things like system effect are less of a challenge
- Pump selections tend to be less than ideal
  - Interaction potential not recognized
  - Required pump heads often are significantly lower than design specifications
- Control systems can play a major role in managing pump performance and optimizing energy use

Bottom Lines

- The operating metrics of our systems have their roots in fundamental physical relationships
- The relationships are not particularly complex in and of themselves
- The dynamics of HVAC systems make the application of the relationships to them complex because of all of the variables
- Control systems play a crucial role in managing these dynamics
- Understanding the underlying relationships will help you do a better job of designing and applying control technology

THE UNIVERSITY of WISCONSIN MADISON



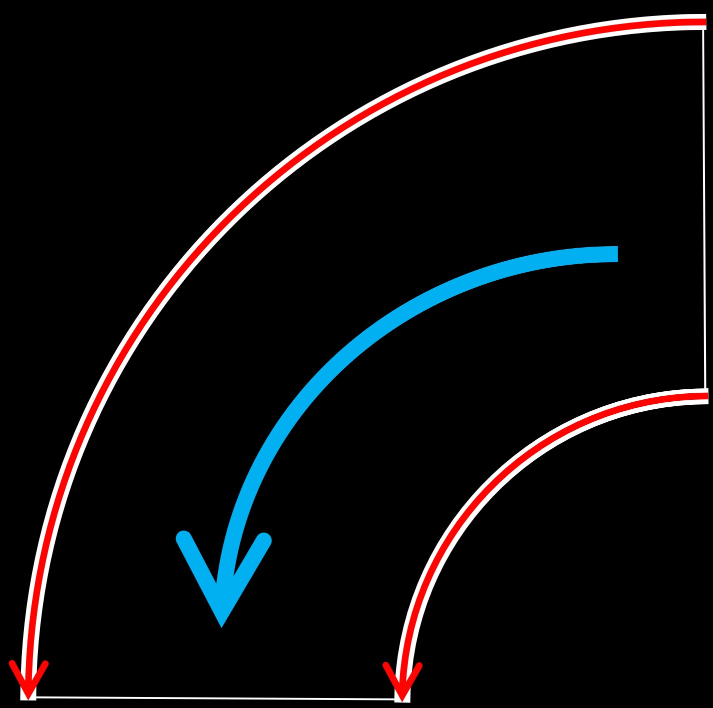

Introduction to the Controlled Systems

Supplemental Information – An Interesting Thing about Elbows

Presented By:
David Sellers, Senior Engineer
Facility Dynamics Engineering

Facility Dynamics ENGINEERING

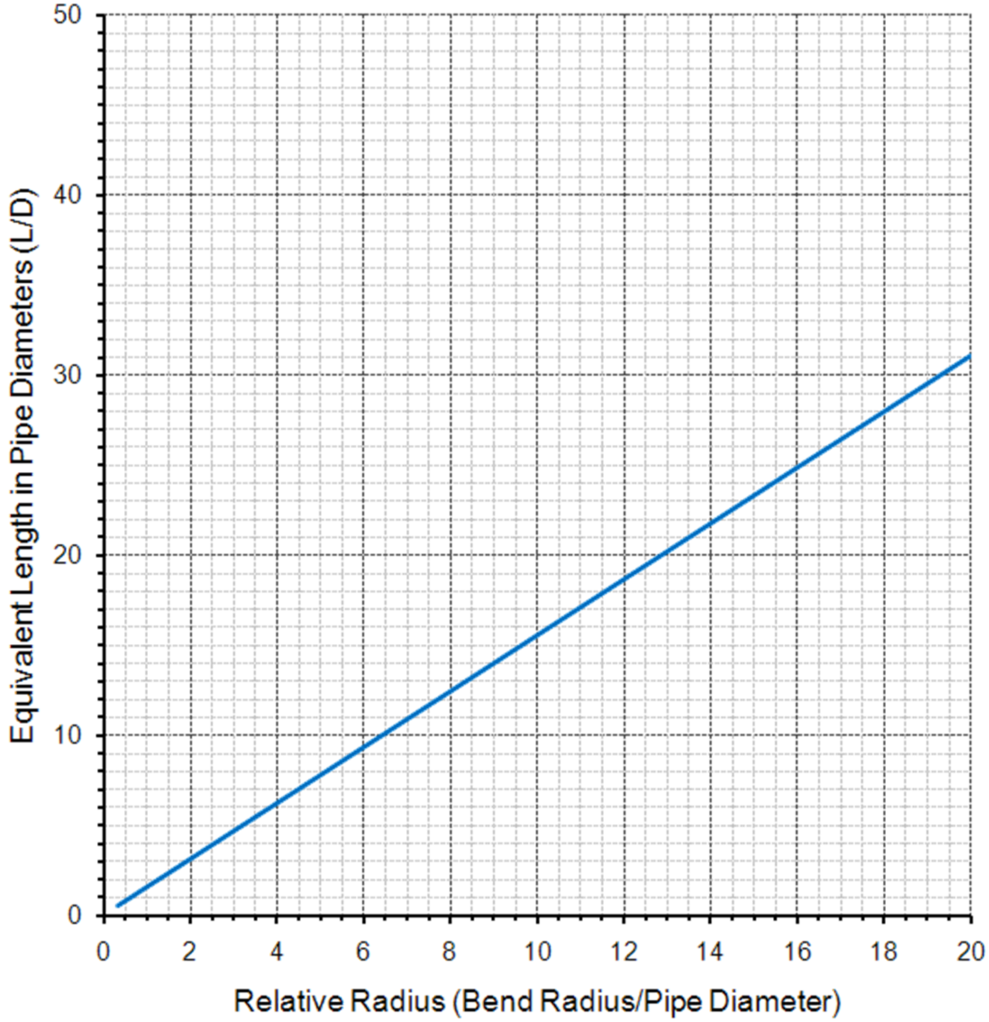
An Interesting Thing about Elbows



One way to think of elbow resistance is to consider it as composed of:

1. Resistance due to interaction with the pipe wall
2. Resistance due to a change in direction

An Interesting Thing about Elbows

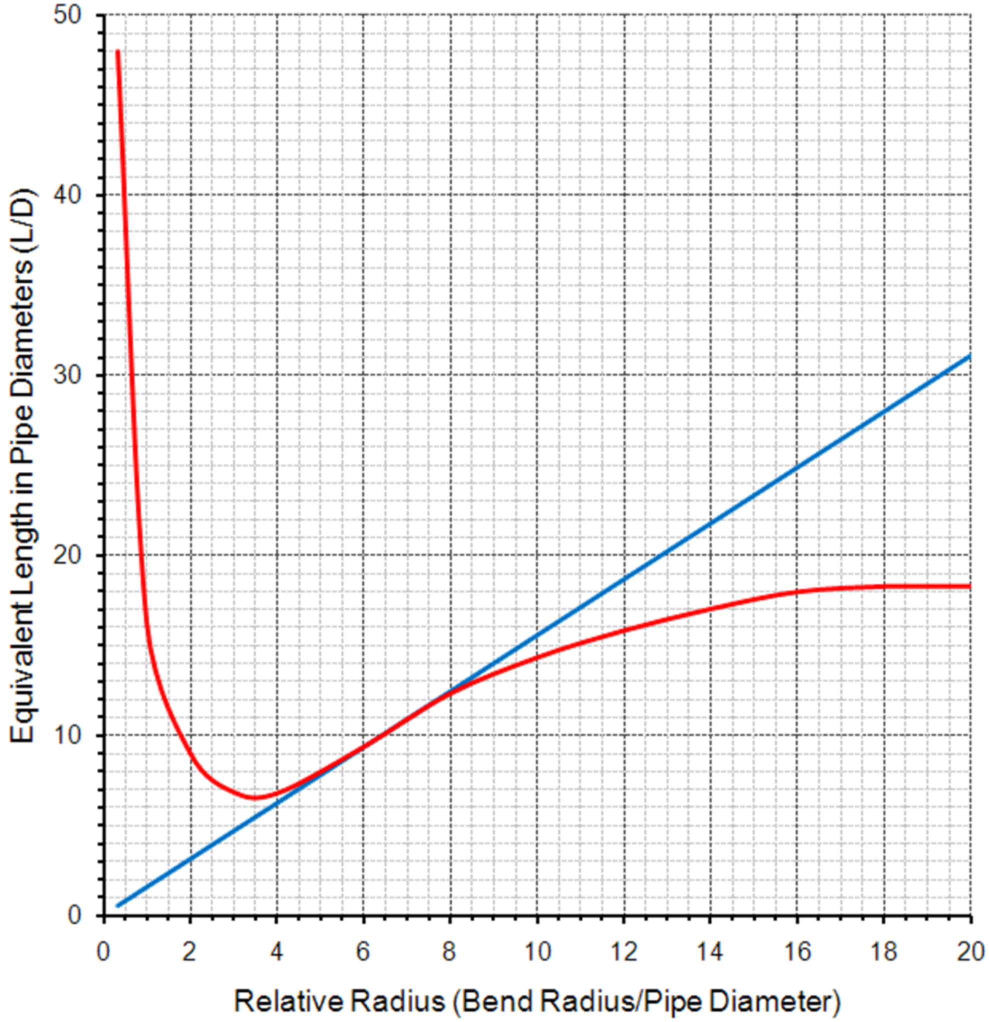


Resistance of 90 Degree Bends

Taken from Crane Technical Paper 410

The Crane paper cites *Pressure Losses for Fluied Flow in 90 Degree Pipe Bends* by K.H. Beij, *Journal of Research of National Bureau of Standards*, Vol. 21, July 1938

An Interesting Thing about Elbows



Resistance of 90 Degree Bends

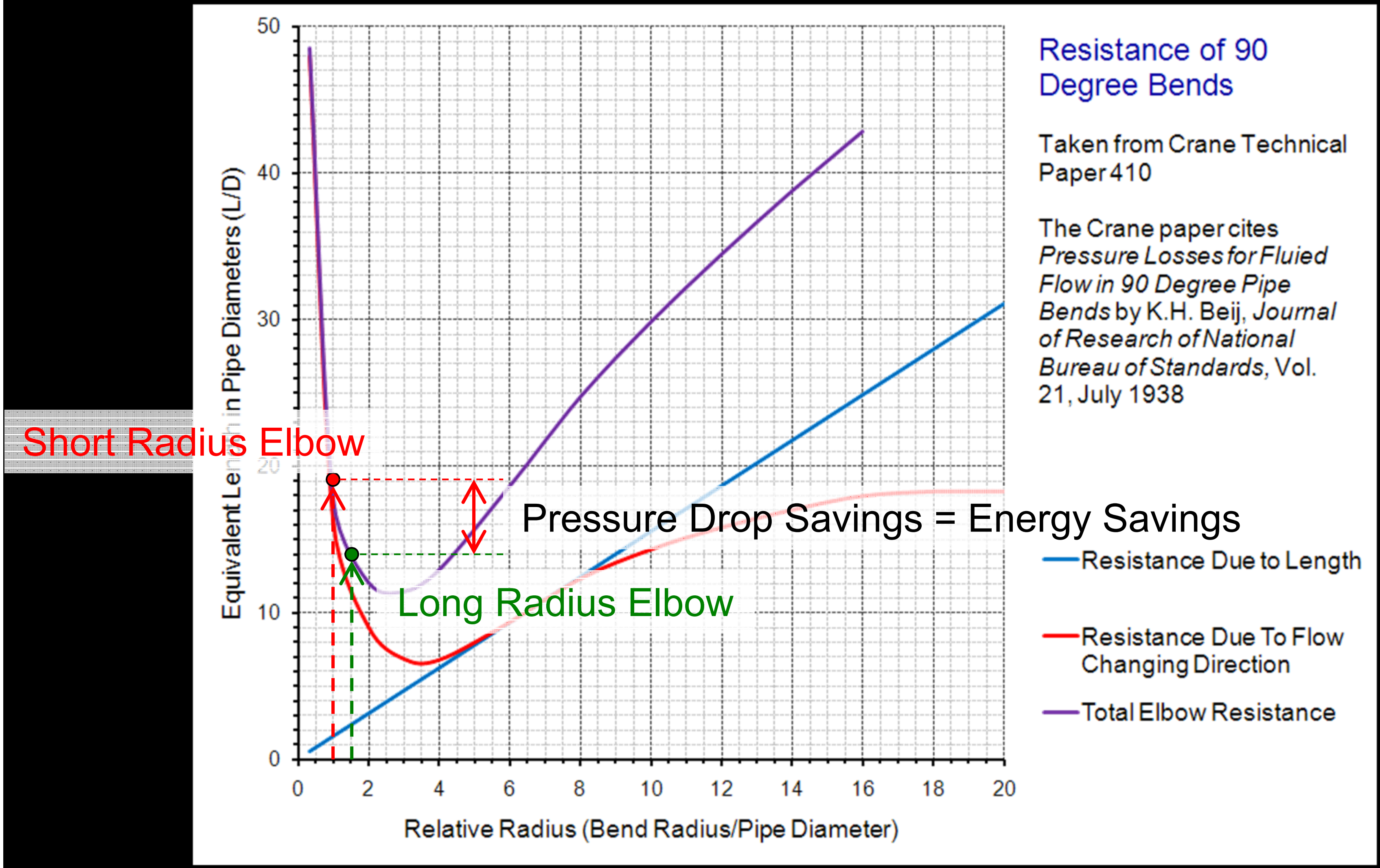
Taken from Crane Technical Paper 410

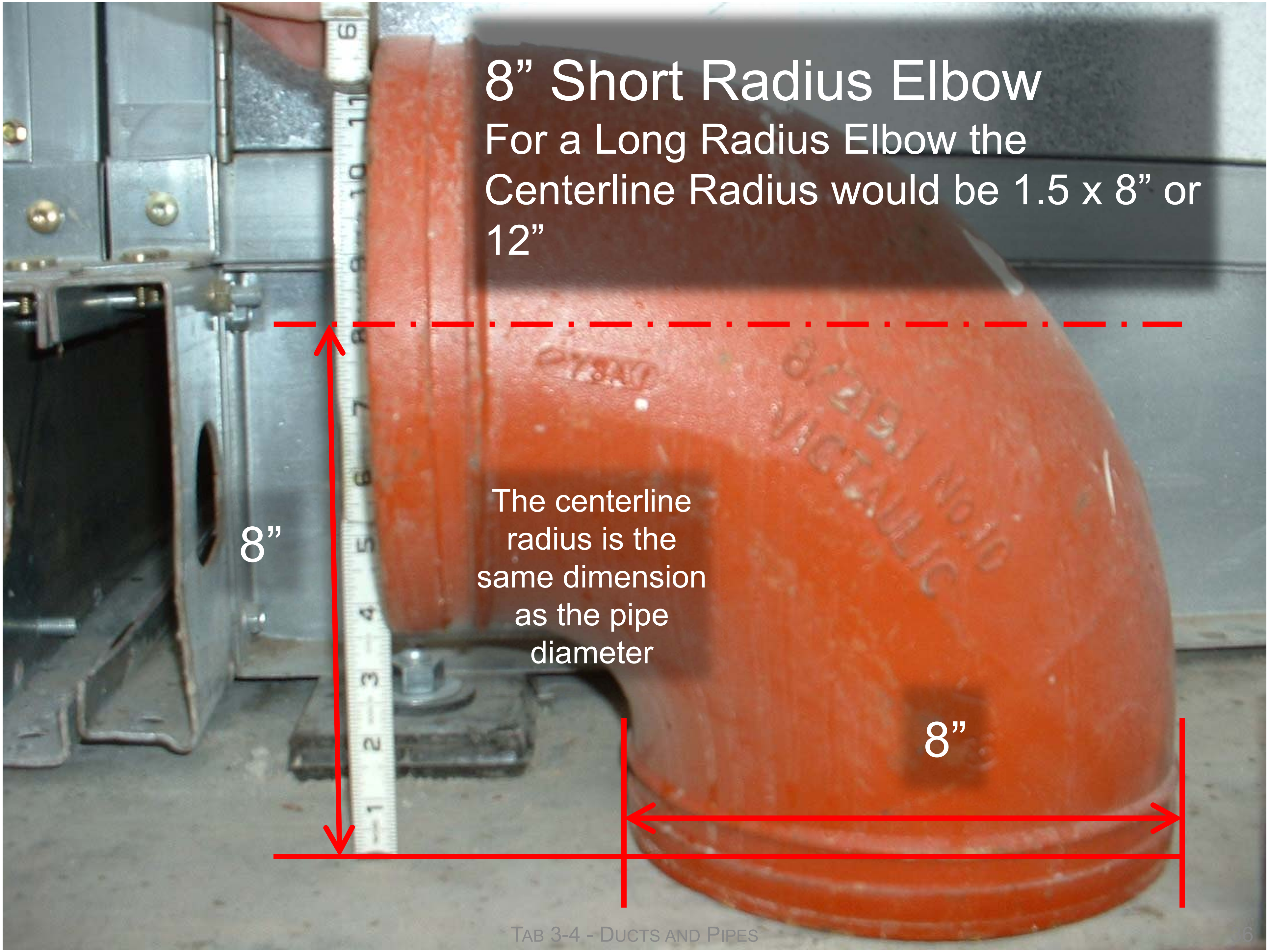
The Crane paper cites *Pressure Losses for Fluied Flow in 90 Degree Pipe Bends* by K.H. Beij, *Journal of Research of National Bureau of Standards*, Vol. 21, July 1938

Resistance Due to Length

Resistance Due To Flow Changing Direction

An Interesting Thing about Elbows




8” Short Radius Elbow
For a Long Radius Elbow the Centerline Radius would be 1.5 x 8” or 12”
The centerline radius is the same dimension as the pipe diameter
8”
8”

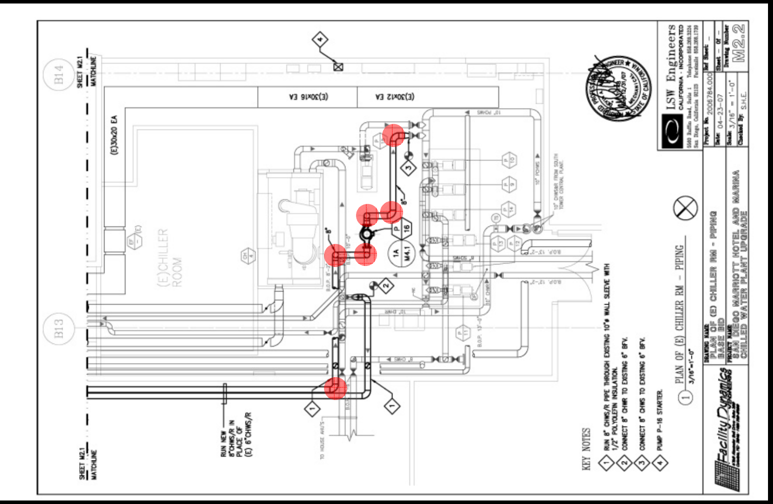
Short Radius vs. Long Radius Elbows

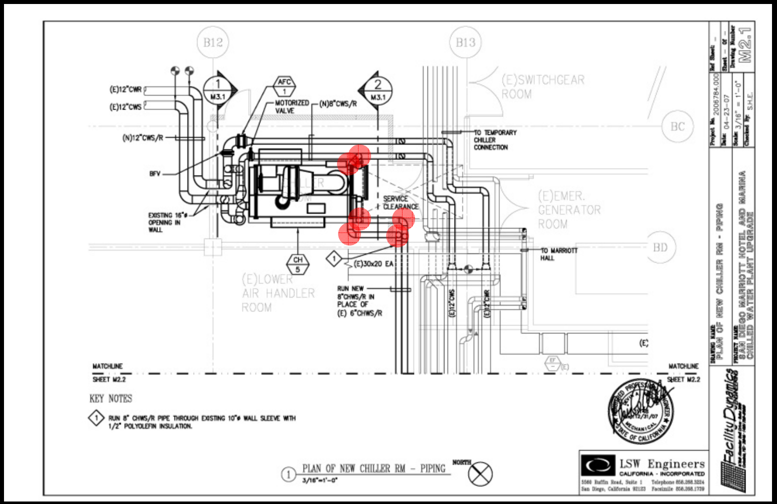
Small differences can add up to significant numbers

Number of elbows = 26

Difference in head for the circuit with long radius versus short radius elbows

- 3+ ft.w.c.
- 9+% difference in pump head (A.K.A. the safety factor)
- .8+ kW
- 7,100 kWh annually
- $850 annually





THE UNIVERSITY of WISCONSIN MADISON

Introduction to the Controlled Systems

Supplemental Information – Calculating kWh into a Pump or Fan

Presented By:
David Sellers, Senior Engineer
Facility Dynamics Engineering

Facility Dynamics ENGINEERING

Calculating Power Into the Fan Motor as kW

$$kW = \left(\frac{Flow \times Static}{6{,}356 \times \ \eta_{Fan} \times \ \eta_{Motor} \times \ \eta_{Drive}} \right) \times .746$$

Where:

kW = Electrical energy into the drive system seving the fan

$Flow$ = Flow rate in cubic feet per minute

$Static$ = Fan static in inches water column

6,356 = A units conversion constant

η_{Fan} = Fan efficiency

η_{Motor} = Motor efficiency

η_{Drive} = Drive efficiency; Don;'t forget about the belts if the motor is not direct drive. Well adjusted belts are 97-98% efficient. Poorly adjusted ones can be as low as 90% or less

.746 = Horsepower to kW conversion constant; there are .746 hp per kW, or stated mathematically with the appropriate units:

$$.746 \frac{\text{kW}}{\cancel{\text{hp}}} \times 1\ \cancel{\text{hp}} = .746\ \text{kW}$$

Calculating Kw Into the Pump Motor

$$kW = \left(\frac{Flow_{gpm} \times Head_{ft.w.c.}}{3{,}960 \times \eta_{Pump} \times \eta_{Motor} \times \eta_{VSD}} \right) \times .746$$

Where:

kW = Input to the system to produce the flow and head.

$Flow$ = Flow rate in gallons per minute. Generally speaking, we try to use a pump test for at least one condition as a basis for this. If that is not available we will use a value from a tab report. Lacking that we will use a design metric from the original drawings or an equipment submittal.

$Head$ = The pump head in ft.w.c. water column, which we usually try to identify from field measurements and pump tests. Lacking those measurements we will use a value derived from a TAB report or the design value.

3,960 = A units conversion constant that is good for water between 40°F and 220°F.

η_{Pump} = Pump efficiency. We usually try to get this number from the pump curve or from the pump's rated brake horse power (bhp), flow and head. Lacking that, we will make a geometrically similar pump selection (same flow rate, head, impeller diameter, and speed) using manufacturer's software and use that efficiency. Lacking that is is reasonable to assume that for a pump rated for 300 gpm or less the efficiency might be in the range of 45-60%. For pumps rated between 300 gpm and 1,500 gpm, efficiencys might range from 60% to 75%. For pumps over 1,500 gpm, efficiencies might range from 75% to as high as 87%. Generally, efficiency will improve with pump size.

η_{Motor} = Motor efficiency. We usually try to get the motor performance curve and select the efficiency from the curve for the bhp that the pump impeller is extracting from it. If we can't get the motor curve, we use a similar motor selected from MotorMaster^{TM} International. In all cases we adjust the efficiency for the motor operating point vs. using the motor's rated nameplate efficiency. Lacking anything else, it is reasonable to assume that the motor efficiency will improve by 1-2% over the nameplate efficiency when the pump is at 65-85% of its rated load, drop back to near nameplate efficiency at around 50% load, and then drop sharply towards 0 at 20-30% of rated load.

Calculating Kw Into the Pump Motor (Continued)

$$kW = \left(\frac{Flow_{gpm} \times Head_{ft.w.c.}}{3{,}960 \times \eta_{Pump} \times \eta_{Motor} \times \eta_{VSD}} \right) \times .746$$

Where:

η_{VSD} = Variable speed drive efficiency. Where possible, we try to get the manufacturer's data for this. But this data is difficult to obtain and not consistent in its development. Lacking manufacture specific data, we use generic data as published by the Department of Energy on their Industrial Best Practices web site. Lacking any other source, it is reasonable to assume there will be at least 4-6% loss in the drive with it at full speed with a gradual decay to 80% efficiency at about 20% load.

.746 = Horsepower to kW conversion constant; there are .746 hp per kW, or stated mathematically with the appropriate units:

$$.746\,\frac{\text{kW}}{\cancel{\text{hp}}} \times 1\,\cancel{\text{hp}} = .746\,\text{kW}$$