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U.S. DEPARTMENT OF ENERGY

Pumping System Assessment

Week 6: Digging Deeper
Valves, Specific Speed, etc.

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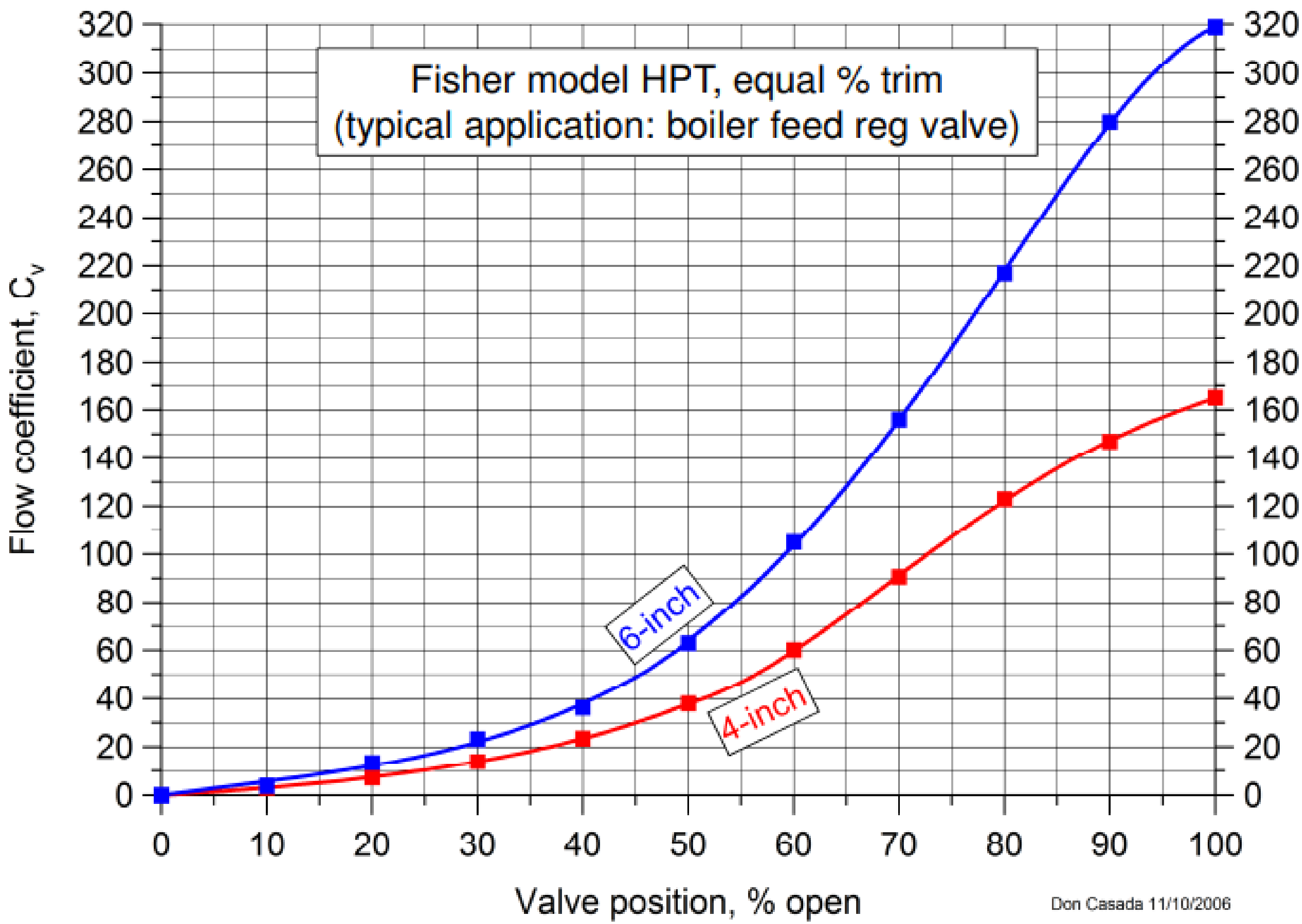
Valve suppliers use the valve flow coefficient, C_v as a primary indicator of valve performance



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The valve flow coefficient is used to predict flow rate, based on differential pressure, fluid SG and adjacent fittings

(Reference: ANSI/ISA S75.01.01-2002, Flow Equations for Sizing Control Valves)

$$Q = F_p C_v \sqrt{\frac{\Delta P}{s.g.}}$$

$$F_p = \left(\frac{C_v^2 \Sigma K}{890 d^4} + 1 \right)^{-0.5}$$

Q	flow rate, *gpm*
F_P	Valve inlet/outlet geometry factor (see below), *dimensionless*
C_v	Valve flow coefficient, *gpm/(psi$^{0.5}$)*
ΔP	Valve unit pressure drop, *psid*
s.g.	Fluid specific gravity, *dimensionless*
K	Loss and Bernoulli coefficients for fittings, *dimensionless*
d	Valve end diameter, *inches*

$\Sigma K = K_1 + K_2 + K_{B1} - K_{B2}$

K_1 = Inlet reducer friction loss (control valves are commonly smaller than adjacent pipe)

K_2 = Outlet expander friction loss

$K_{B1} - K_{B2}$ = adjustment for velocity head differences in inlet, outlet piping

As a limiting case, the friction losses can be treated as sudden contraction and expansion:

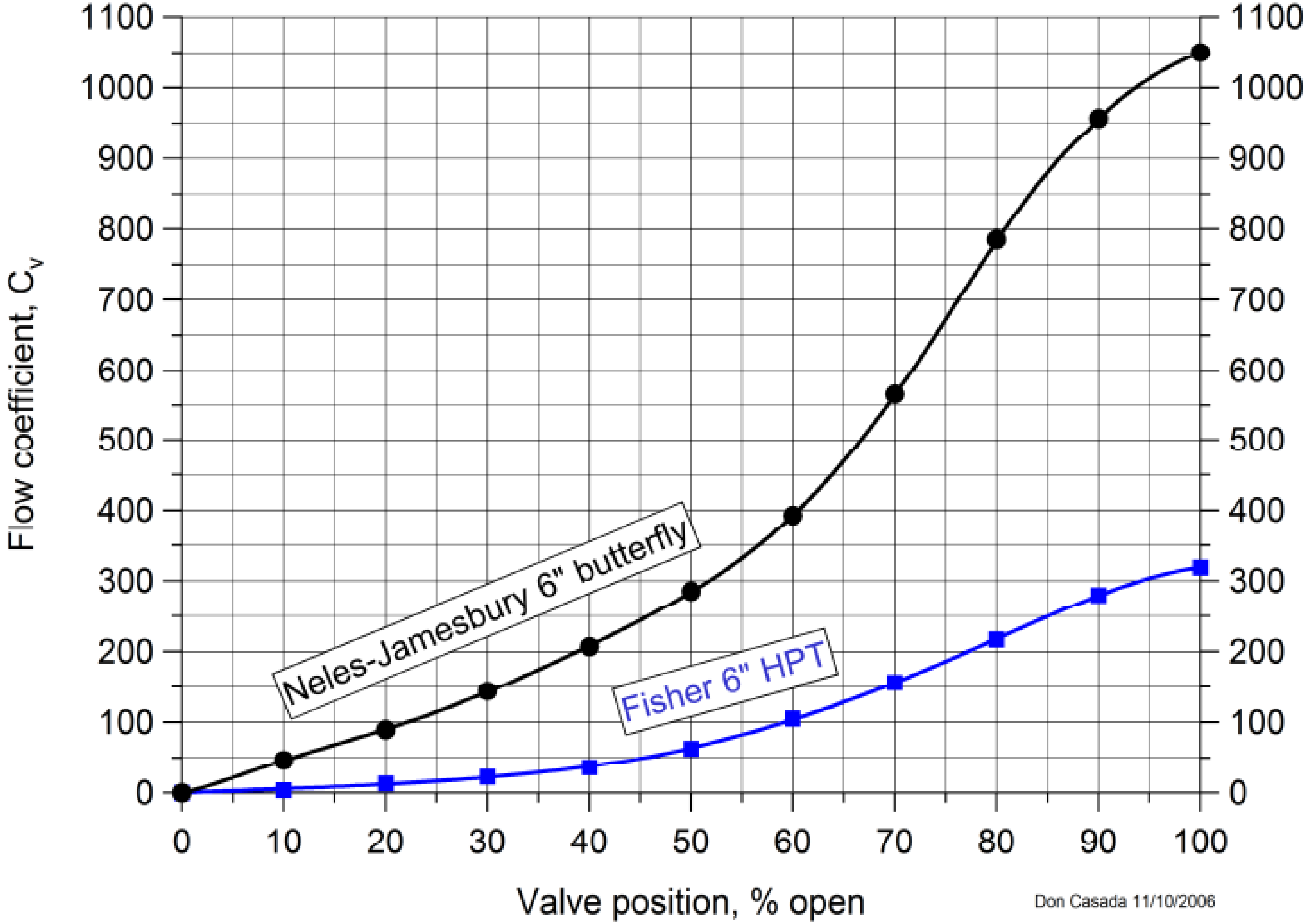
$$K_1 = 0.5\left(1 - \left[\frac{d}{D_1}\right]^2\right) \qquad K_2 = \left(1 - \left[\frac{d}{D_2}\right]^2\right)^2 \qquad K_{Bi} = \left(1 - \left[\frac{d}{D_i}\right]^2\right)^2$$

For most applications, the pipe upstream and downstream of the valve is the same size,
so the K_B terms drop out. This is for estimating only; ISA notes that to meet a 5% accuracy, the valve and fittings should be tested as a unit.

Note: The ISA standard adds a squared exponent to the K_1 term:
Cameron Hydraulics does not include this exponent in their sudden reduction estimating algorithm.

$$K_1 = 0.5\left(1 - \left[\frac{d}{D_1}\right]^2\right)^2$$

Both the shape and magnitude of the coefficient are strongly dependent on valve design



How does the valve flow coefficient relate to the frictional loss coefficient (used in the Darcy-Weisbach equation)?

?

$$Q = C_v \sqrt{\frac{\Delta P}{s.g.}} \qquad H_f = K \frac{V^2}{2g}$$

These algebraic steps show the process of converting from C_v to the head loss coefficient K

$$Q = C_v \sqrt{\frac{\Delta P}{s.g.}} \quad \text{or} \quad \Delta P = \frac{s.g. \cdot Q^2}{C_v^2} \qquad (1)$$

an
d
$$\Delta P = \frac{H_f \cdot s.g.}{2.3108} \quad \text{so} \quad H_f = \frac{2.3108 \text{ x } Q^2}{C_v^2} \qquad (2)$$

Using the modified version of Darcy-Weisbach for friction loss:

$$H_f = K\frac{V^2}{2g} \quad \text{and} \quad V = \frac{q'}{A} = \frac{\frac{Q}{448.8}}{\pi\left(\frac{d}{24}\right)^2} = \frac{Q}{2.448 \cdot d^2} \qquad (3)$$

Substituting, $H_f = K\frac{V^2}{2g} = K\frac{\left(\frac{Q}{2.448 \cdot d^2}\right)^2}{64.348} = K\frac{Q^2}{385.6\, d^4}$ (4)

Combining (2) & (4): $H_f = \frac{2.3108 \text{ x } Q^2}{C_v^2} = K\frac{Q^2}{385.6\, d^4}$ (5)

Rearranging, $K = \frac{891.1\, d^4}{C_v^2}$ or $C_v = \frac{29.851\, d^2}{\sqrt{K}}$ (6)

Also, $Q = 29.851\, d^2 \sqrt{\frac{\Delta P}{K \cdot s.g.}}$ (7)

LEGEND

Symbol	Represents
A	area (ft^2)
C_v	Valve flow coefficient (gpm/(psid$^{0.5}$))
ΔP	Pressure drop across valve, psid
d	pipe inside diameter (inches)
g	gravitational constant (32.174 ft/sec^2)
H_f	Head loss across valve (ft)
K	Loss coefficient (dimensionless)
Q	Volumetric flow rate (gpm)
q'	Volumetric flow rate (ft^3/sec)
s.g.	specific gravity (dimensionless)
V	velocity (ft/sec)

Constants, Conversions used

s.g. = 1.00 reference for 62.316 lb/ft3 (68F)

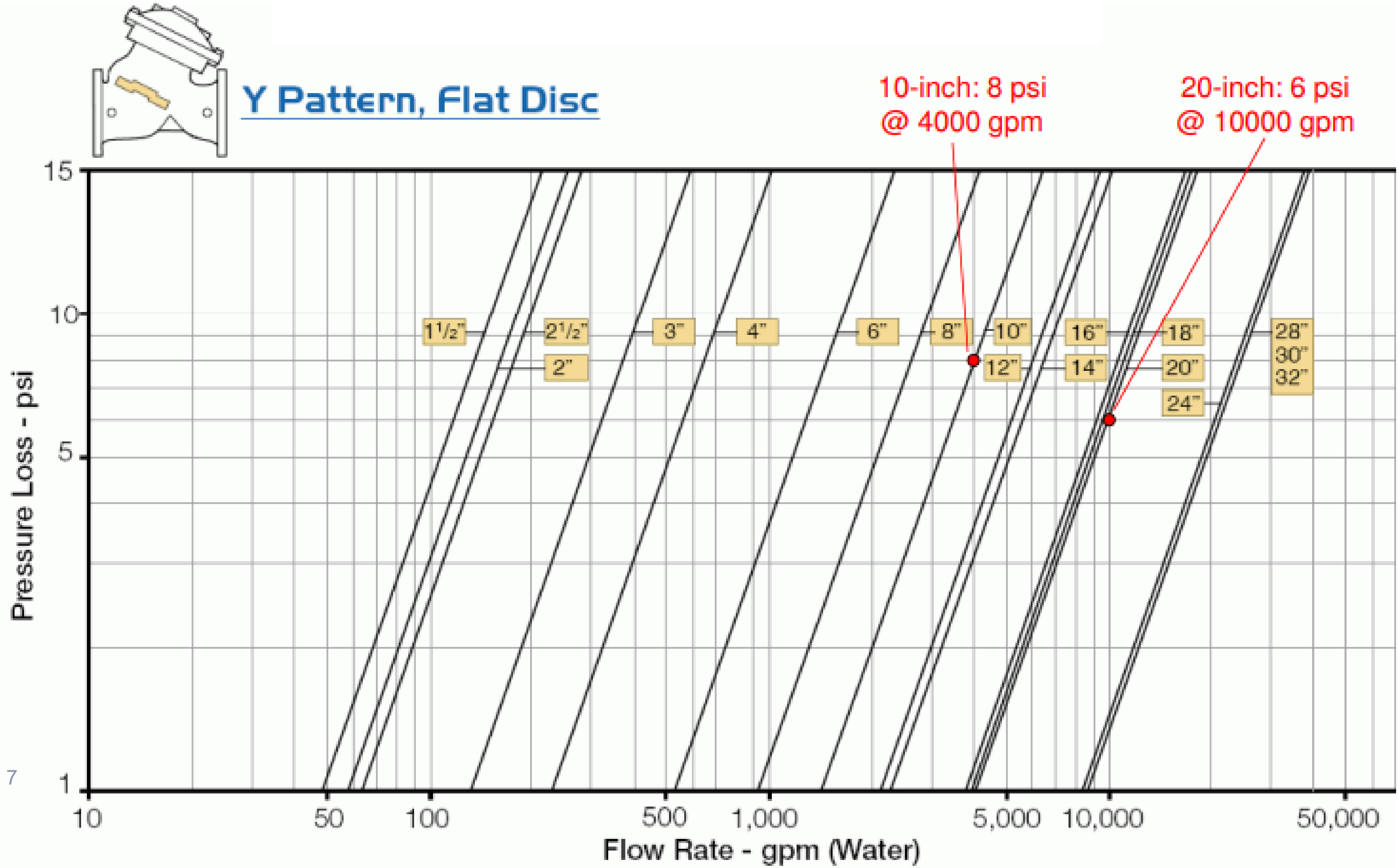
1 cubic foot = 7.48052 gallons

This is tedious work for the algebraically challenged (yours truly), but these last two items are fairly useful.

Note that these relations are for the valve exclusively, and do not include the inlet and outlet losses. When dealing with loss coefficients, the inlet and outlet losses would be added to the valve K.

Note: the value of 891.1 corresponds to the rounded value of 890 in the ISA valve equation

Determine the loss K for 10 and 20-inch valves



Determine the loss K for 10 and 20-inch valves

10-inch valve: 4000 gpm with 8 psi pressure loss

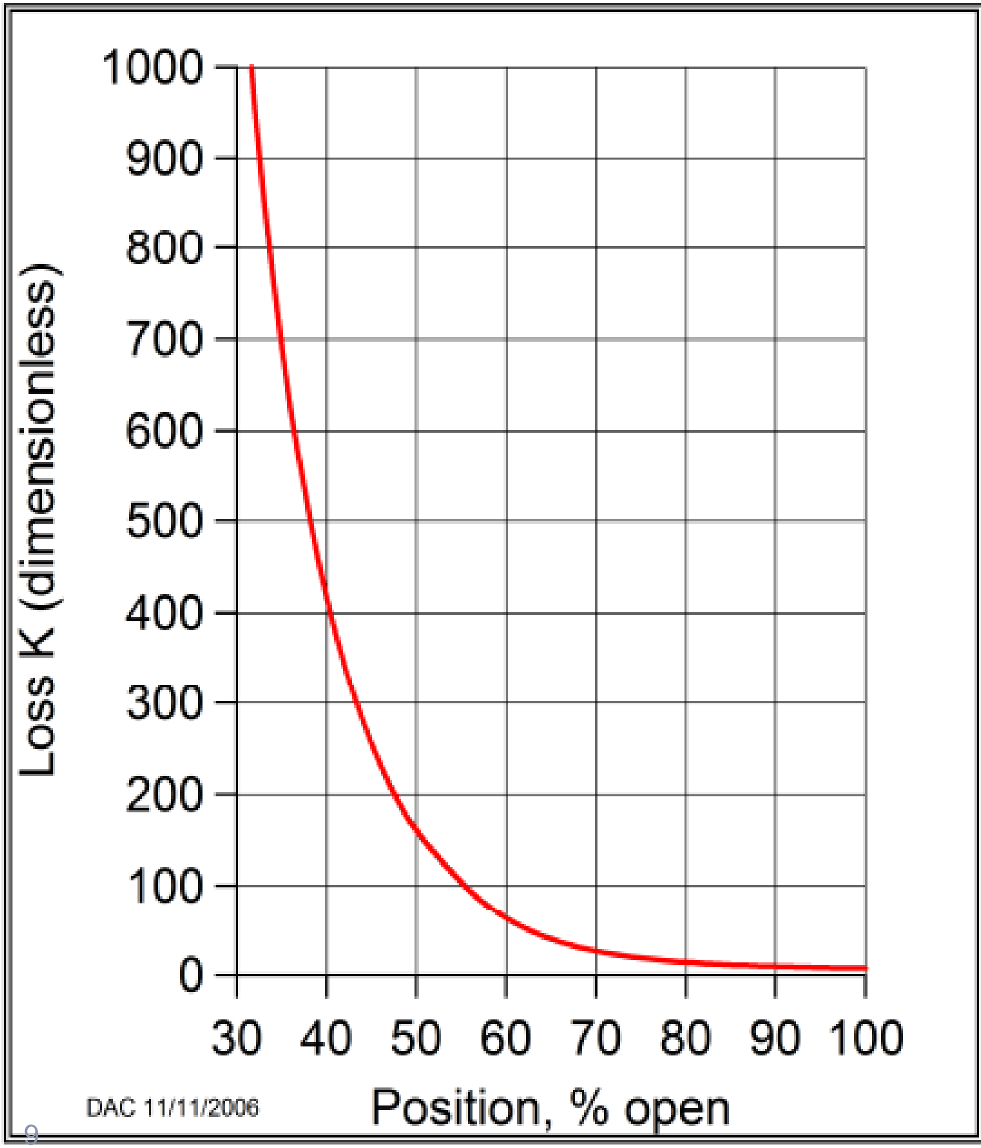
$$Q = C_v \sqrt{\frac{\Delta P}{s.g.}}$$

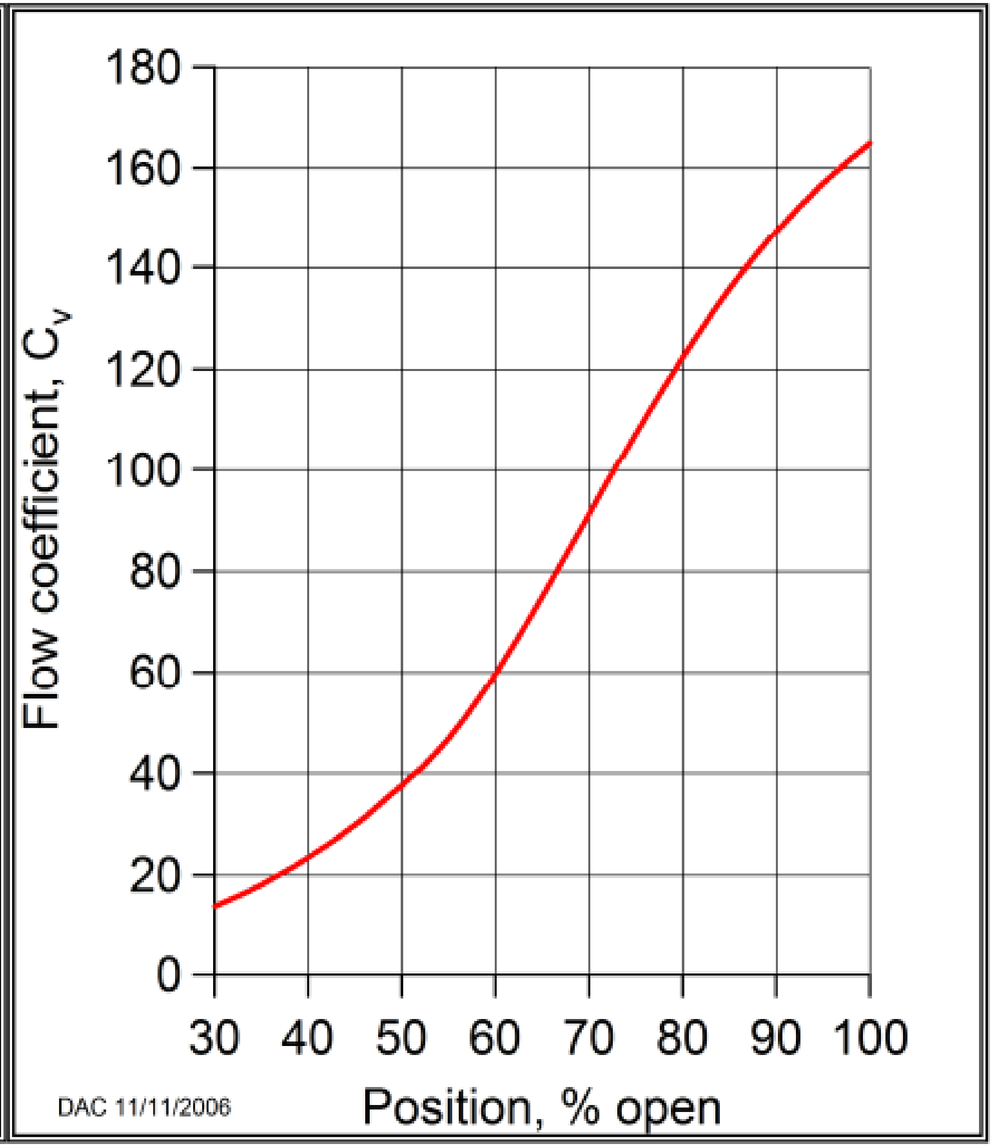
$$C_v = \frac{4000}{\sqrt{8}} = 1414.2$$

$$K = \frac{891.1\,d^4}{C_v^2} \quad \text{or} \quad C_v = \frac{29.851\,d^2}{\sqrt{K}}$$

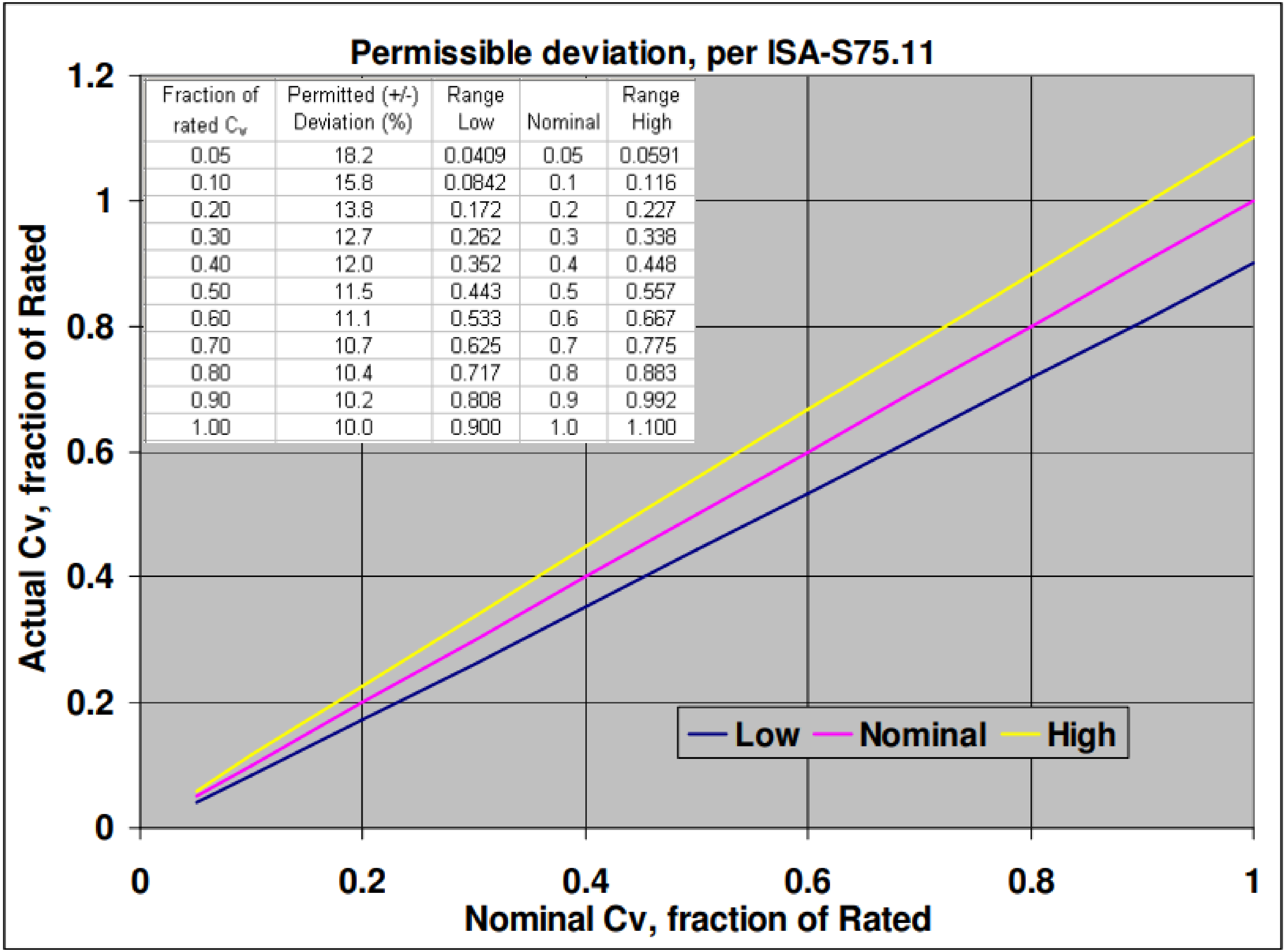
$$K = \frac{891.1\,(10^4)}{1414.2^2} = 4.46$$

A comparison of loss coefficient K and flow coefficient C_v for 4-inch Fisher HPT





Allowable deviation from normal performance can be significant



Fraction of rated C_v	Permitted (+/-) Deviation (%)	Range Low	Nominal	Range High
0.05	18.2	0.0409	0.05	0.0591
0.10	15.8	0.0842	0.1	0.116
0.20	13.8	0.172	0.2	0.227
0.30	12.7	0.262	0.3	0.338
0.40	12.0	0.352	0.4	0.448
0.50	11.5	0.443	0.5	0.557
0.60	11.1	0.533	0.6	0.667
0.70	10.7	0.625	0.7	0.775
0.80	10.4	0.717	0.8	0.883
0.90	10.2	0.808	0.9	0.992
1.00	10.0	0.900	1.0	1.100

Operating basis for the Valve Tool relating valve or other frictional pressure drops to fluid and electric power

$$\text{Fluid power (hp)} = \frac{\text{gpm x ft x s.g.}}{3960} \quad \text{and} \quad \text{ft} = \frac{\text{dP x 2.31}}{\text{s.g.}} \quad \text{so} \quad \text{Fluid power (hp)} = \frac{\text{gpm x psid}}{1714}$$

From the valve characteristic equation,

$$\text{dP (or psid)} = \text{s.g. x} \left(\frac{\text{gpm}}{C_v}\right)^2 \quad \text{so} \quad \text{Fluid power (hp)} = \text{s.g. x} \left(\frac{\text{gpm}^3}{1714\ C_v^2}\right)$$

Note that if a combined pump and motor efficiency of 74.6% is assumed, the fluid horsepower will also be equal to the electric power in kilowatts.

With this simplified (and generally conservative) assumption, the annual electrical energy consumption in kilowatt-hours for full time operation is:

$$\boxed{\text{Annual kWhr} = 5.11 \text{ x s.g. x} \left(\frac{\text{gpm}^3}{C_v^2}\right) = 5.11 \text{ x gpm x psid}}$$

So for a flow rate of 1000 gpm passing through a valve with a 10 psid pressure drop, the annual energy required to overcome the valve friction would be 51,100 kWhr. At an electrical cost rate of 5 cents/kWhr, that translates into an annual cost of $2,555.

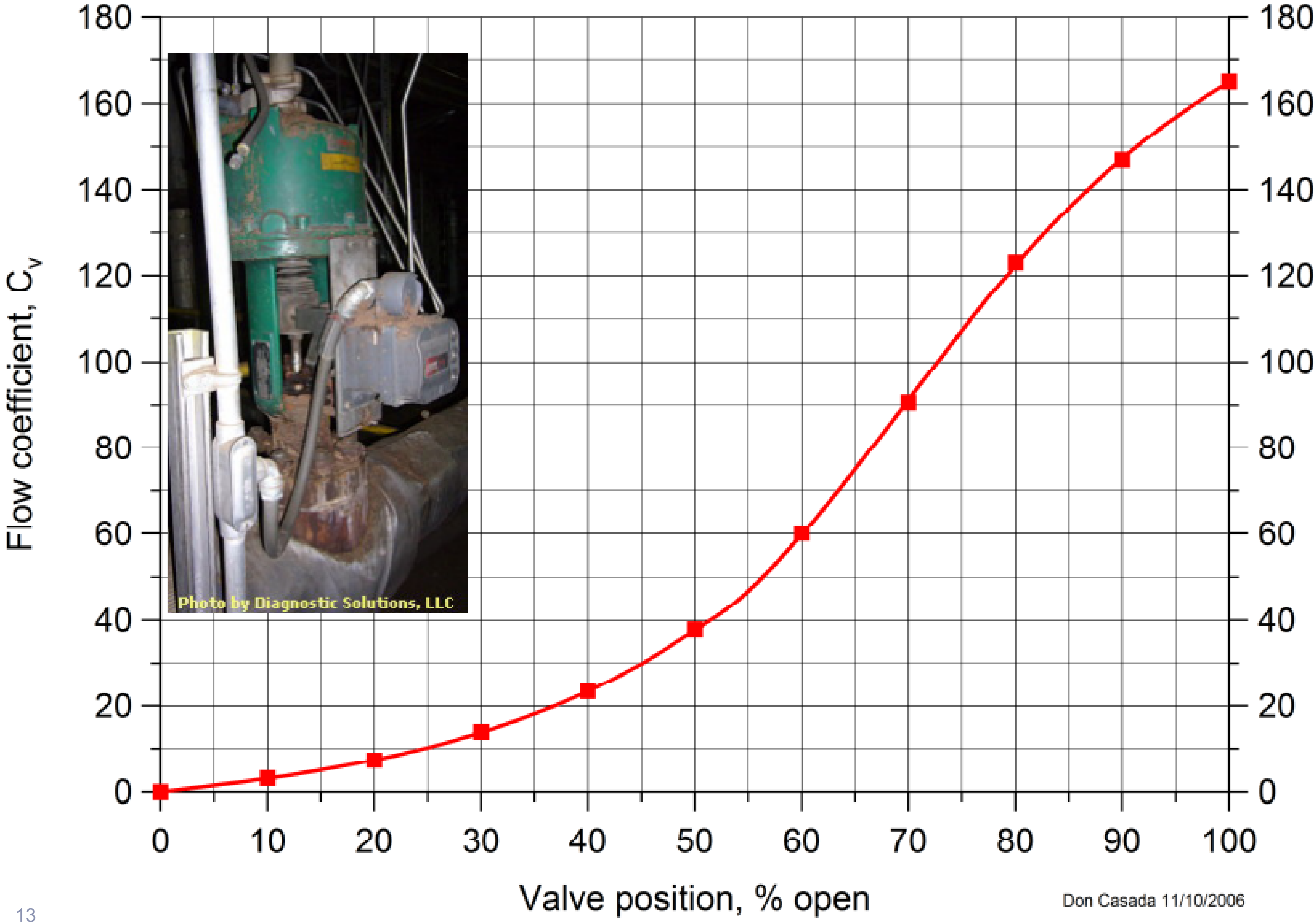
An exercise using valve curves and Valve Tool

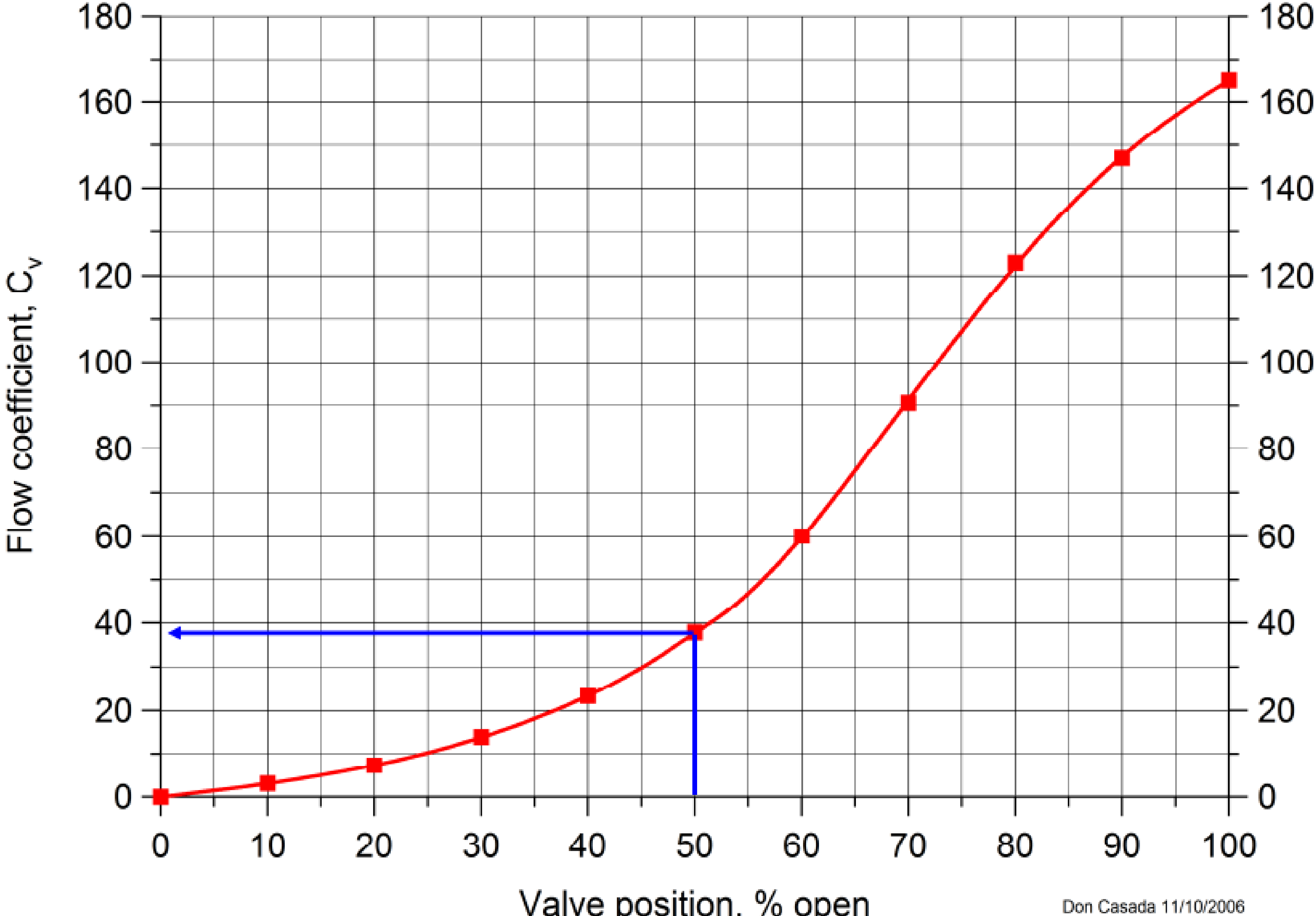
The 4-inch valve is 50% open. Fluid is boiler feedwater at 275 F (specific gravity = 0.931). Pressure just before the valve is 1550 psig. Pressure downstream, about 50 feet in elevation above the upstream gauge is 1100 psig. All piping is 4-inch diameter.

Part 1. Read the nominal valve Cv from the next slide.

Part 2. Use the valve tool to estimate flow rate.

Part 3. Use the permissible deviation chart to estimate the lower and upper C_v values and corresponding flow rates.


Flow coefficient, C_v
Valve position, % open
Photo by Diagnostic Solutions, LLC
0
20
40
60
80
100
120
140
160
180
0
10
20
30
40
50
60
70
80
90
100
Don Casada 11/10/2006



180
160
140
120
100
80
60
40
20
0
Flow coefficient, C_v
0
10
20
30
40
50
60
70
80
90
100
Valve position, % open
Don Casada 11/10/2006

Estimates……….

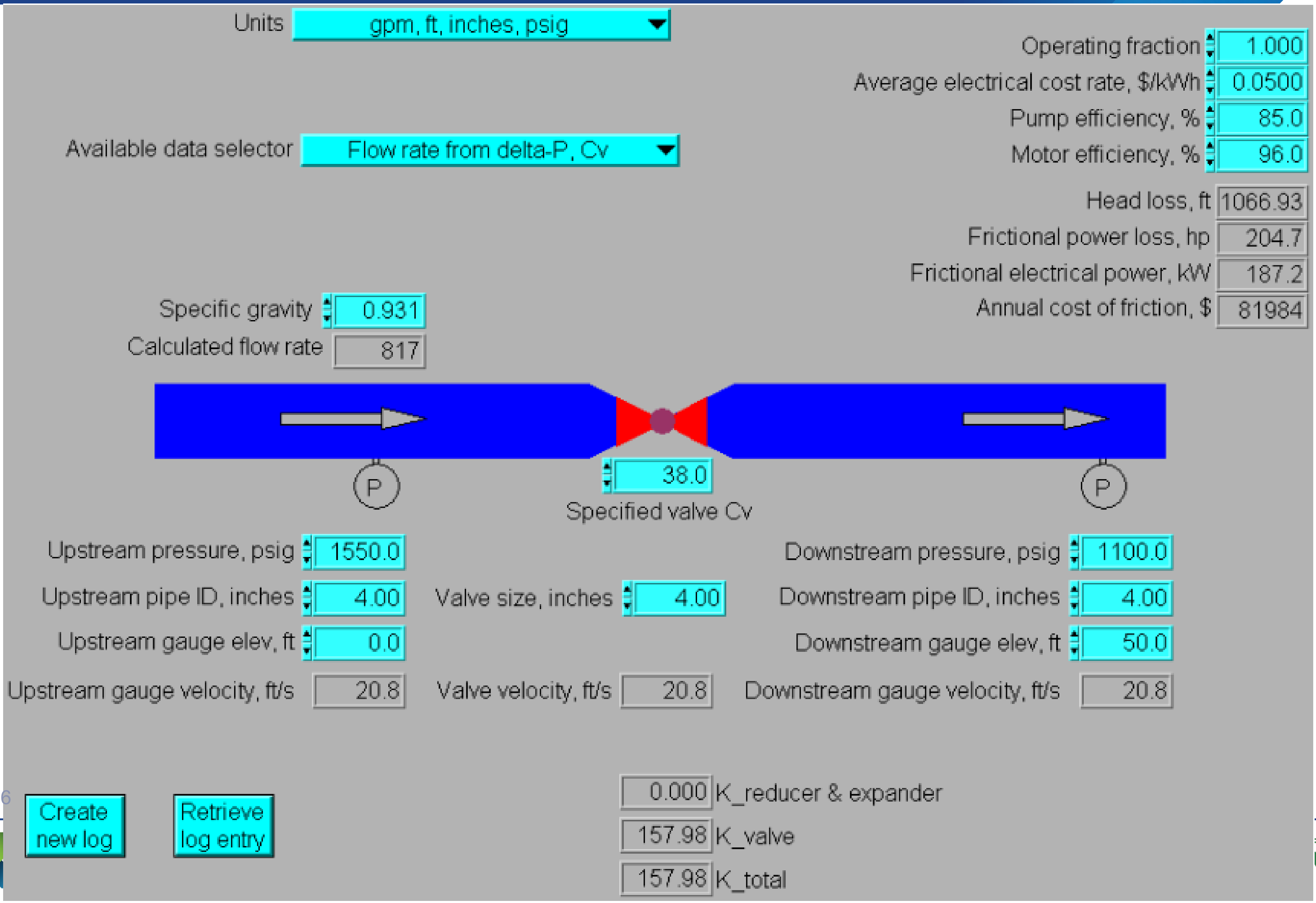
	Cv	Estimated gpm
Nominal	38	820
Low allowable	33	710
High allowable	43	920

$$Ratio = \ 38/165\ (slide\ 10) = 0.23$$

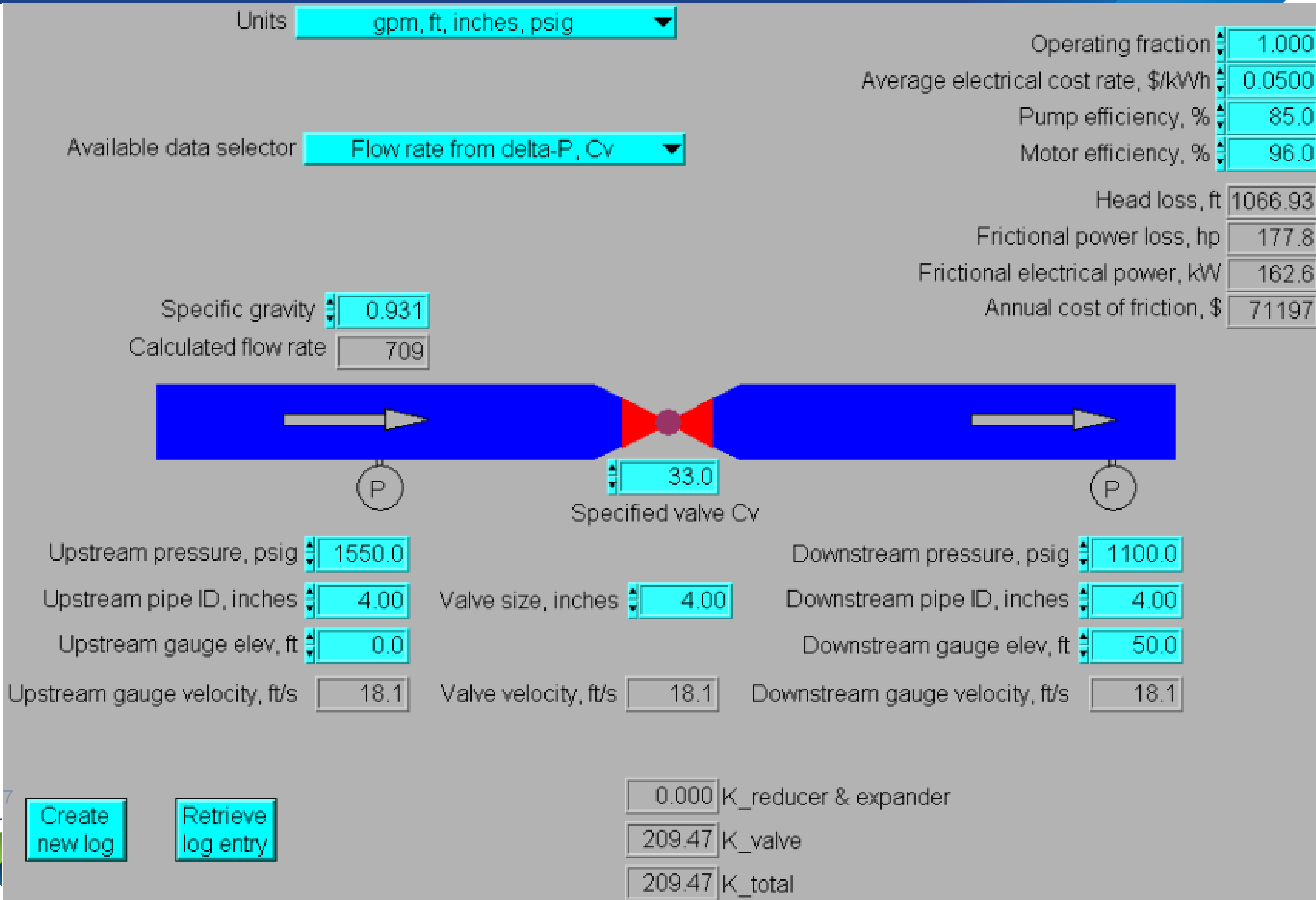
$$Low\ allowable = 38 - ((13.8 - (13.8 - 12.7)0.3)/100)38 = 33$$

$$High\ allowable = 38 + ((13.8 - (13.8 - 12.7)0.3)/100)38 = 43$$

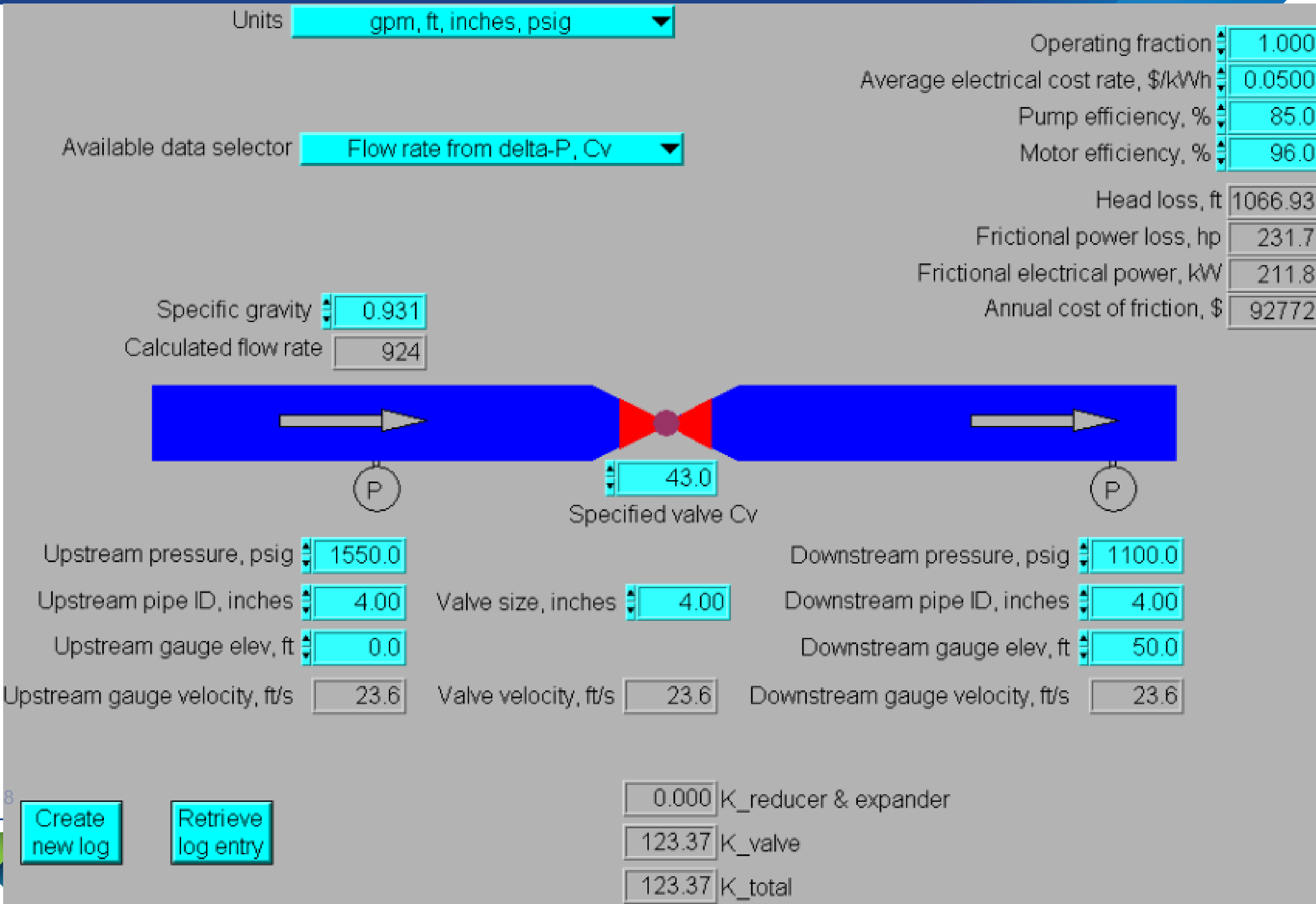
Estimated flow rate (nominal) is around 820 gpm



Estimated flow rate (low) is around 710 gpm



Estimated flow rate (high) is around 920 gpm



Pump specific speed basics, HI efficiency-estimating algorithms, and the MEASUR implementation

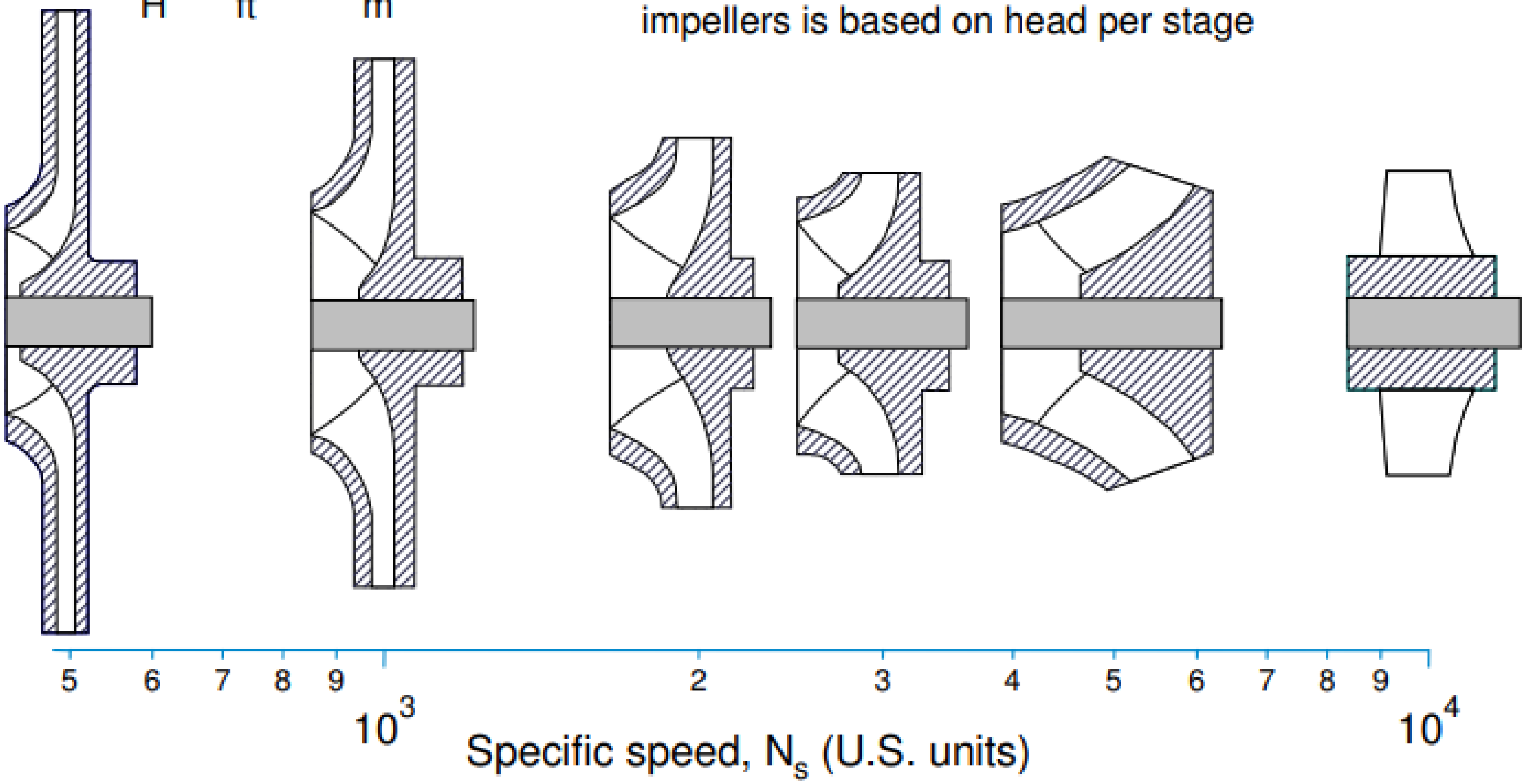
The general impeller shape and specific speed are related

Specific speed, $N_s = \frac{n \times Q^{1/2}}{H^{3/4}}$

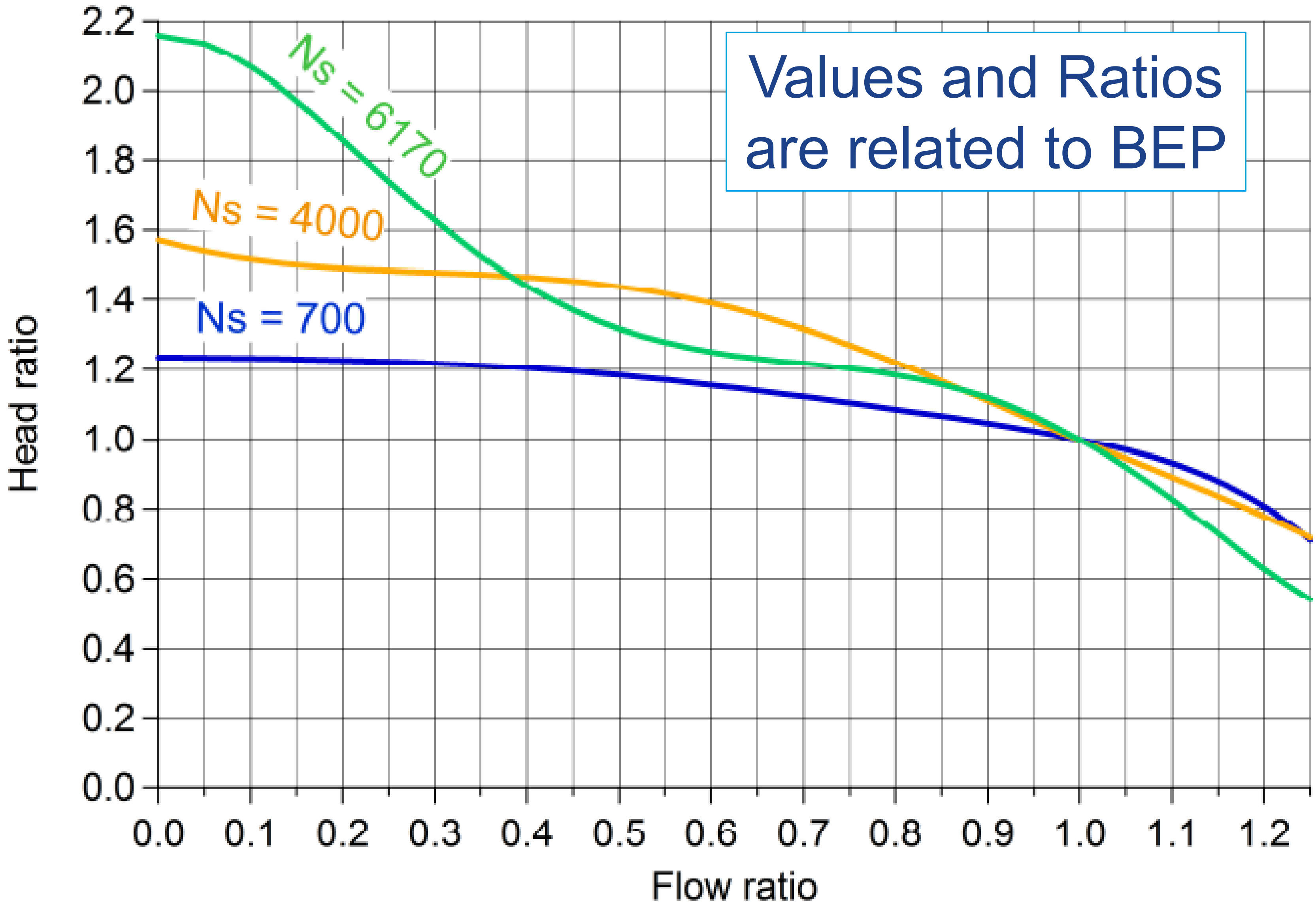
n = speed, Q = flow rate, H = head

	U.S.	Metric
n	rpm	rpm
Q	gpm	m^3/hr
H	ft	m

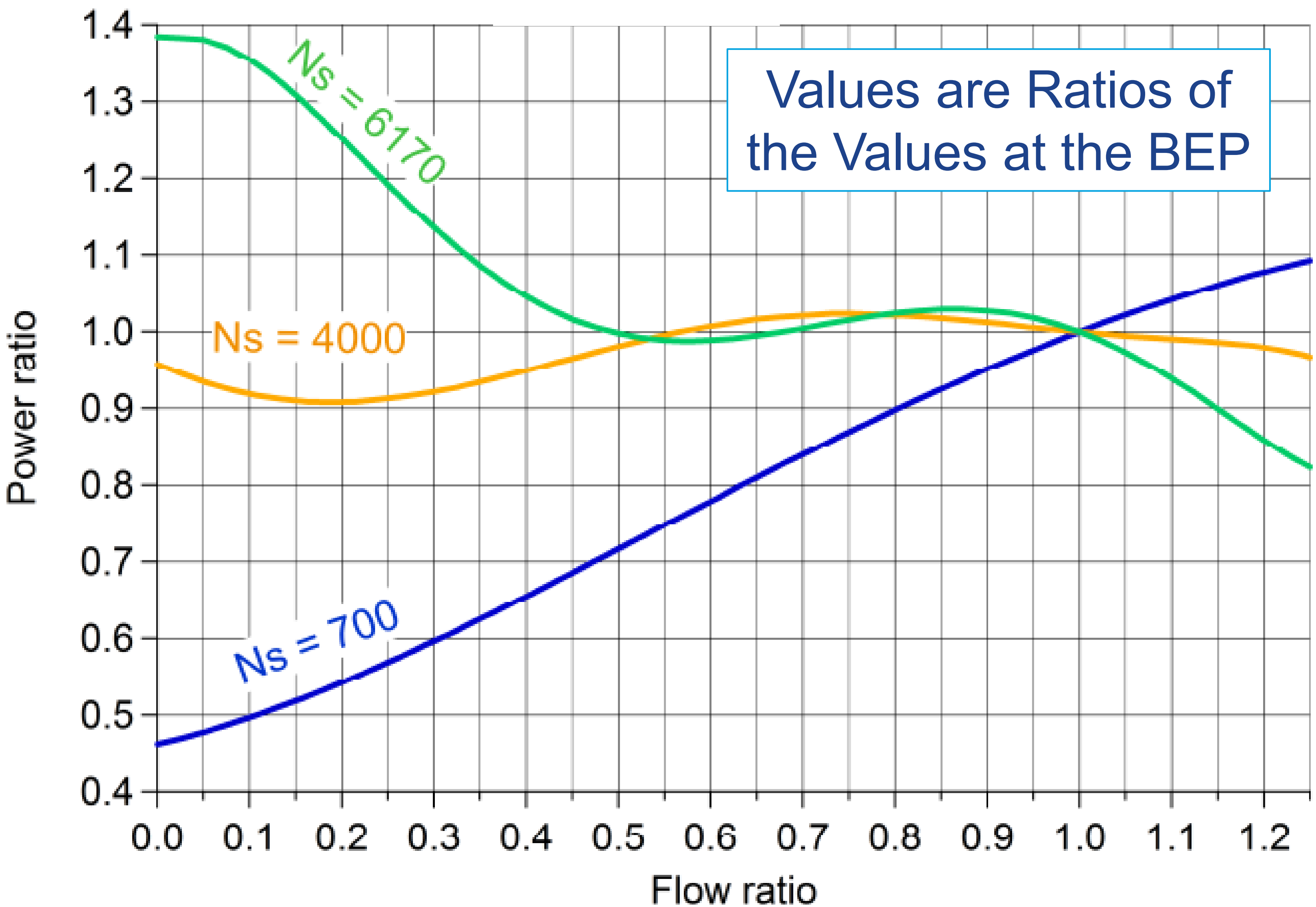
- N_s is ideally a dimensionless parameter that characterizes impeller performance and shape
- N_s is based on best efficiency point conditions
- Neither the U.S. nor metric system users normally employ units that provide dimensionless N_s units
- The specific speed for multistage pumps with similar impellers is based on head per stage



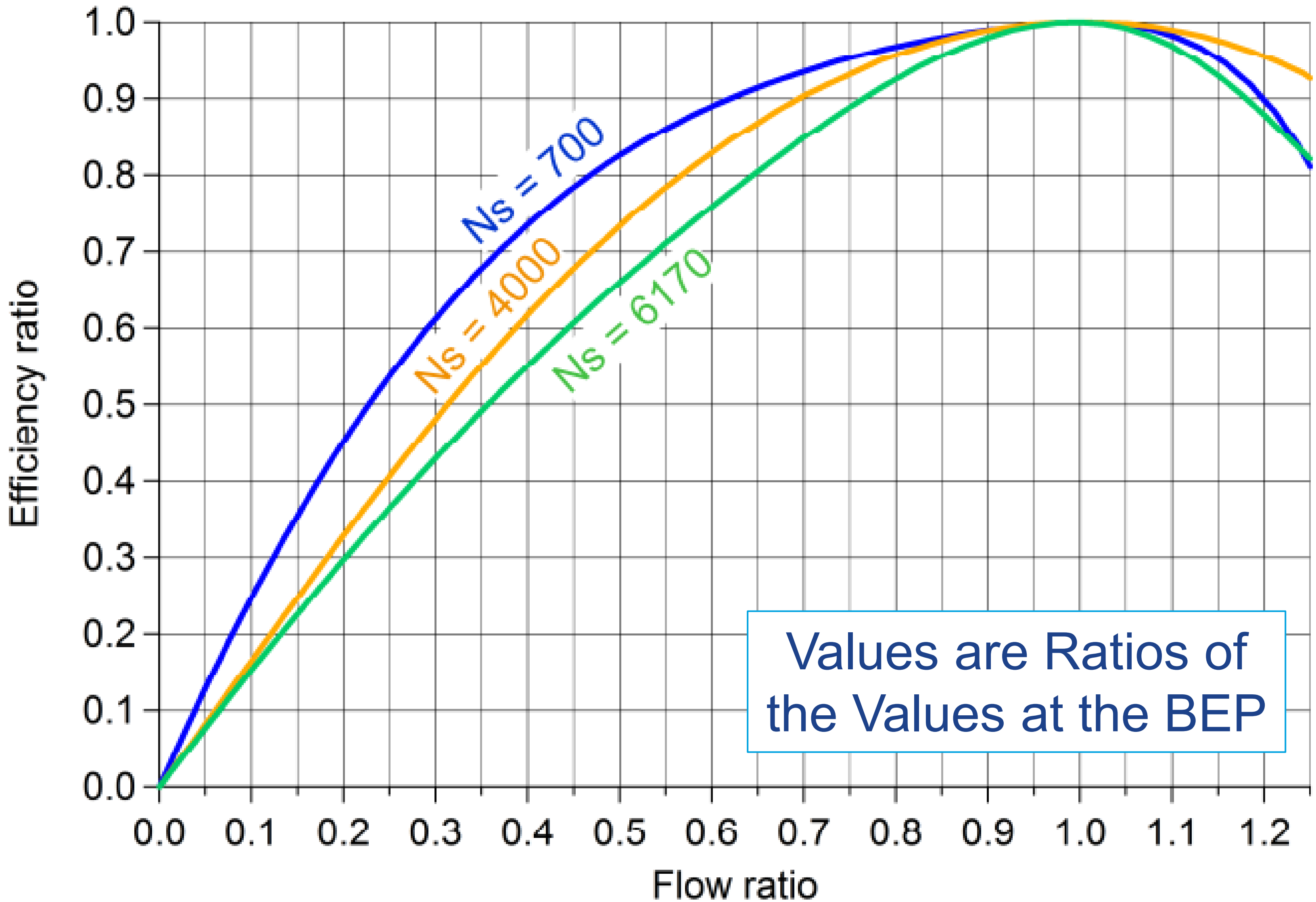
Normalized head-capacity curves for three different pumps



Power-capacity curve shapes vary considerably

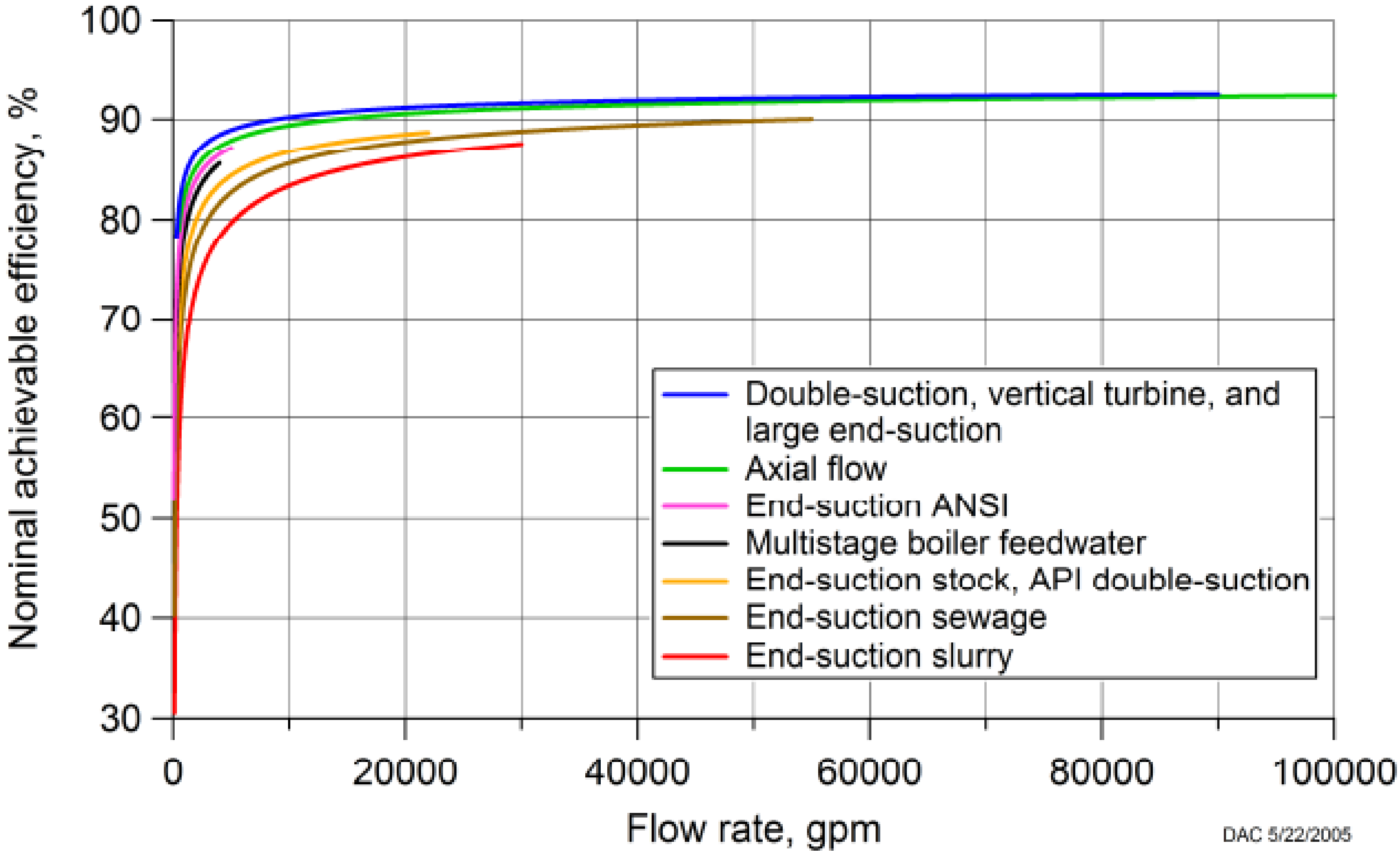


Efficiency-capacity curves

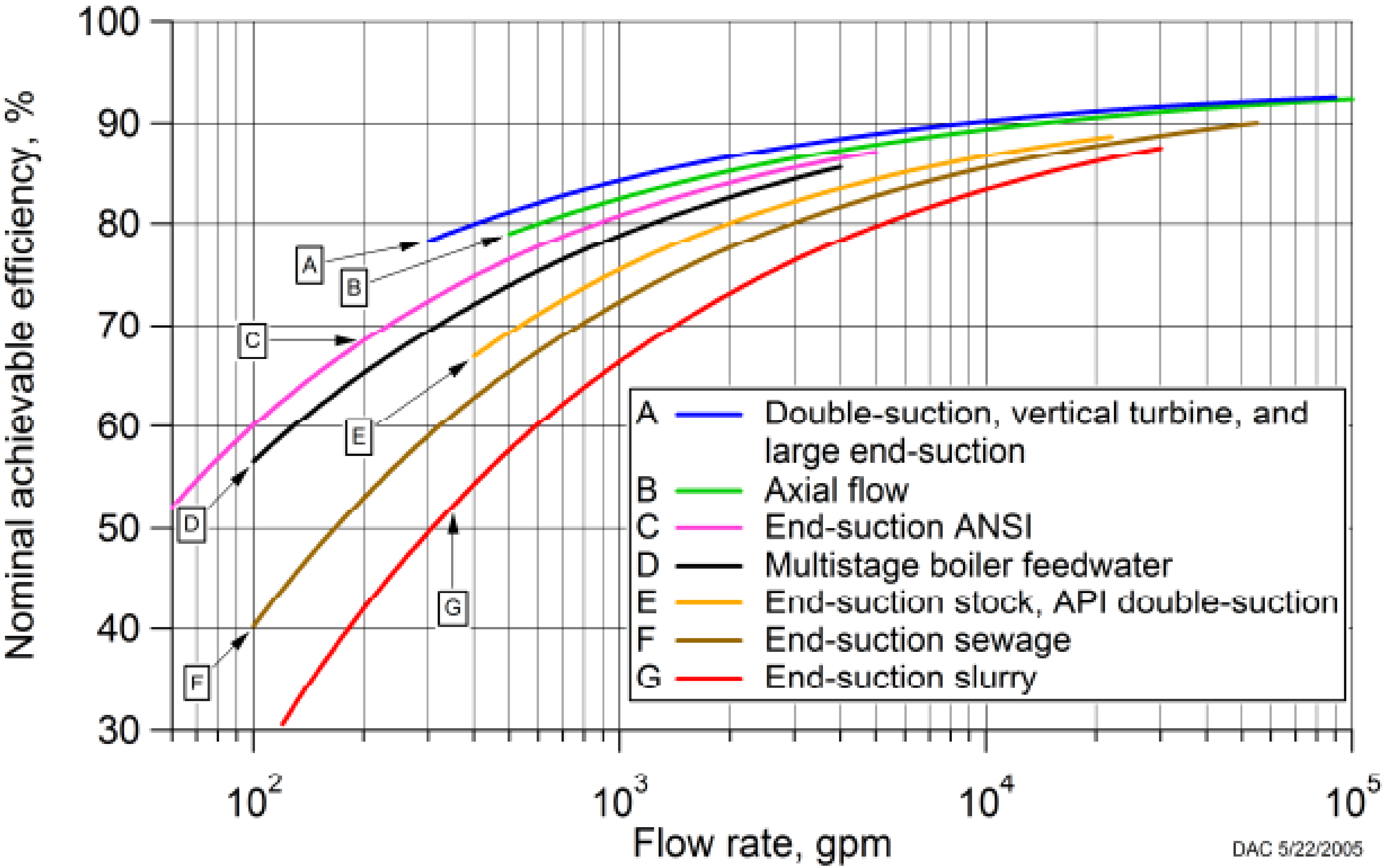


Values are Ratios of the Values at the BEP

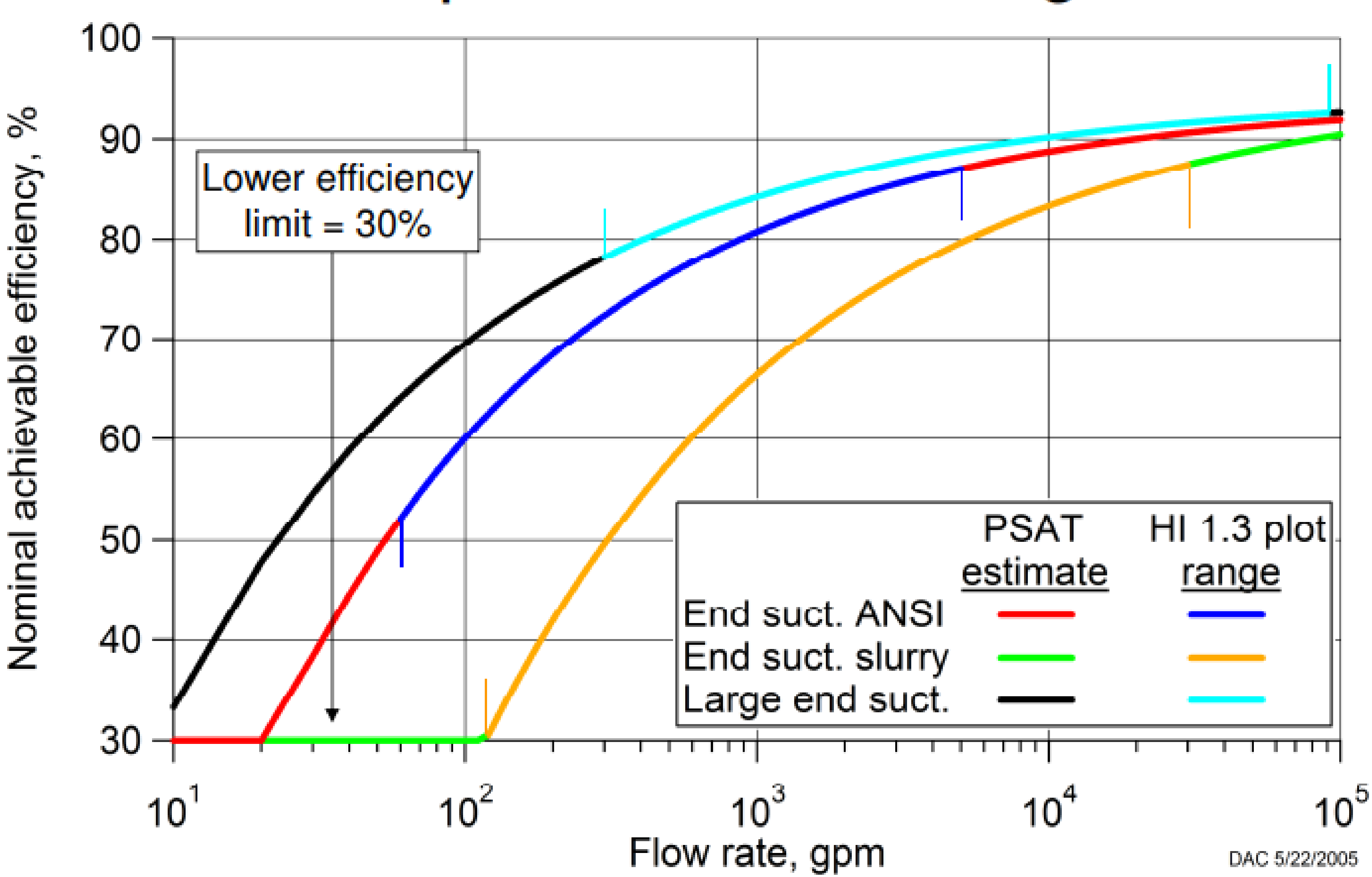
Nominal achievable efficiencies, per HI 1.3



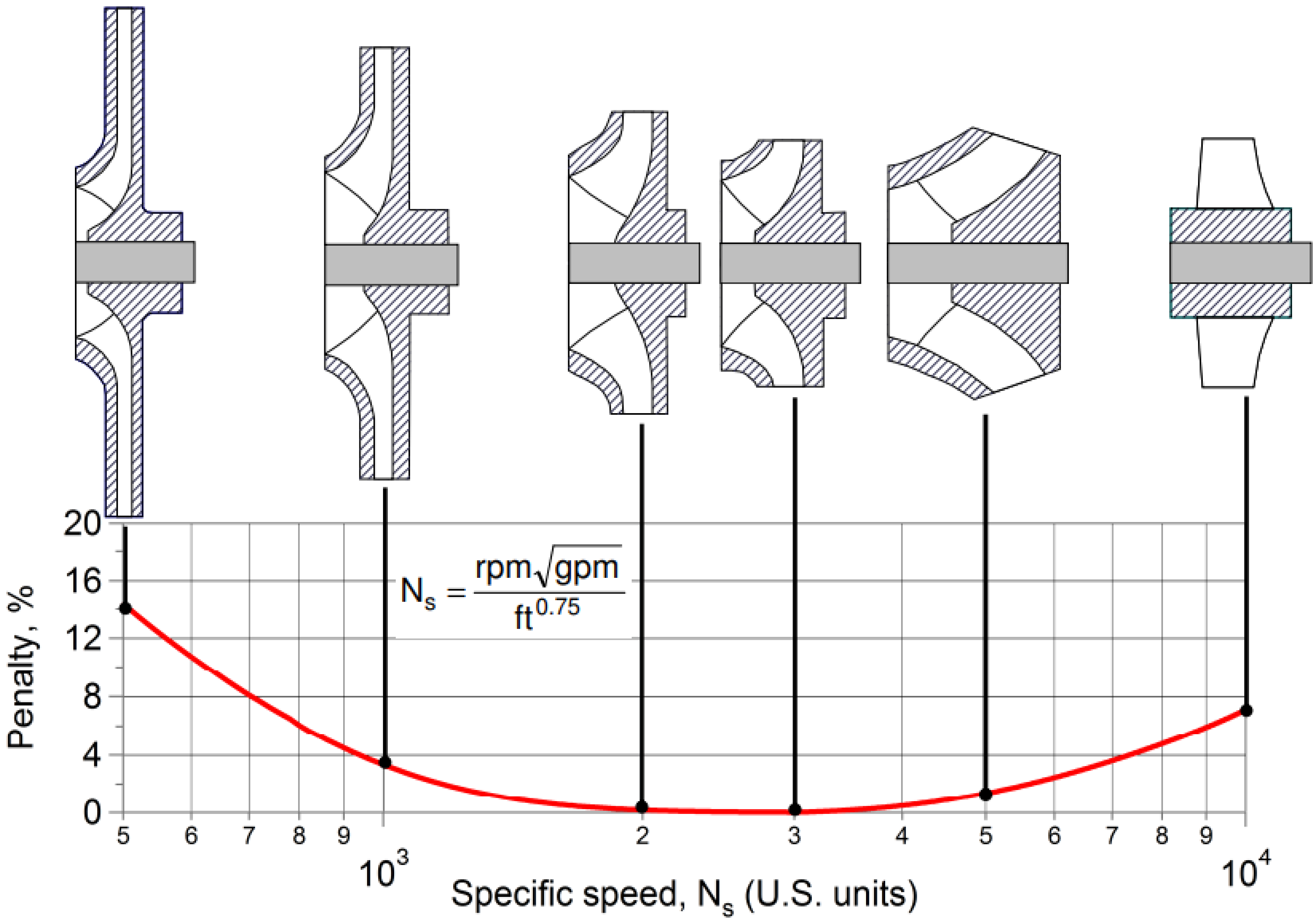
Nominal achievable efficiencies, per HI 1.3



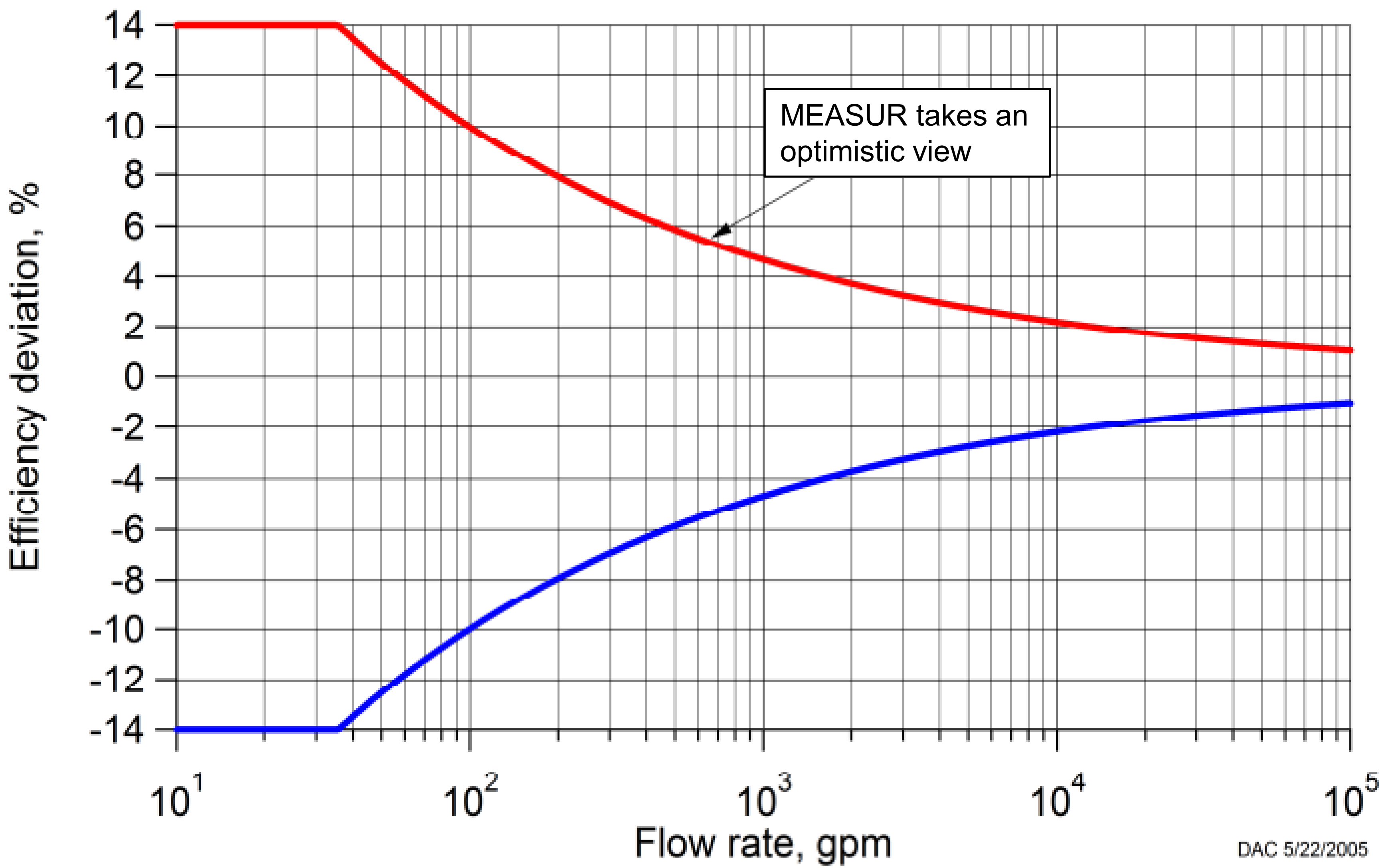
MEASUR takes some liberties and extrapolates the HI ranges



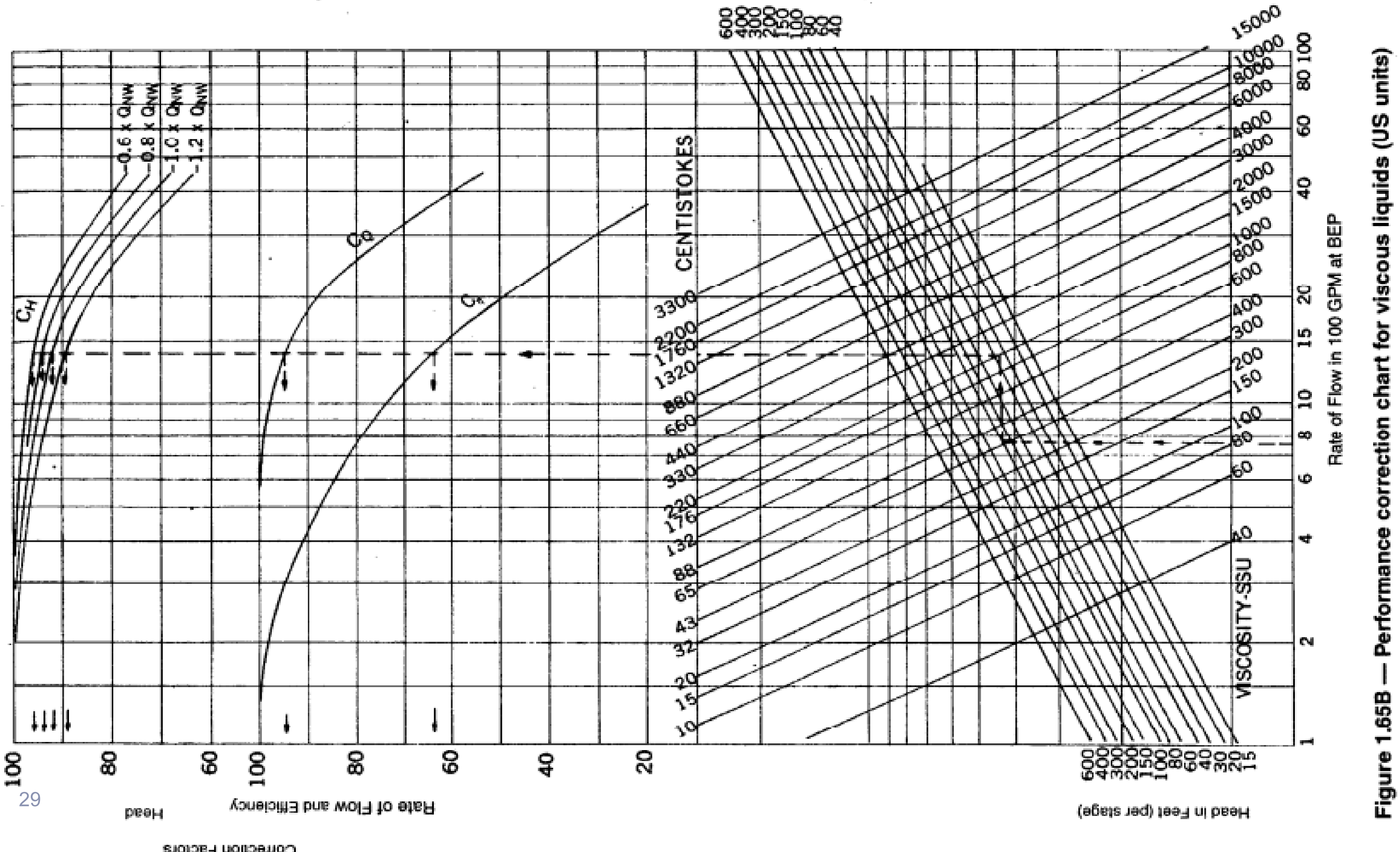
Achievable efficiency is a function of the specific speed



HI – 1.3 has a +/- efficiency deviation (% of generally attainable). MEASUR looks upward



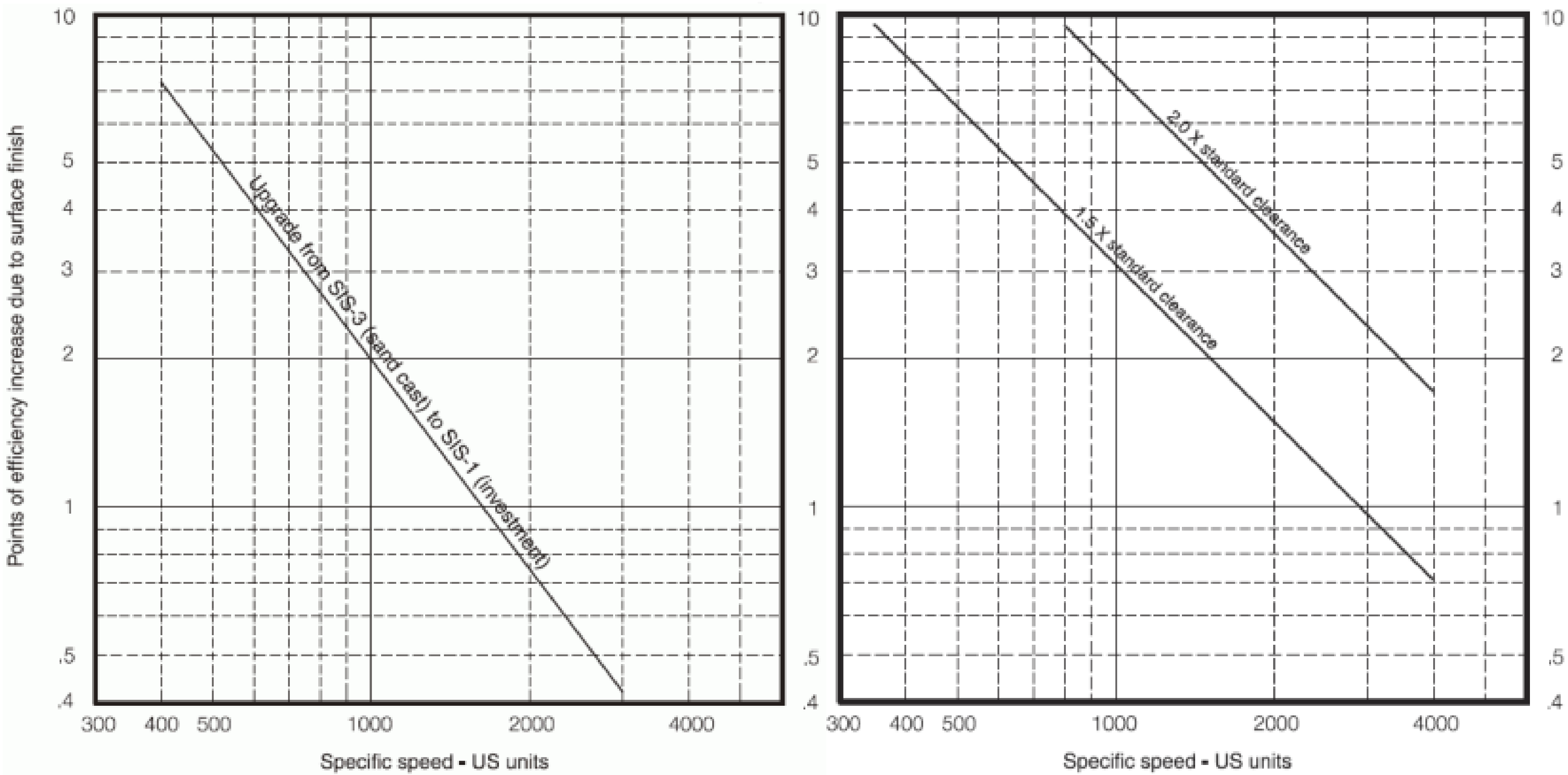
MEASUR uses HI viscosity adjustment



(Scanned from ANSI/HI 1.3)

Other factors: surface finish & wear ring clearance

- MEASUR does NOT apply surface finish adder
- MEASUR does NOT provide for wear ring clearance decrement

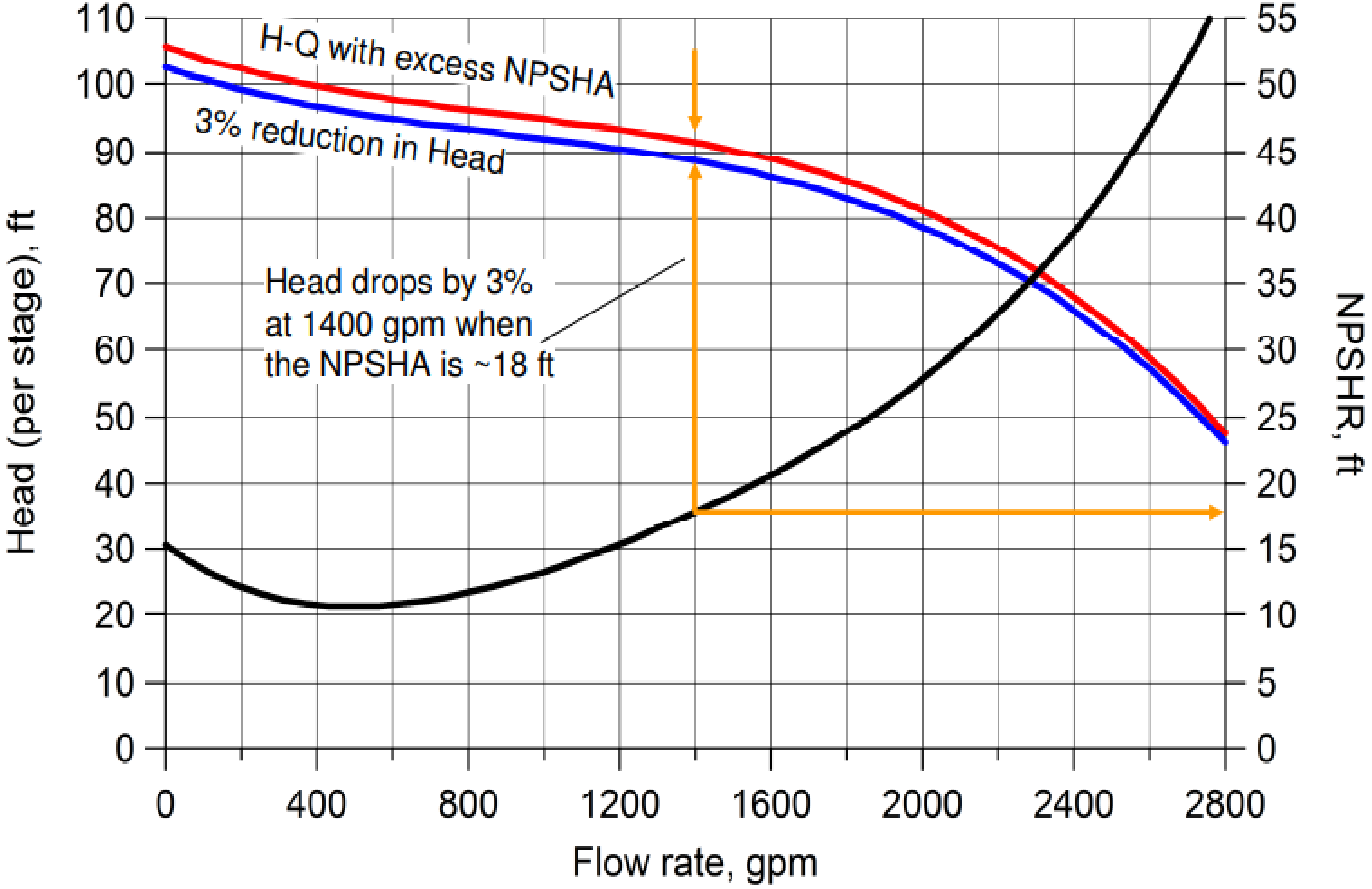


From HI-1.3 Figures 1.77B and 1.78B; pdfs available free from HI at www.pumps.org

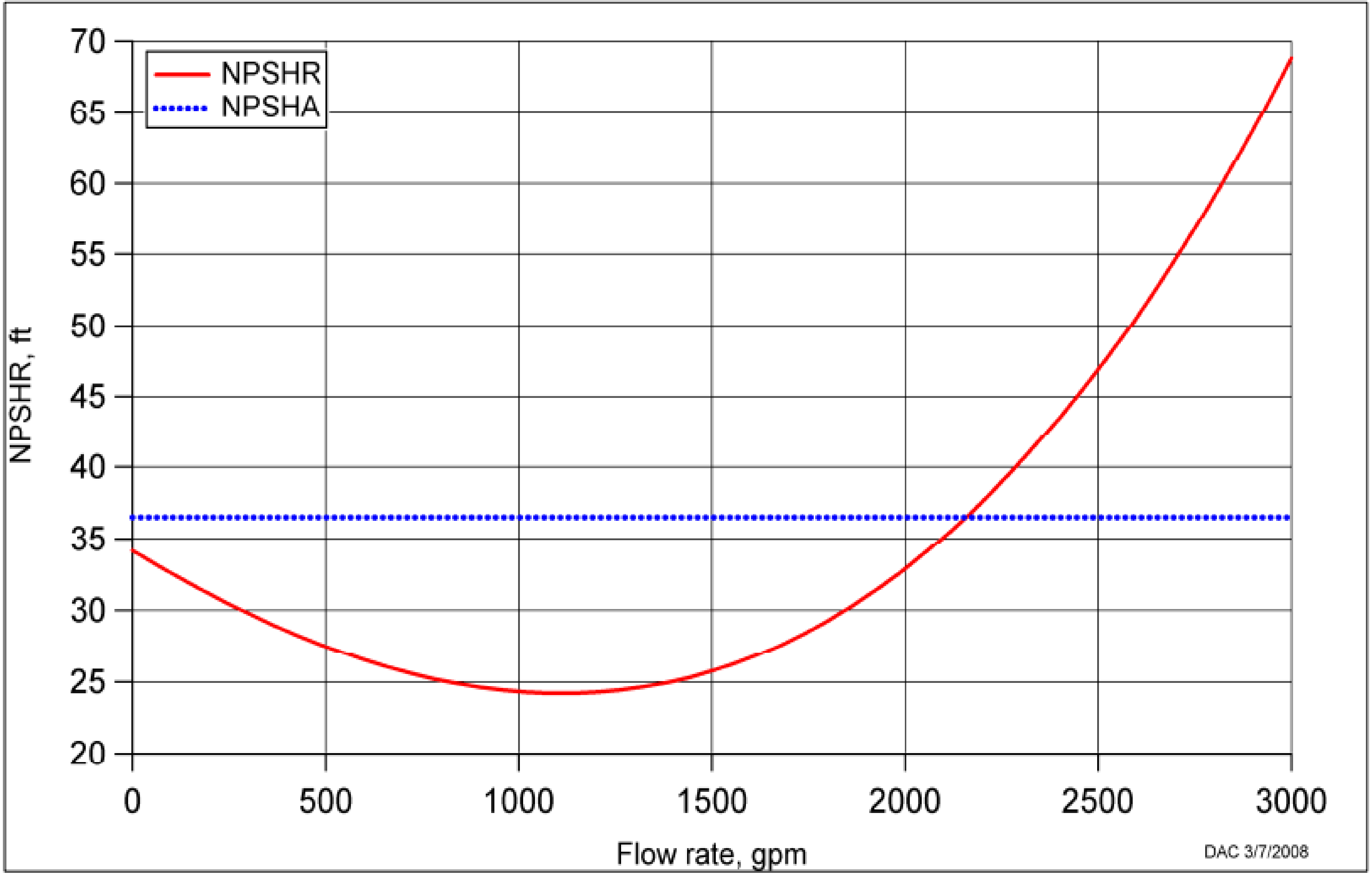
Net Positive Suction Head Required

- NPSHR is, by long-term accepted practice, the available suction head at which the developed pump head has dropped by 3% from the head that it produced with bountiful available suction head
- By definition, then, the pump performance is already degraded due to cavitation-related flow disturbance
- The actual point when cavitation actually begins can be with significantly greater available head than the pump supplier's NPSHR curve
- Two accepted approaches for developing the NPSHR curve:
  - Establish a fixed suction head, then increase flow rate until a 3% reduction in head at a particular flow rate is observed
  - Maintain a constant flow rate and gradually decrease the suction head until the developed head drops by 3%

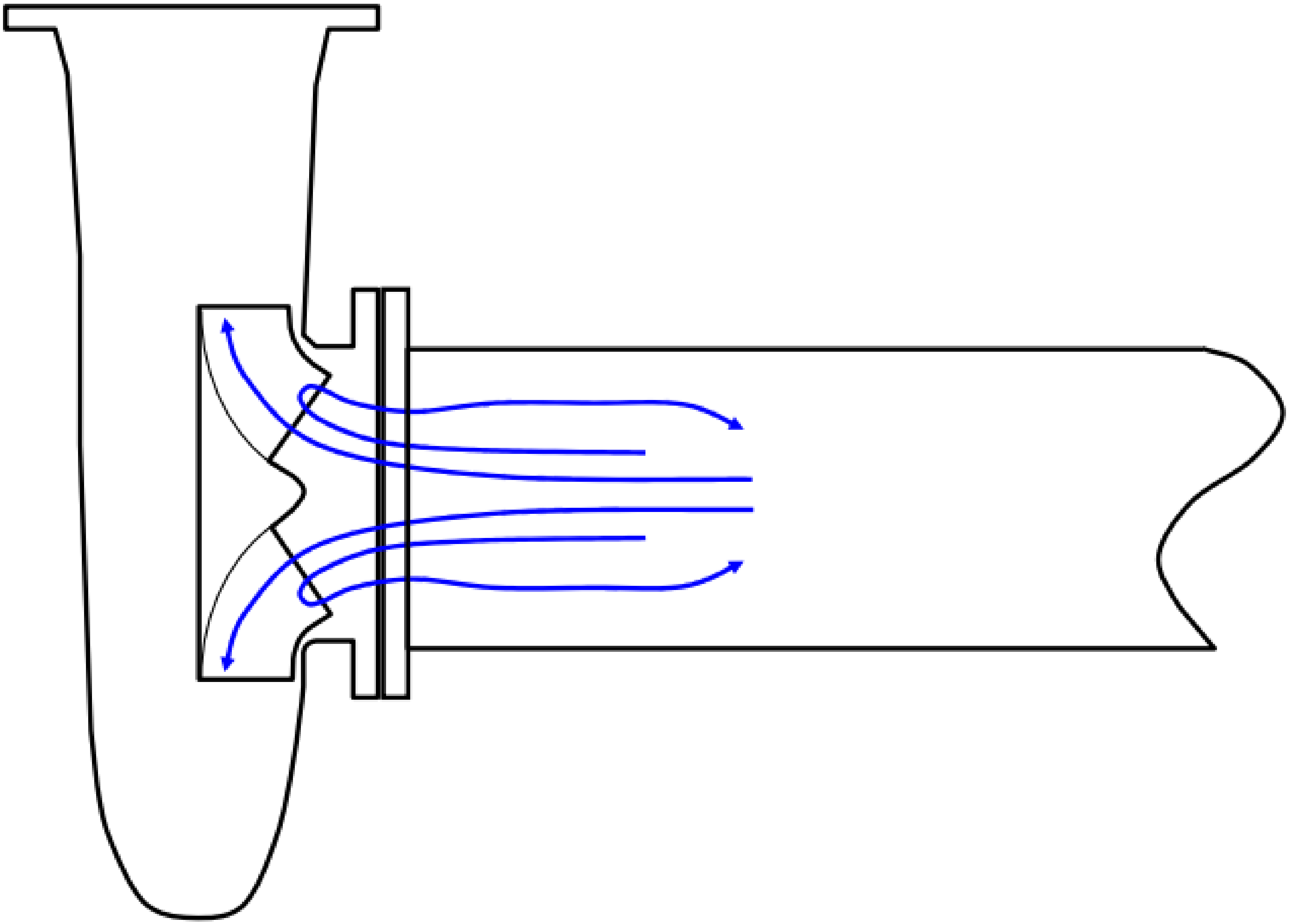
NPSHR: Available suction head with 3% degradation in developed head



NPSHR curve and NPSHA near the low level switch of a submersible lift station pump



Pumps operated away from BEP are more likely to have various problems – including at the suction



Suction specific speed

$$S = \frac{n\sqrt{Q}}{NPSHR^{0.75}}$$

S = Suction specific speed
N = Rotating speed (rpm)
Q* = Flow rate (gpm)
NPSHR = Net positive suction head required (ft)

* Half of the flow rate for double suction pumps

Reducing NPSHR translates into increasing suction specific speed

Suction Specific Speed, S (Suction Specific Speed Required) is another dimensionless ratio and is analogous to specific speed. It describes all the inlet conditions that produce similar flow conditions in geometrically similar inlet passages. Like specific speed it is defined at the BEP of the pump.

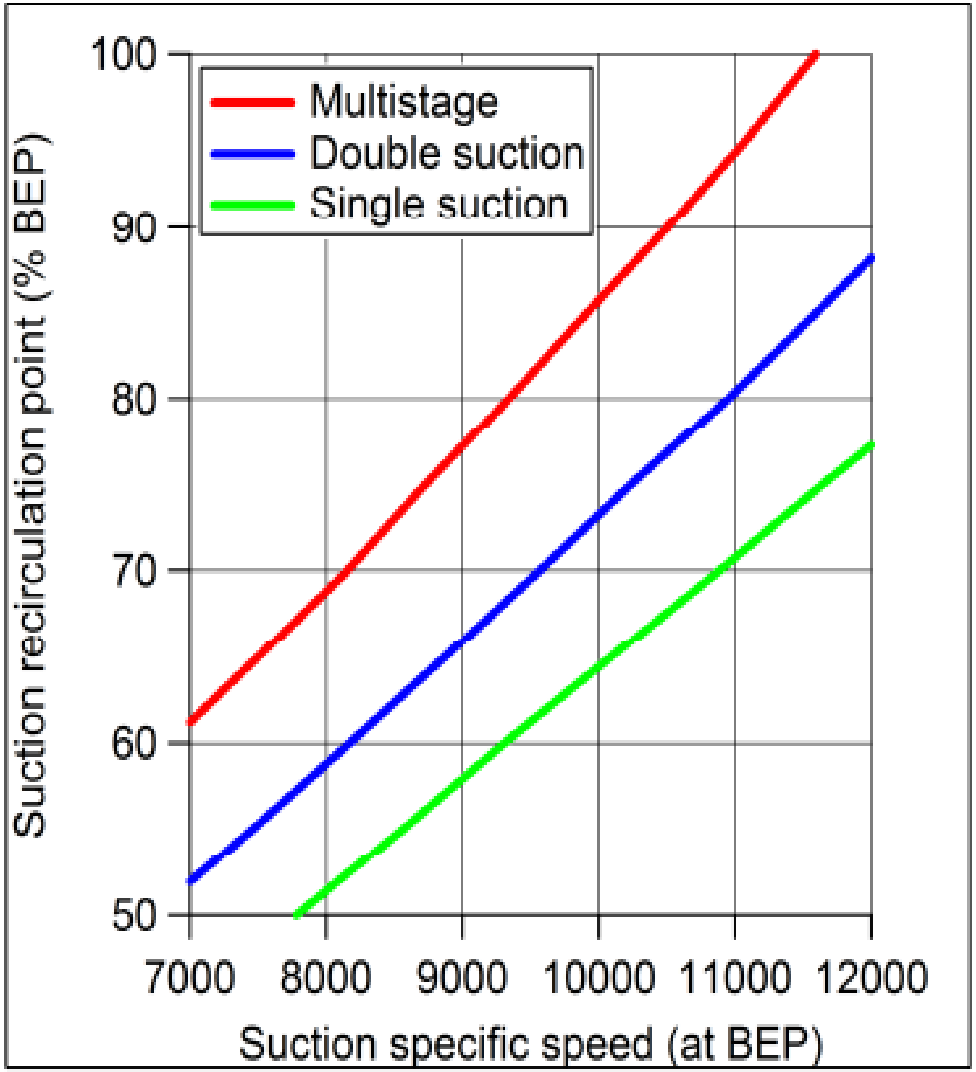
Suction specific speed

- Suction specific speed is an index number for a centrifugal pump similar to discharge specific speed and is used to define its suction characteristic. It is constant for a given pump, regardless of pump speed. Higher numerical values of "S" are associated with better suction capabilities. The numerical value of "S" is mainly a function of the impeller inlet and suction inlet design. For pumps of normal design, values of "S" vary from 6,000 to 12,000.

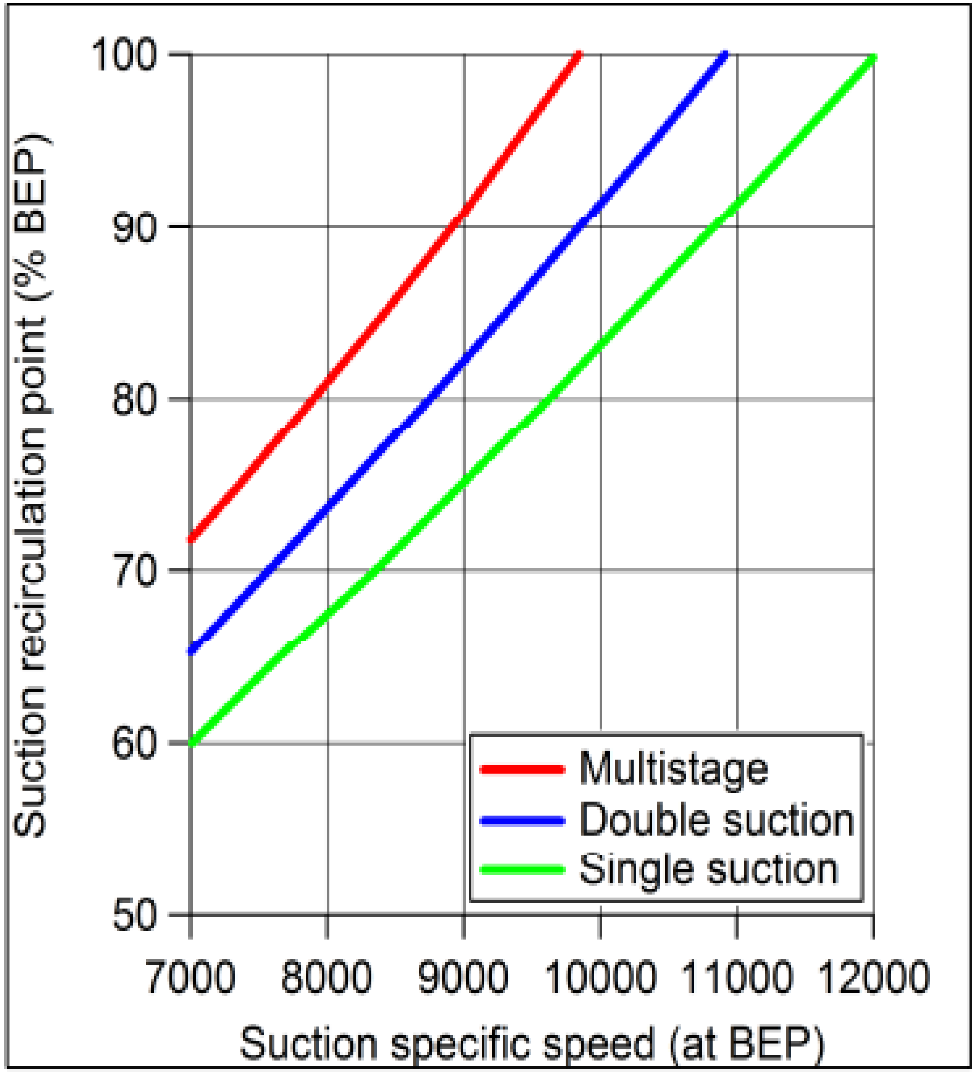
- It shows how aggressive the pump impeller inlet design is (how low is the NPSHR for a given pump speed and BEP flow rate). Higher "S" values mean lower NPSHR and, therefore, greater NPSH Margins.

- Suction Specific Speed is another factor (in addition to Specific Speed) in determining the flow rate at which suction recirculation starts in a pump. Pumps with higher values of Suction Specific Speed typically have narrower "Allowable Operating Flow Regions".

Fraser estimate of suction recirculation onset



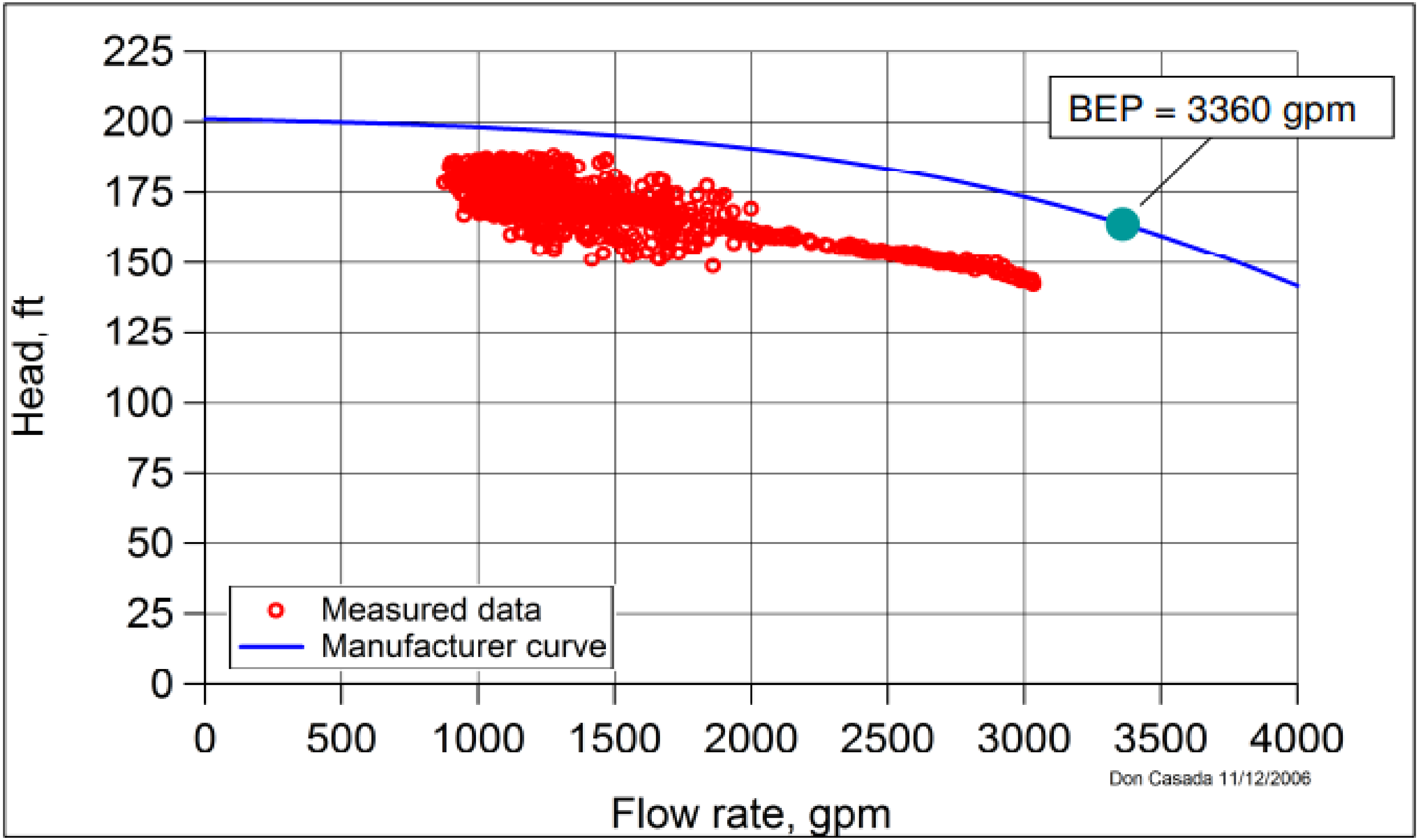
N_s = 500 to 2,500



N_s = 2,500 to 10,000

These are estimates of curves included in a paper by Warren H. Fraser, "Flow Recirculation in Centrifugal Pumps," Proceedings of the 10th Annual Turbomachinery Symposium, Texas A&M University.

A single suction pump with a well-defined point of suction recirculation onset



N_s = 2260
S = 8840

NPSHR is a well-established performance measurement, but it ain't enough

- NPSHA > NPSHR does not guarantee cavitation-free operation
- Higher suction specific speed pumps can, ironically, be more likely to be misbehaved (particularly as flow is reduced)
- Other important factors in determining the energy level and likelihood of damage include:
  - suction geometry
  - operating point (relative to BEP)
  - impeller material
  - fluid characteristics (including corrosive or erosive materials)
  - the pump duty cycle

Suction intake design

- Improper design can, even with adequate NPSHA, cause operating problems
- Example considerations:
  - Non-uniform flow profile
  - Entrained air in the liquid (e.g., from free-fall)
  - Inlet velocity and submergence
  - Spacing relative to the floor, walls, other pump inlet
- Many issues are addressed in ANSI/HI 9.8, Pump Intake Design, such as open structure inlet velocity and submergence

Velocities and vortices: some HI 9.8 guidelines

Flow rate range, gpm	Recommended inlet bell velocity, ft/s	Acceptable velocity range, ft/s
< 5,000	5.5	2 to 9
5,000-20,000	5.5	3 to 8
$\geq$ 20,000	5.5	4 to 7

Recommended minimum submergence to avoid introduction of surface vortices

$S = D + 0.574\ Q/D^{1.5}$

S = minimum submergence (in.)

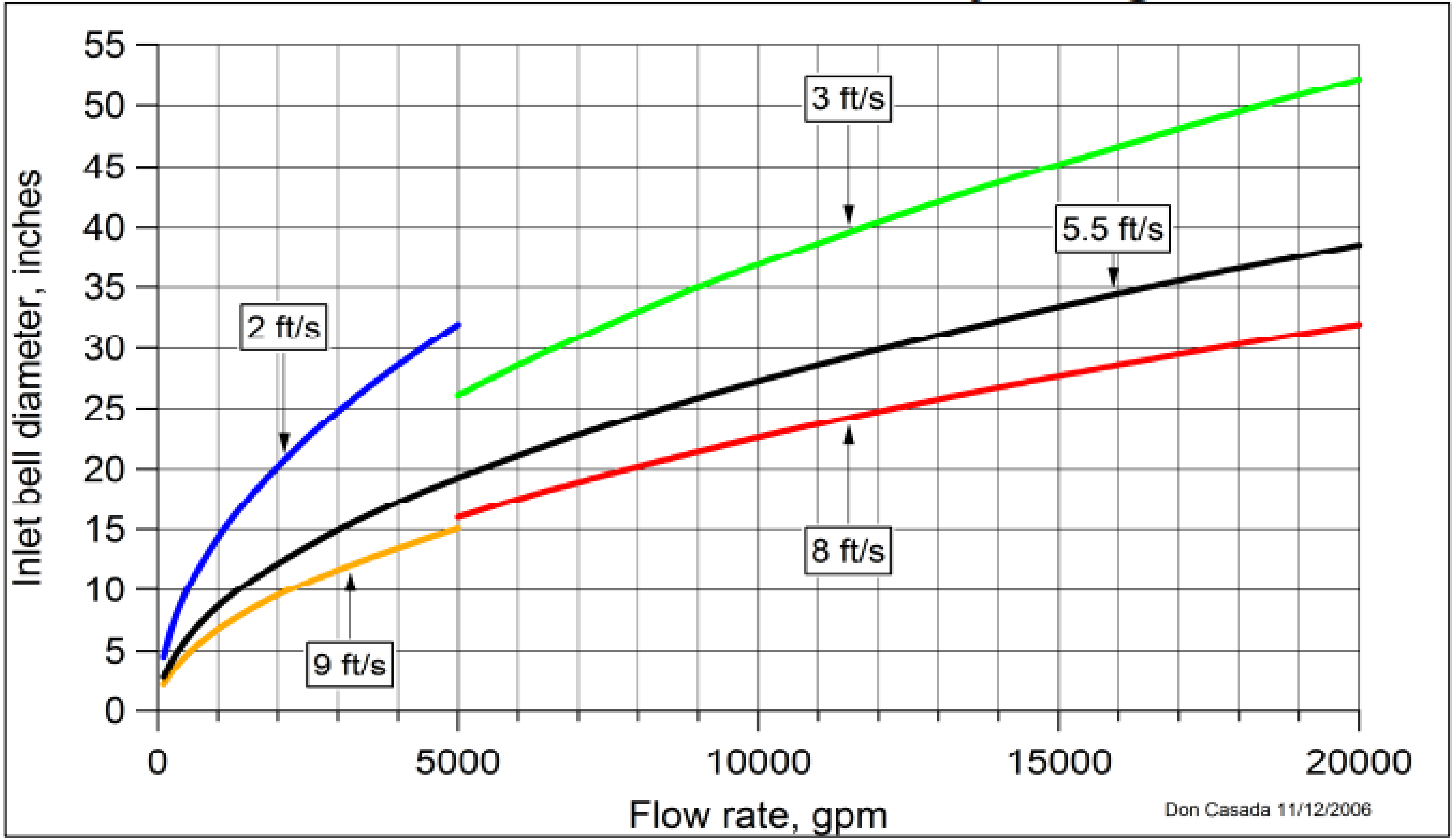
D = bellmouth inlet diameter (in.)

Q = flow rate (gpm)

Source: ANSI/HI 9.8, Pump Intake Design

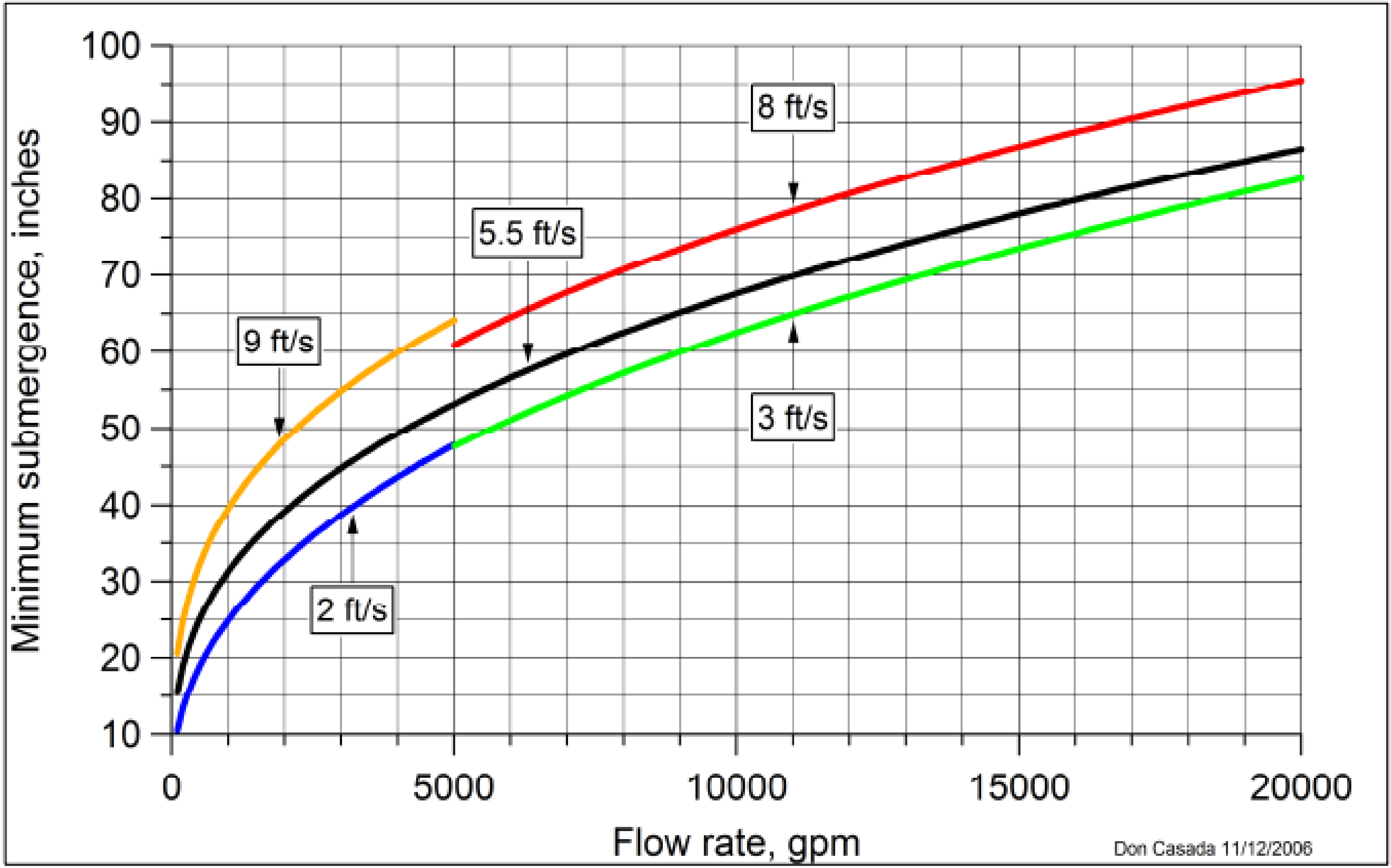
Note: This standard also provides guidance on a variety of suction recommendations, ranging from floor clearance to distances from disturbances to the suction flange

Bellmouth inlet diameters corresponding to the HI-recommended velocity ranges



Note: the lower velocity limits are primarily a practical consideration related to the physical size and economics of the pipe.

Minimum submergence to avoid introduction of free-surface air vortices



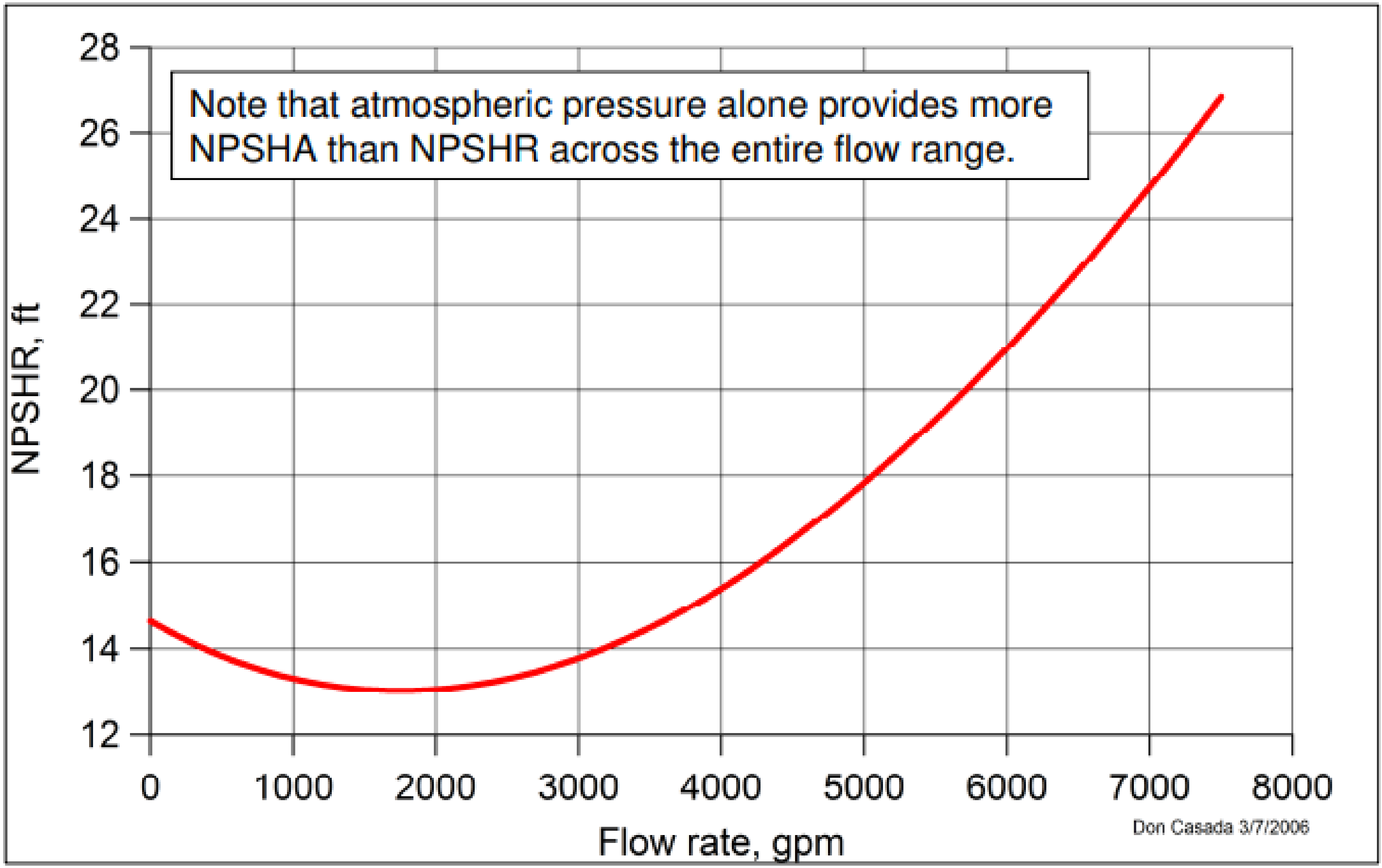
A pump with plenty of NPSHA that was not leading a happy and fulfilling life

Another one……………….

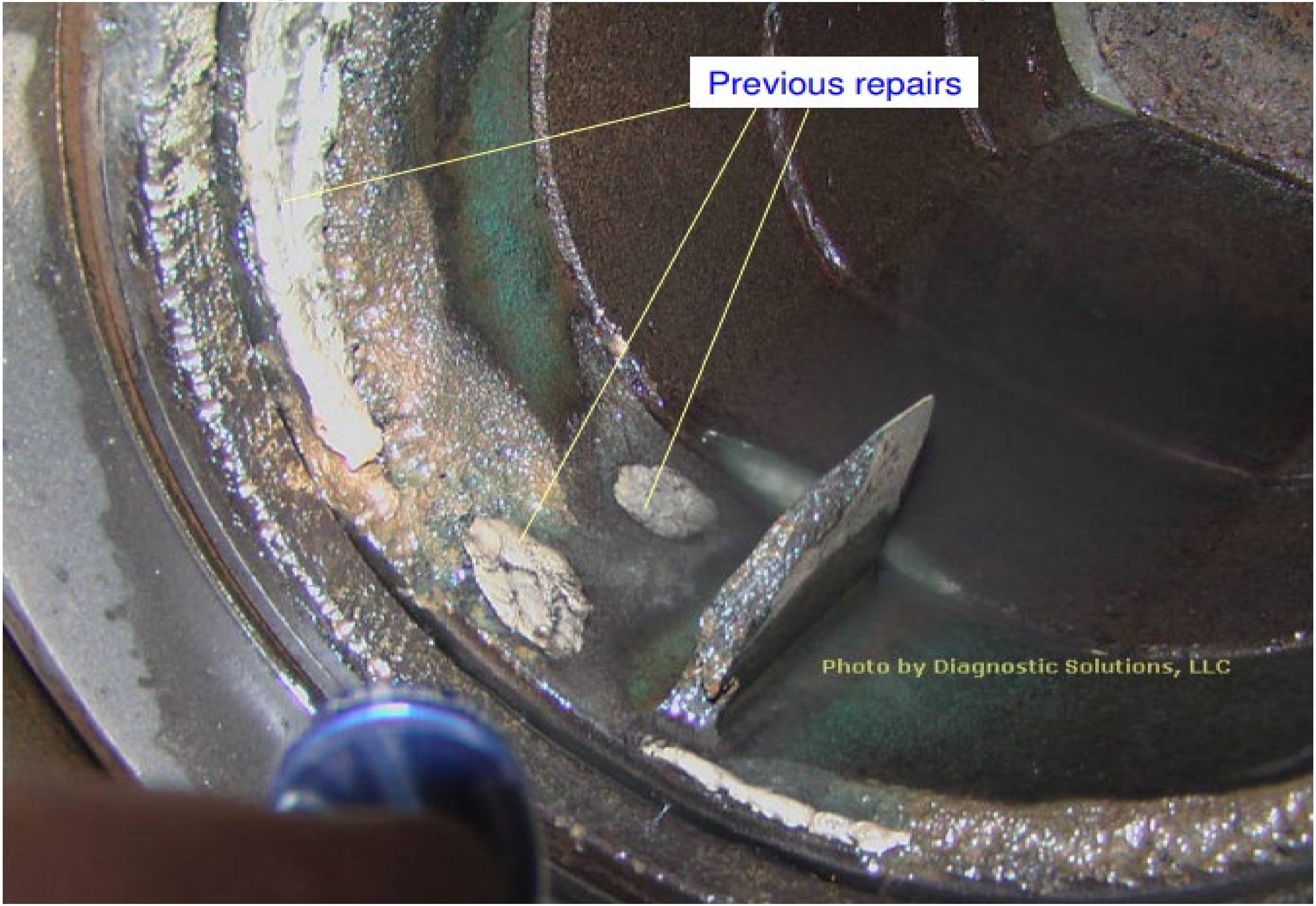
Pump with S = 14,260 (very high)

Compare nose of the impeller on the left (repaired spare unit) with the one on the right, which was being removed from the center pump. But also notice the crack at the vane/shroud of the repaired pump. No significant direct cavitation damage to either impeller was noted.

NPSHR curve for previous pump



Cavitation damage in the pump suction casing (from suction recirculation)




The End