MISTER MECH MENTOR

# HYDRAULICS & PIPING

By

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# CONTENTS

## VOLUME TWO (forthcoming)

Volume Two will consider these main areas of mechanical engineering practice:

- ***ASME Codes***
- ***Pipe stress and strain***
- ***Structural supports***
- ***Pressure vessels***
- ***Jacketed pipes***
- ***Bellows-type expansion joints***
- ***Process piping specialties***

Volume Two involves more professionally-advanced information than Volume One. Its contents were collected to make a companion piece to Volume One, which is more basic and intended for the novitiate, to keep him or her out of trouble on that first important job. Volume Two is perfectly within the capability of the younger engineer to absorb, however, and it should not be considered as limited to the perusal of "senior" people only. If your practice involves Industrial work, especially the ASME Codes for boilers and pressure vessels and piping, and such items of design or analysis as steam piping, hot fluids piping of all types, pipe supports, pipe system structural design, or pressure vessels, you definitely should obtain a copy of Volume Two.

# PREFACE

*Mister $M^{ech}$ Mentor* is a collection of technical articles written in a friendly, first-person style, meant to help new graduates and practicing workers solve certain important mechanical engineering problems. Its title reflects its intent, that is, to augment the reader's necessary training in the way a caring mentor would use. It explains the "why" as well as the "how" in some areas of practice notorious for being misunderstood, with the goal of helping certain potentially dangerous lessons in physics and engineering design application be learned safely, and convincingly, without subjecting the novice to the suffering and embarrassment of learning "the hard way." It seems that life's (and engineering's) most important lessons are learned by making painful mistakes; the author insists that it is far better to learn from the past mistakes of others, than to repeat the mistakes oneself. Whenever actual examples of such accident cases are helpful, they are included in a frank and colorful way most people will appreciate.

Although its primary readership is intended to be mainly younger people who have yet to gain certain vital engineering experiences, and who do not have access to a senior"flesh and blood" on-the-job mentor, the more seasoned engineer may also find it helpful as a quick refresher and source of organized solutions to the ubiquitous problems it embraces. Code references are especially valuable.

Primarily involving mechanical process and utility piping system design and stress analysis, fluids handling (pumping and network flow controls,) real-world hydraulic puzzles and such transient accidents as destructive water-and steam-hammer, plus useful solutions for mechanical stress and strain problems often related to these systems, the book's selected topics are commonly encountered on the job by folks who work in these engineering fields.

- Engineering design/construction firms
- Contract engineers and designer/technicians
- Architect/engineer/planner/consulting firms
- Mechanical contracting firms: process facility utility, HVAC, and plumbing
- Environmental firms especially involved with mechanical equipment and piping
- Forensic loss consultants
- Staff engineers in plant projects, both direct engineering and management, and in utilities, maintenance, safety, and environment departmental duties, and ***especially*** the chemical/manufacturing process HAZOPS team members.

# FOREWORD

**Since this is my book, and since it pretends a claim, however modest, upon mentorhood, then I suppose I am duty bound to offer you my own points of view upon our chosen profession, engineering, and that is what I will try to do here. Of course, my opinions are just that—opinions. Everyone has them. It remains the prerogative, in fact the professional duty, of each of us to strive toward finding his or her own personal truths.**

Any personal guidelines which I share here for your consideration are necessarily taken from my personal experience. Falling personally and quite remarkable short of genius, I have had to master the important lessons of my profession and my life as most folks do; by learning from my own mistakes, when unavoidable, and from the mistakes of others, whenever possible. I sincerely hope that all your learning experiences will be of the latter kind, and that your pilgrimage will be more worthy than was my own.

I like to make simplifying assumptions as much as the next guy does, and my point of view in design will always be conservative, but not ridiculously so. Rather, my degree of conservatism in any technical matter is always assigned in sensible proportion to the particular consequential dangers which might accrue if I made errors of judgment or calculation.

Knowing where to draw the line with yourself is the key. That knowledge will come with practice and observation and experience. You were not born knowing where the line should be drawn; none of us were. But you *were* born with a head full of common sense and valuable human intuition, and a heart full of the inner voice of conscience. Use all of these gifts without hesitation or apology. Weave them into the framework of your professional practice and of your life as well, and the rest will come with time.

I try to be efficient and productive, to create refined systems without putting too fine a point on things, and will avoid gilding lilies and reinventing wheels as best I can. I strive for maximum simplicity and understandability in the things I design, because it seems to me that these are the sources of elegance. They are without question two of safety's necessary ingredients.

If I find that I cannot in plain language explain my design precisely, completely, and clearly enough for its operational physical principles, means of control, range of safe operation, design intent, natural physical limitations, expected service life, and requirements for proper safe operation and maintenance to be thoroughly understandable by its intended owners, builders, operators and maintainers, ***and especially by myself,*** then I go back and simplify the design to the point at which it *will* be 100% understood. If I have to, I will make those changes on my own time and expense. I will never be rich, and don't care; however, I *will* sleep well at night.

As experienced technicians and professionals, we know what we know, and what we can do, and we are expected and paid to do "good engineering" within our range of actual competence. And indeed, we do try our best to be clever and innovative and thrifty and thorough and sophisticated and brave and true-blue and all those other neat things we want our employers and clients to think of us. And being human, many times we are tempted to stretch just a bit beyond, to take a little chance, maybe to want to brag a little, or bite off a tad more than we can comfortably chew. You know the drill. Our nature makes us want to promote ourselves, to continually market our abilities, to advertise our strengths both real and imagined, and by all means to hide our weaknesses and fears at all times, all the while exuding cool self-confidence, and maybe even a general aura of salty seaworthiness. All of which is perfectly natural. Perfectly human.

But we have taken upon ourselves the professional responsibility to do a certain kind of work, ostensibly one which greatly benefits mankind but which, if not done properly, has the potential to do great harm instead. And because of that, we must act professionally, responsibly, at all times in our work. Even when to do so would seem contrary to our own personal advancement. We must not try to practice outside the boundaries of our own limitations, all by our intellectual selves alone. No. To grow our abilities safely, we need to take our first steps on strange new ground with someone else present, someone who knows the ropes, to check us and guide us and keep us as well as our potential benefactors-nee-victims safe from our fledgling efforts. We must swallow our human vanity and ask for help when we need it. And trust me, the oldest and best of us need help much more frequently than you are led to imagine.

Those who blissfully ignore their personal limitations and press on into unknown territory alone, without first achieving a truly satisfactory upgrading in knowledge sufficient to the undertaking, are truly dangerous to themselves and the public whom they are charged with serving.

Every true profession recognizes this principle. Practitioners of education, law, medicine, those who serve us in the military, the guardians of public health and safety, and all the rest; all *know* this, whether or not it receives much public mention.

The key word here is "*alone.*" Do not hesitate to ask for guidance when you sense it is needed. Do not allow yourself to be forced into giving snap explanations or making hasty decisions, thinking that, if you do *not* you may damage your reputation. Far from it! It has been my life experience that really bad screw-ups do not happen unless the opportunity to prevent them falls upon every weak link in the project's entire chain of production.

Each organization involved in that chain will have one de facto decision maker; sometimes, that person will be you. Don't you become a weak link due to fear of speaking out, or shyness about asking questions which you fear might seem "stupid" to the others. When it is your turn to act, when the problem has rolled up to your workstation and you see that a problem exists in the project and that sooner or later there will be trouble if someone doesn't do something to fix it, then by all means, blow the whistle on it! It is your professional responsibility to do so. And, yes, it might cause some "big guy" to look bad somewhere in the chain, and he might cause you trouble and try to get you taken off the job, or even fired.

Well friend, that is just an inevitable human experience. It is a test of your mettle. Sooner or later, it is going to happen to you on the job. When it does, do what you know is right, and stick to your guns. And if you in fact turn out to be correct but lose the argument anyway, if the organization knows you are right but fails to support you or spits you out, whatever the reason, then FINE! It simply proves that they themselves are seriously flawed and not worthy of employing professionals. You shouldn't be working there anyway. Move on to another place where people are willing to act professionally and will invariably do the right thing on principle!

What is a professional engineer? I say he or she is an engineer who possesses necessary minimum levels of professional judgment consistent with adequate computational ability, plus an adequate base of scientific and technical knowledge gained through accredited formal education, plus the ability to master complicated abstract procedures, plus common sense practicality plus emotional maturity plus a well-developed sense of duty and responsibility, plus the ability and willingness always to continue the personal learning and improvement process, not only to teach himself or herself through continual self-study aimed at professional growth but also to seek out the wisdom and valuable experience of those who have been proven to have it.

At present, engineering lacks the grueling internship so justifiably prized and touted by the medical profession. As fledglings we are given typically nowhere near the kind of scrutiny that lawyers invest in their new hires. And we surely don't give our people the kind of gutsy, realistic, no-punches-pulled training that the professional military must receive in order to do their job. with even a prayer of personal survival! No, the "onus" of self-policing is definitely sitting squarely upon our own shoulders. And that is where it should be anyway, if we are to be truly worthy of professional status. We have to make it our own final responsibility to know exactly where our own personal limits of competence end, and where the vast sea of unknowns, our remaining "uncertainties," begins. And believe you me, we all have limits, great bunches and gobs of "uncertainty."

After all, are we not merely human? The more we poor creatures see and experience and learn, the more we realize the true depths of our own ignorance and human frailty. It's just that the professional keeps on trying to improve the situation, realizing his or her quest is finally beyond human endeavour and can never truly end, right up until his day of death.

And truly, the quest needs to come ahead of personal aggrandizement. The best professionals, the best engineers, the best people of all walks of life whom I have been privileged to know during a lifetime of practice have been those who frankly admit their limitations, appearances be damned! They most certainly do not refrain from asking questions or begging assistance when they face something that frightens them, way down inside. They will put the actual welfare of others ahead of their own personal ambition every time.

Finally, I am compelled to give you my honest appraisal of engineering. It is my long-term opinion that this profession demands more actual brain sweat and more acceptance of tangible responsibility, yet repays one's effort with proportionately less money and more grinding of teeth during the late night hours, than any other of which I know. With the single important exception being that of the professional soldier, who can add real mortal danger to the list of professional living conditions, and gets less in return for his tangible sacrifices than do all other men.

I think you have to enjoy this work for its own sake, and have a fair share of intellectual curiosity and the impetus to continually seek more insight into the workings of the universe, for engineering to make sense as a career. And on those terms, I think it does.

CHAPTER

# 1

# WATER AND STEAM HAMMER PHENOMENA

This topic encompasses a rich assortment of transient hydraulic phenomena, every one of which can mean bad news for the design engineer. Water hammer and its cousin steam hammer have well-known potential for causing disastrous accidents.

The topic can be as difficult as it is rich. One could spend a lifetime analyzing endless unique fluid transient situations, wading hip-deep in partial differential equations in the complex domain, or plowing through "methods of characteristics" diagrams and matrices of equations, or setting up more and more complicated 3-D grids for Computational Fluid Dynamics (CFD). Time consuming and expensive!

But thankfully, 99 times out of 100, difficult analysis is unnecessary for our purposes.

Like most of macroscopic physics, the basic cause-effect relationships of water hammer and steam hammer are not so hard to understand. And mathematical difficulties are involved only when we try to quantify exact time-location-velocity and time-location-pressure maps of transient flows in piping systems. There are some excellent guides available in this very specialized subject matter, if you ever need to pursue such tricky quantitative analysis for some reason, and I will mention those sources at the end of this article. Such complexity is clearly beyond the scope of solutions needed by this book, however.

Luckily, our usual job is not to figure out complex point-to-point pressure-velocity-time maps of transient events, but only to check our *hydraulic (i.e., mechanical and civil piping systems)* designs against the possibility of having bad transient effects occur in them at all. It turns out to be relatively easy to assess whether or not a particular fluid system will have potential transient problems. I contend that our responsibility as professionals is to make sure that we eliminate such potentials completely from our designs, at least insofar as possible.

Recognizing a system's potential to produce water/steam hammer effects "boils down" to determining whether or not either of this pair of triggering events can occur in the system's possible operating scenario:

1. **very rapid change in bulk flowrate impressed upon the system by a disturbance such as a sudden valve closure; (see Example Problem 1-1 for complete discussion and definition of what constitutes "rapid" and "sudden");**
2. **rapid phase change (i.e., collapse of a vapor pocket inside the liquid-filled piping system).**

The first of these triggers causes a classical "water hammer" event to happen. Water hammer events are all pretty much the same old thing from case to case, except in degree of damage done to the piping system.

The second is the trigger for "steam hammer," which comes in a variety of costumes and disguises. It can get confusing, so let us define the terms as we go along. The underlying principles are simple and straightforward.

## Plain Vanilla Water Hammer

The principle in avoiding plain vanilla water hammer is to keep rates of changes of momentum in flowing streams of liquid at or below certain quantitative values. The classical physical setup for illustrating water hammer is shown in **Figure 1-1.** Let's look at the upper **Figure 1-1(a)** first.

In **Figure 1-1(a)** a liquid is flowing from left to right, at the same constant velocity **(Uo)** at all points in a pipeline. The pipe is horizontally level and straight, and contains a shutoff valve which is wide open at first, but is capable of being closed extremely rapidly. Upstream of the valve, flow comes from a pump or reservoir. Downstream, the pipe terminates in a free discharge to the atmosphere, or to a large plenum which will remain essentially at a constant total pressure despite influences from the pipe's outflow.

Careful static pressure measurements of the steady flowrate made along the centerline of pipe [**Z-axis in Figure 1-1(a)**] would show a nice, smooth pressure gradient that decreases from left to right, behaving as predicted by the familiar equations: continuity (mass conservation), Bernoulli (hydraulic energy conservation), Darcy-Weisbach (total pressure loss due to friction), and Newtonian mechanics (dynamic equilibrium; vector sum of forces along axis "Z" equals zero).

**Figure 1-1(b)** shows the situation has changed drastically only milliseconds after the valve has very rapidly slammed fully closed. We see the event is not symmetrical in space; there are two different things going on, depending on which side of the valve you examine. Lets take the left side first, because it is much easier to understand and calculate than the right side. We will go through it step by step.

## On the Left Side, Figure 1-1(b)

- Before the instant of rapid valve closure, fluid bulk velocity = constant **Uo** throughout the length of the pipeline.
- During the brief time of valve closing, the bulk water flow from the pump to the left side of the valve decelerates from U = Uo to a full stop, U = 0. Because it has linear "+Z momentum," it tries to pile up on the left face of the valve disc.
- This destroys the "+Z" momentum, and results in microscopic compression of a cross-section of the liquid into a very thin layer on the disc's left face. This is a highly unstable transient condition. Nature deplores it.

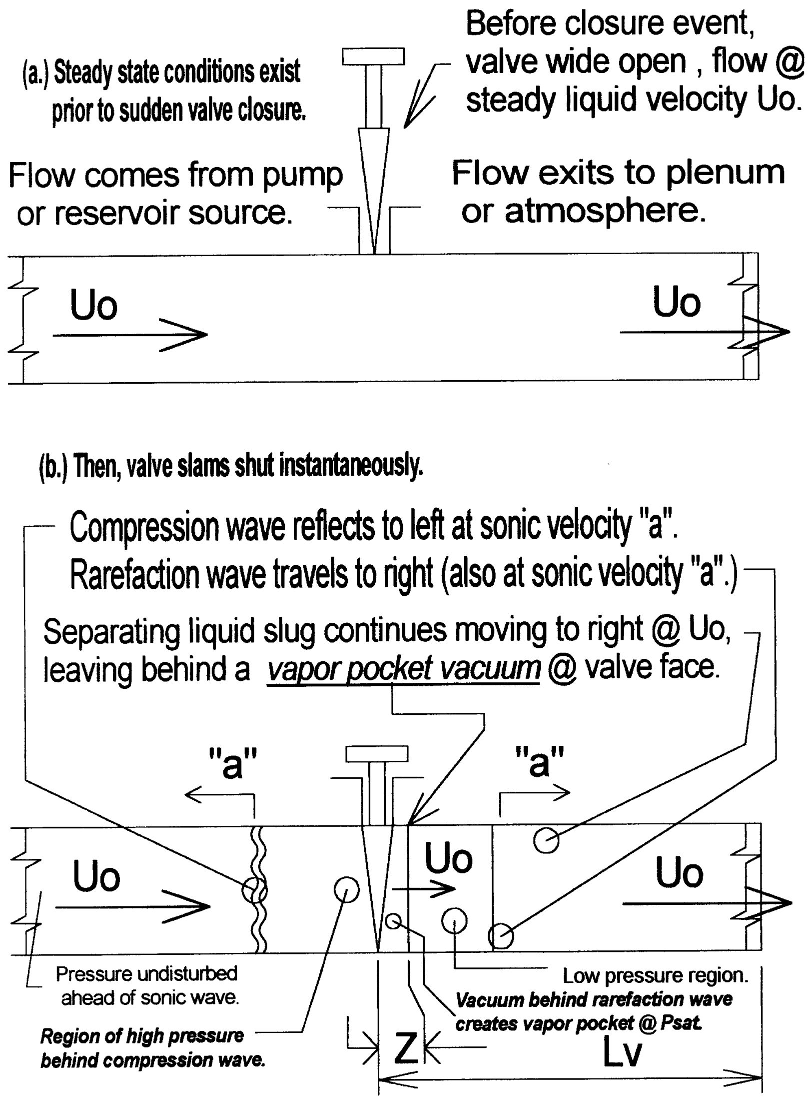



**FIGURE 1-1 RAPID VALVE CLOSURE CAUSING WATER HAMMER**

- As momentum is destroyed, the thin layer of compressed liquid builds up, gaining energy of compression. Its static pressure rises very quickly and a "compression wave" rebounds leftward, traveling at the liquid's prevailing sonic velocity, which we designate = **"a".**
- A region of peak overpressure is reached in that thin compression layer on the valve's left face, and that high-pressure region propagates leftward with the sonic wavefront. The wave travels over the entire length of the left-hand side of the system and finally hits the pump, (In a full-flowing gravity pipe system, it would hit the pipe entrance at the reservoir; same result either way.)
- At this freeze-frame point in time, *for design purposes,* we can consider the whole left-hand volume of piping to be at the same uniform peak pressure. The numerical value of the peak pressure is the sum of pressure rise **ΔP** due to sonic wave passage **plus** the **ambient source pressure** (pump shutoff head, if it is a pumped system, or the depth of gravity pipe entrance submergence beneath the reservoir surface, in feet of liquid converted to psig units).
- *Now the reversal begins.* The microscopically compressed cylinder of liquid expands, trying to eject a little bit of fluid leftward out of the pipe end. This creates a region of reduced pressure at the left end, initiating a rarefaction wave in the opposite direction.
- The rarefaction wave propagates rightward, also at sonic velocity **"a",** relaxing the pressure behind it, finally striking the left face of the valve disc. At this point in time, *for design purposes,*

we can consider the fluid to be at rest and relaxed at the static pressure necessary for static equilibrium to exist henceforth.

- *This description of events was purposefully simplified. Actually several cycles of sonic wave action and secondary reflections will follow passage of the original sonic wave couplet, with diminishing pressure perturbations on each cycle, until all the fluid's original kinetic energy (which it had at "Uo"), has been dissipated into frictional heat in the fluid,* plus work done in permanently deforming the pipe metal, plus mechanical friction resulting from pipe vibratory motion against it restraints, plus energy carried away in sound waves outside the pipe, etc.

And that is all there is to the "upstream" action. A pressure pulse or two travels up and back, and that's it. If it contains enough energy, maybe something gets wrecked.

Now let's put some realistic numbers with our verbal example, and see how we might go about an actual evaluation. In fact, let's do the same problem with two proportional piping configurations, to see how actual scale affects the situation.

## Example Problem 1-1 Water Hammer Compression Wave Travels Upstream from Site of a Rapid Valve Closure

- Given:

Physical layout per Figure 1-1 (check two different pipe diameters of equal length/inside diameter ratios of 1,000:1, and let both pipe sizes be U.S. nominal and have standard wall thickness:)

**L ≡ developed length, pipe left end at pump conn. to valve face in Fig. 1-1(b);**
**D ≡ inside diameter of pipe;**
**L/D ≡ 1,000;**
**pipe #1,** 1-in. size: $\mathbf{D_1}$ = 1.049 in. $\mathbf{L_1}$ = 1.049 × 1000/12 = **87.42 ft;**
**pipe #2,** 12-in. size: $\mathbf{D_2}$ = 12.00 inches, $\mathbf{L_2}$ = 12.00 × 1,000/12 **= 1,000 ft;**

Both pipes to be considered rigid, to avoid having to consider the minor effects of pipe wall flexure.

Let the liquid flowing be fresh water. Its density ρ **= 62.4 lbm/ft$^3$** @ ambient temp., and sonic velocity **"a"=4,800 ft/sec.** Let it be flowing at a steady bulk velocity which is approximately the design-max for each of the given pipe sizes:

**pipe #1,** 1-in. size: $\mathbf{U_1}$ = 3.7 ft/sec, $\mathbf{Q_1}$ ≌ 10 gal/min (gpm);
**pipe #2,** 12-in. size: $\mathbf{U_2}$ = 7.0 ft/sec, $\mathbf{Q_2}$ ≌ 2,500 gpm;
Centrifugal pump causing the flow. Pump has a **shutoff head** (total dynamic head, or TDH, at zero gpm flow) = 80 ft water;
Valve slams shut rapidly enough to cause water hammer. We will calculate the applicable range of closure times after the fact.

## Define Terms and Units:

**Pmax** peak pressure, water hammer plus source pressure, psig
**Pr** Code-rated working pressure, psig
**Pv** vapor pressure, psig
**m** mass of water, lbm
**A** pipe cross-sectional flow area, in.$^2$
$\mathbf{g_c}$ univ. grav. const. **32.2** (lbm ft)/(lbf/sec$^2$)
**L** developed length of piping, ft
**D** pipe inside diameter (ID), ft
**Lc** length of moving water slug, ft
**Uo** initial bulk velocity, ft/sec
**Q** volumetric flow in pipe, gal/min
*ρ* water density @ 60°F., **62.4** lbm/ft$^3$
**a** sonic velocity in liquid, ft/sec
**ΔP** pressure rise @ sonic wave passage, psi
**S** code-allowable tensile stress of pipe material, psi
**E** modulus of elasticity of pipe material, psi
**Y** dimensionless factor, code stress equation
**tmin** minimum allowable pipe wall thickness, code stress equation, in.
**tp** sonic wave propagation time, sec
**td** disturbance (valve closure) time, sec
**Z** axial direction, pipe centerline

## Calculations:

- The idealized maximum pressure rise **ΔP** due to sonic wave passage through liquids in rigid pipes is the worst case for conservative calculation purposes, and it is found from **Joukowsky's equation** in its simplest form:

$\Delta P = (\rho)(a)(Uo - 0)/g_c$;
$\Delta P = (62.4)(4{,}800)(Uo)/(32.2)(144) =$
$= 64.6 \times Uo$ **(psi,** or lbf/in.$^2$);
**Pipe #1, Uo = $U_1$** = 3.7 ft/sec, and **ΔP @ 1 in. pipe** = (64.6)(3.7) = 239 psi
**Pipe #2, Uo = $U_2$** = 7.0 ft/sec, and **ΔP @ 12 in. pipe** = (64.6)(7.0) = 452 psi

- The peak pressure **Pmax** acting internally on the piping system is the sum of the water hammer transient pressure rise **ΔP** and the **source pressure** at the entry (leftmost) end of the pipe, which in our example is the **pump shutoff head:**

**Source pressure** = 80 ft × 0.433 psi/ft water = **34.6 psi** on top of the pump's own inlet pressure; for simplicity's sake, let's use one atmosphere absolute, 0 psig, for pump's own inlet pressure; so,
**Source pressure = 34.6 psig, and:**
**Pipe #1, Uo = $U_1$** = 3.7 ft/sec, ΔP @ **1 in. pipe = (64.6)(3.7) = 239 psi, Pmax** = 239 + 34.6 = 274 psig
**Pipe #2, Uo = $U_2$** = 7.0 ft/sec, ΔP @ **12 in. pipe = (64.6)(7.0) = 452 psi, Pmax** = 452 + 34.6 = 487 psig

- Next we compare these transient maximum pressures to the allowable working pressures for the piping and its joining system's pressure rating, and make an evaluation. Just for illustration, using ASTM A-105 forged steel flanges per ANSI B16.5, and A-53B/A-106B carbon steel piping (seamless) in perfect new condition.

**Per the ASME B31.3 code we would find the allowable working pressure For ASTM Type A-105 forged steel flanges:**

**rated working pressure = (Pr)($\mathbf{S_1}$)**/8,750
where Pr = 115 for Class 150, and

$S_1$ = 60% of min. yield strength at temp. = 0.60 × 36,000 psi = 21,600 psi;
rated Working **P** = (115)(21600)/8,750 = **= <u>284 psig</u>** for any flange pipe size.

**<u>For A-53B/A-106B carbon steel piping (seamless) in perfect new condition:</u>** the B31.3 rated working pressure equation regardless of grade of pipe material is:

**P = (2)(SE + PY)(tmin)/outside diameter (OD)**
which reduces to explicit form
**P = (2tminSE)/(OD – 2Ytmin)**
S = code-allowable tensile stress at temp. =
= 20,000 psi at ambient temp.;
E = seamless pipe quality factor = 1.00;
Y is 0.4 for carbon steel, normal temps;
tmin = wall thickness for new pipe considering mill undertolerance of 12.5% as standard = (0.875)/[(OD – ID)/(2)] =
= (0.4375)(OD – ID);
1 in. Sch.40, OD – ID = 1.315–1.049 = 0.266 in. and **tmin** = 0.4375 × 0.266 = 0.116 in.
12 in. std, OD – ID = 12.75 – 12.00 = 0.75 in. and **tmin** = 0.4375 × 0.75 = 0.328 in.
So, **P** for 1 in. sch 40 pipe =
= (2)(0.116)(20,000)(1.0)/[1.315 – (2)(0.4)(0.116)] **=**
**= <u>3,796 psig.</u>**
And **P** for 12 in. standard wall pipe =
= (2)(0.328) (20,000)(1.0)/[12.75 – (2)(0.4)(0.328)] **=**
**= <u>372 psig.</u>**

## Summary

| **Pipe Size in.** | **P Allowed @Ambient Temp. Sch.40 pipe** | **150# Flange** |
|---|---|---|
| **1 in.** | **3,796 psig** | **284 psig** |
| **12 in.** | **372 psig** | **284 psig** |

***<u>versus the water hammer pressure spikes:</u>***
***Transient Pmax in 1 in. pipe = 274 psig,***
***Transient Pmax in 12 in. pipe = 487 psig***

We would expect no problem with the 1 in. pipe, certainly, or with the 1 in. –150# flange if its bolts were properly torqued. You should recheck the max pressure ratings of any valves or in-line devices such as flowmeters or filter housings, since we are close to the flange pressure limit. If the valve body or in-line device max pressure rating equals or betters Class 150 equivalence, it should be okay.

The 12 in. pipe may be different. It can sustain 372 psig steady internal pressure by Code, but 487 psig impulses will cause hoop stresses that exceed Code allowable by (487 – 372) ÷ (372) = 0.31, that is, by 31%. Would that be enough to break the pipe? Absolutely not, not by itself alone; the hoop stress would still be well below yield, and carbon steel is tough as hell. But if it happened a lot, it would hasten the eventual fatigue failure of the system and by ASME Code would have to be a design factor. Additional longitudinal pipe stress (from bending moments) would exacerbate the problem. All in all, the wise engineer would design the system to prevent the water hammer altogether.

The 12 in. 150# flange will sustain 284 psig at ambient temperature; should we worry about a water hammer shock load of 487 psig? The flange might spring a leak, which is not good. You should certainly check tensile stresses in the flange bolts for overstress due to the shock (impulse) load.

Another real danger would be in causing pipe motions resulting in bending loads at the pipe restraints. As long as any <u>bending stresses in the pipe at the flange connection point</u> which may accompany the water hammer impulse are held below about 6,000 psi, however, there probably would not be a leakage problem even in a flange gasket lacking self-sealing characteristics (ref. *<u>ASME B31.3 Process Piping,</u>* PDP Notes, 1997, courtesy of Glynn E. Woods, P.E., who serves on the B31 Technical Committee.)

(*Note:* a flexibility analysis is necessary to find the forces and moments causing bending stress in realistically complicated piping system geometries. Again, it is wise to avoid that water hammer through careful system design!)

Another consideration is the pipe thrust which water hammer generates; once that subject is broached, then the question of potential damage and available energy usually arises.

**<u>Thrust</u>** = (Pmax) × (Flow Area);
For <u>1 in. pipe</u> = 274 psig × 0.864 in.$^2$ = **<u>237 lbf</u>**
For <u>12 in. pipe</u> = 487 psig × 113.1 in.$^2$ = **<u>55,080 lbf</u>**

**<u>Kinetic energy (KE)</u>** to dissipate = $\mathbf{mU_o}^2/2g_c$;
For 1 in. pipe, m = 87.42 ft. pipe × 0.374 lbm water per ft. pipe = 32.7 lbm water;

**KE <u>1 in. pipe</u>** = (32.7 lbm)(3.7 ft/sec)$^2$/64.4 =
= **<u>7 ft-lbf</u>** energy ***(no problem)***

For 12 in. pipe, m = 1000 ft pipe × 49.0 lbm water per ft pipe = 49,000 lbm water;

**KE** <u>12 in. pipe</u> = (49,000)(7.0 ft/sec)$^2$/64.4 =
= **<u>37,280 ft-lbf</u>** energy ***(dangerous!)***

**<u>Potential thrust work</u> ≃ Dissipated KE;**
Work = Force × Distance = Thrust × Travel;
Travel ≃ KE/thrust;
1 in. pipe travel ≃ (7 ft-lbf/237 lbf) × 12 in/ft ≃ **0.35 in.;**
12 in. pipe travel ≃ (37,280/55,080) × 12 in/ft ≃ **8.12 in.;**

**<u>Relative potential for doing damage:</u>** 1 in. pipe: abruptly lift 237 lb 0.35 in.; **<u>12 in. pipe:</u>** abruptly lift a load of 55,000 lb vertically more than 8 in.!

Thus water hammer in the 1 in. pipe might pose only a nuisance, but in the 12 in. pipeline poses real danger. We must <u>prevent</u> the water hammer, <u>so, we must design our controls to close that valve more slowly</u>!

How slowly? The engineering profession has developed a standard of practice which tells us, and it is beautifully simple. All you do is:

1. Calculate **"Lc",** the length of moving water cylinder whose motion will be stopped abruptly. This is the water slug whose linear momentum and kinetic energy must be dissipated in some way (in our example, it is the developed pipe length from pump discharge to valve. The "developed length" is simply the sum of individual straight lengths in the pipeline. Use the bend radius centerline length for each elbow, and add to the total).
2. Obtain the applicable sonic velocity for the fluid. (**Fresh water's** sonic velocity in rigid pipe is about **4,860 ft/sec.** If you go for conservativism combined with ease of mental calculation, use **4,000 ft/sec** for **"a"** in calculating the sound wave travel time, but use ***a = 5000 ft/sec*** to calculate the value of ***Pmax.***)
3. Calculate the time interval required for the sonic wave to propagate up the pipeline and back; that is, to travel **twice** the

distance **Lc,** the first leg of the trip as a pressure wave and the reverse return trip as a rarefaction wave, i.e., to go **(2 × Lc)** feet in total. Call this time **"tp", the sonic wave propagation time.**

4. To avoid initiating a water hammer event, we must spread out the time interval over which the disturbance event takes place (the valve disc travel time from full open to full closed, in our current example) to make it considerably greater than the wave propagation time. That allows the fluid enough time to make the change in bulk total pressure gradually and smoothly, and therefore in accordance with the normal rules of bulk flow, rather than violently per Joukowsky's equation.

**In engineering practice, the conventional approach is to make the disturbance (valve-in-motion) time "td" *a full order of magnitude*** greater than the sonic wave propagation time "tp".

**So the rule is:**

**By design, make (td) ≧ (10)(tp).***

Now we will see how this applies to our example. Apply this rule to the 1-in. pipeline by following the four steps given above:

1. Lc = 87.42 ft
2. use "a" = 4000 ft/sec—conservative
3. wave propagation time tp = 
   = (2)(87.42)/4000 = 0.044 sec
4. so make the valve closure time td
   ≧ (10)(0.044) ≧ **0.44 sec***

Now the 12-in. line:

1. Lc = 1,000 ft
2. use "a" = 4000 ft/sec—conservative
3. tp = (2)(1,000)/4,000 = 0.50 sec
4. so make td ≧ (10)(.5) ≧ **5.00 sec***

*For globe valves. For gate valve use 100 times tp instead of 10 times tp.

*Note: it turns out that we have solved the general engineering problem of water hammer due to rapid valve closure by examining the left–hand side of the piping system only. One may wish to skip over the right-side analysis. However, it is quite instructive and interesting as well, and is given next for those wishing more understanding of the physics of this transient phenomenon.*

## On the right side of the valve, following closure, Fig. 1-1(b):

- Momentum carries the slug of water onward, away from the valve, toward the open discharge end of the pipeline.
- The closed valve prevents backfilling, and a partial vacuum forms as the tail end of the water slug separates from the face of the valve disc. Simultaneously a rarefaction wave is initiated, traveling rightward at sonic velocity **"a".** This all happens very quickly. (The rarefaction wave of course moves much, much faster at all times than the liquid slug; roughly 1,000 times faster.)
- Since nature will not tolerate the vacuum, it causes liquid to flash into vapor and fill the rapidly expanding void behind the rightward-moving slug of liquid. The vapor in this example is saturated steam, at the partial pressure corresponding to the existing saturated temperature of the liquid water, and since no air or other gas is present, a very low absolute vapor pressure is reached indeed.
- The rarefaction wave reaches the atmosphere and disappears. And just as abruptly, a compression wave moving leftward from the open pipe end at sonic velocity takes its place. (This is a reverse mirror image of what happened on the left-hand side of the valve.) Atmospheric pressure replaces the partial vaccum to the right of the compression wave as it races through the slug, which continues in the original left-to-right trajectory. The wave races toward the closed valve and pocket of water vapor still trapped there.
- The resulting pressure differential acts on the slug of moving water, from high pressure (the atmosphere) on the right, toward the vaporous vacuum on the left. This is a pretty strong differential, on the order of 14.5 psi.
- According to the regular laws of mechanics, this pressure differential acts to retard the slug from its original rightward velocity **(Uo),** finally stopping it altogether then reversing its direction, accelerating the water slug leftward, back toward the valve.
- When the returning wave hits the vapor pocket, the pocket collapses violently back to liquid phase. This is a sonic-velocity event, complete with peak pressure rise which occurs as the rebounding slug of water crashes into the closed valve. As was true with events on the left-hand side of the shut valve, several additional cycles of diminishing wave couplets could be expected to occur before equilibrium is reached in the right-hand side, but only if the free end is submerged in liquid. If the pipe's free end is in the atmosphere, of course, it will drain itself.

Now, what are corresponding values of **ΔP** and **Pmax?** Same as on the left-hand side? Higher? Lower? What? If the answer is immediately obvious to you, then congratulations! Most of us are less fortunate, and have to grind out the answer the hard way. So, here goes.

Incidentally, I am using basically the same analytical procedure that Dr. Frederick J. Moody has given us in his outstanding textbook Introduction to Unsteady Thermofluid Mechanics, John Wiley & Sons, copyright 1990, and in the notes accompanying his truly excellent ASME Professional Development Program Course, How to Predict Thermal-Hydraulic Loads on Pressure Vessels and Piping. I had the distinct privilege of attending that course under Dr. Moody, and I recommend it to you as the best of the best available. Fred did his work in boiling water reactor engineering and containment over a 40-year span with General Electric while serving as adjunct professor in the Mechanical Engineering Department at San Jose State University and as Advanced Engineering Program instructor for 35 of those years. He also served as chairman and co-chairman of ASME's Fluids Engineering and Pressure Vessel & Piping Divisions. In 1980 he received the ASME George Westinghouse Gold Medal Award for his contributions to two-phase flow and reactor accident analysis. His textbook is actually a working engineer's technical manual. It is certainly monumental in scope, and absolutely biblical in value.

Now, on with the mathematical analysis of the transient events on the right-hand side of the valve, after it has slammed shut.

## Define Additional Terms and Units:

$\mathbf{F_z}$ external force along Z-axis, lbf
$\mathbf{V_z}$ bulk slug velocity along Z-axis, ft/sec
**t** time, sec
$\mathbf{t_{end}}$ time of vapor pocket collapse, sec
$\mathbf{P_{in}}$ pressure at pipe discharge, lbf/ft$^2$
**Pv** vapor pressure at liquid temperature lbf/ft$^2$

**A** pipe cross-sectional flow area, $ft^2$
**$g_c$** univ.grav.const., 32.2 (lbm ft)/(lbf $sec^2$)
**Lv** Figure 1-1(b) length of water slug, ft
**Δt** time interval for slug to halt, sec
**tp** sonic wave propagation time, sec
**td** disturbance (valve closure) time, sec
ρ water density @ 60°F, **62.4** lbm/$ft^3$
**ΔP** pressure rise @ vapor collapse, psi
**Pmax** peak pressure, water hammer plus source pressure, psig

## Analysis

We ignore pipe wall friction effects during the transient, since passage of the rarefaction wave destroys the rightward-moving water slug's organized total pressure gradient, which must exist in the direction of frictional flow in pipes. In a sense, during the transient, the pipe is not "flowing full"; pressure falls rapidly towards saturated steam pressure at the liquid's temperature everywhere, and the resulting fluid motion is ballistic.

First we must calculate Δt, the time interval required for the slug to come to a full halt. Vector sign convention shall be:

← **(+Z)**

1. $\Sigma F_z = (m/g_c)(dV_z/dt)$
2. $dV_z/dt \cong \Delta V_z/\Delta t$
3. $\Sigma F_z = (P_\infty - Pv)(A)$
4. $\Delta V_z = (Uo - 0) = Uo$
5. $m = \rho A\ Lv$; substituting,
6. $\Delta t = (\rho Uo\ Lv)/(P_\infty - Pv)(g_c)$

The fact to note at this point is that Δt, the time interval between $\mathbf{V_z}$=(–)Uo and $\mathbf{V_z}$=0, is the disturbance time **td;** that is,

7. $\Delta t \equiv td$

Next, compare the disturbance time to **tp** the sonic wave propagation time, to find out if we can use normal bulk flow equations to find the peak pressure rise.

8. If bulk flow, then **(td)** ≧ **(10)(tp)**
9. **tp** = (2)(Lv)/a
10. **td** = $(\rho Uo\ Lv)/(P_\infty - Pv)(g_c)$ from step 6 & 7;
11. So, is $(\rho Uo\ Lv)/(P_\infty - Pv)(g_c) \geqq (10)\times(2)(Lv)/a$ ???
12. multiply (11), by {a/2Lv} and get

$$(\rho)\ \mathbf{(Uo)(a)/(P_\infty}-Pv)(2)(g_c) \geqq \mathbf{10\ ???}$$

13. Substituting conservative approximate parametric values for the dangerous 12-in. pipe example into this expression yields:

$$\mathbf{(62.4)(7.0)(4{,}000)/(14.5 \times 144)(2)(32.2) =}$$
$$= 13.0 \geqq 10$$

So the 12-in. pipe transient results in a slug flow reversal on the right-hand side of the system, which follows the good old familiar "bulk flow" equations. *(Good*! That gives us a slim prayer of being able to complete the analysis. Otherwise, we've got the general form of the Navier-Stokes equation to deal with, and you would definitely be on your own!)

14. Rewrite (1) in differential form, then integrate:

$$(g_c\ \Sigma F_z/m) \int (dt) = \int (dV_z)^*$$

*Note that (dt) is integrated from zero to "t," and ($dV_z$) from $U_o$ to "$V_z$."

15. $(Pv - P_\infty)(\mathbf{A}g_c)(t - 0)/m = Vz - Uo$
16. Using results of (5) obtain

$$(Pv - P_\infty)(\mathbf{A}g_c)(t - 0)/(\rho A\ Lv) = Vz - Uo$$

17. Simplifying,

$$Vz = Uo + [(g_c)\ (Pv - P_\infty)/(\rho Lv)] \times [t]$$

**Note that the term (Pv – $P_\infty$)** is negative.

18. Since $V_z = dZ/dt$,

$\mathbf{Z} = \int_0^Z (dZ) = \int_0^t V_z(dt) =$

$= \int_0^t [Uo + (g_c)\ (Pv - P_\infty)(t)/(\rho Lv)](dt) =$

$\mathbf{= (Uo)(t) + (g_c)\ (Pv - P_\infty)(t^2)/(2\rho Lv)}$

"Z" will be zero when the slug crashes back into the valve disc. We need to find an expression for the corresponding time "$\mathbf{t_{end}}$". Two solutions of (18) are possible when we set Z ≡ 0. One solution is the trivial one, t = 0. The other solution is:

19. $0 = (Uo)(t) + (g_c)\ (Pv - P_\infty)(t^2)/(2\rho Lv)$

Divide by t: {t ≠ 0}
20. $0 = (Uo) + (g_c)\ (Pv - P_\infty)(t)/(2\rho Lv)$

Solve for $\mathbf{t_{end}}$:
21. $\mathbf{t_{end}} = (-2\rho LvUo)/(g_c)(Pv - P_\infty)$

Substitute this expression for "$\mathbf{t_{end}}$" back into 17 and solve for corresponding terminal velocity at the time of crash, Vz:

22. $Vz = Uo + [(g_c)\ (Pv - P_\infty)/(\rho Lv)] \times [t] =$
$Uo + g_c(Pv - P_\infty)/(\rho Lv)][(-2\rho LvUo)/g_c(Pv - P_\infty) =$
$(Uo - 2Uo) =$ **–Uo**

Well! After all that effort, it turns out that the water slug has initial velocity Uo at time t = 0 when the valve slams shut, and after all that monkey-motion and calculus and stuff we find that it slams back into the disc at exactly the same speed and opposite direction.

Now we can find the pressure rise and maximum pressure on the system's right side. The pressure rise is calculated from the same equation as applied to the left side of the system, namely **Joukowsky's equation** in its simplest form:

$\mathbf{\Delta P = (\rho)\ (a)\ (Vz - 0)/g_c} = (\rho)(a)(-Uo)/g_c$ which is **identical** to in magnitude, but opposite in direction as we would expect, to the pressure rise on the left side of the valve, which we found was **452 psi.** Pretty neat of nature to work it out that way, huh?!

But the final result, **Pmax,** is not the same as on the left side. Remember, we have to add the source pressure to **ΔP** to obtain Pmax, and on the right-hand side of the pipeline, the source pressure is only 1 measly atmosphere, zero psig.

Therefore on the right side,

$$\mathbf{\Delta P = Pmax = 452\ psig.}$$

One last comment about "plain vanilla water hammer" is in order, and it is quite important:

The Joukowsky equation term for pressure rise $\Delta P = (\rho)(a)(Vz - 0)/g_c$ applies to impact of a sonic wave and moving liquid slug into a rigid, completely incompressible and unmovable object. The motionless rigidity of the target assures that no collision energy is removed from the impact zone by the target, thus leaving the fluid mass to absorb it all. However, another common event results in impact with a stationary mass of contained liquid, which compresses slightly upon impact. It turns out that in this case, the pressure rise is exactly one-half of the "full Joukowsky" i.e., $\Delta P = (\rho)(a)(Vz - 0)/(2\ g_c)$. An example of this type is "liquid column separation" in which only gravity and ambient pressure are available as motivational agents for crashing two liquid slugs into each other.

This leads us into the important category of "liquid column separation" in general. I think it best learned by example. However, I have devised a couple of "teaching" examples in which the ΔP results are not "one-half of the full Joukowsky" but are actually worse than "the full Joukowsky." As you will see, it is the additional motivational energy input to the water slug by a pump, added to the usual pressure differential due to gravity + atmospheric pressure alone, that makes the vapor pocket collapse even more violently in such cases.

## Example Problem 1-2

### *Severe Water Hammer*

### *Pump Stop, Liquid Column Separation and Collapse of Vapor Pocket in Piping Upon Restarting Pump*

### *Refer to Fig. 1-2. "Case: Pump's Check Valve Fails"*

### Given:

Physical layout per Figure 1-2.

Note: the vertical pipe loop rising some 60 ft above grade was built to give clearance over an existing obstruction; no suitable alternative to the loop was available.

Pipe size = **8-in. Schedule 40**

Pipe ID = 7.981 in.

Cross-section flow area **"A"** = 50.0 sq in.

Water weight = **21.69** lb/ft. of pipe

Ambient temperature = 60°F

Sonic vel. **"a"** in water = **4,860** ft/sec

Existing pump's shutoff head, i.e., the max. head @ zero flowrate, = **80** ft of water.

Stipulate that in this incident, the check valve was kept lodged open by debris at all times. (*In the next example, we will drop this stipulation and find out what effect the check value has on the system, if any.)*

### Problem:

The pump was run only twice after the system was constructed. The first time was seemingly without incident, and the pump was run until the tank filled with water.

Some water was drained from the tank, as part of a test of the drain valve's remote controls. It was decided to replace that quantity of water and leave the tank filled overnight.

So, about 10 minutes after first shutdown, the pump was restarted.

Immediately, a loud rumbling noise and tremendous bang were heard; the pipe jerked violently and the pump was wrecked as a result. The tank also suffered structural damage (bulged walls, deformed top head).

What happened, and why?

### Physical Analysis:

For convenience, define the water in the left-hand vertical pipe column (the riser which is **31 ft + 29 ft** long) as **"LHC",** & define water in the right-hand vertical column (**9 ft + 34 ft + 17 ft** long) as **"RHC".** The **"top of the pipe loop"** is the **21 ft.** horizontal run of pipe connecting the two risers. We will call the remainder of piping (horizontal, **300 ft.** + tank inlet) the **"main run."**

- When the pump started for the first time, the piping & tank were dry, filled with air.
- Because the piping had no hi-point vent, water flow from the pump eventually purged the whole volume of air out thru the tank's vent. During purge, the pump had to supply a static lift of **63 ft.**
- Once all air was purged from the **top of pipe loop, RHC, and right-hand section of main run,** the piping "ran full" and a stable hydraulic siphon was established. This reduced static lift to only **3–12 ft.** and pump flowrate increased greatly.
- Upon pump shutdown, momentum plus gravity kept the water flowing downward in the **RHC.** But gravity worked against upward flow in the **LHC** and slowed it down.
- As a result, the once-continuous stream was split in two, separated by a void, nearly a perfect vacuum, as shown in the figure. Note that when the split streams came to rest, the "full vacuum" kept the tops of both vertical water columns held at a height ***34 ft above the free surfaces of the pools at each end of the pipeline, which served as liquid vacuum seals.*** The left-hand pool is the open reservoir, in which the pump intake is deeply submerged. The right-hand pool is the storage tank, and the discharge nozzle (tank inlet) is deeply submerged. Submergence maintained the pocket of vacuum as a lasting, stable condition after pump shutdown.
- Actually, it was not a *complete* vacuum. As pressure decreased in the pocket of empty space, water from both columns flashed into vapor, thus filling the pocket with steam saturated at the ambient temperature, which at the time was 60°F (saturation pressure @ 60°F is found in any set of Standard Steam Tables, not included here, to have a value of only 0.25 psia, and that was the actual vacuum pressure.)
- Upon pump re-start, the left hand column **(LHC)** water began rushing upward, being pushed by the spinning impellers. But the right hand column **(RHC)** water remained static, there being nothing to push on it *(yet)*. The net effect was an instant of rising vapor pressure as the lengthening **LHC** acted on the cylinder of trapped vapor like a piston.
- When that happened, the ***"rejoining"*** event resulted, a consequence of the compressing action of the **LHC:**

The vapor condensed *extremely* quickly, at sonic velocity. The pocket literally *imploded* upon itself, and the rapidly accelerating **LHC** crashed into the stationary **RHC** with a *lot* of forward momentum. The impact of water columns created a pair of shock waves moving away from each of other at sonic velocity toward both ends of the pipeline. The energy associated with those shocks manifested as pressure impulses were responsible for wrecking the pump and storage tank.

The ultimate cause of the incident was engineering design error: failure to install a vacuum-breaking vent at the high point of the

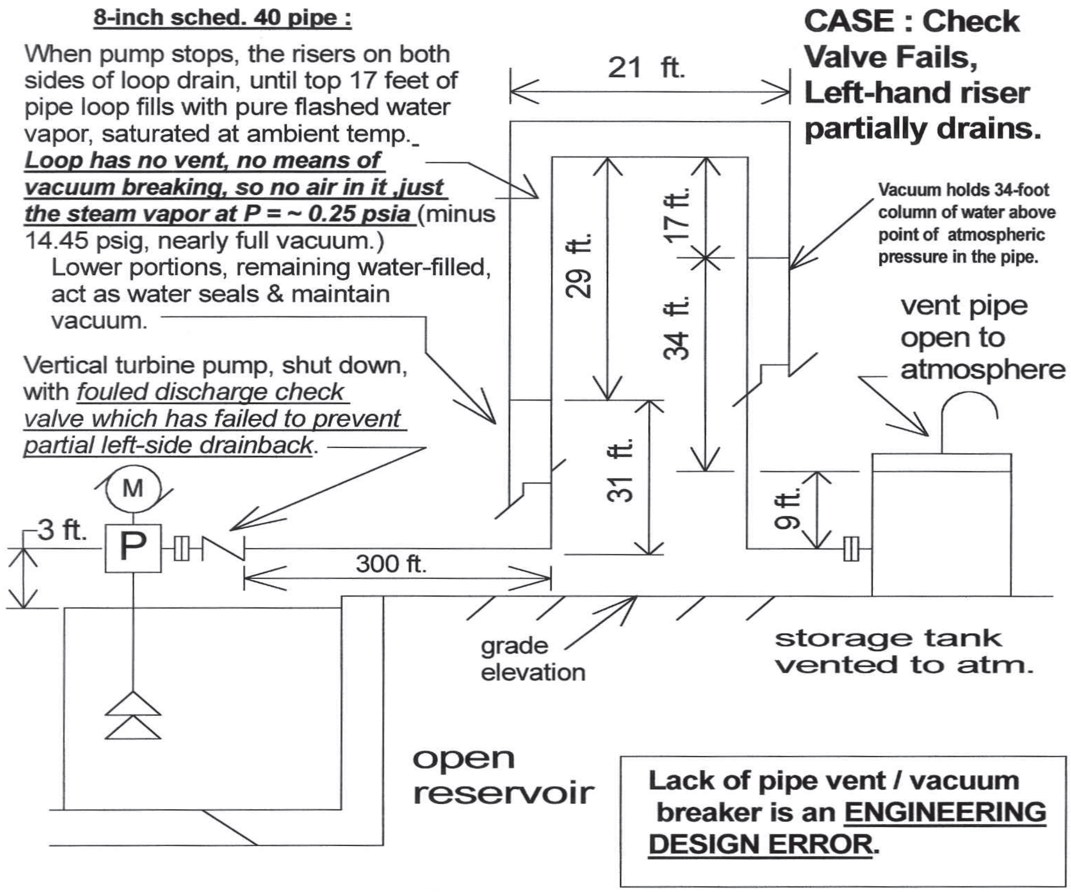



**FIGURE 1-2 VAPOR COLLAPSE: SEVERE WATER HAMMER. LIQUID COLUMN SEPARATION AND REJOINING WHEN PUMP CYCLES "OFF", THEN "ON" AGAIN WITHOUT BREAKING VACUUM WHICH FORMS AT TOP OF PIPELINE LOOP.**

pipe loop. This allowed the vacuum to form in the system when the pump was shut down. The vacuum was beneficial while the pump was running, because the siphon effect which created it made a huge reduction in total dynamic head of the system, hence reduced pump brake horsepower proportionately. But when the pump is shut down, the **top of the pipe loop** must be vented to the atmosphere, preventing vacuum formation.

## Quantitative Calculations:

## Define Terms and Units:

**vapor psig** in pocket before collapse, lbf/in$^2$
**pump psig** disch. @ shutoff head, lbf/in$^2$
**dP** felt by **LHC** @ pump restart, lbf/ft$^2$
**m** mass of **LHC** water, lbm
**A** pipe cross-sect. flow area, in$^2$
**F** motive force on **LHC** @ restart, lbf
**acc** acceleration due to **F,** ft/sec$^2$
$\mathbf{g_c}$ univ. grav.const. **32.2** (lbm ft)/(lbf/sec$^2$)
$\mathbf{T_L}$ elapsed time of **LHC** travel thru **L,** sec
**L** length of stationary vapor pocket, ft
**u** velocity of **LHC** after $\mathbf{T_L}$ passed, ft/sec
**dv** rel. vel. @ impact, **LHC** into **RHC,** ft/sec
**dP** press. rise @ impact shock wave, lbf/ft$^2$
$\rho$ water density @ 60°F., **62.4** lbm/ft$^3$
**a** sonic velocity, 60° water, **4,860** ft/sec
**ΔP** same as dP converted to psi, lbf/in$^2$

## Calculations:

1. Sat. vapor press. @ 60°F = **0.25** psia; **vapor psig** = 0.25 – 14.7 = **(–) 14.45 psig**
2. Pump shutoff head at restart = **80** ft. $H_2O$; **pump psig = (80)** (0.4331 psi/ft) = **34.65 psig**
3. Motive press. diff. **Δp** @ restart, lbf/in$^2$ =
   **= (pump – vapor) psig = 34.65 – (–) 14.45 = 49.1 psi**
4. **m** = [ developed length of **LHC** in feet × water weight per foot of pipe, lb/ft] =
   = **(300 + 31)**ft × 21.69 lb/ft = **7,179 lbm**
5. Unbalanced force acting on LHC at restart
   **= F = Δp × A = (49.1)(50.0) = 2,455 lbf**
6. Acceleration of **LHC** toward **RHC** =
   **= acc = [F ×** $\mathbf{g_c}$**]** / m = (2455)(32.2)/7,179 =
   **= 11.01 ft/sec$^2$**
7. $\mathbf{T_L}$ = [2L/acc]$^{0.5}$; L=(29 +21 + 17)= 67 ft;
   $\mathbf{T_L}$ = [ (2)(67)/11.01]$^{0.5}$ = 3.49 sec
8. **u = acc** × $\mathbf{T_L}$ =(11.01)(3.49) = 38.43 ft/sec

9. Since **RHC** has **zero** velocity at rejoining,
   **dv = (u – zero) = (38.43 – 0) = 38.43 ft/sec**
10. Pressure spike due to impacting water columns (pressure rise across shock wave)
    = **dP** = [ $\rho \times$ **a** $\times$ **dv / 2g$_c$**] =
    = [(62.4)(4860)(38.43)/(2)(32.2)] =
    **= 180,970 lbf/ft$^2$** {note units!}
11. Pressure spike in customary psi units
    $\Delta$P = (**180970** lbf/ft$^2$/**144** sq in per sq ft)
    = **<u>1,257 psi</u>**
12. **Peak axial force on pump outlet nozzle** = (**pump psig** + $\Delta$**P**) $\times$ **flange wetted area**
    Upon investigation, the type of flange gasket used in the system had an ID of **9.375** in., so the wetted area was [$\pi \times$ (**9.375** in.)$^2$ / 4] = **69.0 sq.in.,** and **peak axial force = (34.65 + 1,257 psi)(69 in$^2$)** = **<u>89,124 lbf</u>**!!!

## Closing Comments on this Example:

**A.** The accident report noted *"the pipe jerked violently and the pump was wrecked as a result. The tank also suffered structural damage (bulged walls, deformed top head.")*

It is problematical whether the pump would have suffered structural damage, if only the piping had been rigidly anchored near the pump end of the pipeline. Possibly it could have withstood the pressure transient by itself, if a completely load-resistant pipe anchor, or thrust block, had been there to absorb the axial impulse thrust load. It is clear, however, that the ***<u>pipe supports</u>*** were inadequate to withstand that nearly **45-ton peak axial load.** Remember, it is ***<u>excessive</u>*** strain (stretching, bending, warping, deformation in shear, etc.) that breaks things. <u>Strain within the range of elastic behavior of the material is perfectly okay, and so is the stress that accompanies it.</u>

If we build our structures good and strong, they can absorb the pressure spikes of shock waves <u>many times</u> without failure, as well as terrific levels of sustained fluid pressure, if no accompanying motion or plastic deformation takes place.

In the example problem we just worked through, what actually killed the pump was the <u>huge bending moment</u> in the vertical plane, which was produced by the 45-ton force acting horizontally on the pump's discharge flange, using the length of pump from flange to lower baseplate mounting bolts as a moment arm. Let's say just for illustration that length was **3 ft:** the bending moment was then about **3 $\times$ 90,000 = ~ 270,000 ft-lbf!**

At the risk of overdoing this point, let's say further that we will use a piece of the same 8-in. schedule 40 ASTM A-106/A-53 Gr. B carbon steel pipe from our example pipeline as a restraining anchor to absorb that bending moment; would it do the job? Let's see:

Sect. Mod. **"S"** of 8-in. Schedule 40 = 16.81 in$^3$;
bending stress **"$\sigma$"** = **(M/S)** =
= (**270,000** ft-lbf)(**12** in./ft)/(**16.81** in$^3$) =
= **<u>192,742 psi</u>**; far in excess of the material's ultimate tensile strength, which is only **60,000 psi. Damage should be expected with numbers like these.**

This little exercise with bending moments indicates two further things:

1. to anchor the pump rigidly enough to resist a 45-ton peak thrust load would require a <u>really massive</u> block of concrete and steel for a thrust block, deserving engineering design attention;
2. the <u>pipeline itself</u> probably underwent plastic deformation at the points of maximum bending moment and stress concentration, and may not be fit for further loading in the future without repairs being made to the affected areas. A complete walkdown and close examination of the whole pipeline plus supports would be called for.

**B.** The storage tank is another story. A water tank, designed only for atmospheric pressure on top of its full depth of water, will not survive such a pressure pulse without damage. The rapid cycle from a **<u>1,257 psi</u>** positive pressure spike, when the shock wave enters the tank water, to a spike of **full internal tank vacuum,** when the rarefaction wave rebounds from the tank to the pipe water, will certainly do permanent damage to the tank.

It is not economical or sensible to design the tank, or the piping system for that matter, to withstand such conditions. Therefore, the mechanical design of this system ***<u>must</u>*** prevent the liquid column separation from ever happening in the first place. ***<u>A failure-proof high point vacuum breaker must be included as part of the illustrated system for proper design.</u>***

## Example Problem 1-3
## Severe Water Hammer

## Pump Stop, Liquid Column Separation, and Collapse of Vapor Pocket in Piping Upon Restarting Pump
## Refer to Fig. 1-3.
## "Case: Check Valve Works Okay"

## <u>Given:</u>

Physical layout per Figure 1-3.

Note: except for action of the check valve, this example is identical to Example Problem 1-2, as shown in Figure 1-2.

Pipe size = **8-in. Schedule 40**
Pipe ID = 7.981 in.
Cross-section flow area **"A"** = **50.0** in.$^2$
Water weight = **21.69** lb/ft. of pipe
Ambient temperature = 60°F
Sonic vel. **"a"** in water = **4,860** ft./sec
Existing pump's shutoff head, i.e., the max head @ zero flowrate, = **80** ft of water.
<u>Stipulate</u> that in this incident, the check valve worked as it was supposed to, closing when the pump stopped and thus preventing reverse flow. *(In Example 1-2, we stipulated check valve failure.)*

## <u>Quantitative Calculations:</u>

## <u>Define Terms and Units:</u>

**vapor psig** in pocket before collapse, lbf/in$^2$
**pump psig** disch. @ shutoff head, lbf/in$^2$
**dP** felt by **LHC** @ pump restart, lbf/in$^2$
**m** mass of **LHC** water, lbm
**A** pipe cross-sectional flow area, in.$^2$
**F** motive force on **LHC** @ restart, lbf
**acc** acceleration due to **F,** ft/sec$^2$

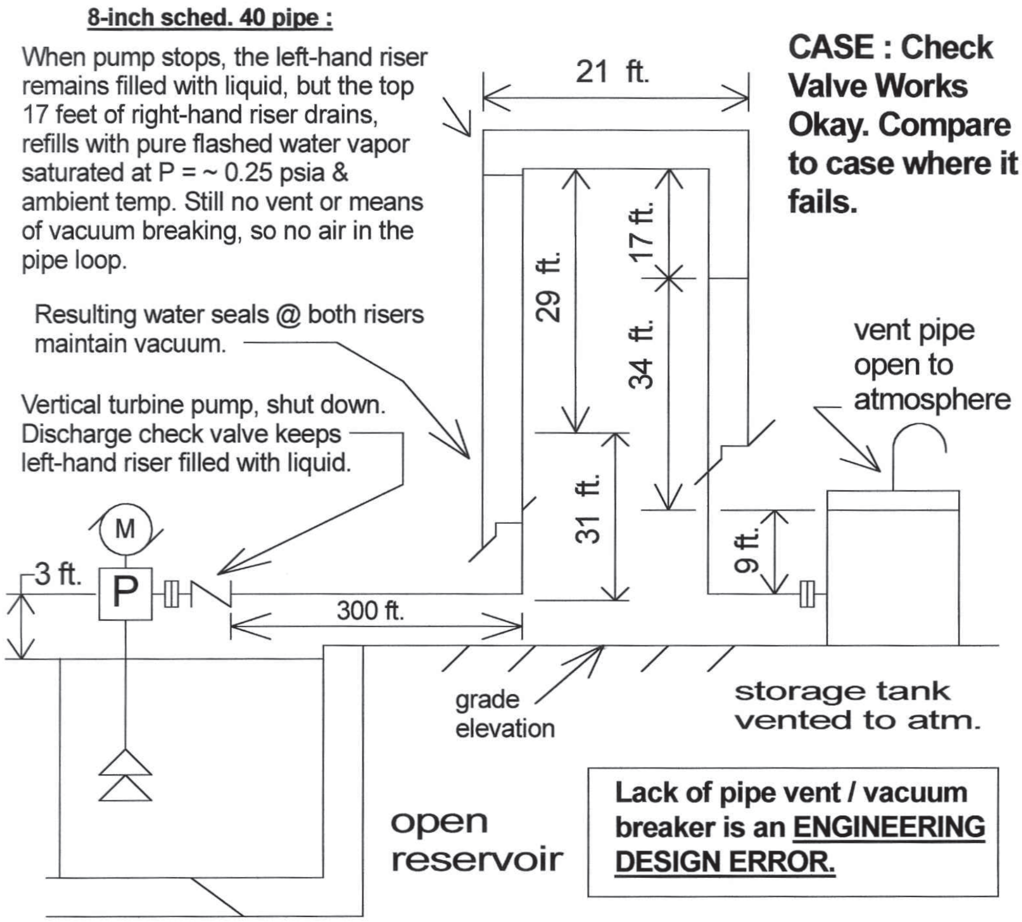



FIGURE 1-3 VAPOR COLLAPSE: SEVERE WATER HAMMER LIQUID COLUMN SEPARATION AND REJOINING WHEN PUMP CYCLES "OFF", THEN "ON" AGAIN <u>WITHOUT</u> BREAKING VACUUM WHICH FORMS AT TOP OF PIPELINE LOOP.

**$g_c$** univ.grav.const., **32.2** (lbm ft)/(lbf sec$^2$)
**$T_L$** elapsed time of **LHC** travel thru **L,** sec
**L** length of stationary vapor pocket, ft
**u** velocity of **LHC** after **$T_L$** passed, ft/sec
**dv** rel. vel. @ impact, **LHC** into **RHC,** ft/sec
**dP** press. rise @ impact shock wave, lbf/ft$^2$
$\rho$ water density @ 60 °F., **62.4** lbm/ft$^3$
**a** sonic velocity, 60 °F water, **4,860** ft/sec
**ΔP** same as dP converted to psi, lbf/in$^2$

## Calculations:

1. saturated vapor press. @ 60°F. = **0.25** psia; **vapor psig** = 0.25 – 14.7 = **(–) 14.45 psig**
2. pump shutoff head at restart = **80** ft $H_2O$; **pump psig** = **(80)**(.4331 psi/ft) = **34.65 psig**
3. motive press. diff. **dP** @ restart, lbf/in$^2$ = **= (pump – vapor) psig = 34.65 – (–) 14.45 = 49.1 psi**
4. **m** = [ developed length of **LHC** in feet × water weight per foot of pipe, lb/ft] = **(300 + 31 + 29)** ft × 21.69 lb/ft = **7,808 lbm**
5. unbalanced force acting on **LHC** at restart **= F = dp × A = (49.1)(50.0) = 2,455 lbf**
6. acceleration of **LHC** toward **RHC** = **= acc = [F × $g_c$]/m = (2455)(32.2)/7,808 = = 10.12 ft/sec$^2$**
7. **$T_L$** = [2**L/acc**]$^{0.5}$; **L = (21 + 17) = 38 ft;** **$T_L$** = [(2)**(38)/10.12]$^{0.5}$** = **2.74 seconds**
8. **u = acc × $T_L$** = (10.12)(2.74) = 27.73 ft/sec
9. since **RHC** has **zero** velocity at rejoining, **dv = (u – zero) = (27.73 – 0) = 27.73 ft/sec**
10. pressure spike due to impacting water columns (pressure rise across shock wave) **= dP = [ ρ × a × dv/2 $g_c$]** = = [(62.4)(4,860)**(27.73)**/(2)(32.2)] = **= 130,585 lbf/ft$^2$** {note units!}
11. pressure spike in customary psi units **ΔP** = (**130585** lbf/ft$^2$/**144** in.$^2$ per ft$^2$) = <u>**907 psi**</u> {*vs.* <u>***1,257 psi***</u> with no check valve.}
12. **peak axial force on pump outlet nozzle = (pump psig + ΔP) × flange wetted area;** Upon investigation, the type of flange gasket used in the system had an ID of **9.375** in., so the wetted area was **[π × (9.375 in.)$^2$/4]** = 69.0 in.$^2$, and **peak axial force = (34.65 + 907 psi)(69 in.$^2$)** = <u>***64,974 lbf***</u> (vs. <u>***89,124 lbf***</u> without check valve)

## Closing Comments on this Example:

In the first liquid column separation problem we worked (Example 1-2, above), the <u>huge bending moment</u> on the pump flange was produced by a **45-ton** force acting horizontally, using a

3-ft length from frange to lower baseplate mounting bolts as a moment arm.

But in this problem, proper action of the check valve reduced that force to *(just!)* **33 tons** of thrust. The bending moment was thus reduced to about **3 × 65,000 = ~ 195,000 ft-lbf,** less than the first example's 270,000, but *still* quite a healthy load!!

To continue our comparison with the previous case, let's use the same piece of 8-in. Schedule 40 carbon steel pipe as a restraining anchor to absorb that bending moment; would it do the job this time? Let's see:

Sect. Mod. **"S"** of 8-in. Schedule 40 = 16.81 in$^3$;
bending stress **"σ"** = **(M/S)** =
= (**195,000** ft-lbf)(**12** in./ft)/(**16.81** in$^3$) =
= ***139,203 psi*** vs. the **192,742 psi** in the previous example; *STILL* far in excess of the material's ultimate tensile strength, which is only **60,000 psi. Damage should be expected with either one of these numbers.**

This second example's exercise with bending moments shows that the check valve failure did not cause the problem, but did exacerbate it somewhat. The same final conclusions still apply, regardless of presence of the check valve.

## Steam Hammer

In the previous sections of this topic, we have gone in depth into "water hammer" transients of various types. We found that water hammer actually involves formation and rapid collapse of pockets or bubbles of flashed vapor within rigid enclosures. The motion induced in the liquid phase surrounding the "bubble collapse zone" is the phenomenon that catches our interest as "hammer," but is clearly just an "effect."

The "cause" of all the trouble is the condition, whatever it may be, that creates the local zone of low static pressure in the pipe, valve throat, pump intake, etc., that allows the liquid to flash into vapor in the first place. If we eliminate the phase change *"liquid-flashing-to-vapor,"* we eliminate the more dangerous aspects of waterhammer.

Rapid valve motions may still create higher transient pressure rises in our piping systems than we would wish for, but without the "flashing" and inevitable "crashing" which follows, the pressure rise is smaller in magnitude and much more "spread out" in time (less violent).

Before going further, it is important to note that the simple collapse of a tiny vapor bubble, all by itself, can be quite damaging under certain circumstances. The example to prove this assertion is ubiquitous: **cavitation.**

*Cavitation* is the name given to small, highly localized and semipermanent zones of low pressure in the flow, where the liquid flowing velocity has accelerated due to a sharp decrease in the cross-sectional area normal to the flow stream. The ***velocity pressure*** increases as velocity squared, and when it rises sharply in a piece of obstructed piping or in a valve throat restriction, the ***static pressure*** must fall proportionately, since their sum (i.e., the ***total pressure***) cannot rise in the process . No energy is added from outside the stream, so the total pressure will in fact drop a little bit over the short travel distance (due to friction.)

Cavitation also results when the accelerating liquid flashes into a localized region of mixed phase flow in the *intake section of a pump,* with little vapor bubbles being created in the liquid. As the stream continues into a region of lower velocity (as happens in a valve) or into a pump impeller, where energy is added to the stream, the static pressure rises above the saturation point once more, and the little bubbles collapse. In the steady state this results in a stationary zone of continual bubble collapse. Many of the bubbles will be on or near the pipe or pump casing or valve body's metal surface when they implode. You can hear cavitation easily from outside, because it sounds just like stone gravel being pumped through the system, a loud rattling noise.

Carefully instrumented tests have shown that when the little bubbles collapse, microjets of liquid implode on the scene, butting heads at what was the centroid of the now-vanished vapor bubble. **The local pressure spikes may attain instantaneous values in the range of *1,000,000 psi.*** The resulting shock waves are confined to tiny little spaces. Tiny little craters are formed in the metal surfaces by the microblasts, over time, and a valve or pump impeller will eventually be destroyed by the erosion. That's why one must provide sufficient Net Positive Suction Head (which is universally referred to as 'NPSH AVAILABLE') for every pump, and why one should avoid throttling any valve down too far for very long. Change inlet conditions, or get a pump with less NPSH required, or split-range the control valve, or whatever it takes, but do not let the cavitation condition get formed in the first place.

## Now for steam stuff:

We have learned that "water hammer" events are initiated by some action which results in a very much unplanned-for situation somewhere in the piping system or pressure vessel, namely, the formation of a pocket of saturated vapor of the liquid being piped, existing at the saturation pressure corresponding with the liquid's temperature. If, for example, the liquid is water at 60°F, the Steam Tables tell us the vapor pressure will be less than 0.26 psia, a pretty darn good vacuum. And vacuums always attract company, specifically "flying water slugs" in the context of our current topic.

We saw that the Joukowsky water hammer equation describes the bulk pressure rise in a slug of water that accompanies the violent impact typical of collapsing pockets of vapor inside rigidly constraining piping and vessel walls. According to the Joukowsky equation the fluid bulk kinetic energy converted to additional fluid static pressure involves the ***product of fluid bulk velocity × sonic velocity in the liquid*** under the flowing conditions. Since the latter is about 1,000 times greater than the former, the product is numerically quite large. The resulting pressure rise is not always negligible, and may be great enough to do damage to the mechanical system.

Different physical situations were seen to give rise to pressure spikes ranging from the "half-Joukowsky" typical of liquid column separation where no pump energy is involved in the vapor collapse, through the "full-Joukowsky plus motive head" typical of rapid valve closures in pumped flow streams and piped gravity flow systems "flowing full," to superdangerous pump startups and sudden valve openings under high pressure when a vapor pocket awaiting collapse is lurking downstream of the pump or valve (Examples 1-2 and 1-3).

In my view, steam hammer and water hammer are different only in the total energy content and magnitude of pressure differentials available to do damage when vapor pockets collapse and "instantaneous" phase changes occur.

Very hot pressurized water flashes partially into "live" steam when its overhead static pressure drops, a fact we will illustrate later with a typical industrial "flash tank" design exercise. If uncontrolled, the rapid expansion can cause a distinct nuisance: loud bangs, rumbling noises, pipe motions, vibration, and upset flowrates.

And "live steam" certainly collapses back to the liquid state, when its pressure rises without additional heat being added, or when pressure is held steady but heat is transferred away from the hot vapor to its colder surroundings, whatever they may be. The collapse event can be just as dangerous, and potentially is even more so, than large-scale water hammer. In fact it can closely resemble cavitation events, only on a potentially much larger scale, and we just saw how dramatic the damage from sustained small-scale cavitation can be.

We will discuss that aspect of steam hammer, its potentially large-scale effects, in very plain English, because it is a phenomenon that demands clear understanding if it is to be avoided; it has killed people in the past, and can do so again if old engineering mistakes are allowed to be repeated.

## Is It the Steam? Or Is It the Condensate?

If steam hammer happens, you can bet safely that the answer to the above question is **"both."** The best way to get a working understanding of steam hammer, which can be applied to design of a steam system and its piping, is to examine several common types of steam hammer events, searching for the "Cause vs. Effect" relationships inherent in them.

It will become obvious that calculating exactly accurate numerical values of pressure differentials, motive forces, and fluid velocities in realistic steam hammer situations is impractically difficult for everyday purposes. There are just too many variables and complexities involved in the mathematical modeling and in the resulting equations to be solved, especially those of heat transfer, mixed-phase and two-phase flow, complex container geometry, friction effects, and fluid-structure interactions.

***But it doesn't matter what the exact numbers really are. We can't afford to let the conditions for causing steam hammer exist in the first place! So <u>through good design,</u>*** we must <u>avoid</u> the steam hammer events (and their accursedly laborious numerical predictions) altogether.

## One Common Garden Variety

Once upon a time I got called in, by an engineering firm's field startup assistant, as a consultant during startup of a small pilot chemical plant, which had been built alongside an existing production plant. They were having several mechanical problems, including conspicuous and embarrasing levels of steam hammer in the new steam system which had been built for the pilot plant addition.

Upon examination, it turned out to be a dandy illustration of the more common types of steam hammer one might encounter. Please refer to **Figure 1-4** for an illustration of the system. It consisted of:

1. New plant steam supply line running on an elevated steel pipe rack from the plant boiler room to the new pilot process area, where the new main steam header branched several times to serve several steam-to-product feedstock heat exchangers and steam-jacketed *<u>batch-reactor</u>* vessels (not shown in the figure.)
2. In the process area, an atmospheric-vented *<u>packaged</u>* condensate receiver and submersible return pump collect the steam condensate, and lift it vertically upward for the trip back to the plant boilers. The new main condensate return header (points "h" to "j") parallels the new steam header on the pipe rack, and terminates in an existing atmospheric-vented receiver back in the boiler room.
3. Per common practice, the newly constructed steam header's drip leg and trap assemblies collect condensate formed in the steam header. In this particular installation, the design engineer had lifted the condensate, via steam trap motive pressure alone, *directly back into the new condensate return line.* The actual plant configuration had about 1,000 linear feet of outdoor steam header piping on the rack, and had two drip-trap legs about 500 ft apart (for simplicity of illustration I only drew one drip-trap leg in **Figure 1-4.** The condensate return main was also about 1,000 ft long and located outdoors.
4. Each drip leg and condensate trap assembly consisted of the usual drip leg pipe with blowdown (off bottom of header @ "a"), horizontal side takeoff pipe with gate valve, strainer with blowdown, trap with unions ("b"–"c"), check valve, and gate valve. The trap discharge pipe rises from points "d" to "e", runs above the condensate return header's elevation from points "e" to "f", and terminates by means of a vertical drop "f"–"g" into the top of the return header.
5. Finally:
   a. The pilot plant's new steam supply header was small (either 4 in or 6 in pipe size, I forget which), and the plant boilers generated 150 psig saturated output.
   b. The condensate return header was likewise small, only 1-1/2 in pipe size as I recall, and possibly smaller. The two traps and their discharge lines were 1/2-in size.
   c. The new packaged condensate receiver outlet was equipped with a good-quality external swing check valve (shown at point "h".) The receiver pump's packaged controller was standard, level-controlled on-off switches; *on* @ high level and *off* @ low level. It was properly sized, both in tank volume and pump size, for the application.
   d. Pilot plant startup was begun in cold winter weather, and it was a batch process anyway, so it was normal and expected that system flows and temperatures would not remain constant but would fluctuate all the time, and would go from "full on" to "off" and later back to "full on" periodically.
   e. There were a few other steam consumers in the new pilot plant besides the pilot reactors and heat exchangers which utilized the packaged condensate receiver, including some building space unit heaters, so the packaged return pump did not sit there like a bump on a log; it ran fairly often, especially during startup as would be expected.

*Now here was the problem, the reason why they called me in. They would shut off the new pilot plant's main feed steam shutoff valve (which was located in the boiler room, not shown in **Figure 1-4)*** every now and then, to make equipment adjustments and minor construction changes during the pilot plant's startup and shakedown effort (which apparently did not go flawlessly, so there were a few false starts). It seemed that whenever they started back up, not very long after the main steam valve was opened, some steam hammer noises—pops and bangs—would begin. These noises varied in strength and were pretty random in timing; after the system had heated up somewhat, occasionally some

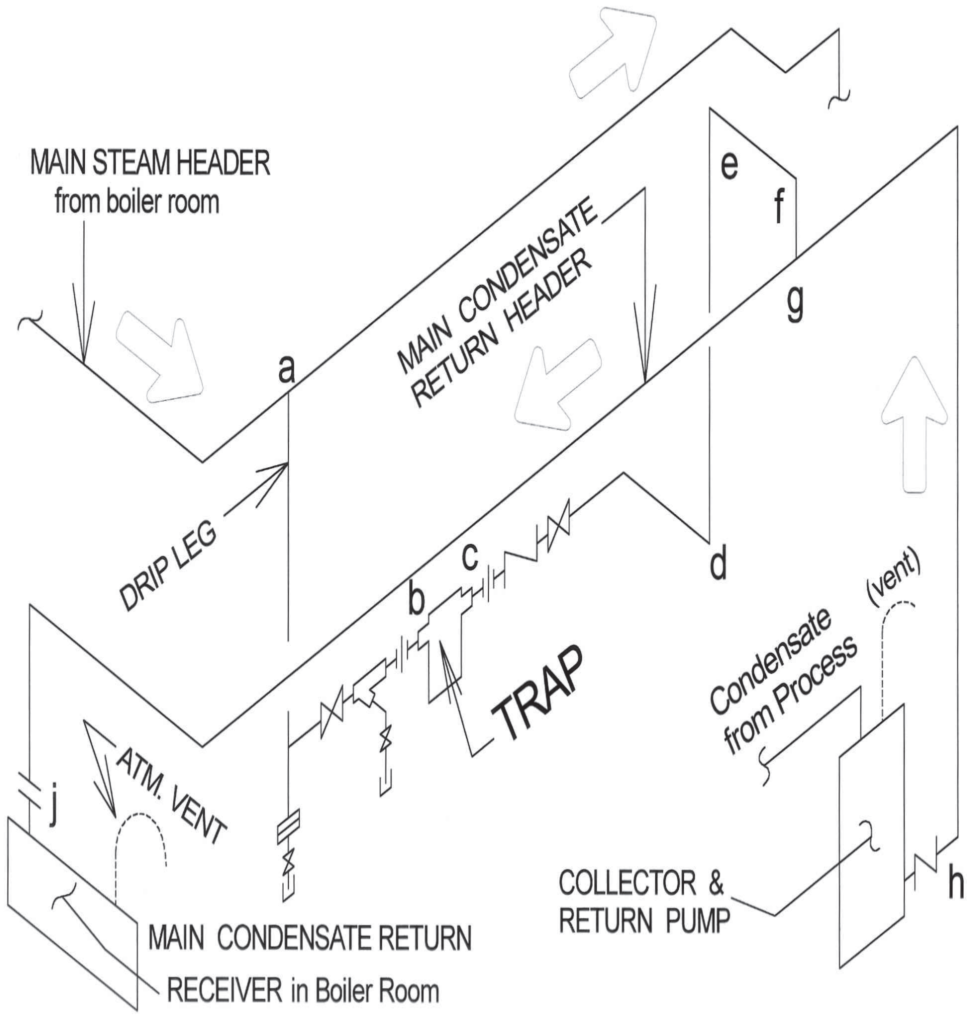



**FIGURE 1-4 STEAM HAMMER DRIP LEG AND TRAP ON STEAM MAIN. "GARDEN VARIETY"**

really loud bangs would issue forth here and there, plus some piping shakes were going on, and the mechanical crew, startup assistant and owner's rep were getting testy about it. I went, saw, heard, found that it was an excessive amount of "steam hammer" activity, and agreed with those who summoned me that some improvements were needed. The causes, effects, and remedies are discussed next.

## Cause and Effect

The problem had been described to me in terms of loud, frequent random noises (coming from the new steam piping system), of frightened pipefitters, welders, and instrument mechanics trying to finish their work in the vicinity of those noises, of sometimes noticeable pipe shaking, and last, but certainly not least, a growing alarm on behalf of the owner's folks who were observing all of the above. These are quite common symptoms associated with steam hammer, and I saw that the problem could be solved by realistic means *because the new steam system was simple, easy to get to, relatively small and cheap, and was not yet committed to production. No major shutdown would be necessary to fix it, and the pilot plant schedule was not in serious trouble as yet. There was time, labor, and impetus available on all sides to nip the problem in the bud, right then and there.*

So the first thing I did, after taking my initial look at the piping construction drawings, making a site walkdown, and then making a mental comparison of "as-built vs. as-drawn," was to try to calm everyone down and assure them that it could be fixed in one of several different ways, and was not at that point a safety problem. (The shaking motion amplitudes were not actually going to lead to early pipe fatigue failure, in my opinion, based on ASME B31.3 Course movie presentations I had studied of actual full-scale "steel pipe shaking failures" made under controlled lab test conditions; and, the piping supports seemed adequate to stand up to what shaking motion I could see for a good while—perhaps a year or more at the level of shaking I observed at that time.)

The basic underlying **cause** of steam hammer in that piping system was the same as it always is in every piping system: **rapid, uncontrolled change of phase of the fluid. The rapid phase change and large significant spikes (transients) in fluid pressure always accompany each other.** Depending on what the pressurized fluid has available to it to act upon, within the confines of the pipe or pressure vessel, we can be sure that transient pressure forces will accelerate slugs of condensate (water) into motion of some kind, as projectiles of some mass, velocity and energy level. The phenomenon from that point onward resembles severe water hammer as we previously studied it.

To sufficiently understand what was happening in the pilot plant steam system, all one has to do is to imagine being a small particle of the liquid, riding along in a "thought experiment" inside the piping with the bulk flow of the water, as it begins its journey as saturated steam from the boiler **(see point "a")** and ends up back in the boiler room's main receiver as hot, near-saturated liquid condensate **(point "j" in Figure 1-4):**

- At point **"a"** some of the 150-psig steam which has condensed on the pipe steel flows by gravity into the drip leg. The flow is continual; the colder the pipe steel, the greater the condensate formation and flow rate.
- The drip leg fills up to the elevation of the horizontal takeoff to the steam trap. Further flow then goes through the open gate valve and strainer and begins to spill into the condensate holding chamber of the trap.
- Slowly, the trap fills up with condensate. (Air, which initially filled the trap and leg, has already been purged automatically by the trap.) When full of liquid condensate, which is *saturated water existing at about 150* ***psig and 366°F,*** the trap mechanism goes to work and the trap opens. The trapped condensate feels a sudden drop in pressure ahead of it, because pressure downstream of the closed trap is a whole lot lower than the steam pressure, and when the trap opens, the trap's exposed discharge orifice becomes a bullseye. The pressure differential across the now-open orifice is on the order of **150 psig,** and the orifice area is on the order of **0.1 in$^2$**, so a net force of about **15 lbs** rapidly blows the condensate, which has a volume of perhaps a tea cup, out of the trap.
- Moving like a bat out of Hades, the slug of hot condensate zips through the check valve and discharge gate, losing only a little bit of its pressure. The trap slams shut behind it, but its momentum carries the slug forward. Let's say the condensate slug hits the discharge piping at **point "d" at 366°F** (the same temperature as inside the trap body, since not nearly enough time has elapsed for sensible heat transfer to take place … heat transfer takes time! … and at a pressure of about **145 psig.** Now what happens?
- **The slug finds itself suddenly surrounded by practically no fluid pressure at all! The thermodynamics of phase change now takes over, and things get interesting very quickly…**

*So that we can get a feel for the magnitude of things let's do a freeze-frame type of assessment from this point onward, and stipulate for argument's sake the following existing conditions in the condensate return piping* <u>downstream of the trap leg,</u> where nothing has as yet been disturbed:

1. Assume a little bit of "colder" condensate (maybe at 0 psig and 210°F) was in the horizontal discharge piping when the trap blew open. It was displaced immediately into the riser **"d"–"e",** which is 24 in. length, where, in our freeze-frame picture of the event, the "colder" condensate fills about, let's say, the first 3 vertical inches of the riser's base. The rest of the riser and legs **"e"–"g"** are filled with a **saturated mixture of air and water vapor, at atmospheric pressure, and at a mixture bulk temperature of about 190°F.** (The *exact* pressure and temperature values aren't important.)
2. Assume the horizontal portions of the condensate return main, from points **"g" to "j",** are about **1/4 full of stationary 190°F water, with the upper 3/4 filled with the saturated mixture of air and water vapor at atmospheric pressure and about 190°F.** The first riser, beginning at check valve "h," is essentially **full of stationary water at 190°F.** The downcomer attached to the vented receiver at **point "j"** contains only **the air-water vapor mixture.**
3. The return pump is OFF, its check valve is closed, and *no water flow is taking place downstream of point* ***"d"*** … in any direction.

Now, we return to the newly discharged slug of hot condensate just ahead of the elbow at **"d"** and <u>end the freeze-frame. As the action resumes</u> we would see a sequence unfold something like this:

…**beginning with some of the hot condensate flashing from the saturated liquid phase to saturated vapor.** The mass of water that flashes to vapor is a lot less than the total quantity ejected from the trap (see sample calculation of flash tank volumes later herein). However, the increase in volume occupied by the total mass of the trap ejecta after flashing is really large. **To illustrate, when the ejecta was still saturated liquid at 165 psig, its specific volume $v_f$** was about <u>0.0183 ft$^3$</u>/lbm. Immediately upon flashing to saturated vapor at about zero psig, the specific volume $v_g$ of the fluid which flashed was about <u>26.80 ft$^3$</u>/lbm. That's an increase of 26.8/.0183 = 1,465 times. Each cubic inch of hot condensate that flashed, produced a burst of expanding vapor that reached at its maximum extent a volume of about 1,465 in.$^3$ of 15 psia steam.

Whoa, hoss! Think about that a second:

**1465 in.$^3$** is the internal volume provided by a length of <u>520 ft of 1/2-in.</u> Schedule 80 carbon steel pipe! But the 1/2-in. trap discharge pipe is no more than, say, 10 developed ft long! So, assuming our guess of 1.0 in$^3$ of flashed liquid is about right, then all that flash steam blows into the condensate return header, which we said was 1 1/2-in. pipe size (Schedule 40).

**1465 in.$^3$** of expanding water vapor fills <u>60 ft of empty 1–1/2-in.</u> Schedule 40 pipe, but our condensate main is one-fourth full of liquid and three-fourths filled with moist air. The moist air is easily displaced by the explosive expansion of flash steam; the displaced volume of air, assuming the preexisting liquid stays put in the bottom of the condensate header, is about (60)(4/3) ~ <u>80 linear ft of pipe,</u> which constitutes about <u>8% of the whole condensate main.</u>

Now we can understand what was causing all those crazy sounds! It takes only a few moments after the first <u>*"bang,"*</u> which was the arrival of the trap ejecta hard against the elbow at **"d,"** and the accompanying *"<u>WHOOOOOOOM</u>"*, which was the flash vapor expansion into the header, for the hot flash vapor to lose heat via conduction to the cooler pipe steel and previously undrained condensate. Just as suddenly as it had flashed into vapor, after a few moments have passed *it suddenly condenses into practically no volume at all, leaving a nearly full vacuum in a stretch of pipe about 80 ft long* (Remember, the flash steam displaced all the air in its way!)

Thanks to our study of water hammer (*see the section based on* Figure 1-1), we know what can happen when only a small vapor pocket collapses. So, the very loud and truly upsetting **"<u>KAAA-*PIE*</u>-**YOW!!!", which was the crashing impact of separated water columns rushing together to eliminate that hateful 80-foot-long vacuum in the 1,000-foot-long header, should come as no surprise whatsoever.

And neither should the series of "POP-BANG-RATTLE-CLANG-RATTLEs" which finally fade away, after a few repetitions, because all that linear momentum has to go to some kind of final resting place, and that place is the mechanical piping system! Cyclic bending and torsional stress reversals occur in the piping, and maybe in the pipe supports too, as the steam hammer vibes shake the system back and forth. If uncorrected, sooner or later fatigue failure will occur at the point of maximum cyclical stress range intensity in the pipe or hanger steel. Not good.

It should be pretty apparent, by this point, that exact numerical prediction of "steam" hammer overpressures, during flashed steam expansions and subsequent condensation collapses, would be next to impossible to make, in any but the simplest controlled laboratory setups, and truly impossible in real life piping systems. We can see, however, that the potential for damage is exactly the same as in "water" hammer.

Now, the worst case:

The worst steam hammer event occurs when motive pressure from some outside source collapses the pocket of flashed steam (crushes the vapor back into liquid phase) *before* natural collapse due to cooling & condensation can take place (a gentler process, by comparison.) Per **Figure 1-4,** this would be the case if the condensate return pump happened to switch "on" while a pocket of flashed trap ejecta occupies part of the return header. For example, assume the shutoff head of that pump happens to be 50 ft of water, 21.7 psi. The motive differential for accelerating the water slug toward the vapor pocket = (21.7 – 0) = 21.7 psi. If we use a "liquid mover" device *(in which live steam admitted thru a control valve replaces the mechanical pump as motive source in the condensate collector),* the initial differential can be much larger, namely, the pressure downstream of the automated steam control valve or regulator feeding the "liquid mover" device during its operation). So! The "common garden variety" steam hammer noises are coming from the condensate return system, and are due to the uncontrolled expansion of pressurized condensate into flash steam, with subsequent collapse of the flashed vapor which causes a local vacuum, followed by water slug acceleration toward the vacuum. Its impact on pipe steel is what we hear, and what eventually can lead to fatigue failure (or worse, if the supports are inadequate or the pressure integrity of the system is sub-par).

**To avoid steam hammer of this kind, simply refrain from connecting steam trap discharge directly to the condensate return piping header. Instead, collect the trap ejecta into an engineered atmospheric receiver, where its flashing to 0 psig and 212°F. is controlled by chamber volume and vent pipe size. Then, use either a "liquid mover" type device, or a mechanical pump, to lift the still-hot but now atmospheric-pressure and thus non-flashing liquid condensate into an overhead sloped gravity return main, sized to flow only half full or less at any time, and let gravity bring that condensate safely and quietly back to the boiler room. The sloped gravity flow piping should have periodic vents rising to atmosphere at intervals, just like a gravity sewer line, to avoid "plugging and chugging" problems.**

**Figure 1-5 illustrates my suggestion. (Chapter Two of this book covers design of gravity flow piping systems.)**

A different, initially cheaper, and over some time period more costly, but equally effective way to solve garden variety steam hammer is to *throw the condensate away.* Instead of collecting steam trap ejecta, waste it (safely) to a suitable drain. Give it a place to flash without causing any problems or hazards, and let it disappear from your life forever (and take its perfectly reusable bought and paid-for mass, heat content, and feedwater chemical treatment with it, placing additional burden on the plant chemical waste treatment system in the process.)

***Recommended? Economical? Good practice? Hardly.***
***Effective at preventing the noise and pipe shaking? Very.***

So much for the "garden variety" steam hammer nuisance. Now we must become aware of a different, and much more dangerous variety. This next one is a proven killer.

## One Uncommon Fatal Variety:

Please note: in suggesting fixes for "garden variety" steam hammer, I did **not** offer as a third alternative the omission of drip leg traps from the steam main. The following will make it quite clear that my failure to do so was **most definitely intentional.**

This uncommon variety is similar to all the other flavors of "fluid hammer" in many ways, but it has three distinct differences:

1. It takes place inside the steam supply pipe.
2. It involves potentially catastrophic levels of motive pressure differential and energy set free to cause damage.
3. It may happen only once, in a system that has run just fine for years, without giving any prior warning whatsoever, with tragic consequences resulting.

## Cause and Effect

The cause is extremely simple: ***the collection of liquid, in a steam header connected to a branch network or another parallel header.***

The effect, when live steam flow is admitted from a dry header into the waterlogged pipe, is mechanical interaction between steam and water that goes something like this:

- Pool of cold water fills lower portion of cold waterlogged pipe. There may be a closed valve in the waterlogged pipeline, or an undrained low point acting as a sort of unintended P-trap which collects the water. It could be an undrained vessel or knockout pot, too. *Whatever;* a pool of liquid there in the waterlogged pipe is directly in the path of the motive steam coming from the dry header connected to it.
- Live steam hits cold water and cold steel, and condenses, at first. The motive steam just keeps on coming from the dry steam source, and part of it just keeps on condensing, for a while.
- The newly formed condensate adds to the mass of liquid already trapped in the waterlogged pipe, and gradually warms the previously existing water and the pipe steel that contains it.
- Eventually, the water and steel approach, and finally reach, the saturation temperature corresponding to the initial pressure of the motive steam being admitted into the waterlogged pipeline. For example, if the boilers are sending 150 psig steam to this section of pipeline, the trapped water and pipe steel temperatures will continually rise until they hit the saturation point for 150 psig: *366°F.*
- Meanwhile, the motive steam in the *dry header* connected to the waterlogged pipe blows fast, by design, in a velocity range usually around 80–150 ft/sec.
- The liquid trapped below the flying steam is now saturated at the steam pressure and temperature, and thus quits condensing

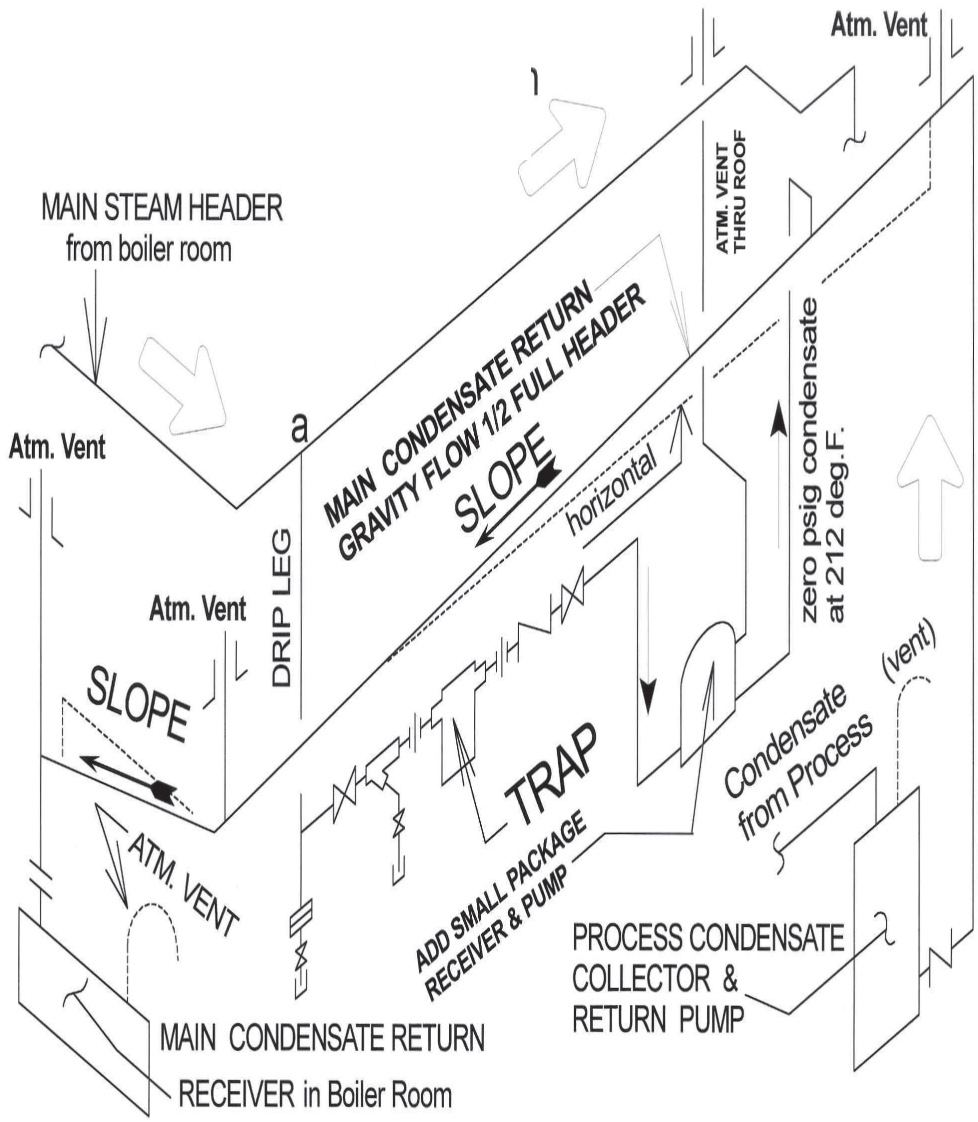



**FIGURE 1-5 NO STEAM HAMMER FROM TRAPPING INTO CONDENSATE MAIN. "GARDEN VARIETY PROBLEM SOLVED"**

any more of the steam, and it drags at the dry steam flying overhead. **Waves** form on the liquid surface in the water-logged pipe. **They grow,** *and the wave tops reach for the top of the pipe.*

- Finally, a few of the waves reach the top, forming 100% blockages of the steam's flow path. The **slug flow** regime has begun. The slugs of liquid are subjected to large body forces, and become serious projectiles.

The actual piping system configuration determines the final outcome of this drama abuilding. While the various possible outcomes differ in severity, none of them are good for anyone or anything!

- At the very least, if the pipeline owner is really lucky, the water shots will simply hammer the piping so roughly and violently that a human will take notice, figure out that something has gone badly wrong, and will turn the steam supply valve nearly, but not completely, shut *{to avoid causing additional "shock" while effectively reducing the fresh energy input to a safe level, and thus allow the system gyrations to damp out naturally … mechanically and thermodynamically.}*
- With a little less luck, the human will quickly shut the valve completely off. The action may not stop immediately, however. And mechanical damage may or may not occur, or be visibly evident, if the human acts soon enough. But one way or another, the gyrations *will* cease.

*If the human unfortunately does not respond by shutting off the steam supply, but sticks around the area long enough,* he may live to witness a **true steam hammer event:**

- The flying waves of water slugs will continue to impact the physical barrier (probably a closed valve or pipe riser) which keeps the trapped water from being blown away and out of harm's way in the first place. The impacts will be more or less cyclical in nature. A rhythmical motion will begin; back-and-forth pressure pulses will build.

At some point the pressure swings will get quite large, and cause local liquid column separation effects at the physical barrier wall. Each rebounding slug departure from the barrier will now cause a vapor pocket creation, followed by collapse and yet again another separation, cycle after cycle, building, building, with the live steam pressure as the motivator! ***Wicked! Wicked! But wait … it can get even worse!!!***

- Geometry and pipe restraints permitting, the impact frequency may build, and build, until it corresponds to one of the fundamental natural frequencies of the piping system. Acoustic resonance of the fluid column (organ pipe vibration), and mechanical vibratory resonance of the pipe within its restraints (guitar string vibration) can both come into play. Both modes of vibratory resonance will cause mechanical energy to be stored, and to increase rapidly without dissipation to ward off the coming disaster. ***And if our bad luck is world-record class all-time bottom-of-the-barrel supreme,*** the natural acoustic resonance frequency will overlap the mechanical pipe resonance natural frequency, HUGE doses of fluid energy will go to creation of linear momentum of the piping system component masses, and then it will all be over very quickly.

This is what has killed people, right up to the present day. The energy stored in those fluid pressure waves is so huge, and the blows delivered to the piping and its supports by those cyclical impacts are so violent, that the mechanical system will be wrecked. People may be struck by the flying metal when brittle iron valves disintegrate like grenades, or when pipe hangers break and the piping falls to earth, or be scalded to death, or literally cut in half, as neatly and quickly as by any sawmill blade, by a jet of high pressure steam escaping through a blown-away flange gasket space at sonic velocity. A live steam jet from a high-pressure source is an incredible cutting tool.

In his really neat ASME Professional Development Series piping lectures, George Antaki, P.E., of the Savannah River Plant details a recent case like this that took place in Japan. The man who was killed happened to be standing beside a steam crossover pipe riser inside a valve pit, during startup of a cold buried steam line in parallel with a live hot one. The cold line, 8-in pipe size and 840-ft long, had been isolated for 8 months before startup, and had ***not*** been drained. It had a slope *toward the closed isolation valve* of 11 ft in that 840-ft run. The low end was full of water when the isolation valves were opened for startup. You can guess the rest.

The man was crushed by sudden rupture of the valve he was tending. The valve was on the end of the header, where the action was. To determine the accident's cause for sure, the Japanese engineers made a hydrodynamically scaled (approximately 1/6-size) model of the failed system, and used glass pipe for visibility (the system piping had been low carbon steel, the proper material). Their scale tests permitted the steam hammer buildup, which takes time to develop and maximize, to be observed. No doubt about it. The ruptured valve had been at the location of significant transduced pressure spikes. (I do not know what the calculated magnitude of the real event spike was, but that is neither here nor there. The lesson was learned!)

You know, what impressed me the most about George's documented account, besides its tragic result, was the small size of the failed steam pipe … *only 8-in.* … and the operating motive steam condition, which was ***only 115 psig!!*** Not an unusual system in *any* respect, in anyone's book! If it happened to those guys, it can happen to anyone!

If the cold steam line had contained only air, prior to startup, nothing out of the ordinary would have occurred. No steam hammer. No damage. No death.

## The Steam Hammer Resulted from Human Failure to Keep Liquid Out of the Steam Pipe

This would have come as no surprise to old timers. Once upon a time, some cities received their space heating media from miles-long networks of underground municipal steam lines. Those long, relatively flat underground utility pipes were "naturals" for steam hammer, a fact the engineers of those days soon noted.

With Mr. Antaki's kind permission, because this is a true safety issue and needs to be familiar to all engineers who may have to become involved in steam piping, I am reprinting two pages from George's fine ASME Course Manual, "Operation, Maintenance and Repair of Plant Piping Systems." The first page contains an excellent appraisal of the matter which was published by the good old ASME in 1883. Not *1993*——**1883.**

## Condensable Two-Phase Flow

### Mixing of Steam and Water

*Robert H. Thurston, Hoboken, N.J., Transactions of the American Society of Mechanical Engineers, 1883*

*"When a pipe is filled with steam, and then has introduced into it a quantity of cold water, or when a pipe, itself cold, and containing cold water, even in very small quantity, and without pressure, has steam turned into it, the first contact of the two fluids is accompanied by a sudden condensation which causes a sharp blow to be struck, usually at the point of entrance; and sometimes a succession of such blows, which are the heavier as the pipe is large, and which may be startling, and even dangerous."*

*"The steam, at entrance, passes over, or comes in contact with the surface of the cold water standing in the pipe. Condensation occurs, at first very slowly, but presently more quickly, and then so rapidly that the surface of the contact between the two fluids is broken, and condensation is completed with a suddenness that produces a vacuum. The water surrounding this vacuum is next projected violently from all sides into the vacuous space, and, crossing it, strikes upon the surface surrounding it. As water is nearly incompressible, the blow thus struck is like that of a solid body…"*

*"Where pipes are not burst by this action, it is common to see them sprung and twisted out of line, torn from their connections and, when a succession of shocks occur, as is often the case, the whole line writhes and jumps lengthwise to an extent that is sufficiently serious to cause well-grounded alarm."*

*"It seems very certain that we may consider it as proven that the pressures produced by "water-hammer" are often enormously in excess of those familiar to us in the use of steam, and that they have, in many cases exceeded 1000 pounds per square inch. It is, then, evident that it is not often safe to calculate upon meeting these tremendous stresses by weight and thickness of metal, but that the engineer must rely principally. If not solely. upon complete and certain drainage of the pipe at all times as the only means of*

*safely handling steam in long pipes, such, especially, as are now coming into use in the heating or cities by steam led through the streets in underground mains."*

***U.S. Nuclear Regulatory Commission, NUREG-0927, 1984.***

*"State-of-the-art mechanistic or quantitative two-phase analysis of water hamer phenomena is not a practical means of resolving all water hammer … the extensiveness of possible plant conditions, alignments, and computer code calculational limits preclude analyzing all possible scenarios … Anticipated water hammer events, caused by components performing in their intended manner should be included as occasional loads in the design basis of piping and their support systems."*

***Notes: Rules of good practice to avoid waterhammer induced by steam condensation.***

*(a) Place steam traps and drains (free blows) at low points.*
*(b) Vent high points when draining low points.*
*(c) Properly size steam traps.*
*(d) Drain condensate from both sides of closed isolation valves.*
*(e) Slope lines to allow draining the condensate.*
*(f) Avoid pockets that could trap condensate, such as low points in valve bonnets or strainers.*
*(g) Place relief valves on low-pressure side of pressure regulators.*
*(h) Drain the trap discharge condensate.*
*(i) If water-steam mixing is intentional in certain processes, mix through a specially designed mixing valve.*
*(j) Periodically inspect the condition of pipe supports.*
*(k) Familiarize operators with the causes of waterhammer, its symptoms, and corrective actions should it occur.*

## Controlling Flash Steam When You Let Liquid Condensate Drop Down to a Lower Pressure or to Atmospheric Pressure Immediately: Using a Flash Tank Vessel to Produce Intermediate Pressure Steam from High Pressure Condensate Discharged from a Trap

"Wait just a doggone minute," you say?

"I am <u>not</u> allowed to just dump the liquid steam condensate into a big old open waste pit in the ground somewhere, to let it expand down to atmospheric pressure and 212°F after it leaves the trap!! I would be fired in a heartbeat!! (True.) I must recover that condensate, and as much of the heat it contains as possible! (You bet!) Now what can I do to avoid all those dangerous steam hammer expansion effects you have warned us about so vehemently, if my project's process design requires converting all the hot condensate from the high pressure traps to steam vapor at a lower steam pressure for further use? Or if the condensate is to be reduced in pressure before being recycled back to the steam boiler's condensate receiver vessel as really hot water?"

Well, as you doubtlessly already know, there is a good answer already worked out for those contingencies. It involves a very simple device called, appropriately enough, a "flash tank." We will cover its features and design approach now.

Steam hammer is an awful nuisance and we have already learned how very dangerous it can be. It is avoided by proper engineering design of the steam condensate handling system and associated equipment. Therefore the "flash tank" or "condensate receiver" is a necessary ingredient of any condensate handling system.

The key ingredient is a pressure vessel which receives liquid condensate from one or more steam traps, at the saturated temperature and pressure of the live steam. The vessel has an open-sight side liquid inlet, a trapped or pumped-out bottom liquid outlet, and a top center vapor/air outlet. The vapor outlet is connected to a receiving volume of steam at a constant, lower saturated pressure. See **Figure 1-6.**

The vessel acts as a ***control chamber,*** letting the portion of depressurized condensate, which flashes to steam at the reduced outlet vent pressure, separate safely from the remaining liquid. ***No slug formation, no condensate projectiles whizzing through the piping system, no steam hammer!***

In industrial and power plant applications, the vessel is called a ***flash tank.*** A typical example might receive saturated liquid from a **300 psig system steam trap,** at about **300 psig and 422°F.** About **15.4%** of the liquid would **flash to 40 psig steam,** and would **vent to a 40 psig steam header.** The remaining **84.6%** of the condensate would collect in the bottom of the tank, at its **new** saturated liquid condition: **40 psig, 287 °F.**

*Note that such a vessel is subject to the ASME Section VIII Div. 1 Unfired Pressure Vessels Code and must bear the ASME Registry and U-Stamp.*

In heating, ventilation, and air conditioning (HVAC) applications, the vessel is called an **atmospheric, condensate receiver.** A typical example might receive saturated liquid from a bunch of **150 psig system steam traps,** at about **150 psig and 366°F.** About **16.3%** of the liquid would **flash to zero psig steam,** and would **vent to the atmosphere.** The remaining **83.7%** of the condensate would collect in the bottom of the receiver, at its **new** saturated liquid condition: **0 psig, 212°F.** *Since the receiver is not pressurized, if properly vented directly to the atmosphere (no back-pressure buildup) it can be fabricated of non-Code construction.*

## Calculating Flash Steam Quantity

Flash steam calculations are easy. Let subscript **"H"** stand for high-pressure condensate in its equilibrium state before flashing, and **"L"** for Low, also an equilibrium state, after flashing has occurred.

From steam tables, **hf** = enthalpy of saturated liquid, **hg** = enthalpy of saturated vapor, and **hfg = (hg – hf),** the enthalpy of phase change, all at a given steam pressure. **In a flashing event:**

1. A mass of liquid condensate $\mathbf{mf_h}$ at saturation state $\mathbf{P_H}$, $\mathbf{T_H}$ rapidly depressurizes ("flashes") to a new lower state $\mathbf{P_L}$, $\mathbf{T_L}$ when it leaves a liquid trap and enters into a new spatial volume which is at some lower pressure.
2. Flashing is adiabatic and occurs inside a closed volume (closed vessel.) By the first law of thermodynamics and the law of conservation of mass, we find that after flashing, the mass of liquid condensate is reduced, to $\mathbf{mf_L}$, by precisely the amount of flash steam produced, **mg,** at the new lower pressure $\mathbf{P_L}$.

Therefore, the continuity law says that *flash vapor mass = reduction in liquid mass;*

$$\mathbf{mg = mf_H - mf_L}$$

3. The energy content of the flash steam can only come from the energy contained in the original condensate mass $\mathbf{mf_H}$. The first law tells us that total energy of the closed system also remains constant (no work or heat crosses the system boundary), and therefore

$$\mathbf{(mf_H)(hf_H) - (mf_L)(hf_L) = (mg)(hfg)}$$

***Note:*** since the end state is at thermal equilibrium, the term **"hfg"** is evaluated at the **new lower state $\mathbf{P_L,T_L}$** which is the state shared by $\mathbf{mf_L}$ and **mg.**

4. All we have to do is rearrange equation (3) to obtain the simple formula for finding the mass percent of condensate which flashes to steam:

$$\mathbf{\%\ Flash\ steam} \equiv \mathbf{(mg \div mf_H)} \times 100\% = $$
$$= \{(hf_H - hf_L) \div hfg\} \times 100\%$$

The steam trap manufacturers publish tables and plots based on this relationship, so you do not have to look up the enthalpies or use Eq. 4, above to find the mass flow of flashed steam coming off a given mass flow of high-pressure condensate. You can also use their data to physically dimension the flash tank vessel.

The most excellent Armstrong International, Inc. company has furnished the following pages of data on flash tanks (all except the crude figure on the very next page, which is my own unworthy work.)

## Sizing the Flash Tank Is Simple:

**The object is to keep the upward velocity of flash steam inside the vessel low, to minimize liquid droplet entrainment in the steam. For the same reason, the tank is vertical and has a minimum vertical height and diameter. The following Armstrong data illustrate the procedure quite well. It is *reprinted with the permission of Armstrong International, Inc. 2005.***

## Methods of Using Flash Steam {Armstrong Data}

When traps drain medium and high-pressure process equipment, substantial amounts of flash steam may be formed. The heat content of this flash is identical to that of live steam at the same pressure. But, this valuable heat content is wasted when an excess amount of flash is allowed to escape through the vents in the receiver.

Note: In smaller plants not employing deaerators, it is desirable to vent some flash from the receiver.

This venting releases corrosion-producing oxygen and carbon dioxide commonly existent in steam systems.

The latent heat content of flash steam may be used for space heating; for heating or preheating water, oil and other liquids; and for low-pressure process heating.

The flash steam recovered may be used by itself to take care of the demands of one or more steam heated units or systems. Or, if exhaust steam is available it may be combined with the flash. In other cases, the flash will have to be supplemented by live make-up steam at reduced pressure.

**Flash Tank Hookup.** Condensate return lines contain both flash steam and water. To recover the flash steam, the return header may be run to a flash tank, where the condensate is drained off. The steam is piped from the flash tank to point of use.

## Combining Flash and Live Steam

The demand for low-pressure steam should correspond with the availability of flash steam, but should preferably be greater, the difference being made up with live steam through a reducing valve. The reducing valve is set to maintain the required low steam pressure and insures a continuous supply.

Fig. CG-57 illustrates how a reducing valve is used to combine flash and live steam. Remember that the amount of flash will never be as great as theoretical. Therefore the reducing valve should be sized to passs enough make-up live steam under conditions of maximum demand and minimum supply of flash.

Fig. CG-57 also shows the proper way to hook up a flash tank to combine flash and live steam. Notice how the pipe-work is arranged so that the flash tank can be conveniently bypassed and condensate taken direct to the receiver. This is essential if flash steam cannot always be used when it is available.

Notice the "required" relief valve on the flash tank. It prevents pressure from building up and interfering with the operation of the high pressure steam traps. A pressure gauge on the tank is also advisable.

## Sizing Flash Tanks

The flash tank can usually be conveniently constructed from a piece of large diameter piping with the bottom ends welded or bolted in position. The tanks should be mounted vertically. A steam outlet is required at the top and a water outlet a few inches from the bottom. The condensate inlet connection should be 6 to 8 inches above the water outlet. No internal fittings are needed.

The important dimension is the inside diameter. This should be such that the upward velocity of flash to the outlet is low enough to insure that the amount of water carried over the flash is small. If the upward velocity is kept low, the height of the tank is not important, but good practice is to use a height of two to three feet.

It has been found that a steam velocity of about 10 feet per second inside the flash tank will give good separation of steam and water. On this basis, proper inside diameters for various quantities of flash steam have been calculated; the results are plotted in Chart CG-22. This curve gives the smallest recommended internal diameters. If it is more convenient, a larger size of tank may be used.

Chart CG-22 does not take into consideration pressure – only weight. Although volume of steam and upward velocity are less at a higher pressure, because steam is denser, there is an increased tendency for priming. Thus it is recommended that, regardless of pressure, Chart CG-22 be used to find the internal diameter.

## Practical Considerations in Installation of a Flash Tank

Bearing in mind that a flash tank causes back pressure on the steam traps feeding into the tank, the traps should be checked to

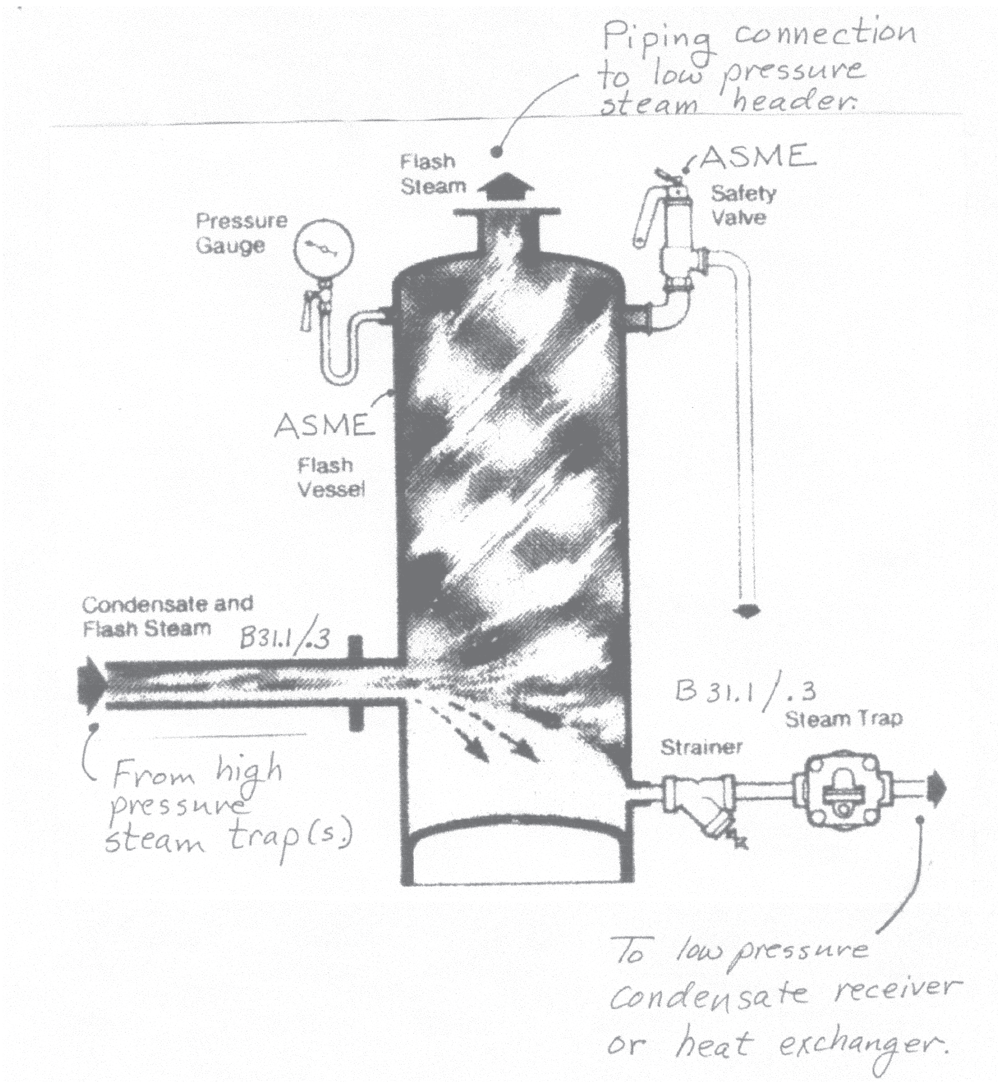



**FIGURE 1-6 FLASH TANK: SIMPLIFIED SCHEMATIC**

see if they still have sufficient capacity at the reduced differential pressure.

Condensate lines should be pitched down towards the flash tank. Where more than one line feeds into a flash tank, each line should be fitted with a swing check valve. Then, if any line is not in use, it will be isolated from the others and will not condense and waste flash steam.

The steam trap draining the flash tank should be large enough to handle the maximum condensate load, which will usually occur when starting up. As this may be three times normal load, the trap should be sized to meet these conditions. Because the traps will work at low pressure, gravity drainage to the receiver should be provided.

Generally, the location chosen for the flash tank should meet the requirement for maximum quantity of flash steam and maximum length of pipework.

Condensate lines, the flash tank, and the low pressure steam lines should be insulated, so as to prevent waste of flash through radiation.

The fitting of a spray nozzle on the inlet pipe inside the tank is *not* recommended. It may become choked, stop the flow of condensate, and set up a back pressure to the traps.

Condensate lines from low pressure equipment using flash steam should not be connected to the lines feeding the flash tank. The reason is that the pressures will be about equal, and a slight drop in the condensate line pressure will cause back-up.

In some cases large volumes of air may need to be vented from the flash tank. A thermostatic air vent of adequate capacity will vent the air from the tank and keep it from passing through the low pressure heating or process units.

## HOW TO TRAP FLASH TANKS

When hot condensate or boiler water, under pressure, is released to a lower pressure, part of it is re-evaporated, becoming what is known as flash steam. The heat content of flash is identical to that of live steam at the same pressure, although this valuable heat is wasted when allowed to escape through the vent in the receiver.

**Figure CG-58. Flash Steam Recovery from an Air Heater Battery**
Flash is taken from the flash tank and combined with live steam, the pressure of which is reduced to that of the flash by a reducing valve.

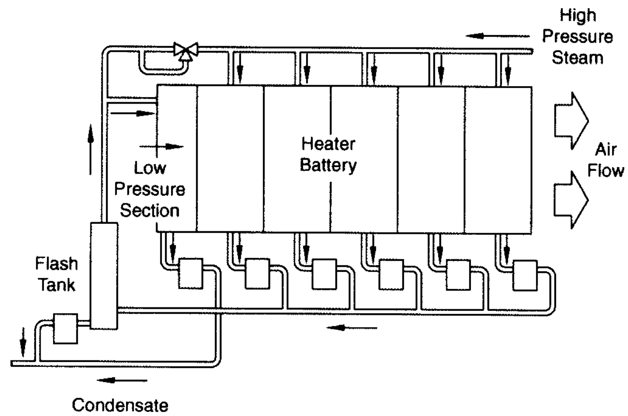



**Chart CG-21. Recommendation Chart**
(See Page CG-2 for "Feature Code" References.)

| Equipment Being Trapped | 1st Choice and Feature Code | Alternate Choice |
|---|---|---|
| Flash Tanks | IBLV<br>B, E, M, L, I, A, F | F&T or *DC |

*Recommended where condensate loads exceed the separating capability of the flash tank.

**Figure CG-57. Typical Flash Tank Piping Sketch**

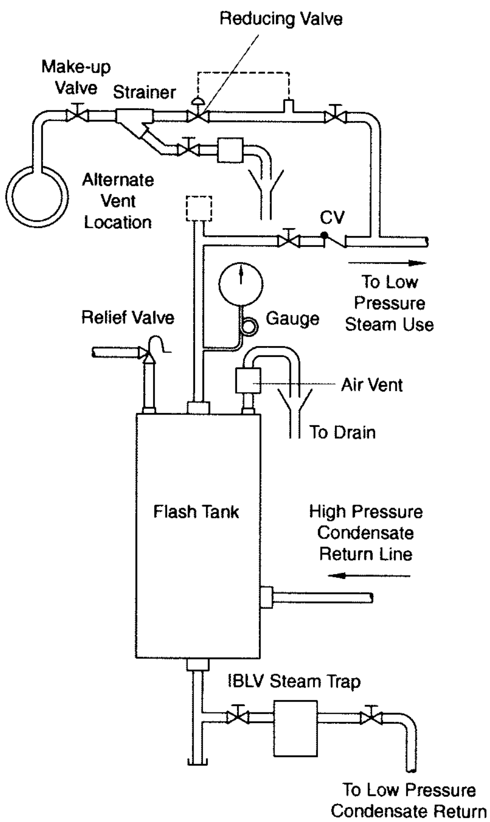
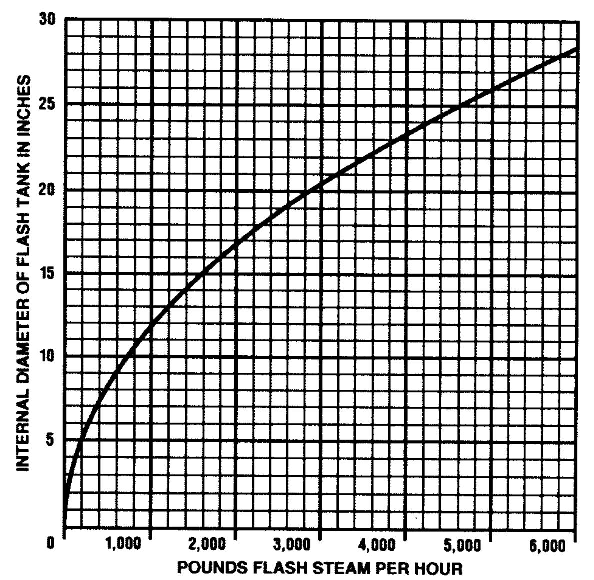



**Chart CG-22.**
**Determination of Internal Diameter of Flash Tank to Handle a Given Quantity of Flash Steam**
Find amount of available flash steam (in pounds per hour) on bottom scale, read up to curve and across to vertical scale, to get diameter in inches.



With proper sizing and installation of a flash recovery system, the latent heat content of flash steam may be used for space heating; heating or preheating water, oil and other liquids; and low pressure process heating.

If exhaust steam is available it may be combined with the flash. In other cases, the flash will have to be supplemented by live make-up steam at reduced pressure. The actual amount of flash steam formed varies according to pressure conditions. The greater the difference between initial pressure and pressure on the discharge side, the greater the amount of flash that will be generated.

### Trap Selection

The condensate load can be calculated using the following formula:

$$Q = L - \frac{L \times P}{100}$$

Where:

Q = Condensate load in lbs/hr
(to be handled by steam trap)
L = Condensate flow into flash tank in lbs/hr
P = Percentage of flash

EXAMPLE: Determine the condensate load of a flash tank with 5,000 lbs/hr of 100 psig condensate entering the flash tank held at 10 psig. From elementary calculations, the flash percentage is P = 10.5%. Using the formula:

$$Q = 5{,}000 - \frac{(5{,}000 \times 10.5)}{100} = 4{,}475 \text{ lbs/hr}$$

Due to the importance of energy conservation and operation against back pressure, the trap best suited for flash steam service is the inverted bucket type with large bucket vent. In addition, the IB operates intermittently while venting air and CO2 at steam temperature.

In some cases, the float and thermostatic type trap is an acceptable alternative. One particular advantage of the F&T is its ability to handle heavy start-up air loads.

A third type of device that may be the preferred selection in many cases is the automatic differential condensate controller. It combines the best features of both the IB and F&T and is recommended for large condensate loads that exceed the separating capability of the flash tank.

### Safety Factor

The increased amount of condensate at start-up and the varying loads during operation accompanied by low pressure differential dictates a safety factor of 3:1 for trapping flash tanks. Flash steam tank with live steam make-up, showing recommended fittings and connections, were shown in Figure CG-57. The check valves in the incoming lines prevent waste of flash when a line is not in use. The by-pass is used when flash steam cannot be used. Relief valves prevent pressure from building up and interfering with the operation of the high pressure steam traps. The reducing valve reduces the high pressure steam to the same pressure as the flash, so they can be combined for process work or heating.

## HOW TO TRAP FLASH TANKS: SUMMARY

### Installation

Condensate return lines contain both flash steam and condensate. To recover the flash steam, the return header runs to a flash tank, where the condensate is drained, and steam is then piped from the flash tank to points of use, Fig. CG-57. Since a flash tank causes back pressure on the steam traps discharging into the tank, these traps should be selected to ensure their capability to work against back pressure and have sufficient capacity at the available differential pressures.

Condensate lines should be pitched toward the flash tank, and where more than one line feeds into a flash tank, each line should be fitted with a swing check valve. Then, any line not in use will be isolated from the others and will not be fed in reverse with resultant wasted flash steam. If the trap is operating at low pressure, gravity drainage to the condensate receiver should be provided.

Generally, the location chosen for the flash tank should meet the requirement for maximum quantity of flash steam and minimum length of pipe.

Condensate lines, the flash tank, and the low pressure steam lines should be insulated to prevent waste of flash through radiation. The fitting of a spray nozzle on the inlet pipe inside the tank is not recommended. It may become choked, stop the flow of condensate, and produce a back pressure to the traps.

Low pressure equipment using flash steam should be individually trapped and discharged to a low pressure return. Large volumes of air need to be vented from the flash tank; therefore, a thermostatic air vent should be used to remove the air and keep it from passing through the low pressure system.

### Flash Tank Dimensions

The flash tank can usually be conveniently constructed from a piece of large diameter piping with the bottom ends welded or bolted in position. The tank should be mounted vertically. A steam outlet is required at the top and a condensate outlet at the bottom. The condensate inlet connection should be 6″–8″ above the condensate outlet.

The important dimension is the inside diameter. This should be such that the upward velocity of flash to the outlet is low enough to ensure that the amount of water carried over with the flash is small. If the upward velocity is kept low, the height of the tank is not important, but good practice is to use a height of 2′–3′.

It has been found that a steam velocity of about 10′ per second inside the flash tank will give good separation of steam and water. On this basis, proper inside diameters for various quantities of flash steam have been calculated; the results are plotted in

**Table CG-1. Properties of Saturated Steam**
(Abstracted from Keenan and Keyes, THERMODYNAMIC PROPERTIES OF STEAM, by permission of John Wiley & Sons, Inc.)

| | Col. 1 Gauge Pressure | Col. 2 Absolute Pressure (psia) | Col. 3 Steam Temp. (F°) | Col. 4 Heat of Sat. Liquid (Btu/lb) | Col. 5 Latent Heat (Btu/lb) | Col. 6 Total Heat of Steam (Btu/lb) | Col. 7 Specific Volume of Sat. Liquid (cu ft/lb) | Col. 8 Specific Volume of Sat. Steam (cu ft/lb) |
|---|---|---|---|---|---|---|---|---|
| Inches of Vacuum | 29.743 | 0.08854 | 32.00 | 0.00 | 1075.8 | 1075.8 | 0.096022 | 3306.00 |
| | 29.515 | 0.2 | 53.14 | 21.21 | 1063.8 | 1085.0 | 0.016027 | 1526.00 |
| | 27.886 | 1.0 | 101.74 | 69.70 | 1036.3 | 1106.0 | 0.016136 | 333.60 |
| | 19.742 | 5.0 | 162.24 | 130.13 | 1001.0 | 1131.0 | 0.016407 | 73.52 |
| | 9.562 | 10.0 | 193.21 | 161.17 | 982.1 | 1143.3 | 0.016590 | 38.42 |
| | 7.536 | 11.0 | 197.75 | 165.73 | 979.3 | 1145.0 | 0.016620 | 35.14 |
| | 5.490 | 12.0 | 201.96 | 169.96 | 976.6 | 1146.6 | 0.016647 | 32.40 |
| | 3.454 | 13.0 | 205.88 | 173.91 | 974.2 | 1148.1 | 0.016674 | 30.06 |
| | 1.418 | 14.0 | 209.56 | 177.61 | 971.9 | 1149.5 | 0.016699 | 28.04 |
| PSIG | 0.0 | 14.696 | 212.00 | 180.07 | 970.3 | 1150.4 | 0.016715 | 26.80 |
| | 1.3 | 16.0 | 216.32 | 184.42 | 967.6 | 1152.0 | 0.016746 | 24.75 |
| | 2.3 | 17.0 | 219.44 | 187.56 | 965.5 | 1153.1 | 0.016768 | 23.39 |
| | 5.3 | 20.0 | 227.96 | 196.16 | 960.1 | 1156.3 | 0.016830 | 20.09 |
| | 10.3 | 25.0 | 240.07 | 208.42 | 952.1 | 1160.6 | 0.016922 | 16.30 |
| | 15.3 | 30.0 | 250.33 | 218.82 | 945.3 | 1164.1 | 0.017004 | 13.75 |
| | 20.3 | 35.0 | 259.28 | 227.91 | 939.2 | 1167.1 | 0.017078 | 11.90 |
| | 25.3 | 40.0 | 267.25 | 236.03 | 933.7 | 1169.7 | 0.017146 | 10.50 |
| | 30.3 | 45.0 | 274.44 | 243.36 | 928.6 | 1172.0 | 0.017209 | 9.40 |
| | 40.3 | 55.0 | 287.07 | 256.30 | 919.6 | 1175.9 | 0.017325 | 7.79 |
| | 50.3 | 65.0 | 297.97 | 267.50 | 911.6 | 1179.1 | 0.017429 | 6.66 |
| | 60.3 | 75.0 | 307.60 | 277.43 | 904.5 | 1181.9 | 0.017524 | 5.82 |
| | 70.3 | 85.0 | 316.25 | 286.39 | 897.8 | 1184.2 | 0.017613 | 5.17 |
| | 80.3 | 95.0 | 324.12 | 294.56 | 891.7 | 1186.2 | 0.017696 | 4.65 |
| | 90.3 | 105.0 | 331.36 | 302.10 | 886.0 | 1188.1 | 0.017775 | 4.23 |
| | 100.0 | 114.7 | 337.90 | 308.80 | 880.0 | 1188.8 | 0.017850 | 3.88 |
| | 110.3 | 125.0 | 344.33 | 315.68 | 875.4 | 1191.1 | 0.017922 | 3.59 |
| | 120.3 | 135.0 | 350.21 | 321.85 | 870.6 | 1192.4 | 0.017991 | 3.33 |
| | 125.3 | 140.0 | 353.02 | 324.82 | 868.2 | 1193.0 | 0.018024 | 3.22 |
| | 130.3 | 145.0 | 355.76 | 327.70 | 865.8 | 1193.5 | 0.018057 | 3.11 |
| | 140.3 | 155.0 | 360.50 | 333.24 | 861.3 | 1194.6 | 0.018121 | 2.92 |
| | 150.3 | 165.0 | 365.99 | 338.53 | 857.1 | 1195.6 | 0.018183 | 2.75 |
| | 160.3 | 175.0 | 370.75 | 343.57 | 852.8 | 1196.5 | 0.018244 | 2.60 |
| | 180.3 | 195.0 | 379.67 | 353.10 | 844.9 | 1198.0 | 0.018360 | 2.34 |
| | 200.3 | 215.0 | 387.89 | 361.91 | 837.4 | 1199.3 | 0.018470 | 2.13 |
| | 225.3 | 240.0 | 397.37 | 372.12 | 828.5 | 1200.6 | 0.018602 | 1.92 |
| | 250.3 | 265.0 | 406.11 | 381.60 | 820.1 | 1201.7 | 0.018728 | 1.74 |
| | | 300.0 | 417.33 | 393.84 | 809.0 | 1202.8 | 0.018896 | 1.54 |
| | | 400.0 | 444.59 | 424.00 | 780.5 | 1204.5 | 0.019340 | 1.16 |
| | | 450.0 | 456.28 | 437.20 | 767.4 | 1204.6 | 0.019547 | 1.03 |
| | | 500.0 | 467.01 | 449.40 | 755.0 | 1204.4 | 0.019748 | 0.93 |
| | | 600.0 | 486.21 | 471.60 | 731.6 | 1203.2 | 0.02013 | 0.77 |
| | | 900.0 | 531.98 | 526.60 | 668.8 | 1195.4 | 0.02123 | 0.50 |
| | | 1200.0 | 567.22 | 571.70 | 611.7 | 1183.4 | 0.02232 | 0.36 |
| | | 1500.0 | 596.23 | 611.60 | 556.3 | 1167.9 | 0.02346 | 0.28 |
| | | 1700.0 | 613.15 | 636.30 | 519.6 | 1155.9 | 0.02428 | 0.24 |
| | | 2000.0 | 635.82 | 671.70 | 463.4 | 1135.1 | 0.02565 | 0.19 |
| | | 2500.0 | 668.13 | 730.60 | 360.5 | 1091.1 | 0.02860 | 0.13 |
| | | 2700.0 | 679.55 | 756.20 | 312.1 | 1068.3 | 0.03027 | 0.11 |
| | | 3206.2 | 705.40 | 902.70 | 0.0 | 902.7 | 0.05053 | 0.05 |

Chart CG-22. This curve gives the smallest recommended internal diameters. If it is more convenient, a larger tank may be used.

Chart CG-22 does not take into consideration pressure—only weight. Although volume of steam and upward velocity are less at a higher pressure, because steam is denser, there is an increased tendency for priming. Thus it is recommended that, regardless of pressure, Chart CG-22 be used to find the internal diameter.

CHAPTER

# 2

# GRAVITY FLOW OF LIQUIDS IN PIPES

My favorite method for designing gravitational flow of Newtonian liquids in mechanical and civil piping systems is covered here. It is very good for designing steam condensate return systems, gravity-flow portions of cooling tower water systems, and industrial process drain systems (but not sanitary waste drains, which are designed to local building codes.)

It is quick, a widely accepted old standard, easy to understand, and accurate enough for any purposes I've encountered so far. It combines a simple equation with a couple of normalized graphs, which are included.

*If you will adopt my design philosophy for gravity flow piping, and ensure your design is followed and its intent is realized in the constructed system, your gravity systems will work as planned every time.*

To use this method, you can work a problem in either direction:

1. Start forward, with a known or specified necessary maximum flowrate and specified allowable maximum fluid depth in the pipe (i.e., the "% full"), and use the method to find the required pipe diameter, slope, and actual flowing velocity.
2. Specify some combination of pipe size, slope, velocity, and "% full" parameters, and back-calculate the resulting flowrate. We will look at a few simple illustrative example calculations later in this section.

## Equation and Nomographs

**The basic equation I like to use is the Manning (Chezy-Manning) equation:**

### Chezy-Manning Equation

**Q = (A) × (1.486/*n*) × [(HR) ^ 0.6667] × [$S^{0.5}$]**

**where:**

**Q** = gravity flowrate in sloped pipe, **100% full** of liquid, in units = **$ft^3$/sec**

**A** = 100% of pipe or conduit flow area, the total cross section, in **$ft^2$** units

**n** = empirical roughness coefficient: perfectly smooth pipe, $n$ = 0.010; but recommended by engineers, ***n*** = 0.013 ($n$ = 0.013 is properly conservative)

**S = slope** of the pipe or conduit's bottom, i.e., the slope of flowpath's invert, in **feet of fall per foot of horizontal run *(example: for a 1% grade, which is a vertical fall of 1 ft per 100 ft of horizontal run, S = 1/100 = 0.01)***

**HR** = hydraulic radius in **feet** (see below)

When the pipe is not full, but has airspace above the liquid, we have *partial flow,* and instead of using inside diameter (ID) "D" for finding the pipe flow area, we have to use an equivalent factor called the "hydraulic radius" **(HR),**

*where, by definition:*

**q** = gravity flowrate in sloped pipe which is only **partly filled** by liquid, **$ft^3$/sec**

**a** = actual cross-sectional area of the flowing partial stream **"q",** in **$ft^2$** units

**D** = regular inside diameter of round pipe per usual, in **feet**

**Pwet** = actual length of wetted perimeter of the pipe or conduit corresponding to the partial flowrate **"q",** in **feet**

Then **HR = (a/Pwet);**

When the pipe actually flows **100% full,** the HR term reduces to HR = **D/4,** because then HR = (a)/(Pwet) = ($\pi D^2$/4) / ($\pi$D) = D/4. Note that the fully wetted perimeter is the inside circumference ($\pi$D).

The **Chezy-Manning** formula predicts flowrate of water in a pipe running full. The value it yields represents the flow one would expect for the given pipe diameter, slope, and pipe roughness.

When used with $n$ = 0.013, it is conservative, but not overly so. (Smooth pipe $n$ is 0.010, and since $n$ appears in the denominator and in the first power, "smooth pipe $n$ = 0.010" yields a difference of only 30% more flowrate than the recommended "average pipe condition $n$ = 0.013," for the same head loss, hardly an excessive factor of safety.)

By comparison, the ***Hazen-Williams "C-factor"*** (familiar to most mechanical piping folks thanks to the excellent Cameron Pump Division "Condensed Hydraulic Data" handbook from Ingersoll-Rand) uses **C = 100** for "old design value" steel pipe roughness, and **C = 140** for "good, clean, new" steel pipe. It turns out that ***C = 140 new pipe*** predicts a head loss of only ***54%*** of the head loss derived from a ***C = 100 old pipe*** carrying the same flowrate. This works out to an equivalent statement that **C = 140** produces about **50%** more flowrate in a pipe of given head loss (slope) than does the Hazen-Williams recommended **C = 100.** Compare the Hazen-Williams **"50%"** more flow through new pipe" to the Manning **"30%** more flow" when all other factors are equal, and you see what I mean.

*(The most-accurate Darcy-Weisbach formula, which I use for all forced Newtonian fluid flow pipe calculations, requires calculation of the exact experimentally-proven friction factor "f", by Colebrook's equation or from the equivalent Moody diagram, so "safety factor" is more of a controlled parameter when using the Darcy-Weisbach formula.)*

Bottom line: for gravity flow, Use Manning's formula with ***n*** = 0.013 *minimum.* (I use $n$ = 0.013 always.)

**<u>Example Calculation:</u> Chezy-Manning.**
**<u>Given:</u>**

**Pipe Size** = 18-inch std. thickness (ID = 17.25 in)
**Slope** = 1% (run 100 ft, fall 1 ft.)
**Gravity flow, average pipe, running full.** <u>**Find:**</u> full-pipe flowrate in <u>gpm.</u> <u>**Solution:**</u>
$\mathbf{Q = (A) \times (1.486/n) \times [(HR)\ \hat{}\ 0.6667] \times [S^{0.5}]}$
$\boldsymbol{n} = 0.013$
$\mathbf{A} = (\pi D^2/4) = [(\pi/4)(17.25 \text{ in.})^2] / 144 =$
$= \mathbf{1.623\ ft^2}$
**HR** = full circle, D/4 = 17.25 in (4 × 12) =
= **0.3594 ft**
**S** = 1/100 = **0.01**
**Q** = (1.623) × (1.486/0.013) × [(0.3594) ^ 0.6667] × [(0.01) ^ 0.5] =
= (1.623)(114.308)(0.50548)(0.1) =
= 9.37776 $ft^3$/sec;
**gpm** = (9.37776 $ft^3$/sec) × (60 sec/min) × (7.4805 gal/$ft^3$)
= 4,209; gpm;
**so the full-pipe flowrate = <u>4,200 gpm.</u>**

## Designing for Partly Filled Pipe Flow: the Chezy – Manning Nomographs

Gravity piping is designed such that under maximum flowrate conditions, the pipe is not full, but has an airspace along the top of the pipe. The airspace is tall enough (has sufficient vertically upward radial dimension) to prevent slugs from forming. Slugs would cause local vacuums to form, resulting in water hammer. For the same reason, to prevent pressure differentials from forming, both pipe ends (at least), the entry and exit ends, are vented to the atmosphere (or, rarely, to closed system vapor or gas accumulators in special chemical plant arrangements). Periodic line vents and special vents for abrupt turns, tees, etc. are added to the basic end-of-line vents when needed, to ensure smooth, even, steady flow, without waves which might build up into slugs. These are the secrets for designing trouble-free gravity flow piping.

My strongest suggestion is that the pipe be sized and sloped such that the maximum flow it is intended to carry shall fill no more than 50% of the total pipe flow area, that is, that the water depth not exceed half of the pipe inside diameter, ever. If the reasoning behind this recommendation does not seem self-evident, please read and reread the preceding paragraph until you fully understand and appreciate this concept. We must not let the water fill the pipe, else we will get water hammer, possibly destructive water hammer. Only vented, partially filled and slug-free pipe flow at atmospheric pressure and reasonably mild velocities will prevent water hammer. *Note: this is vitally important in very large gravity flow water piping design, because the inertial forces of potential water hammer are so large in absolute magnitude, and in steam condensate return piping, because of the potential danger associated with condensate flashing to steam inside closed piping or non-vented containment (see Chapter #1, Steam Hammer).*

<u>This is where the Chezy-Manning nomographs come into play.</u> They plot three different normalized parametric ratios which permit us to design for controlled situations over the required range of pipe flows, <u>all</u> versus the ratio of flowing depth **"d"** to ID **"D".** (Example: if the pipe ID = 12 in and the flowing stream is <u>6 in deep</u> (d = 6), then **d/D** = (6/12) = 0.50, and the pipe is *"flowing half-full."* If flowing depth **"d"** is only <u>4 in,</u> then the **d/D ratio** = (4/12) = 0.33, and the pipe flows *"one-third full,"* or *"33% full."*

The three parametric ratios that are plotted against **"d/D"** are:

- **"q/Q,"** actual flowrate vs. full-pipe flow; see **Figure 2-1**
- **"a/A,"** actual stream to full-pipe area; see **Figure 2-2**
- **"v/V,"** actual stream to full-pipe velocity which can exceed 1.00 (above d/D = .91); see **Figure 2-3**

Those three plots were made by my ***<u>TK Solver</u>*** program, using data points lifted from a published all-in-one single-graph plot of all the parameters, plus more, given as Figure 10 on page 18–17 of my fifth edition of Crocker and King's excellent, well-distributed standard reference, ***Piping Handbook*** (McGraw-Hill). I always found the published plot too small (and book too thick) for three-figure accuracy in reading it, and too "busy" to suit me, so I made the enlarged stand–alone plots given herein, **Figures 2-1, 2-2, and 2-3.**

Now to demonstrate their use by continuing the previous example:

**<u>Example Calculation, (cont'd):</u>**
**<u>Previously Given:</u>**

**Pipe size** = 18-inch std. Thickness (ID = 17.25 in.)
**Slope** = 1% (run 100 ft, fall 1 ft)
**Gravity flow, average pipe, running full.** <u>**Previously Found:**</u>
**Q = 4,200 gpm** (= 9.37776 $ft^3$/sec) in the full-flowing pipe.

If our task had been to design a pipe to carry ***<u>0–1,400 gpm</u>*** of water, gravity flow, in a pipe having a slope no greater than 1%, here is a procedure to follow:

1. First, since **we want d/D ≦ 0.50, for the pipe to flow no more than 50% full at maximum gpm,** enter <u>**Figure 2-1 @ d/D = 0.50.**</u> Read "ratio of actual to full-pipe flowrate," which is the ratio q/Q ; the value I read is ≌ 0.40.
2. Since our desired maximum flowrate "q" = 1,400 gpm, calculate Q = (1,400/0.40) =
= 3,500 GPM. So the pipe size we select must carry **at least 3,500 gpm** at a **1% slope (with S=0.01,** and the usual $\boldsymbol{n}$ = 0.013).
3. The next step is <u>trial & error,</u> by necessity, because we cannot independently find the values of "A" and "HR" from what we are given. The way I like to do it is to pick up my trusty little <u>Cameron's Condensed Hydraulic Data</u> booklet, which is always by my side on the job, and <u>start looking through the C = 100 Steel Pipe columns, by increasing pipe size, until I hit the first one whose flow, at head loss = 1 ft per 100 ft, equals or slightly exceeds 3500 gpm.</u> ***(A gravity slope of 1% in a full C = 100 pipe is just about the same thing as a friction head loss of 1 ft of water per 100 ft of n = 0.013 pipe run.)***

   This is a quick and easy way to go. When I do this, I find that a 16-in pipe is too small, having a head loss at 3500 gpm = **1.31 ft water/100 ft pipe,** exceeding my **1.00 limit** by too much. The next size, **18-in,** fits the bill with a head loss per hundred feet of pipe run = 0.735 ft of water. So my <u>tentative</u> selection is for 18-in nominal pipe size.
4. The next step is to find **Q** using the **Chezy-Manning** formula for **18-in pipe, 1% slope flowing full.** Since we have already done this in the first part of our example, we know that Q = 4,200 gpm. This step verifies that 18-in pipe is indeed adequate. (You can go back and check the actual capacity of the

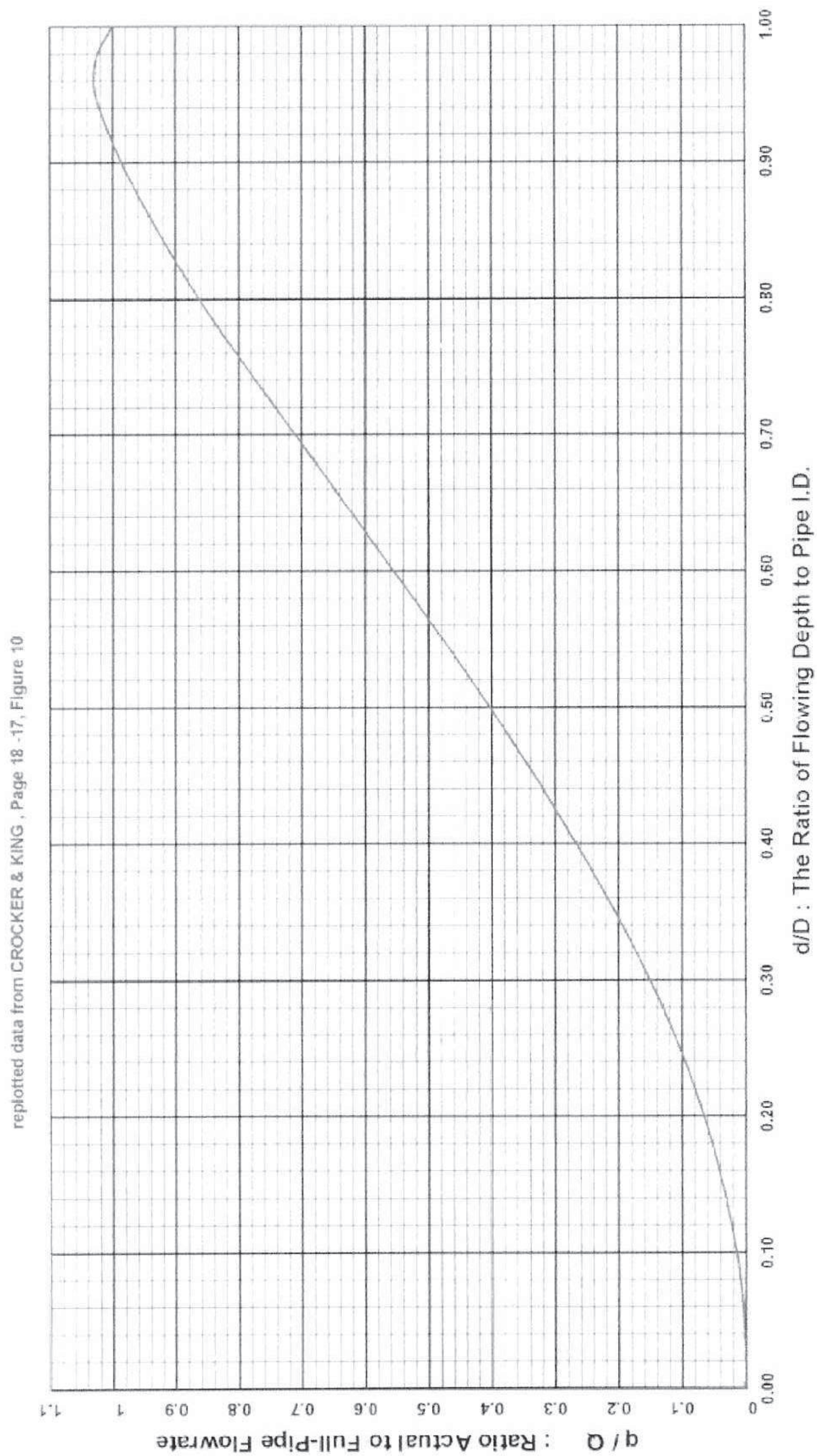



**FIGURE 2-1 NORMALIZED FLOWRATE**

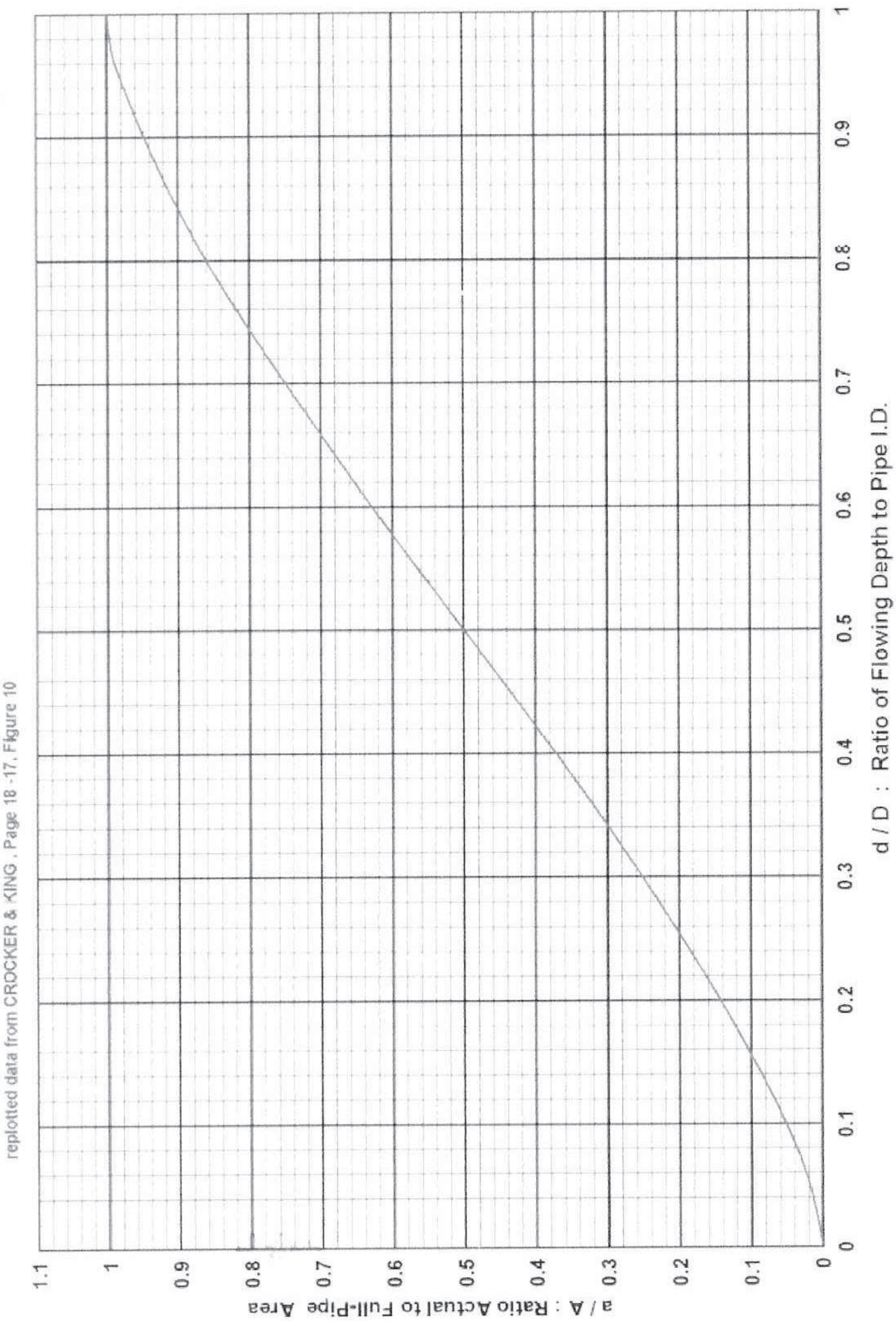



**FIGURE 2-2 NORMALIZED AREA**

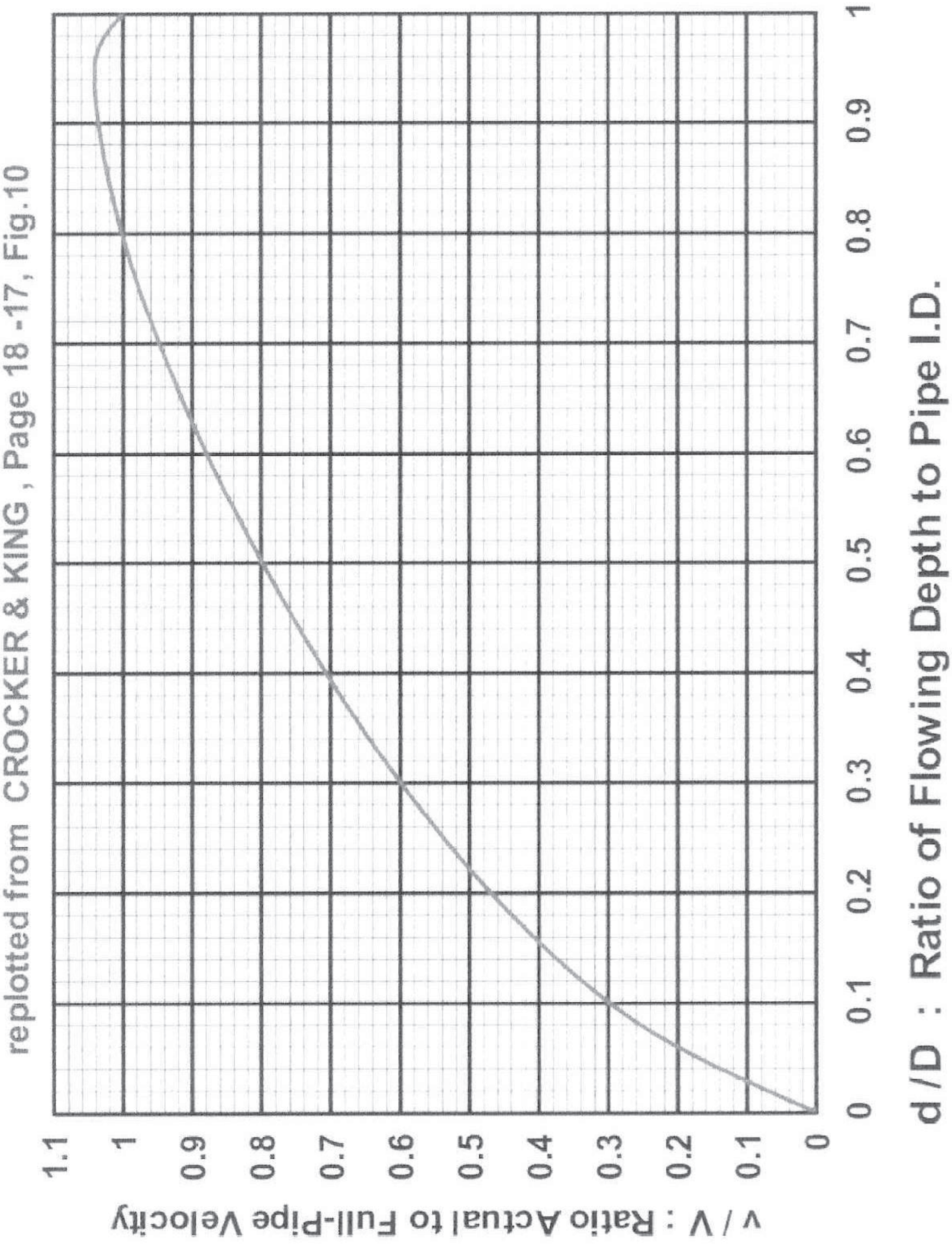



**FIGURE 2-3 NORMALIZED VELOCITY**

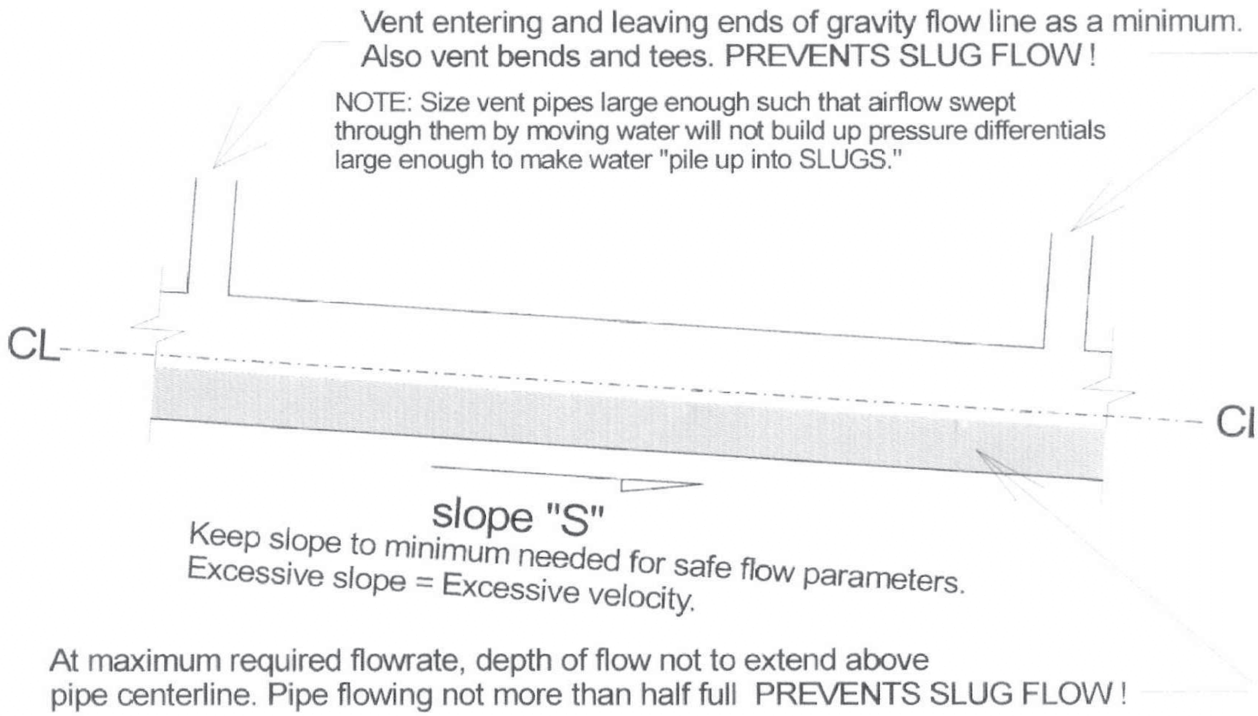



**FIGURE 2-4 SIZE, SLOPE & VENT!!**

16-in pipe for comparison. Maybe it is big enough after all; just stay below the magic d/D = 0.50 limit. I will not bother with that here, but I would in practice.)

5. If I wish, I can now find the actual flowing velocity very easily:

   @ Q=4,200 gpm = 9.37776 $ft^3$/sec in the full 18-in pipe, for which A = 1.623 $ft^2$, we use the simple relationship for full-pipe velocity V = Q/A, and obtain:

   Full-pipe V = (9.37776/1.623) = 5.778 ft/sec

   The desired actual value of **q/Q** in our problem statement = *1,400*/4,200 = **0.333**

   Now in **Figure 2-1** @ **q/Q** = 0.333, read the key parameter **d/D** = 0.448 ≌ 0.45 ; place this value in **Figure 2-3** and read **v/V** = 0.75,

   and obtain the **actual flowing velocity value v = (0.75)(5.778) = 4.33 ft/sec.**

   Similar shenanigans will yield actual cross-sectional flowing area (via **Figure 2-2**), should you need to find it.

## Postscripts

If you are fortunate enough to own or have free access to the declarative rules-based scientific/engineering mathematics solving program named **"TK Solver,** which is a truly great calculating machine that every engineer should discover, I think you will appreciate my next offering.

Namely, the **TK Solver** "Rule Sheet" and "Variable Sheet" for the entire **Chezy-Manning equation, and all three nomographs which go with it: (q/Q), (a/A), and (v/V) versus (d/D).** So if you set up a TK Solver application using this "source code" for your Rules and Variables sheets, you can solve **any Chezy-Manning gravity flow problem** very, very quickly and accurately, without having to make a single calculation or having to look up anything in **Figures 2-1, 2-2, or 2-3.**

The source code filename which I used, shown in the "footer" of the one rule sheet page and two variable sheet pages, is MANNING_GRAV_FLOW_SOLVER. The code notes make the material self-explanatory. Also, I illustrated the use of MANNING_GRAV_FLOW_SOLVER by means of a sample problem (two pages of variable sheet output whose footers read "MANNING_EXAMPLE"), included with the source code pages just described. The sample problem is for 10-in pipe size, 2% slope, and (d/D) = 0.5; the comments on its first page serve to explain it fully.

Let me mention one other thing. Although I have absolutely no vested interest in the software, I do use the **TK Solver** program a lot, and have made a good bit of profit using it to crunch numbers on engineering consulting contracts, thanks to its speed and absolute utility. It list solves, forward solves, back-solves, plots, uses implicit formulas as simply as explicit ones, and does a bunch more. I am publicizing it herein because it is a darn fine program and deserves a mountain of meritorious praise. It turns a PC into the first-rate engineering computation and design tool it should be, and serious engineers really ought to look into it. *I got my copy from UTS Software, Universal Technical Systems, Inc., 1220 Rock Street, Rockford, IL 61101, USA.* As this is written their World Wide Web address is http://www.uts.com and they are in the UK and India also.

Last but not least, Figure 2-4 is included as a reminder to VENT THOSE GRAVITY LINES! Good luck, my friend.

**MANNING_GRAV_FLOW_SOLVER**

| St | Rule |
|---|---|
| | ; Manning Formulae for Partially Filled Gravity Flow of WATER in Pipes. |
| | ; Reference Crocker & King PIPING HANDBOOK 5th Ed. page 18-16. |
| | ; Also Ref. Vennard FLUID MECHANICS 4th Ed. page 353. |
| | |
| * | PipeID / 12 = D |
| * | pi() * (D^2) / 4 = A |
| * | D / 4 = Rh |
| * | n = .013 |
| * | FALL / RUN = So |
| * | Q = A * (1.486/n) * ((Rh)^0.666667) * ((So)^0.5) |
| * | GPM = Q * 448.83117 |
| * | b1q = -.0003112559 |
| * | b2q = .1379154408 * (d/D) |
| * | b3q = -3.1282221134 * ((d/D)^2) |
| * | b4q = 41.4405104262 * ((d/D)^3) |
| * | b5q = -165.6683283258 * ((d/D)^4) |
| * | b6q = 360.7764665962 * ((d/D)^5) |
| * | b7q = -444.35594284 * ((d/D)^6) |
| * | b8q = 289.1863133029 * ((d/D)^7) |
| * | b9q = -77.3857162683 * ((d/D)^8) |
| * | (q / Q) = b1q+b2q+b3q+b4q+b5q+b6q+b7q+b8q+b9q |
| * | Actual_gpm = q * 448.83117 |
| * | Flow_Depth = 12 * d |
| * | Depth_Fraction = d/D |
| * | b1v = -.0006638638 |
| * | b2v = 4.5622912608 * (d/D) |
| * | b3v = -21.9449956396 * ((d/D)^2) |
| * | b4v = 92.9784954785 * ((d/D)^3) |
| * | b5v = -266.0739418982 * ((d/D)^4) |
| * | b6v = 486.7077053627 * ((d/D)^5) |
| * | b7v = -539.5274178442 * ((d/D)^6) |
| * | b8v = 327.6064709394 * ((d/D)^7) |
| * | b9v = -83.3058692703 * ((d/D)^8) |
| * | V = Q / A |
| * | (v / V) = b1v+b2v+b3v+b4v+b5v+b6v+b7v+b8v+b9v |
| * | Actual_velocity = v * 1.00 |
| | |

Rule Sheet

MANNING_GRAV_FLOW_SOLVER

| Sta | Input | Name | Output | Unit | Comment |
|---|---|---|---|---|---|
| | | | | | Gravity Flow in Partially Filled Pipes. |
| | | | | | "Save As" some job-related filename |
| | | | | | before using ; keep this file pristine. |
| | | | | | USER INPUTS SHALL BE : |
| | | | | | PipeID , FALL , RUN , Depth_Fraction . |
| | | | | | TK will find all the rest. |
| | | | | | << SOLVE BY PRESSING F9 >> |
| | 0 | PipeID | | in | User lookup & input always. |
| | | D | | ft | "PipeID" converted to feet by TK. |
| | | A | | ft^2 | 100% of pipe cross-section area. |
| | | Rh | | ft | Hydraulic radius of full pipe : (D/4). |
| | | n | | | Manning roughness coeff ; .01<n<.013. |
| | | | | | VALUE .013 IS BUILT-IN , LEAVE AS-IS! |
| | | | | | (.01=smooth; .013=std. value for calc.) |
| | 0 | FALL | | ft | User input overall decrease of invert ; |
| | 0 | RUN | | ft | Input horizontal straight run of pipe. |
| | | So | | ft/ft | Invert slope calc by TK. ("FALL/RUN"). |
| | | Q | | ft^3/s | TK solution of Manning Eq. for 100% |
| | | | | | full-pipe gravity flow of water in pipe. |
| | | GPM | | gal/min | Full-pipe"Q" convert. to gal/min by TK. |
| | | b1q | | | (curve-fit parameter : ditto all bxq.) |
| | | b2q | | | |
| | | d | | ft | TK solution of curve-fit equation for |
| | | | | | depth of flow in pipe (free surface of |
| | | | | | stream to pipe invert perp. dim.) |
| | | b3q | | | |
| | | b4q | | | |
| | | b5q | | | |

| Sta | Input | Name | Output | Unit | Comment |
|---|---|---|---|---|---|
| | | b6q | | | |
| | | b7q | | | |
| | | b8q | | | |
| | | b9q | | | |
| | | q | | ft^3/s | TK solution of part-filled pipe flowrate. |
| | | Actual_gpm | | gal/min | "q" converted to gal/min by TK. |
| | | Flow_Depth | | in | "d" converted to inches by TK. |
| | 0 | Depth_Fraction | | in/in | Decimal fraction "d/D". |
| | | b1v | | | (curve-fit parameter : ditto all bxv.) |
| | | b2v | | | |
| | | b3v | | | |
| | | b4v | | | |
| | | b5v | | | |
| | | b6v | | | |
| | | b7v | | | |
| | | b8v | | | |
| | | b9v | | | |
| | | V | | ft/s | Full pipe velocity = "Q/A". |
| | | v | | ft/s | Calc by TK : actual velocity of the |
| | | | | | flow in part-filled pipe. |
| | | Actual_velocity | | ft/s | "v" given unambiguous name. |

Variable Sheet

MANNING_EXAMPLE

| Sta | Input | Name | Output | Unit | Comment |
|---|---|---|---|---|---|
| | | | | | Gravity Flow in Partially Filled Pipes. |
| | | | | | "Save As" some job-related filename |
| | | | | | before using ; keep this file pristine. |
| | | | | | USER INPUTS SHALL BE : |
| | | | | | PipeID , FALL , RUN , Depth_Fraction . |
| | | | | | TK will find all the rest. |
| | | | | | << SOLVE BY PRESSING F9 >> |
| | 10.02 | PipeID | | in | User lookup & input always. |
| | | D | .835 | ft | "PipeID" converted to feet by TK. |
| | | A | .547599234474786 | ft^2 | 100% of pipe cross-section area. |
| | | Rh | .20875 | ft | Hydraulic radius of full pipe : (D/4). |
| | | n | .013 | | Manning roughness coeff ; .01<n<.013. |
| | | | | | VALUE .013 IS BUILT-IN , LEAVE AS-IS! |
| | | | | | (.01=smooth; .013=std. value for calc.) |
| | 1 | FALL | | ft | User input overall decrease of invert ; |
| | 50 | RUN | | ft | Input horizontal straight run of pipe. |
| | | So | .02 | ft/ft | Invert slope calc by TK. ("FALL/RUN"). |
| | | Q | 3.11509084600994 | ft^3/s | TK solution of Manning Eq. for 100% |
| | | | | | full-pipe gravity flow of water in pipe. |
| | | GPM | 1398.14986907093 | gal/min | Full-pipe"Q" convert. to gal/min by TK. |
| | | b1q | -.0003112559 | | (curve-fit parameter : ditto all bxq.) |
| | | b2q | .0689577204 | | |
| | | d | .4175 | ft | TK solution of curve-fit equation for |
| | | | | | depth of flow in pipe (free surface of |
| | | | | | stream to pipe invert perp. dim.) |
| | | b3q | -.78205552835 | | |
| | | b4q | 5.180063803275 | | |
| | | b5q | -10.3542705203625 | | |

| Sta | Input | Name | Output | Unit | Comment |
|---|---|---|---|---|---|
| | | b6q | 11.2742645811313 | | |
| | | b7q | -6.943061606875 | | |
| | | b8q | 2.25926807267891 | | |
| | | b9q | -.302287954173047 | | |
| | | q | 1.24780356627565 | ft^3/s | TK solution of part-filled pipe flowrate. |
| | | Actual_gpm | 560.053134581672 | gal/min | "q" converted to gal/min by TK. |
| | | Flow_Depth | 5.01 | in | "d" converted to inches by TK. |
| | .5 | Depth_Fraction | | in/in | Decimal fraction "d/D". |
| | | b1v | -.0006638638 | | (curve-fit parameter : ditto all bxv.) |
| | | b2v | 2.2811456304 | | |
| | | b3v | -5.4862489099 | | |
| | | b4v | 11.6223119348125 | | |
| | | b5v | -16.6296213686375 | | |
| | | b6v | 15.2096157925844 | | |
| | | b7v | -8.43011590381563 | | |
| | | b8v | 2.55942555421406 | | |
| | | b9v | -.325413551837109 | | |
| | | V | 5.68863257998833 | ft/s | Full pipe velocity = "Q/A". |
| | | v | 4.55338240551135 | ft/s | Calc by TK : actual velocity of the |
| | | | | | flow in part-filled pipe. |
| | | Actual_velocity | 4.55338240551135 | ft/s | "v" given unambiguous name. |

Variable Sheet

## HAZEN & WILLIAMS FLOW DATA for 60 DEG.F. WATER
### PIPES FLOWING FULL : adapted from *Cameron's Hydraulic Data*
### OLD STEEL PIPE : C = 100 : STANDARD WALL THICKNESS
### VELOCITY in FT./SEC. & HEAD LOSS in FT. WATER per 100 FT. of PIPE

| GPM | VEL. | H.L. | GPM | VEL. | H.L. | GPM | VEL. | H.L. |
|---|---|---|---|---|---|---|---|---|
| **FOR 1/2-INCH PIPE SIZE** | | | | | | **FOR 2-INCH PIPE SIZE** | | |
| 0.5 | 0.53 | 0.58 | **FOR 1-INCH PIPE SIZE** | | | 14 | 1.34 | 0.803 |
| 1.0 | 1.06 | 2.10 | 2 | 0.74 | 0.60 | 16 | 1.53 | 1.03 |
| 1.5 | 1.58 | 4.44 | 3 | 1.11 | 1.26 | 18 | 1.72 | 1.28 |
| 2.0 | 2.11 | 7.57 | 4 | 1.49 | 2.14 | 20 | 1.91 | 1.55 |
| 2.5 | 2.64 | 11.4 | 5 | 1.86 | 3.24 | 22 | 2.10 | 1.85 |
| 3.0 | 3.17 | 16.0 | 6 | 2.23 | 4.54 | 24 | 2.29 | 2.18 |
| 3.5 | 3.70 | 21.3 | 8 | 2.97 | 7.73 | 26 | 2.49 | 2.52 |
| 4.0 | 4.23 | 27.3 | 10 | 3.71 | 11.7 | 28 | 2.68 | 2.89 |
| 4.5 | 4.75 | 33.9 | 12 | 4.46 | 16.4 | 30 | 2.87 | 3.29 |
| 5.0 | 5.28 | 41.2 | 14 | 5.20 | 21.8 | 35 | 3.35 | 4.37 |
| 5.5 | 5.81 | 49.2 | 16 | 5.94 | 27.9 | 40 | 3.82 | 5.60 |
| 6.0 | 6.34 | 57.8 | 18 | 6.68 | 34.7 | 45 | 4.30 | 6.96 |
| 6.5 | 6.87 | 67.0 | 20 | 7.43 | 42.1 | 50 | 4.78 | 8.46 |
| 7.0 | 7.39 | 76.8 | 22 | 8.17 | 50.2 | 55 | 5.26 | 10.1 |
| 7.5 | 7.92 | 87.3 | 24 | 8.91 | 59.0 | 60 | 5.74 | 11.9 |
| 8.0 | 8.45 | 98.3 | 26 | 9.66 | 68.4 | 80 | 7.65 | 20.2 |
| 8.5 | 8.98 | 110.0 | 28 | 10.4 | 78.5 | 100 | 9.56 | 30.5 |
| | | | 30 | 11.1 | 89.2 | 120 | 11.5 | 42.7 |
| | | | 35 | 13.0 | 119.0 | 140 | 13.4 | 56.9 |
| **FOR 3/4-INCH PIPE SIZE** | | | | | | 160 | 15.3 | 72.8 |
| 1.5 | 0.90 | 1.13 | | | | 180 | 17.2 | 90.5 |
| 2.0 | 1.20 | 1.93 | **FOR 1-1/2 INCH PIPE SIZE** | | | 200 | 19.1 | 110.0 |
| 2.5 | 1.51 | 2.91 | 6 | 0.95 | 0.565 | | | |
| 3.0 | 1.81 | 4.08 | 7 | 1.10 | 0.751 | **FOR 2-1/2 INCH PIPE SIZE** | | |
| 3.5 | 2.11 | 5.42 | 8 | 1.26 | 0.962 | 20 | 1.34 | 0.654 |
| 4.0 | 2.41 | 6.94 | 9 | 1.42 | 1.20 | 30 | 2.01 | 1.39 |
| 4.5 | 2.71 | 8.63 | 10 | 1.58 | 1.45 | 40 | 2.68 | 2.36 |
| 5.0 | 3.01 | 10.5 | 12 | 1.89 | 2.04 | 50 | 3.35 | 3.56 |
| 6.0 | 3.61 | 14.7 | 14 | 2.21 | 2.71 | 60 | 4.02 | 4.99 |
| 7.0 | 4.21 | 19.6 | 16 | 2.52 | 3.47 | 70 | 4.69 | 6.64 |
| 8.0 | 4.82 | 25.0 | 18 | 2.84 | 4.31 | 80 | 5.36 | 8.50 |
| 9.0 | 5.42 | 31.1 | 20 | 3.15 | 5.24 | 90 | 6.03 | 10.6 |
| 10.0 | 6.02 | 37.8 | 22 | 3.47 | 6.25 | 100 | 6.70 | 12.8 |
| 11.0 | 6.62 | 45.1 | 24 | 3.78 | 7.34 | 120 | 8.04 | 18.0 |
| 12.0 | 7.22 | 53.0 | 26 | 4.10 | 8.51 | 140 | 9.38 | 23.9 |
| 13.0 | 7.82 | 61.5 | 28 | 4.41 | 9.76 | 160 | 10.7 | 30.7 |
| 14.0 | 8.43 | 70.5 | 30 | 4.73 | 11.1 | 180 | 12.1 | 38.1 |
| 16.0 | 9.63 | 90.2 | 40 | 6.30 | 18.9 | 200 | 13.4 | 46.3 |
| 18.0 | 10.80 | 112.0 | 50 | 7.88 | 28.5 | 240 | 16.1 | 66.4 |
| | | | 60 | 9.46 | 40.0 | 280 | 18.8 | 86.3 |
| | | | 70 | 11.0 | 53.2 | 300 | 20.1 | 98.1 |
| | | | 80 | 12.6 | 68.1 | 350 | 23.5 | 130 |
| | | | 90 | 14.2 | 84.7 | | | |
| | | | 100 | 15.8 | 103.0 | | | |

**HAZEN & WILLIAMS FLOW DATA for 60 DEG.F. WATER**
**PIPES FLOWING FULL : adapted from *Cameron's Hydraulic Data***
**OLD STEEL PIPE : C = 100 : STANDARD WALL THICKNESS**
**VELOCITY in FT./SEC. & HEAD LOSS in FT. WATER per 100 FT. of PIPE**

| GPM | VEL. | H.L. | GPM | VEL. | H.L. | GPM | VEL. | H.L. |
|---|---|---|---|---|---|---|---|---|
| **FOR 3-INCH PIPE SIZE** | | | | | | **FOR 10-INCH PIPE SIZE** | | |
| 30 | 1.30 | 0.481 | **FOR 6-INCH PIPE SIZE** | | | 800 | 3.25 | 0.660 |
| 40 | 1.74 | 0.820 | 100 | 1.11 | 0.162 | 900 | 3.66 | 0.821 |
| 50 | 2.17 | 1.24 | 200 | 2.22 | 0.584 | 1000 | 4.07 | 0.998 |
| 60 | 2.60 | 1.74 | 300 | 3.33 | 1.24 | 1100 | 4.48 | 1.19 |
| 70 | 3.04 | 2.31 | 400 | 4.44 | 2.11 | 1200 | 4.89 | 1.40 |
| 80 | 3.47 | 2.96 | 500 | 5.55 | 3.19 | 1300 | 5.30 | 1.62 |
| 90 | 3.91 | 3.67 | 550 | 6.11 | 3.80 | 1400 | 5.70 | 1.86 |
| 100 | 4.34 | 4.47 | 600 | 6.66 | 4.46 | 1500 | 6.10 | 2.11 |
| 110 | 4.77 | 5.53 | 650 | 7.22 | 5.17 | 1600 | 6.51 | 2.38 |
| 120 | 5.21 | 6.26 | 700 | 7.77 | 5.93 | 1700 | 6.92 | 2.66 |
| 130 | 5.64 | 7.26 | 750 | 8.34 | 6.74 | 1800 | 7.32 | 2.96 |
| 140 | 6.08 | 8.32 | 800 | 8.88 | 7.60 | 2000 | 8.14 | 3.60 |
| 150 | 6.51 | 9.48 | 850 | 9.45 | 8.50 | 2200 | 8.95 | 4.29 |
| 160 | 6.94 | 10.7 | 900 | 9.99 | 9.44 | 2400 | 9.76 | 5.04 |
| 200 | 8.68 | 16.1 | 1000 | 11.1 | 11.5 | 2600 | 10.6 | 5.84 |
| 300 | 13.0 | 34.1 | 1200 | 13.3 | 16.1 | 2800 | 11.4 | 6.70 |
| 400 | 17.4 | 58.0 | 1400 | 15.6 | 21.4 | 3000 | 12.2 | 7.61 |
| 550 | 23.9 | 105 | 1600 | 17.8 | 27.4 | 3200 | 13.0 | 8.58 |
| | | | 2000 | 22.2 | 41.4 | 3600 | 14.6 | 10.7 |
| **FOR 4-INCH PIPE SIZE** | | | 2400 | 26.7 | 58.0 | 4000 | 16.3 | 13.0 |
| 80 | 2.02 | 0.788 | | | | 5000 | 20.3 | 19.6 |
| 100 | 2.52 | 1.19 | **FOR 8-INCH PIPE SIZE** | | | 6000 | 24.4 | 27.5 |
| 120 | 3.02 | 1.67 | 400 | 2.57 | 0.554 | | | |
| 140 | 3.53 | 2.22 | 500 | 3.20 | 0.838 | **FOR 12-INCH PIPE SIZE** | | |
| 160 | 4.03 | 2.84 | 600 | 3.85 | 1.17 | 800 | 2.27 | 0.275 |
| 180 | 4.54 | 3.53 | 700 | 4.49 | 1.56 | 1000 | 2.84 | 0.415 |
| 200 | 5.05 | 4.29 | 800 | 5.13 | 1.99 | 1200 | 3.41 | 0.581 |
| 220 | 5.55 | 5.12 | 900 | 5.77 | 2.48 | 1400 | 3.98 | 0.773 |
| 240 | 6.05 | 6.01 | 1000 | 6.41 | 3.02 | 1600 | 4.55 | 0.990 |
| 260 | 6.55 | 6.97 | 1100 | 7.05 | 3.60 | 1800 | 5.11 | 1.23 |
| 280 | 7.06 | 8.00 | 1200 | 7.69 | 4.23 | 2000 | 5.68 | 1.50 |
| 300 | 7.57 | 9.09 | 1300 | 8.33 | 4.90 | 2200 | 6.25 | 1.78 |
| 320 | 8.07 | 10.2 | 1400 | 8.97 | 5.62 | 2400 | 6.81 | 2.10 |
| 400 | 10.1 | 15.5 | 1500 | 9.61 | 6.39 | 2600 | 7.38 | 2.43 |
| 500 | 12.6 | 23.4 | 1600 | 10.3 | 7.20 | 2800 | 7.95 | 2.78 |
| 600 | 15.1 | 32.8 | 1800 | 11.5 | 8.95 | 3000 | 8.52 | 3.17 |
| 700 | 17.6 | 43.6 | 2000 | 12.8 | 10.9 | 3500 | 9.95 | 4.21 |
| 800 | 20.2 | 55.8 | 2200 | 14.1 | 13.0 | 4000 | 11.4 | 5.39 |
| 900 | 22.7 | 69.3 | 2400 | 15.4 | 15.2 | 5000 | 14.2 | 8.15 |
| 1000 | 25.2 | 84.3 | 2600 | 16.7 | 17.7 | 6000 | 17.0 | 11.4 |
| 1100 | 27.7 | 101 | 2800 | 18.0 | 20.3 | 8000 | 22.7 | 19.4 |
| | | | 3000 | 19.2 | 23.0 | 10000 | 28.4 | 29.4 |
| | | | 4000 | 25.6 | 39.2 | 12000 | 34.1 | 41.2 |
| | | | 5000 | 32.0 | 59.3 | | | |

## HAZEN & WILLIAMS FLOW DATA for 60 DEG.F. WATER

**PIPES FLOWING FULL : adapted from *Cameron's Hydraulic Data***

**OLD STEEL PIPE : C = 100 : STANDARD WALL THICKNESS**

**VELOCITY in FT./SEC. & HEAD LOSS in FT. WATER per 100 FT. of PIPE**

| GPM | VEL. | H.L. | GPM | VEL. | H.L. | GPM | VEL. | H.L. |
|---|---|---|---|---|---|---|---|---|
| **FOR 14-INCH PIPE SIZE** | | | | | | **FOR 24-INCH PIPE SIZE** | | |
| 1000 | 2.33 | 0.256 | **FOR 18-INCH PIPE SIZE** | | | 2000 | 1.42 | 0.051 |
| 1100 | 2.56 | 0.306 | 1600 | 2.21 | 0.173 | 2400 | 1.70 | 0.071 |
| 1200 | 2.79 | 0.359 | 1800 | 2.49 | 0.215 | 2700 | 1.92 | 0.089 |
| 1300 | 3.02 | 0.416 | 2000 | 2.77 | 0.261 | 3100 | 2.21 | 0.114 |
| 1400 | 3.26 | 0.477 | 2500 | 3.46 | 0.394 | 3400 | 2.41 | 0.136 |
| 1500 | 3.49 | 0.542 | 3000 | 4.15 | 0.553 | 3800 | 2.70 | 0.113 |
| 1600 | 3.72 | 0.611 | 3500 | 4.85 | 0.735 | 4200 | 2.99 | 0.200 |
| 1800 | 4.19 | 0.760 | 4000 | 5.54 | 0.941 | 4500 | 3.20 | 0.226 |
| 2000 | 4.65 | 0.924 | 4500 | 6.23 | 1.17 | 4800 | 3.41 | 0.255 |
| 2500 | 5.81 | 1.40 | 5000 | 6.92 | 1.42 | 5200 | 3.69 | 0.298 |
| 3000 | 6.98 | 1.96 | 6000 | 8.31 | 1.99 | 5500 | 3.91 | 0.329 |
| 4000 | 9.31 | 3.32 | 8000 | 11.1 | 3.39 | 5900 | 4.19 | 0.377 |
| 6000 | 14.0 | 7.05 | 10000 | 13.8 | 5.12 | 6200 | 4.41 | 0.413 |
| 8000 | 18.6 | 12.0 | 12000 | 16.6 | 7.18 | 6500 | 4.62 | 0.450 |
| 10000 | 23.3 | 18.1 | 14000 | 19.4 | 9.55 | 6900 | 4.90 | 0.502 |
| 15000 | 34.9 | 38.4 | 20000 | 27.7 | 18.5 | 7600 | 5.40 | 0.603 |
| 20000 | 46.5 | 65.4 | 30000 | 41.5 | 39.1 | 8300 | 5.90 | 0.713 |
| | | | 40000 | 55.4 | 66.6 | 9000 | 6.40 | 0.820 |
| | | | 46000 | 63.7 | 86.3 | 10000 | 7.11 | 1.00 |
| **FOR 16-INCH PIPE SIZE** | | | | | | 12000 | 8.55 | 1.40 |
| 1400 | 2.46 | 0.241 | | | | 14000 | 9.95 | 1.86 |
| 1600 | 2.81 | 0.308 | **FOR 20-INCH PIPE SIZE** | | | 20000 | 14.2 | 3.61 |
| 1800 | 3.16 | 0.383 | 2000 | 2.22 | 0.153 | | | |
| 2000 | 3.51 | 0.466 | 2500 | 2.78 | 0.231 | ***FOR 30-INCH PIPE SIZE*** | | |
| 2500 | 4.39 | 0.704 | 3000 | 3.33 | 0.323 | 2000 | 0.91 | 0.017 |
| 3000 | 5.27 | 0.987 | 3500 | 3.89 | 0.430 | 2400 | 1.10 | 0.023 |
| 3500 | 6.15 | 1.31 | 4000 | 4.45 | 0.551 | 2700 | 1.23 | 0.030 |
| 4000 | 7.03 | 1.68 | 5000 | 5.55 | 0.832 | 3100 | 1.43 | 0.039 |
| 5000 | 8.79 | 2.54 | 6000 | 6.67 | 1.17 | 3400 | 1.55 | 0.046 |
| 6000 | 10.5 | 3.56 | 7000 | 7.78 | 1.55 | 3800 | 1.74 | 0.057 |
| 7000 | 12.3 | 4.73 | 8000 | 8.89 | 1.98 | 4100 | 1.87 | 0.065 |
| 8000 | 14.1 | 6.06 | 10000 | 11.1 | 3.00 | 4500 | 2.05 | 0.077 |
| 9000 | 15.8 | 7.53 | 12000 | 13.3 | 4.20 | 4800 | 2.19 | 0.087 |
| 10000 | 17.6 | 9.15 | 14000 | 15.5 | 5.59 | 5500 | 2.51 | 0.112 |
| 15000 | 26.3 | 19.2 | 16000 | 17.8 | 7.15 | 6200 | 2.83 | 0.139 |
| 20000 | 35.1 | 33.0 | 18000 | 20.0 | 8.90 | 7600 | 3.47 | 0.203 |
| 25000 | 43.9 | 49.9 | 20000 | 22.2 | 10.8 | 8300 | 3.79 | 0.240 |
| 30000 | 52.7 | 69.8 | 30000 | 33.3 | 22.9 | 9700 | 4.42 | 0.319 |
| | | | 40000 | 44.5 | 39.0 | 11000 | 5.01 | 0.401 |
| | | | 50000 | 55.5 | 58.9 | 16000 | 7.30 | 0.810 |
| | | | 60000 | 66.7 | 82.3 | 20000 | 9.12 | 1.22 |
| | | | 70000 | 77.8 | 110 | 28000 | 12.75 | 2.27 |

## HAZEN & WILLIAMS FLOW DATA for 60 DEG.F. WATER
### PIPES FLOWING FULL : adapted from *Cameron's Hydraulic Data*
### OLD STEEL PIPE : C = 100 : STANDARD WALL THICKNESS
### VELOCITY in FT./SEC. & HEAD LOSS in FT. WATER per 100 FT. of PIPE

| GPM | VEL. | H.L. | GPM | VEL. | H.L. | GPM | VEL. | H.L. |
|---|---|---|---|---|---|---|---|---|
| **FOR 36-INCH PIPE SIZE** | | | | | | | | |
| 2000 | 0.63 | 0.007 | | | | | | |
| 4000 | 1.26 | 0.026 | | | | | | |
| 6200 | 1.95 | 0.057 | | | | | | |
| 8300 | 2.61 | 0.098 | | | | | | |
| 9700 | 3.05 | 0.131 | | | | | | |
| 10000 | 3.14 | 0.139 | | | | | | |
| 12000 | 3.78 | 0.193 | | | | | | |
| 14000 | 4.40 | 0.260 | | | | | | |
| 16000 | 5.03 | 0.330 | | | | | | |
| 18000 | 5.66 | 0.412 | | | | | | |
| 20000 | 6.30 | 0.504 | | | | | | |
| 22000 | 6.92 | 0.590 | | | | | | |
| 24000 | 7.55 | 0.695 | | | | | | |
| 28000 | 8.80 | 0.935 | | | | | | |
| 30000 | 9.44 | 1.065 | | | | | | |
| 34000 | 10.7 | 1.34 | | | | | | |
| 42000 | 13.2 | 1.99 | | | | | | |
| | | | | | | | | |
| | | | | | | | | |
| **FOR 42-INCH PIPE SIZE** | | | | | | | | |
| 4000 | 0.92 | 0.012 | | | | | | |
| 5000 | 1.16 | 0.018 | | | | | | |
| 6000 | 1.39 | 0.025 | | | | | | |
| 7000 | 1.62 | 0.033 | | | | | | |
| 8000 | 1.85 | 0.043 | | | | | | |
| 9000 | 2.08 | 0.053 | | | | | | |
| 10000 | 2.31 | 0.065 | | | | | | |
| 12000 | 2.78 | 0.092 | | | | | | |
| 14000 | 3.24 | 0.122 | | | | | | |
| 16000 | 3.70 | 0.157 | | | | | | |
| 18000 | 4.16 | 0.194 | | | | | | |
| 20000 | 4.62 | 0.238 | | | | | | |
| 24000 | 5.55 | 0.332 | | | | | | |
| 28000 | 6.48 | 0.432 | | | | | | |
| 32000 | 7.40 | 0.566 | | | | | | |
| 38000 | 8.80 | 0.778 | | | | | | |
| 46000 | 10.6 | 1.10 | | | | | | |
| 50000 | 11.6 | 1.28 | | | | | | |
| 56000 | 12.9 | 1.60 | | | | | | |
| | | | | | | | | |
| | | | | | | | | |
| | | | | | | | | |
| | | | | | | | | |
| | | | | | | | | |

CHAPTER

# 3

# SIPHON SEALS AND WATER LEGS

This topic is somewhat arcane, and yet we shall give it quite a lot of attention. The reason is not because siphons are necessarily any great design product to be sought after. The reason is because siphons are potentially quite dangerous and destructive, if they occur unexpectedly and are not properly controlled. *(As used in this context, a siphon is a flow condition, not a particular piece of hardware.)* Siphons are formed by some type of sealing action at the two ends of an open liquid circuit, as a combined consequence of the pipe geometry and flowing condition, and they most certainly can occur accidentally and totally unseen. So consider siphons a safety issue, and this section as primarily "preventative" in nature.

In this topic, we will discuss the ways siphons can form in piping systems, the necessary conditions for maintaining a stable siphon, both with and without seal legs, and especially we shall emphasize how to prevent siphons from existing. The pros and cons of letting a siphon exist are fully explained.

A siphon effect may be defined as creation of flow-induced regions of subatmospheric pressure, partial vacuums, at high spots in the liquid-carrying pipeline.

In some piping arrangements, a siphon is formed as a result of the geometry alone. In others, it depends upon a combination of piping geometry and operating conditions, especially the fluid flowrate, velocity, and friction effects. During the design process, it is easy to overlook the issue of whether or not any of the possible operating conditions of a particular piping system might cause siphon effect to form. We must consider periods of pump shutdown, whether intentional or due to power failure, as a normal operating condition, which must not be allowed to result in harmful transient effects, including unplanned siphonage.

Overlooking vacuum formation can lead to profound surprises for the unwary engineer. "Liquid column separation" and the violent water hammer it causes are probably the worst. In-line cavitation is not far behind; we all know it will ruin a pump inlet, but rarely worry about the deep pitting and erosion damage it can cause in the pipelines themselves. Other nasty surprises include causing big errors in pump head calculations, and allowing air entrainment with subsequent pocketing at vacuum-prone high spots through leaky pipe joints, cracks in dried-out valve stem seals and gaskets, etc.

We must be able to recognize the potential for a siphon to form in piping we design, and to do so during the design stage, not after. "After" could be too late.

As is usual in fluid mechanics, as well as all the other branches of physics which involve potential gradient-flow resistance principles, such as heat transfer and electromagnetics, building complexity into the geometry of the physical system can result in real difficulty and uncertainty in analysis, so please use due care. You may end up making the system design so complicated that you can't predict all its possible behaviors and consequences, or have the time available to make the necessary analyses prior to construction.

***And therein lies the danger.*** From that hard-learned principle, as well as for many other practical reasons, learned over too many years of bearing too much engineering responsibility, my own personal design credo has distilled down to this little truism:

***"Simple is good; simpler is better; simplest is best."***

My personal approach to siphoning and liquid sealing is purely conservative, because of the potentially drastic consequences of being wrong in prediction.

Of course, there are occasions when designing for planned, controlled siphon effect can yield excellent economic benefit, especially when piping system design flowrates and total dynamic heads (TDH) are very large, because a siphon effect downstream of the pump discharge may reduce the TDH for a given flowrate, and thus reduce the pump driver's horsepower requirement. Siphons can save lots of costly kilowatt-hours this way. Of course, siphons upstream of pump inlets have the opposite effect, reducing inlet pressure and promoting cavitation in the pump inlet; very bad, indeed.

The downside of planned siphon operation is that careful engineering and operating attention must be paid to two important functions when designing, and when operating, siphon-based systems:

1. Venting all air from the high pockets in the piping upon pump startup. Air pockets trapped in high points of the piping system prevent formation of the partial vacuum necessary to form the intended siphon action. Air pockets are difficult to clear by liquid flow alone, and almost always call for mechanical vents of some sort.

   Simple self-contained "automatic" air vent devices are notoriously prone to develop liquid leaks under pressure and air leaks under vacuum, and are susceptible to freezing. They are at least in the same problems-league as simple in-line check valves, and may be worse. For my money, they should be avoided, except possibly in absolutely noncritical HVAC applications. They are not really suitable as "process equipment" and if used as such, demand frequent maintenance checks. The cost of attendant maintenance labor and the nuisance of uncertainty of their reliability tends to outweigh their lower first cost.

   Instead, I recommend one either utilize tight shutoff manual vent valves, of suitable materials and class of construction and with leakproof seals, or else use the equivalent in "full-on full-off" vapor-tight automatically operated control valves designed for the appropriate services. Manual vents as well as automated vents should be depicted on all P&IDs where they occur, and be treated with the same attention as a pump, process control valve or any other piece of process equipment, both engineering-wise and maintenance-wise.

2. <u>Breaking the vacuum in the high pockets upon pump shutdown.</u> As we will note later, to <u>*not*</u> do so creates a dangerous unstable physical situation.

Everything just said regarding <u>air vents</u> goes double for <u>vacuum breakers.</u> The only real difference between the two pieces of equipment is the direction of air flow through them when they operate.

Of course, reliable setups utilizing one automated control valve mounted at the piping high point to serve as air-relief on pump startup and vacuum-breaker on shutdown, and incorporating the necessary instrumentation and controls, can easily serve both purposes, venting in both directions as needed. Appropriate sensors and control logic are essential to this approach. However, an "engineered from scratch" combination air vent/vacuum breaker arrangement which will not leak, freeze, or fail without warning under specified operating conditions requires care in design and will increase the piping system's first cost. Therefore it replaces the cheaper off-the-shelf units' uncertainty and poor long-term performance with added complexity and higher first cost.

Liquid seals are also potential trouble spots in a system. They require liquid makeup systems to prevent loss of seal through evaporation or other drops in seal pot fluid level. Also, seal pot liquids can absorb vapor and solid contaminants from their surroundings, changing pH and becoming corrosive as time passes.

## Discussion of Examples (Figures 3-1–3-3)

In the next pages we have collected vacuum-forming and siphon-breaking information on some of the more common "simple" piping arrangements. More complicated piping systems can generally be analyzed by envisioning them as a series of these simple arrangements, and following direct cause-effect logic from bottom to top, one "simple" step at a time.

## <u>Figure 3-1:</u> Open-Sight Downcomer, Downward Vertical Discharge from the Pipe End

One encounters the equivalent of this type of piping arrangement frequently, the only physical difference being that the actual free discharge illustrated in Figure 3-1 is usually hidden from human view inside a tank or process vessel (i.e., a "blind" discharge, instead of taking place verifiably in plain view, which is the intended meaning of "open sight"—an easily visible air gap).

Of course, the physics does not depend on whether or not the jet of discharged liquid can be *seen,* but is determined by whether or not it leaves the pipe at an air gap above the free liquid surface, and is thus free to admit air (or gas from the vessel headspace) into the open pipe end, to break up any vacuum that might try to form in the pipe. *(For piping arrangements that terminate <u>below</u>* the free surface of a liquid volume, please refer to the "dip leg" illustration in Figure 3-3.)

Someday you may be called upon to troubleshoot an existing system, and it may become apparent that existence of a siphon condition plays a role in the problem. If you can't personally verify the actual operating discharge conditions of an existing system visually, in all of its pertinent start-run-stop operating conditions, or else find an absolutely trustworthy, reliable and responsible person who knows the full range of actual discharge conditions for sure and does not mind being quoted in your technical report, then I recommend that before proceeding with analysis you dig up the <u>**final revision approved for construction (NOT the as-built) P&IDs**</u> on which the vessel appears and that reflects the physical operation as intended by the responsible engineers, and **cross-check against the as built piping design drawings, vessel ordering data sheets, and certified vessel shop prints** to see whether or not the formation of a siphon effect was (1) planned for by the process and mechanical design engineers, and (2) would deliver the intended effect in the as-built condition.

All too often, the engineer(s) who would recognize a potential operating problem, accruing from a construction deviation from the original system mechanical design, will not be in the review and approval loop for deviations or changes made after the formal, preconstruction P&ID issuance has taken place. The as-built system may not be capable of producing the physical behavior intended by the designers. By checking the as-builts against the "as-intendeds," you will often gain valuable clues as to the causes of the existing system operational problem.

I also recommend reading the closely related **Chapter #1: Water Hammer,** subtopic **"liquid column separation"** in this book.

**Figure 3-1** illustrates that under "normal" conditions, those in which fluid velocities are kept reasonably low, a vertically downward liquid discharge from an open pipe end into a large plenum volume which contains only a gas or vapor, will be in freefall. Open-sight discharge into an air gap satisfies this criterion.

That is to say, the downward motion of the liquid column is purely ballistic, perfectly analogous to rainfall in the sky. It responds to gravity by breaking up into droplets and accelerating downward, feeling no motive pressure gradient from behind. The freely falling vertical "column" is actually a mixture of water droplets with gas and vapor, everywhere in the free jet and throughout the entire downcomer pipe as well. The bulk mixture from top to bottom inside the downcomer pipe will have the same total pressure as that of the atmosphere, or as the vessel headspace gas or vapor, which it enters as a free jet at the pipe end exit plane.

The gas or noncondensing vapor at the pipe's open end is free to enter the pipe and travel counterflow, upward, through the open channel in the pipe. This prevents any siphon effect, that is, formation of a stable partial vacuum condition in the high point of the piping, from ever forming in the first place. <u>No siphon will exist in **Figure 3-1**</u> under normal flowing conditions.

***Now we will examine the possibilities of "abnormal" flow conditions. What will it take to form a siphon in this piping arrangement?***

To do so we first must <u>assume</u> that <u>"pressure flow"</u> exists in the downcomer, from **points "c" through "g";** in other words, we assume that the pipe is uniformly "flowing full" of fluid, that the fluid is everywhere in single phase and all liquid, with no air or vapor pockets counterflowing up the pipe, no local vaporization or cavitation or bulk flashing, or breakup of the continuous pressurized stream into droplets, and having a continuous downward-acting motive total pressure gradient keeping the flow moving <u>at constant unaccelerated velocity against the retarding friction drag from the pipe inside walls</u> per our familiar **Darcy-Weisbach** or equivalent **Hazen-Williams** pipe friction formulae.

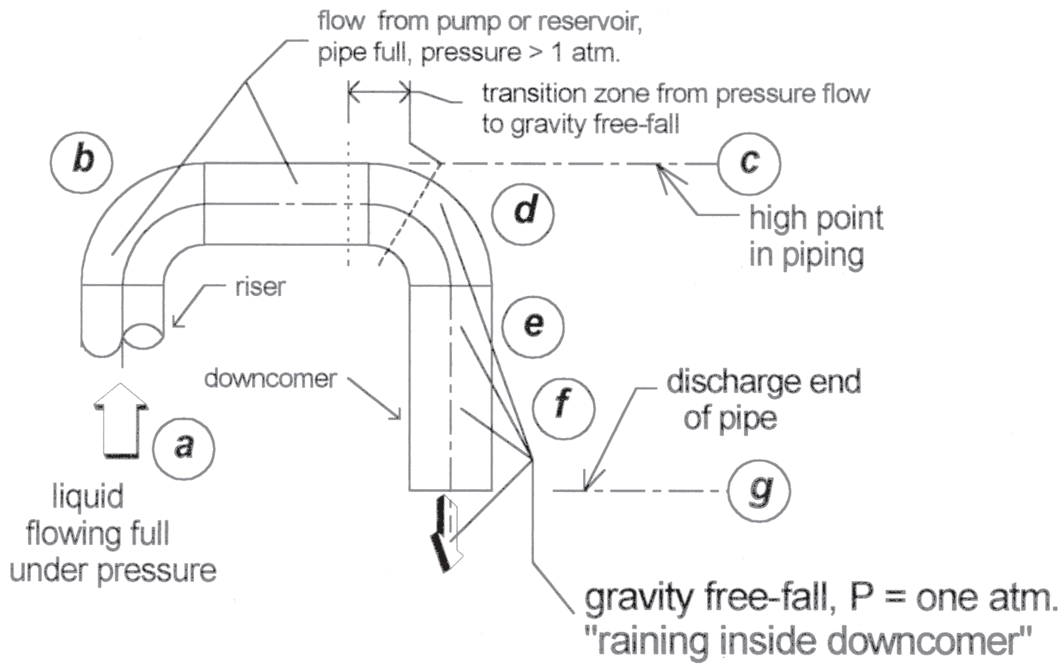



**FIGURE 3-1 ELEVATION VIEW PUMPED LIQUID FLOW WITH OPEN-SIGHT DISCHARGE “NORMAL” CONDITIONS: NO SIPHON EXISTS.**

**Velocity of pump-driven flow within “normal” range, ~ 7 ft/sec max. Friction loss also “normal”, less than ~ 10 ft. head loss per 100 feet of pipe. Flow “breaks up” in transition zone, becomes gravity-driven; No siphon formed, because breakup destroys vacuum at point “c”; so Pc = Pd = Pe = Pf = Pg = 1.0 atm. absolute = zero psig.**

**For siphon to form, liquid downflow velocity must be constant, not accelerating. Only force available to retard the flow is pipe friction. To have a siphon condition, the frictional head loss between “c” and “g” must exceed 100 feet per 100 feet of piping!!**

Application of **Bernoulli’s equation** to **Figure 3-1** yields:

$$(\mathbf{P_c}/\gamma) + (V_c)^2/2g + Z_c =$$
$$= (\mathbf{P_g}/\gamma) + (V_g)^2/2g + Z_g + HL_{\text{friction c – g}}$$

It is the term “$\mathbf{HL_{friction\ c-g}}$” that we seek to evaluate. “P” units above are $\mathbf{lbf/ft^2}$.

We define **point “c”** as the high point in the downcomer pipe, and **point “g”** as in a plane normal to the free jet, outside the pipe, just barely below the pipe’s terminal end.

***At this point it bears review that all of the Bernoulli equation terms, including the “head loss due to friction” term “$HL_{friction\ c-g}$”, are in fact specific energy terms. The P, V, and Z terms apply numerically only to the specific points in the run of pipe whose nodal numbers they bear as subscripts. They are point properties only. Those “H” terms, including “$HL_{friction\ c-g}$,” apply to the entire subscripted stretch between the specifically identified points in the flowing stream, in our case all points between and including “c” and “g.” The “head” unit of “feet” is actually “ft/lbf of energy associated with the fluid per lbm of fluid flowing, at the local value of gravity.” We use the specific weight, $\gamma$, in $lbf/ft^3$ units for convenience in Bernoulli’s equation; it is simpler to write than the understood actual term “$\rho g/g_c$,” where g is numerically the local acceleration due to gravity in $ft/sec^2$, $g_c$ is numerically = 32.174 with units $lbm\text{-}ft/lbf\text{-}sec^2$, and $\rho$ is the mass density in units of $lbm/ft^3$.***

Now continuing our analysis of Figure 3-1: We know that $\mathbf{P_g}$ = 1 atm absolute pressure. Outside the confinement of the pipe walls, beyond its discharge end, there is no continuous water column or pressure gradient, and the flow is ballistic “rain.”

Our assumption of unaccelerated flow inside the pipe means that $(\mathbf{V_c})^2/2g = (V_g)^2/2g$ ft liquid, so the velocity head terms cancel out of the equation. (This would not be true if, contrary to our initial assumption, the actual condition in the downcomer were “gravity flow” as illustrated in **Figure 3-1,** because acceleration due to gravity and breakup of the water column would occur. The fluid would be a raining mixture of liquid, flashed vapor, and sucked-in air from the atmosphere.

Finally, no matter what reference system you choose for the “Z” elevation head terms, you arrive at:

$(\mathbf{Z_c} - Z_{g-})$ = vertical length “$L_{down}$” of the downcomer pipe, measured from point “c” to point “g”, in units of feet. So, making the substitutions for these expressions back into the Bernoulli’s equation, and from this point forward using pressure term **“P”** units of $\mathbf{lbf/in^2}$ absolute, psia, we obtain:

$$\mathbf{(144)}[(\mathbf{P_c} - P_g)/(\gamma)] + L_{down} = HL_{\text{friction c – g}}$$

Since by our own assumed rules we cannot allow cavitation or flashing to occur, and must maintain pressurized, constant-velocity

flow, then a gradient of static pressure which decreases in the direction of flow must exist. In mathematical form, then,
$\mathbf{P_c} - P_g > O$; so,

$$(\mathbf{144})[(\mathbf{P_c} - \mathbf{P_g})/(\gamma)] > 0$$

Now divide both sides of our Bernoulli's derivation by the term $\mathbf{L_{down}}$, to put our equation on a "per foot of downcomer pipe length" basis:

$$(\mathbf{1/L_{down}})(144)[(P_c - P_g)/(\gamma)] + (1/L_{down})(L_{down}) = (1/L_{down})(HL_{friction\ c-g});$$

Now since $(\mathbf{144})[(\mathbf{P_c} - P_g)/(\gamma)] > 0$, then so is $(\mathbf{1/L_{down}})(144)[(P_c - P_g)/(\gamma)] > 0$; call this last term **"press. grad."**; then, **(press. grad.) + 1.00 = Friction Loss/$\mathbf{L_{down}}$** in feet of liquid per foot of pipe length; And since **(press. grad.) > 0,** then we have obtained our solution: the head loss gradient due to pipe friction must be: **$(1/L_{down})(HL_{friction\ c-g}) > 1.00$ ft of head loss per ft of pipe travel.**

• **This is a very handy relationship.**

Tabulations of the pipe friction head loss gradient versus volumetric flowrate for the range of commercially available pipe sizes and most common material surface roughness conditions are commonly published in various formats. Some widely available favorites I've used since 1973 are *Condensed Hydraulic Data* published by the Cameron Pump Division of Ingersoll-Rand Corp, and equivalent data found on the circular slide rule-type B&G System Syzer Calculator which has been distributed by ITT Bell & Gossett for many years. These data happen to use the Hazen-Williams "C-factor" pipe hydraulic roughness formula, which is completely equivalent to the Darcy-Weisbach hydraulic formula and Fanning Friction Factor "f", obtained from Colebrook's empirical equation and/or the ubiquitous Moody Diagram, and the velocity head multiplier loss coefficients "K" for dynamic fitting losses, which are explained clearly and elegantly in the classic Crane Company *Technical Paper No. 410, Flow of Fluids through Valves, Fittings, and Pipe.* All of the above should have permanent places on your work desk.

The Darcy-Weisbach hydraulic formula is:

**pipe wall friction head loss =**
**= (f)(L/d) (V)$^2$**/2g, and for fittings
**dynamic head loss = $(\mathbf{K})_{\mathbf{fitting}}(V)^2/2g$.**

The value of "f" is typically **$0.01 < f < 0.03$** for turbulent flow in pipes, and although it does not vary especially strongly in normal practice, its precise value depends on Reynolds number and degree of pipe wall surface roughness. Usually you can't be too far wrong if all your figuring results in **$f \sim 0.018$** or thereabouts, with $10^5 < NRE < 10^6$. The equivalent Hazen-Williams "C" will range from about **C = 140** for new clean steel pipe to **C = 100** for rough old iron walls.

The Darcy-Weisbach equation shows us that for a given pipe size, length, and condition of our downcomer pipe, if we can specify the friction head loss gradient, we can go to the pipe flow tables and find the minimum velocity and flowrate required to satisfy our relationship,

$$(\mathbf{1/L_{down}})(HL_{friction\ c-g}) > 1.00 \text{ ft/ft pipe.}$$

If unaccelerated flow is to be our net result, then this must be the result of our minimum total pressure gradient which must "overpower" gravity at least sufficiently to keep the "falling" water column "pushed, i.e., compressed" into an all-liquid continuum, in which no breakup occurs. The downward flow velocity must therefore stabilize at a value abnormally high compared to bulk flow velocity in normal "pumped" piping systems. And therefore we must expect abnormally high flowrates for any given pipe size. But what are the numerical values of velocity, flow and pressure? Well, according to my battered old Cameron's tables it is highly unlikely that so grossly oversized a pump, or undersized a pipeline, or combination of both, could come into being as to cause flowrates and velocities high enough to meet the **"1 ft of head loss due to friction per foot of downcomer pipe length" criterion.** That figure is more than 10 times the value of head loss gradient normally used in pipe hydraulics design. We certainly would not do so on purpose. To illustrate that fact quite brutally:

- In an old, rough C = 100 size 1/2 in iron pipe, the velocity would have to exceed 8.5 ft/sec to exceed 1.0 ft/ft friction loss; that's more than 8 gpm, four times the maximum we would design for—in half-inch pipe, that is.
- In an old, corroded 4-in line, we are talking over 27.7 ft/sec, *or* 1,100 gpm, whereas prudent design would keep the flow less than 300 gpm—in 4-inch piping.
- And in a *new,* 8 in Schedule 40 steel downcomer, that's more than 57 ft/sec., a minimum flow of about 9,000 gpm. Normally, I hold flow in an 8 in line to not exceed 1,100 gpm.

So with an open-sight freely-discharging-to-the-atmosphere downcomer, we do not have to worry too much about accidental siphon legs forming; one would "really have to work at it" with brute force to make it happen.

But if the pipe's terminal air gap is hidden from view inside a vessel, how are we to know if the vessel gets flooded and eliminates the air gap, sealing off the downcomer? And what if the pump then trips off, or we shut it off? And what if the suction end is sealed off (against airflow which could otherwise break the vacuum) also? And what if there is no vacuum breaker valve or device in the pipeline? Think about it.

## Figure 3-2: Open Horizontal Discharge Exits from the Pipe End

The open horizontal pipe end affects discharge flow in a weir-like manner. The free surface of a small flowrate which cannot completely fill the exit pipe, will display a downward slope toward the exit. The air gap between free surface and highest point in the pipe cross section will increase as flow nears the exit. This would be gravity flow in the partially full run of horizontal pipe; the pressure at the free surface will equal that of the air or gas above it regardless of horizontal location point along the pipeline.

**Note that increasing the flowrate will narrow the height of air gap at the top of the open pipe end, and that decreasing the flowrate will increase the gap height.** If flow drops below a certain value, the gap will enlarge such that the sloping free surface will extend all the way back to the downcomer's elbow, thus producing a continuous air channel from the atmosphere, breaking the liquid seal and destroying any siphon effect previously existing in that downcomer.

The necessary exit velocity to produce 100% full flow at the exit plane, sealing the pipe completely against air entry, can be found by careful experiment. King and Brater published comprehensive results of minimum sealing flowrate of water versus pipe diameter in 1976, which are reproduced in **Table 3-A** below

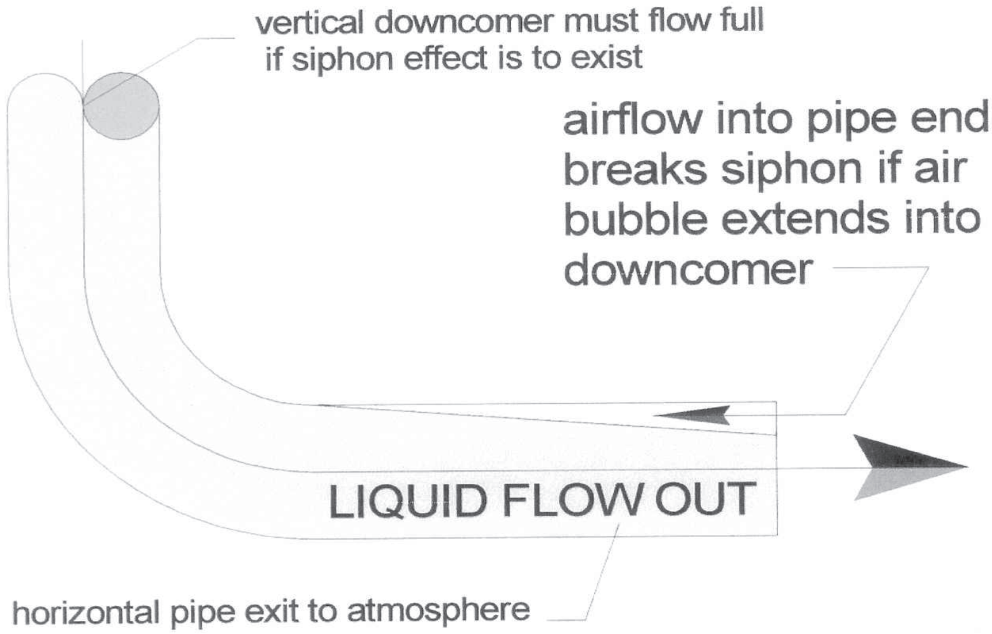



**FIGURE 3-2 ELEVATION VIEW : PIPE FREE DISCHARGE TO ATMOSPHERE ILLUSTRATING DISCHARGE VELOCITY EFFECT ON SIPHON SEAL**

**TABLE 3-A MINIMUM REQUIREMENTS TO MAINTAIN SIPHON SEAL IN FIGURE 3-2 ABOVE.**

| Nominal Pipe Size, standard schedule, inches | Minimum Flowrate Required to Seal, gallons/ minute | Approximate Min. Equivalent Velocity feet/ second | Approx. Head Loss ft. loss/ 100 ft. pipe New Pipe, C= 140 | Approx. Head Loss ft. loss/ 100 ft. pipe Old Pipe, C= 100 |
|---|---|---|---|---|
| 1/2″ | 3.2 | 3.3 | 9.79 | 18.12 |
| 1″ | 11.5 | 4.3 | 8.22 | 15.23 |
| 1-1/2″ | 35 | 5.3 | 7.97 | 14.75 |
| 2″ | 63 | 6.0 | 7.01 | 12.98 |
| 3″ | 169 | 7.3 | 6.39 | 11.83 |
| 4″ | 340 | 8.4 | 6.21 | 11.50 |
| 6″ | 930 | 10.3 | 5.35 | 9.90 |
| 8″ | 1,840 | 11.8 | 5.04 | 9.34 |
| 10″ | 3,300 | 13.2 | 4.91 | 9.09 |
| 12″ | 5,100 | 14.5 | 4.57 | 8.46 |

**Figure 3-2.** These flowrates are greater than normal design practice, getting abnormally higher with increasing diameter. For what it's worth, the K&B data derive from a curvefit of their experimental data, namely,

$$\mathbf{GPM/(d)^{2.50}} \geqq 10.2$$

In practice, I use extreme caution with this data, especially if the horizontal exit run is straight, relatively short, relatively large in pipe diameter, when the flowrate is prone to vary, is critical, or potentially dangerous. There are too many uncontrollables to allow me to be comfortable with these numbers, and calculations regarding the actual physics are economically intractable for usual project design budgets. When in doubt, it is far better to assume gravity flow when designing the downcomer.

## Figure 3-3: Liquid Legs and Static Siphon Seals

About all that needs adding to the figure's notes is a general caution about corrosion at the waterline in the dip leg. All it takes to break the liquid seal is a pinhole within sucking distance of the air-water interface. Try changing wetted portions of dip tube material from carbon steel to stainless steel or plastic, or whatever makes sense for the situation.

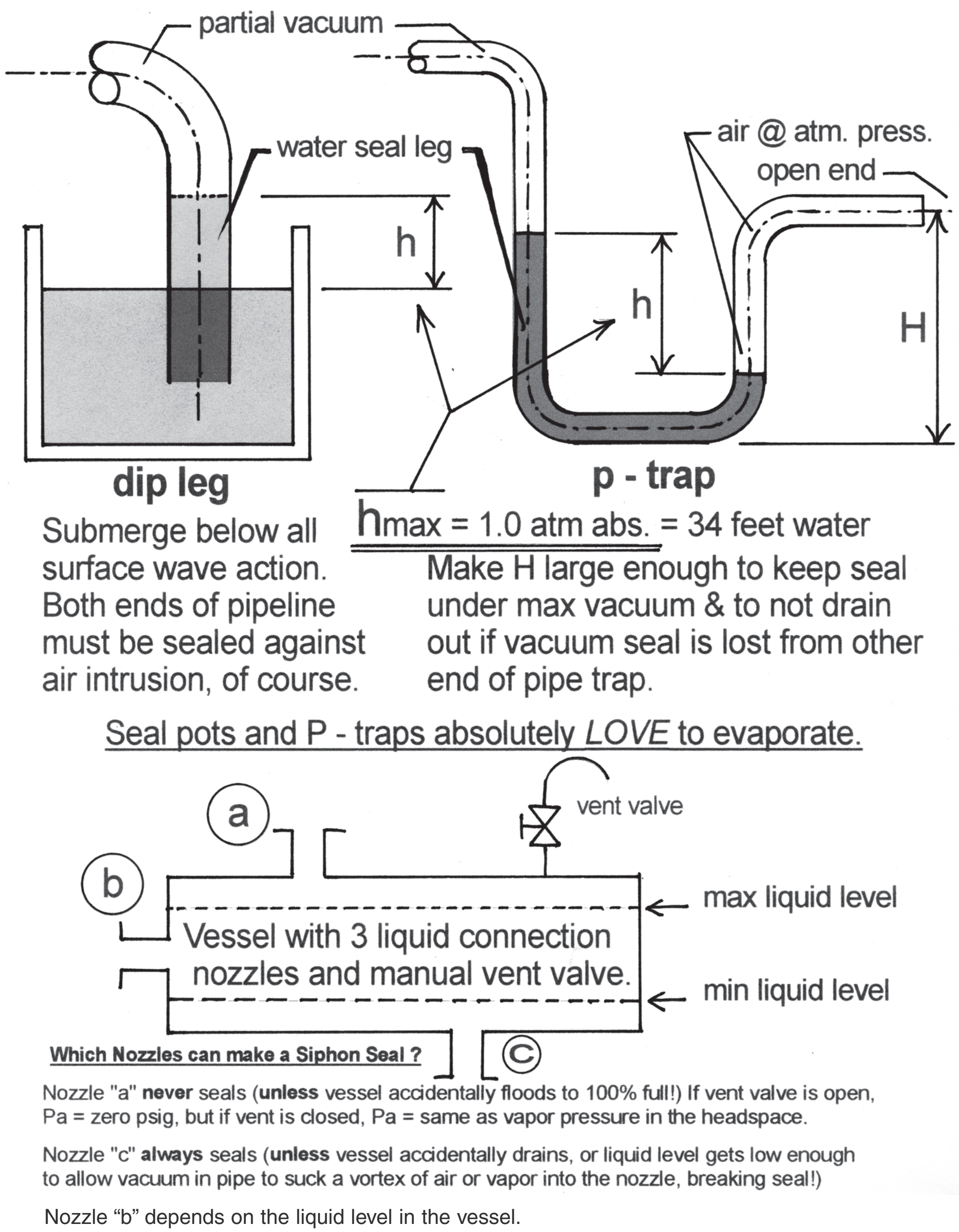



**FIGURE 3-3 LIQUID LEGS & STATIC SIPHON SEALS**

CHAPTER

# 4

# REGULATING STEAM PRESSURE DROP

## Via Drop Across Turbine vs. Drop Across Throttling Valve

In industry, a "steam study" requires the engineer to determine the thermodynamic conditions of various streams of flowing steam. Changes of pressure, temperature, and physical state are not always evident, and numerically depend strongly on the actual path followed, especially when expansion is the pertinent process.

Many years ago, to answer some management inquiries arising from such a study, I wrote up the included example problem, to illustrate the factors involved. To obtain clarity and contrast, I invented an extreme example comparing expansion of superheated steam from an initial condition of 300 psia and 700°F, to a final pressure of 16 psia and unknown temperature.

- In one case, the expansion-pressure drop took place through a power turbine, and in the other the pressure drop took place across a throttling valve.
- The mass flow and the inlet and exit velocities were the same for both cases: 10,000 lbm/hr entering @ 200 ft/sec. and exiting @ 600 ft/sec.
- The object was to demonstrate the calculation of all of the problem's two exit streams' pertinent thermodynamic properties, and compare results from the two cases.

The reference text given in the analysis was G.J. Van Wylen, Thermodynamics, 1st ed., University of Michigan Department of Mechanical Engineering, John Wiley & Sons, New York, 1959.

### Example Problem: Chapter 4

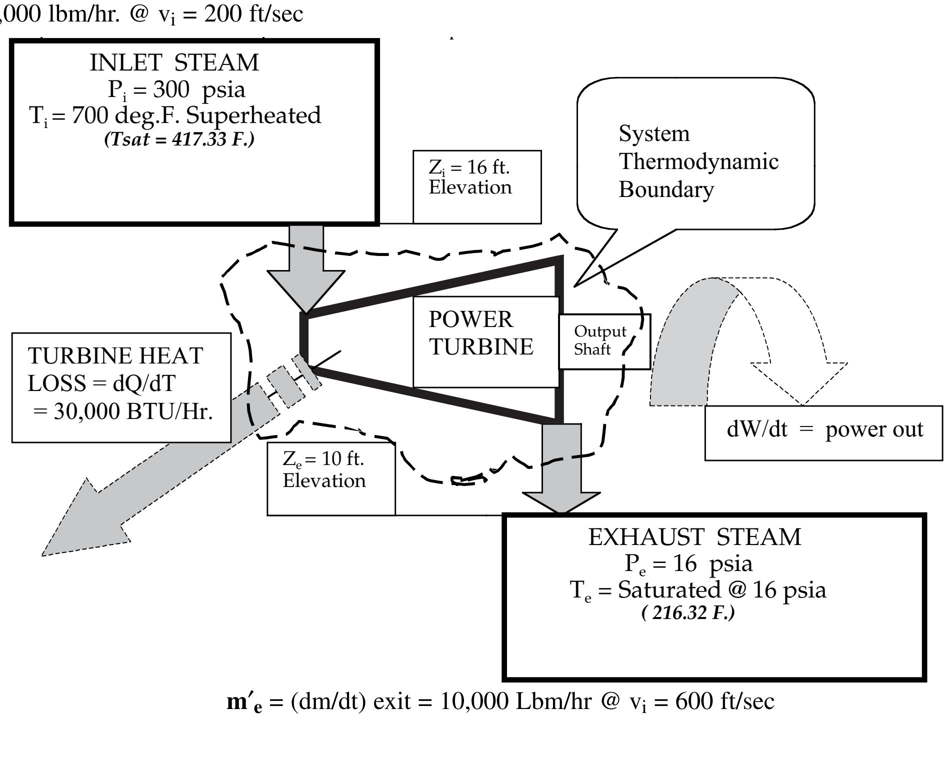
Compare end states of steam having the same entering conditions and exit pressure, when:

1. "dropped" through a ***turbine,*** versus
2. "dropped" through a ***pressure-reducing valve.***

### Schematic Flow Diagram: 1. Dropped Through Turbine

(Reference Example 5.6, page 87 of Van Wylen's classic text.)

**$m'_i$** = (dm/dt) inlet = 10,000 lbm/hr. @ $v_i$ = 200 ft/sec




**$m'_e$** = (dm/dt) exit = 10,000 Lbm/hr @ $v_i$ = 600 ft/sec

**This is a steady-state equilibrium First Law solution.**
**Algebraically, heat loss from system (dQ/dT) is negative (–); power out (dW/dt) is positive (+).**
**Work @ Btu per hour basis; g = 32.17 ft/sec$^2$**; gc = 32.174 lbf-ft/lbf-sec$^2$; J = 778 ft-lbf/Btu.

$$\begin{array}{cccccccccccccccc}
 & & & & & 200^2 & & 32.17 \times 16 & & & & & & 600^2 & & 32.17 \times 10 \\
\mathbf{Q'_{net}} & + & \mathbf{(M'_i)} & \mathbf{(h_i} & + & \mathbf{V_i^2/2Jg_c} & + & \mathbf{gZ_i/Jg_c)} & = & \mathbf{W'_{net}} & + & \mathbf{(M'_e)} & \mathbf{(h_e} & + \;\mathbf{V_e^2/2Jg_c} & + & \mathbf{gZ_e/Jg_c)} \\
-30{,}000 & & 10{,}000 & 1{,}368.3 & & 2 \times 778 \times 32.17 & & 778 \times 32.17 & & \text{Btu/hr} & & 10{,}000 & 1{,}152.0 & 2 \times 778 \times 32.17 & & 778 \times 32.17
\end{array}$$

To better visualize the terms' order of magnitude, divide each term by **m′** = 10,000 lbm/hr. This places the First Law on a "Btus per pound mass" basis. Also, reduce the terms:

$$\overset{q_{loss}}{\mathbf{-3.0\ Btu/lbm}} + (\overset{h_i}{\mathbf{1368.3}} + \overset{k.e._i}{\mathbf{0.799}} + \overset{p.e._i}{\mathbf{0.026}})\ \mathbf{Btu/lbm} = (\overset{h_e}{\mathbf{1{,}152.0}} + \overset{k.e._e}{\mathbf{7.2}} + \overset{p.e._e}{\mathbf{0.0128}})\ \mathbf{Btu/lbm} + \overset{W_{out}}{\mathbf{w\ Btu/lbm}}$$

*Note:* Potential energies **p.e.$_i$** and **p.e.$_e$** are entirely negligible. Although kinetic energy at the exit is 7.2 Btu/lbm, about 10 times the kinetic energy (KE) at inlet, the exit KE is only (7.2/(1152 + 7.2)) × 100% = = **0.62%** of the total exit stream energy; thus it could be neglected in initial calculations.

Solving for the turbine work $\mathbf{W_{out}}$ term, we obtain $\mathbf{W_{out}}$ = 206.9 Btu/lbm

Also, the WORK is ***much*** greater than the HEAT LOSS, i.e., **206.9 >> 3.0,** so we could have neglected the heat loss without serious error. Note that the heat loss and exit kinetic energy were of the same order of magnitude, less than 1% of the enthalpy terms. While maybe not always the case, this gives us a practical look at heat loss from axial flow machinery; heat transfer takes *time,* and the steam doesn't hang around for very long inside the turbine! So each pound of steam loses very little of its heat through the machine metal. Thermal insulation on this turbine is more of a personnel safety issue than a thermal efficiency issue.

Finally, solve $W_{out}$ in horsepower units:

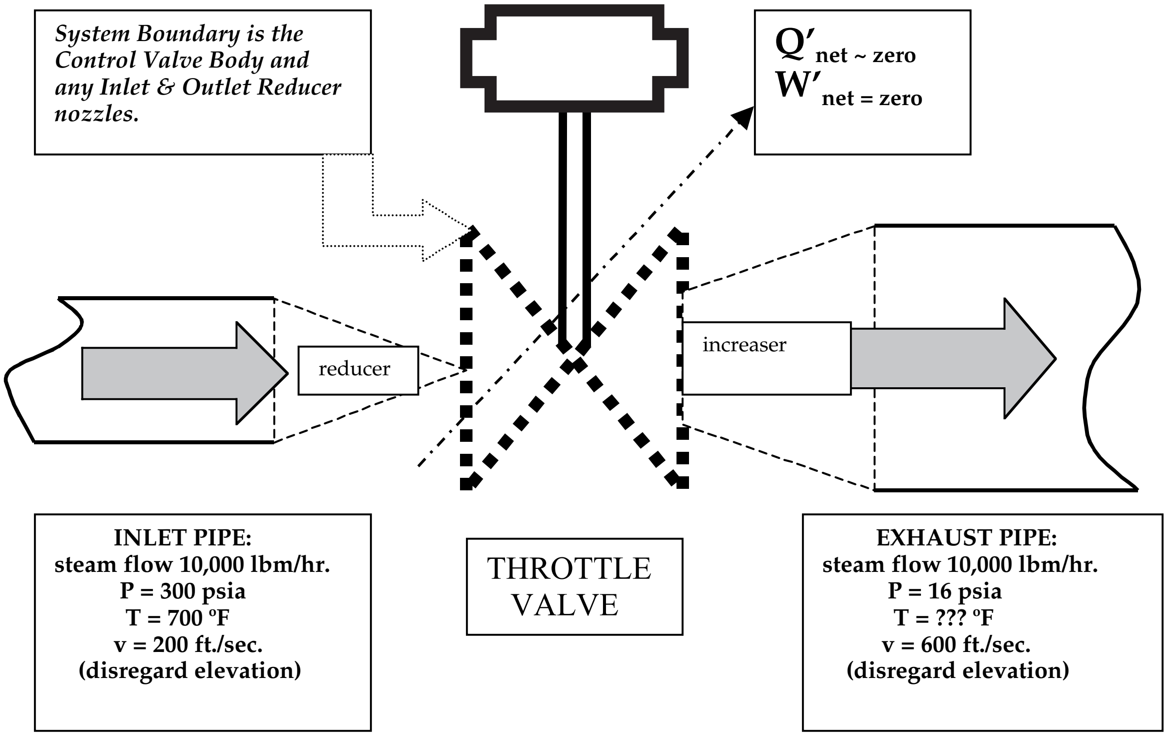
$$\mathbf{m'_e} \times w_{out} = (10{,}000 \text{ lbm/hr} \times 206.9 \text{ Btu/lbm})/(2545 \text{ Btu/hp-hr}) = \underline{\mathbf{813\ hp}}$$

### Summary of Steam Conditions from Turbine Example:

| Property | Inlet | Exit |
|---|---|---|
| Pressure, psia | 300 >> | 16 |
| Temperature, °F | 700 >> | 216.32 |
| State | Superheated | Saturated (100%) |
| Enthalpy, Btu/lbm | 1,368.3 > | 1,152.0 |
| Specific volume, ft.$^3$/lbm | 2.227 | << 24.75 |
| Entropy, Btu/lbm-°F | 1.6751 | < 1.7497 |

Now let's see what happens when the steam pressure is reduced the same amount by a throttling valve (Reference Example 5.15, page 103 of Van Wylen's text).

**Schematic Flow Diagram: 2. Dropped Through Pressure-Reducing Valve**




Because there is no work crossing the system boundary, and the heat transfer (loss) from the valve body is negligible, and the change in elevation is negligible (here zero), the First Law reduces to the "throttle" form. Pressure drops radically across the valve, and the pressure drop is a nonrecoverable loss. Specific volume increases radically as a result, and by conservation of mass law, the exit velocity must exceed the inlet velocity sufficiently to keep the mass flowrate constant. Therefore the enthalpy (energy) of the steam is decreased by the same amount that the kinetic (energy) increases.

$(\mathbf{h_i + v_i^2/2Jg_c}) = (\mathbf{h_e + v_e^2/2Jg_c})$; numerically, from the preceding Turbine example,
$(1368.3 + 0.799) = (\mathbf{h_e} + 7.2)$, and $\underline{\mathbf{h_e}}$ **= 1361.9 Btu/lbm**

Interpolating the steam superheat tables for the remaining exit properties @ 16 psia:

$$\textbf{Temperature: } \mathbf{T_e} = 600°\text{F} + \overbrace{\frac{(1361.9 - 1334.7)}{(1383.1 - 1334.7)}}^{\{ratio\ =\ 0.561983471\}} \times (700°\text{F} - 600°\text{F}) = \mathbf{656.2°F}$$

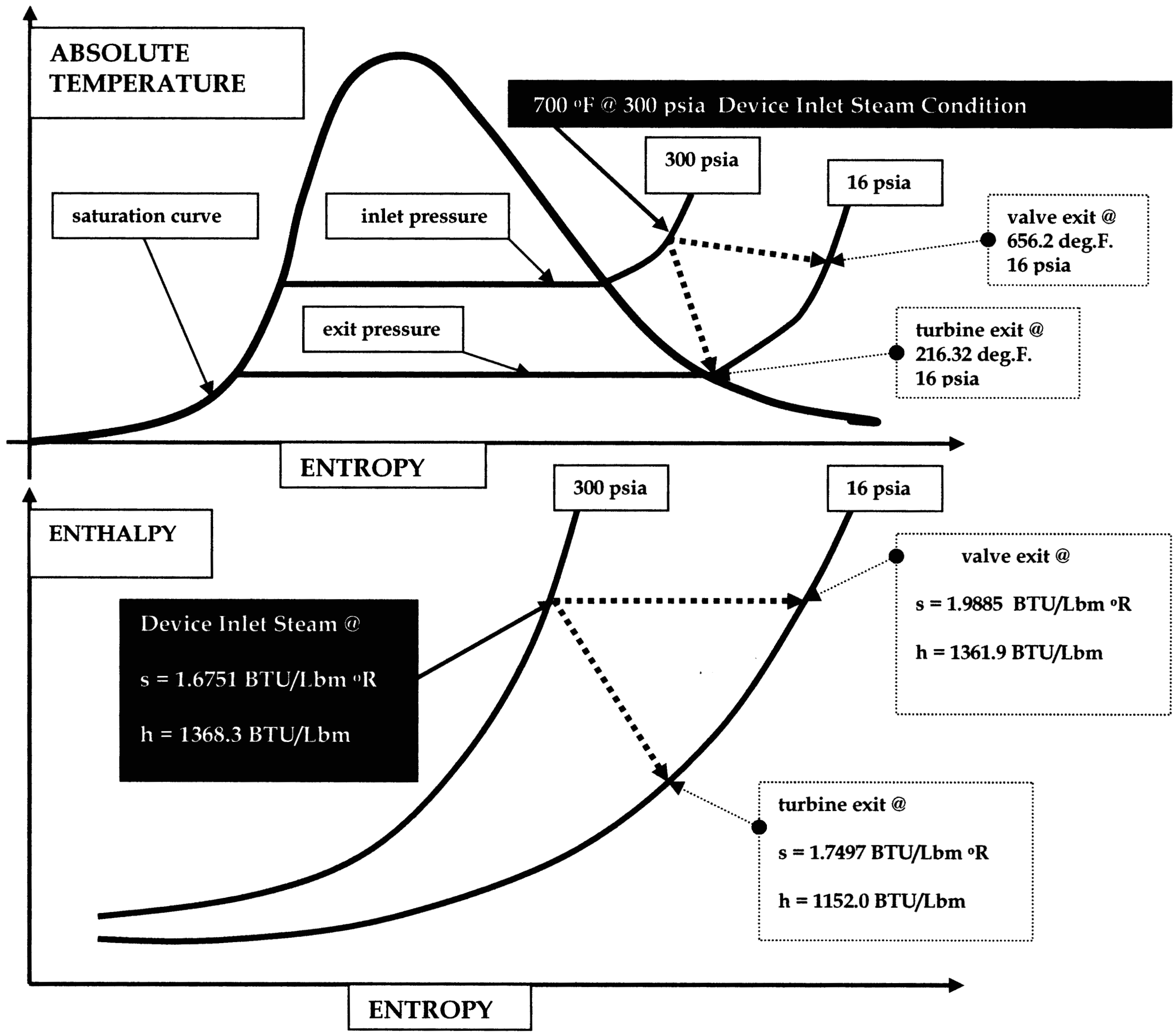
**Specific volume:** $\mathbf{V_e} = 39.36 + (0.561983471)(43.10 - 39.36) = \mathbf{41.462\ ft_3/lbm}$

Expand the summary to include results of the throttle valve expansion process:

**Comparison of Steam Exit Conditions from the Two Examples:**

| Property | Inlet | Turbine Exit | Throttle Valve Exit |
|---|---|---|---|
| Pressure, psia | 300 | 16 | 16 |
| Temperature, °F. | 700 | 216.32 | 656.2 |
| State | Superheated | Saturated (100%) | Superheated |
| Enthalpy, Btu/lbm | 1,368.3 | 1,152.0 | 1,361.9 |
| Specific volume, $ft_3$/lbm | 2.227 | 24.75 | 41.462 |
| Entropy, Btu/lbm-°F. | 1.6751 | 1.7497 | 1.9885 |

***Temperature-Entropy Diagram of Processes and Enthalpy-Entropy Diagram of Processes***




**Specific entropy:** $s_e = 1.9639 + (0.561983471)(2.0076 - 1.9639) =$ **1.9885 ft$_3$/lbm**

Now calculate:

1. Availability property of entering steam (same at turbine and reducing valve entry);
2. Availability property of steam leaving the turbine exhaust;
3. Availability property of steam leaving the pressure reducing valve;
4. Ratio of actual turbine work to (reversible) available work and the thermodynamic Irreversibility for that change of state;
5. Lost Work property and Irreversibility of the flow across the pressure reducing valve. *Note: for text reference, see Van Wylen pp. 237–246.*

1. Availability of superheated steam flowing at P = 300 psia, T = 700°F, ignoring the kinetic and gravitational potential energies of the flow stream. Ref. datum = 14.7 psia @ 77°F.

   Availability = $\psi = (h - h_o) - (T_o)(s - s_o)$ *{Van Wylen Eq. 10.12 and p. 242};*

   h = 1368.3 Btu/lbm @ 300 psia/700°F (superheated vapor, abs. temp. = 1160°R);

   $h_o$ = 45.02 Btu/lbm @ state (abs. temp. = 537°R)

   To = 537°R;

   s = 1.6751 Btu/lbm-°R @ 300 psia/700°F;

   $s_o$ = 0.0876 Btu/lbm-°R @ 77°F/saturated liquid;

   $\psi_{entering} = (1368.3 - 45.02) - (537)(1.6751 - 0.0876) = 470.7925$ Btu/lbm

2. Availability of steam leaving the turbine (same formula, see summary for input values):

   $\psi_{leaving\ turbine} = (1152.0 - 45.02) - (537)(1.7497 - 0.0876) = 214.4323$ Btu/lbm

3. Availability of steam leaving the P.R. valve (same formula, see Summary for input values):

   $\psi_{leaving\ valve} = (1{,}361.9 - 45.02) - (537)(1.9885 - 0.0876) = 296.0967$ BTU/lbm.

4. To find the ratio of actual turbine work to (reversible) available work for the calculated change of state, we have to know the actual turbine work, reversible turbine work, and the turbine's thermodynarnic efficiency, and from those we can also find the Irreversibility:

   Efficiency ($\eta_{\ turbine}$) = $W_{act}/W_{rev} = W_{act}/(\psi_{entering} - \psi_{leaving})$ = 206.9 Btu/lbm/(470.7925 – 214.4323 Btu/lbm) = 206.9/ 256.3602 = = 0.807; in usual parlance, the turbine efficiency $\eta_{turbine}$ = 80.7%

   Irreversibility$_{turbine}$ *(Van Wylen Eq. 10.16)* = $(W_{rev}^{-} W_{act})_{turbine}$ = (256.3602 – 206.9) =
   = 49.4602 Btu/lbm

   For you seekers of deep knowledge, and other masochistic gluttons for punishment, we can derive the *turbine's irreversibility factor* as follows:

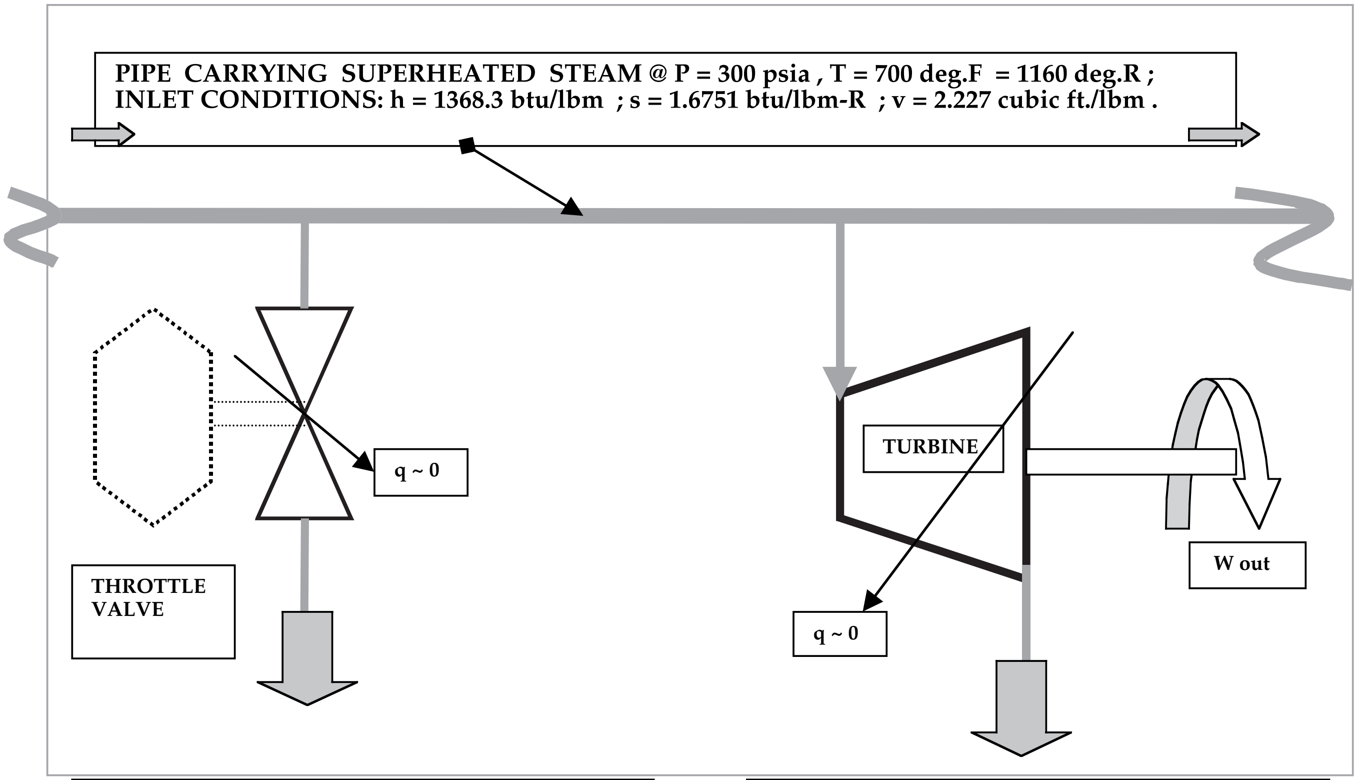
   Irreversibility = (1 – $\eta_{\ turbine}$) = (1.000 – 0.807) = 0.193 (*19.3% of availability*); i.e., (Irreversibility)/{Loss of availability} = (49.4602/256.3602) = 0.193

5. To find the lost work and Irreversibility across the throttling valve *(Van Wylen, Ref. 10.10, p. 245):*

   Irreversibility$_{P.R.\ valve}$ = $(T_o)(s_{exit} - s_{inlet})$ = (537°R) (1.9885 – 1.6751) =
   = 168.2958 Btu/lbm

   Lost work$_{P.R.\ valve}$ = $(T_i)(S_{exit} - S_{inlet})$ = (1160°R)(1.9885 – 1.6751) =
   = 363.544 Btu/lbm

**Schematic Summary**




Pressure = 16 psia
Temperature = 656.2 ºF
*(440 deg. superheat)*
enthalpy h = 1361.9 btu/lbm
*(1368.3 if ΔVelocity = 0)*
entropy s = 1.9885 btu/lbm-deg.R
specific volume v = 41.462 $ft^3$/lbm

irreversibility = 168.30 btu/lbm

actual work = zero

*reversible work = $\psi_{entering} - \psi_{leaving}$ =
470.7925 – 296.0967 =
= 174.7 btu/lbm

***Reversible work between entering and leaving states theoretically available. Since the valve process does no work at all, the formal "Lost Work" is 363.544 btu/lbm as calculated above.*

thermodynamic efficiency η = zero.

P = 16 psia
T = 216.32 ºF
*(saturated vapor)*
h = 1152.0

s = 1.7497
v = 24.75

irreversibility = 49.46 btu/lbm

actual work = 206.9 btu/lbm

reversible work = 256.36 btu/lbm

thermodynamic efficiency η = 80.7%

CHAPTER

# 5

# IRI Fuel Gas Burner Piping Valve Train

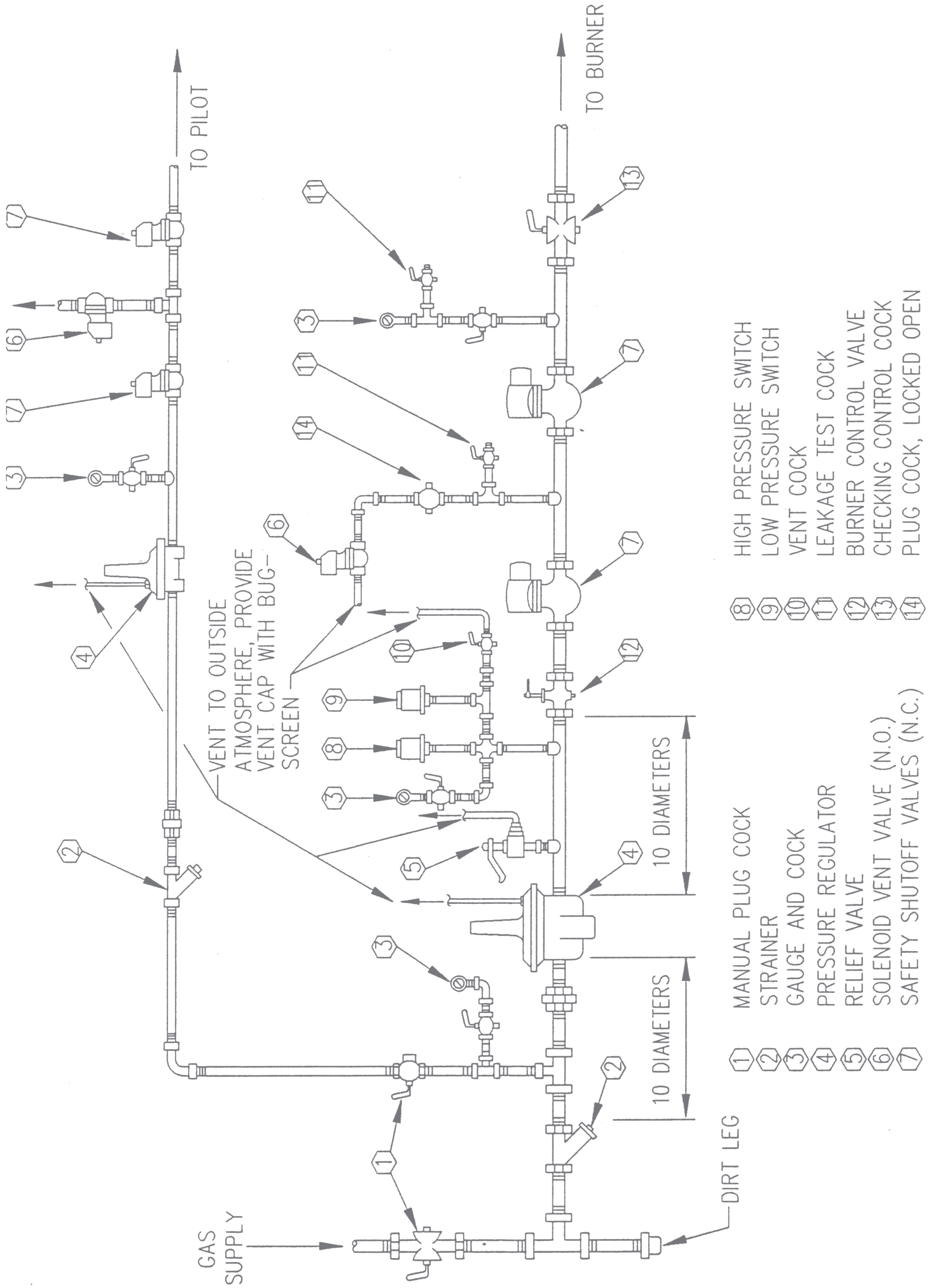
Several insurers of industrial, institutional, and commercial facilities and physical plants have established standards of safe design / construction practice; the ***Factory Mutual*** **(FM)** and ***Industrial Risk Insurers*** **(IRI)** are a couple of those best known in U.S. work.

One handy article to keep around is a good schematic of a safe approved piping arrangement, including pilot flame and gas burner control valves, for supplying fuel gas to boilers, furnaces, and the like.

A (circa 1994) schematic of the **IRI gas train** is included here to assist you. You will wish to verify its currency and applicability to your jobsite before committing it to design approval.

Remember, it is your responsibility as design engineer to comply with all local, state and federal regs for gas burning equipment and its piping and controls.

My thanks to the long-forgotten CAD drafter who originally produced the graphic of the gas train.




**I.R.I. GAS TRAIN**
**SCALE: NONE**

CHAPTER

# 6

# Controlling Differential Air Pressure of a Room with Respect to Its Surroundings

Our goal here is to be able to make a suitably accurate estimate of the amounts of airflow entering and leaving a particular room in order to design a heating, ventilating, and air conditioning (HVAC) system that can maintain the air pressure in that room at a predetermined constant value with respect to the pressure(s) of its surrounding boundary air volumes. We will not concern ourselves with the heating, cooling, or humidity control of any of those air volumes in this section; just the room's relative air pressure.

The candidate air streams into and out of a room are: supply air, return air, exhaust air, makeup air, infiltration into a negative pressure airspace and exfiltration from a positive pressure airspace. Not all streams exist in every case.

The example problem solved later in this chapter is illustrated and defined in the calculation. Although the example selected was rather simple, the same procedure applies to any enclosed volume of air, and to positive differentials as well as negative ones. Just formulate the algebraic sum of all airstreams in and out, as a function of the stream's pressure drop due to flow, solving finally for the unknown stream flowrate.

Don't try to make this into a complicated thing. It's not. The concept is very simple. Begin with a room at equilibrium. If all at once you begin ramming more pounds of air into a room (from outside of it, by turning on the motor driving a ducted pressure fan or pressure blower discharging into the room) than is initially leaving it, then compression will immediately begin to take place; the total air mass in the room will begin to increase, and the room air pressure will start to rise. This happens very rapidly; the pressure changes in waves moving at sonic velocity in the air, which is about 760 miles per hour at sea level. So it doesn't take very long!

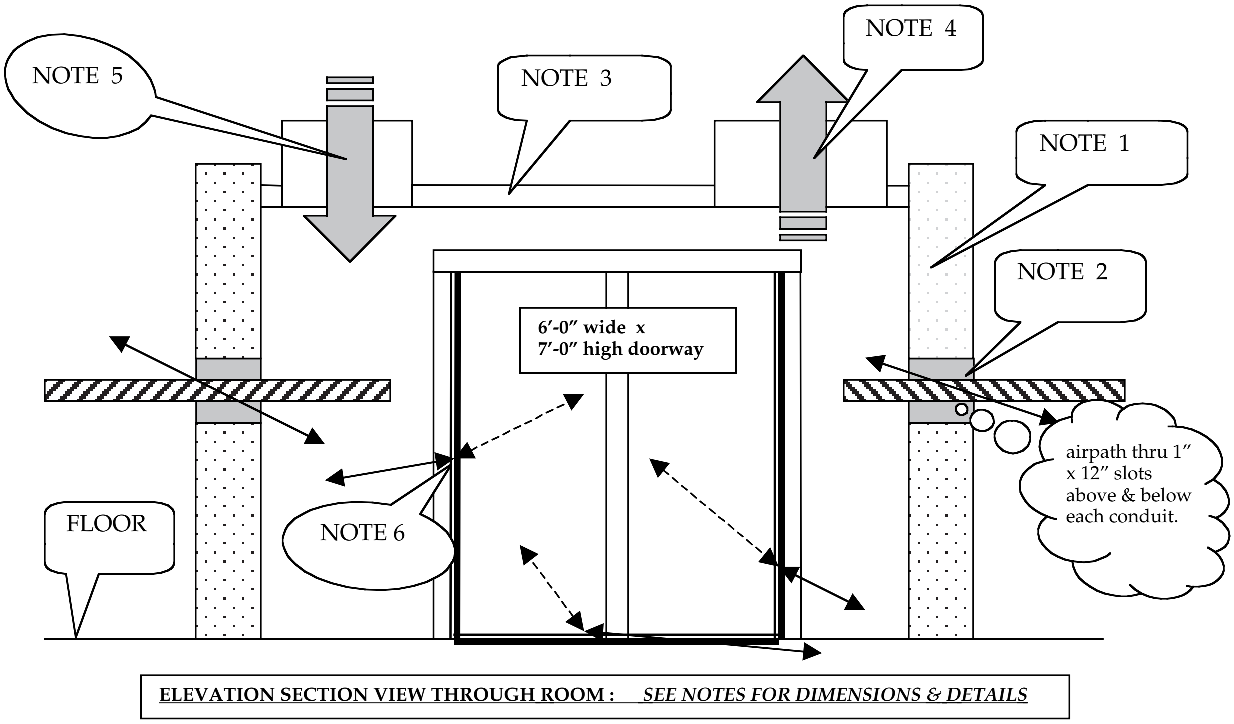
Let the supply fan continue to run with no speed or damper changes. <u>What happens is that as the room pressure rises, the mass flowrate of air leaving the room</u> (whether by separate exhaust vents, or through cracks under the door or open windows, or through a leaky ceiling tile system into the attic, or however air is leaving the room, it makes no difference) <u>will begin to increase.</u> Eventually, air mass flowrate equilibrium will be reached, and the pounds of air entering the room per unit of time through the supply fan will precisely equal the pounds leaving it. When this mass flowrate balance is attained (which happens naturally and inevitably if no fan speeds are changed or dampers reset or windows closed) then the compression process ends, and the room is at mechanical equilibrium with its surroundings once more. Only now, the room exists at a <u>**higher**</u> (positive) bulk static air pressure relative to its surroundings, than it did before we turned the supply fan on. There will be a uniform net outflow of air from the room to surrounding spaces. This is a desired condition for clean rooms, since it prevents airborne contaminants from infiltrating.

Naturally, if our initial action is to turn on an exhaust fan instead of a supply fan, the mirror image of the process occurs, and the room will come to equilibrium at a <u>**lower**</u> (negative) relative static pressure than before. Balance of the individual airstream flowrates by physical means gives us ultimate control, easily automated.

### Statement of Problem

The required quantity of conditioned air supplied to the room, which we will denote as <u>**"Supply Standard Cubic feet per minute *(SCFM)***</u>", has been calculated, measured, and set up through the supply duct and its room discharge grille. The exact supply air quantity to the room is considered a very important parameter for process quality control reasons, and therefore the supply air duct contains an air mass flowrate transducer, in series with a high-precision laboratory quality flow control damper and dedicated single-loop high-precision control system. As changes in the outdoor air static pressure and/ or changes in the room bulk static air pressure *relative to the outdoors* occur, as they naturally will, then the resulting supply air mass flowrate change is sensed, and automatically corrected back to the desired "Supply SCFM setpoint value" by repositioning of the damper blades via the automated damper actuator/positioner-controller. If the continuously monitored "Supply SCFM" mass flowrate quantity drifts below the mandated required setpoint, the loop controller will automatically reset the damper to the "more open" position needed to precisely regain the setpoint flowrate. ***(On a rise above setpoint, the reverse is true.)***

The air flowrate <u>***"Exhaust SCFM"***</u> is created by a ducted exhaust fan with variable speed fan drive-motor controller, which is operated automatically by an equally precise dedicated room bulk static differential-pressure loop controller. The room static air pressure differential relative to the ambient outdoor atmosphere is equally important to the room's process, and is continuously monitored by a high-precision pressure comparator transducer-indicator-controller installation, dedicated to that specific duty.




**<u>Example Problem</u>: *To maintain a room at <u>0.05 in of water gauge</u>* higher pressure than its surrounding environment, calculate the air flowrates through the various air paths into and out of the room to achieve a correct air balance resulting in the desired room relative air pressure.**

NOTES:

1. Walls are 12 in thick, solid masonry, with all surfaces including the slots above and below the conduit penetrations being finished <u>very smooth</u> and epoxy-painted.
2. In the two places shown, a 12-in-wide, solid metal-skinned conduit penetrates the wall horizontally. The conduit edges are sealed airtight to the wall, but there are 1-in-high full-width openings in the wall above and below each conduit. Air can flow through those 1 in × 12 in openings, but not through the metal conduit itself.
3. Ceiling is airtight gasketed tiles system which can stand the differential pressure and remain airtight.
4. Exhaust air flow thru ceiling grille (all wasted to outdoors, no return air to HVAC unit).
5. Conditioned supply air = 100% fresh outdoor air through supply duct filter grille.
6. Air leakage cracks at doors consist of two 1/32 in wide × 7 ft 0 in. long cracks, one at each vertical side of the doorframe, plus one 1/16 in wide by 6 ft 0 in long crack under the doors at the floor line. The doors open inward, and are normally kept closed at all times the room HVAC unit is running and the room is in "clean" operation mode.

Therefore, the exhaust air mass flowrate quantity, created by the exhaust fan, *will be maintained at whatever it takes for the room static air pressure to remain constant at the required pressure differential with respect to the outdoors.* If the continuously monitored room relative pressure differential drifts below the mandated required setpoint, the loop controller will automatically decrease the exhaust fan rotational speed (rpm) to precisely regain the setpoint differential pressure. ***(On a rise above setpoint, the reverse is true.)***

**<u>"Exfiltration *SCFM*</u>"** is the total flow from inside the room to the surroundings through the wall conduit slots, the door cracks, and any other structural *air leakage paths* that might exist. In our example, there are only the wall slots and door cracks to be concerned with. (See NOTES 2 and 6.) These leakage paths behave essentially as "fixed area orifices"; if the room pressure differential **($P_{room}$ – $P_{atm}$)** rises, the leakage rate will increase accordingly, and vice versa.

1. ***What is the required exhaust fan air flowrate "<u>Exhaust SCFM</u>"*** to maintain the room at a measured bulk static pressure of <u>0.05 in of water positive</u>, relative to the surrounding corridors, which stay at atmospheric ambient pressure and to the atmosphere itself? Note that we have <u>not</u> specified the <u>numeric value</u> of "<u>Supply SCFM</u>."
2. *How much force due to that differential pressure is holding the doors, which open inward, closed against the doorframe? This needs to be known so that in an emergency nobody can get trapped inside the room. (Sufficiently high $\delta P$ can "nail" the doors shut if we are not careful.)*

**Solution:**

(Exhaust SCFM) = (Supply SCFM) – (Exfiltration SCFM)

To solve for the *Exfiltration CFM,* (note: *CFM* = volumetric flowrate, whereas *SCFM* = mass flowrate) equate the controlled room pressure differential (0.05 in of water greater than the surroundings) to the head loss due to friction through each of the exfiltration routes, *in terms of air velocity*. Then, we can say:

$n$ = number of separate routes

Exfiltration CFM = $\Sigma$ (Exfiltration route area $\times$ Velocity)$_i$ ft$^3$/min., i = 1 through n,
and

Exfiltration SCFM = ($\rho \times$ Exfiltration CFM) $\div$ 0.0763, where $\rho$ is the exfiltration air density in Lb$_m$/ft$^3$ units.

**Step #1**

Pressure drop (i.e., head loss) relationship:

**Friction loss = (Entry loss) + (Duct loss) + (Exit loss)**

$$= (K_{entry} \times V^2/2g) + ([f][L/D] \times V^2/2g) + (K_{exit} \times V^2/2g)$$

**Note**: to ensure proper units, I like to use the standard American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) conversion from velocity to velocity head for air @ *standard conditions:* ($\rho_{STD} \cong 0.0763$ lbm/ft$^3$):

**Std. air velocity head, *in of $H_2O$* gauge ≅ (Air velocity in ft/min/4,005)$^2$**

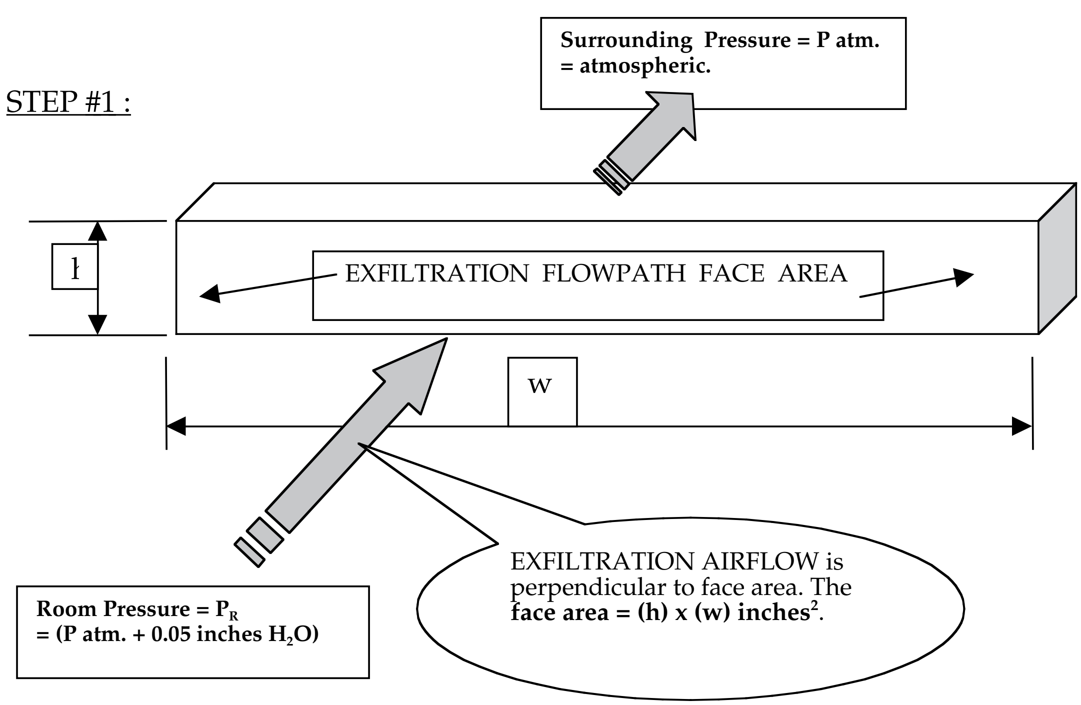
**For any** gas at any density $\rho$, *in lbm/ft$^3$*, to obtain gas velocity head *("VH")* in the convenient ASHRAE units *"in $H_2O$ gauge"* where velocity is still in ft/min,

$$\text{VH, } in\ of\ H_2O \text{ gauge} \cong (8.30592 \times 10^7) \times (\rho, \text{lbm/ft}^3) \times (V^2) \cong (\rho)\,(V/1{,}097.25)^2$$

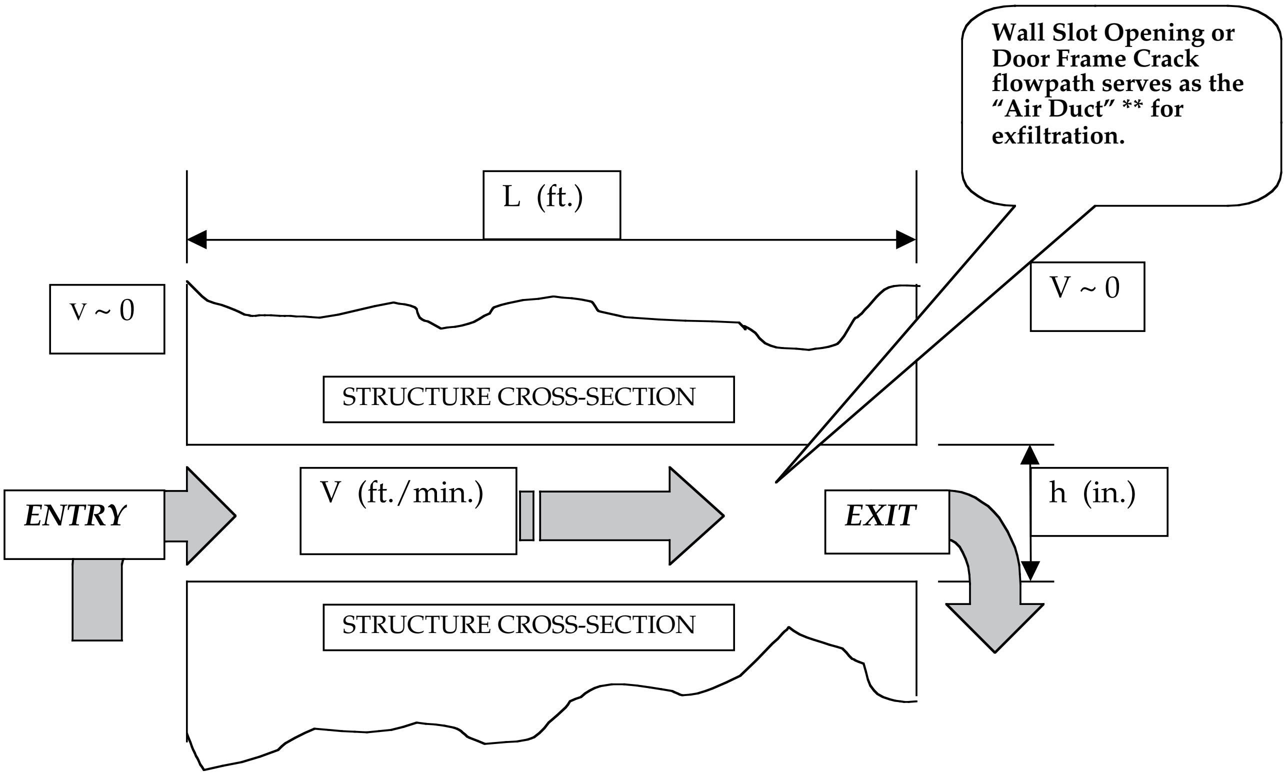
Room air conditions can be corrected to standard density in the usual "perfect gas laws" way. For illustrative purposes, I assume here that no correction is required in our example problem. That is because, in the long run, we will be specifying a mass-constructed Original Equipment Manufacturer (OEM) variety of centrifugal exhaust fan, whose flow versus speed versus total head characteristics are too imperfectly characterized for our variance of a few measly percent in specified air density to make any measurable difference in final system performance. Besides, the variable speed control fan is being driven by an automated loop controller, which will make the room differential pressure remain as desired, so we only need to get "pretty close" with the calculated exhaust fan flow (scfm) to make this thing work satisfactorily. By that, I mean within, say, 5–7% of "exact" on the calculated "air mass flow @ total fan head" requirement for our system design specification. The more precise the measurements of exfiltration crack area and friction loss coefficients, the closer we can come in our SCFM/fan head calcs.

So, for room air near standard conditions, we write:

**Friction head loss across each flow path = {$K_{entry}$ + f(L/D) + $K_{exit}$} × {V/4,005}$^2$ in *in of $H_2O$* gauge units, where velocity V is in units of ft/min.**



**{STEP #1 con't.}**




***Empirical duct wall friction Factor***

## Step #2

Find the Head loss coefficients $\mathbf{K_{entry}}$, $\mathbf{f \times (L/D)}$ **&** $\mathbf{K_{exit}}$

***Duct length to diameter ratio, dimensionless***

From the ASHRAE Guide fundamentals volume, we find:

For a flush entry, $\mathbf{K_{entry}} \cong 0.50$, an empirically determined number, and

For *any type of exit*, by definition, $\mathbf{K_{exit}}$ = exactly 1.00

We must do some work to obtain **f** and **D.** We will read the friction factor **"f"** from the Moody diagram for pipe flow. The Moody diagram requires that we know the *Reynolds number for the flow* $\mathbf{N_{re}}$, and the *dimensionless ratio of duct surface roughness to duct diameter* **ε / D.**

For a flow path having a very large aspect ratio (high value of width to height of the flow face opening), we must use the *hydraulic radius* to define the duct diameter. In our example there are three different exfiltration "ducts" to be found; the wall slots above and below the conduit tray penetrations, the 1/16-in horizontal crack under the door, and the 1/32-in vertical cracks beside the doors in their frames.

The *hydraulic radius*, you may recall, is defined as the Equivalent round duct diameter **"D"** found from: **"D"** = 4 × (flow area/flow passage wetted *(face or cross section)* perimeter). We shall use this definition to find the hydraulic diameter for each exfiltration "duct."

For each of the two rectangular conduit penetrations through the room walls, the cross-sectional flow picture is as shown below. There are two slots per conduit, each having a 12 in × 1 in face dimension. There is no flow through the shaded area, which is a solid metal box. Each slot has an equivalent round duct diameter "D" which equals:

$$\mathbf{D_{wall\ slots}} = 4 \times \text{area/perimeter} = 4 \times (1\text{ in} \times 12\text{ in})/(1\text{ in} + 1\text{ in} + 12\text{ in} + 12\text{ in}) = 48/26 = 1.846\text{ in} = (1.846/12)\text{ ft} = \underline{\mathbf{0.154\ ft}}$$

For the whole room, then, there are four slots, each represented as a round duct of diameter 0.154 ft and of length 1.00 ft (since the wall thickness = flow passage length = 12 in.)

For each of the two vertical door cracks, which are 1/32 in wide and 7 ft long, the equivalent round duct diameter "D" is:

$$\mathbf{D_{vert.door.crack}} = \{ (4)(1/32 \times 12)(7.00)\} \div \{(2)[(1/32 \times 12) + 7]\} = \underline{\mathbf{0.0052\ ft}}$$

For the two horizontal cracks under the door, which are 1/16 in wide and 6 ft long, the equivalent round duct diameter "D" is:

$$\mathbf{D_{horiz.door.crack}} = \{(4)(1/16 \times 12)(6.00)\} \div \{(2)\{(1/16 \times 12) + 6]\} = \underline{\mathbf{0.0104\ ft}}$$

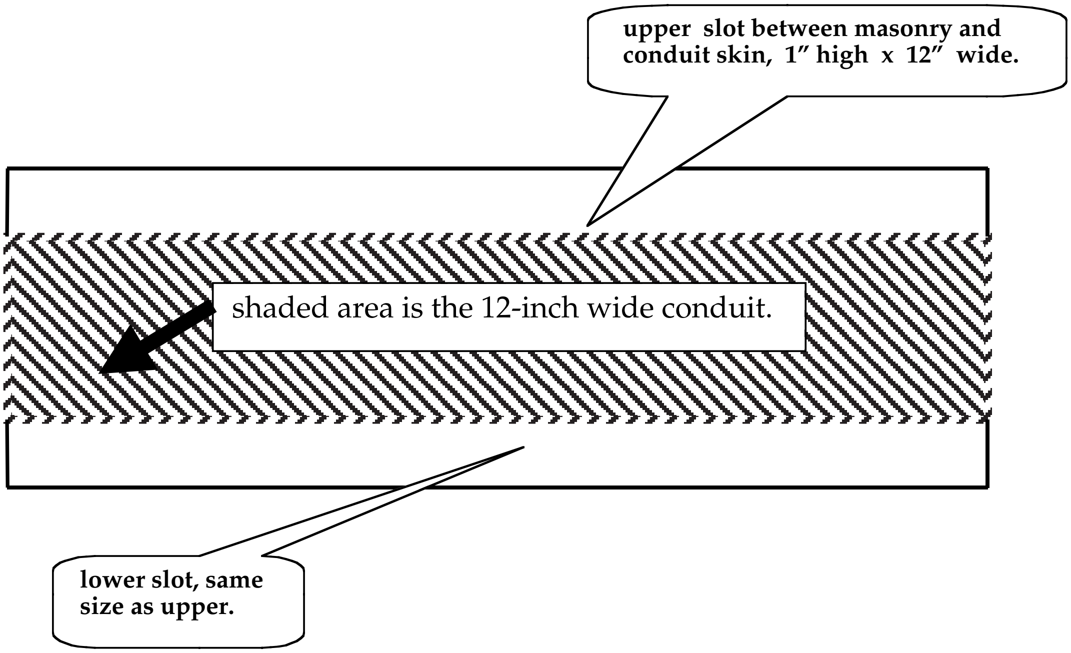
The flow length "L" for the wall slots is 1 ft, the measured wall thickness. The door crack flow length is the thickness of a door leaf, which we will measure and find is $1^5/_8$ inch = 0.1354 ft. Tabulate findings thus far:

**Wall slots: quantity 4**: D = 0.154 ft: L = 1.000 ft

**Vert. door cracks: quantity 2**: D = 0.0052 ft: L = 0.1354 ft

**Horiz. door cracks: quantity 1**: D = 0.0104 ft: L = 0.1354 ft

Next we need the roughness to diameter ratio (**ε/D**) for each exfiltration flowpath. Actual surface roughness **"ε"** (of masonry




plaster, door leaf edges, etc.) depends on material finish and is not easily measured. Maybe you can find data somewhere, or make a decent estimate by comparing your material and finish to the listed values for various pipe and duct materials which often accompany the Moody diagram in reference texts.

We will use a representative value $\varepsilon$ = 1/100 in = 0.01 in, or in foot units

= (0.01 in/12 in per ft) = 0.000833 ft

$(\varepsilon/\mathbf{D})_{\text{wall slots}}$ = (0.000833 ft/0.154 ft) = 0.0054

$(\varepsilon/\mathbf{D})_{\text{vert.door.cracks}}$ = (0.000833 ft/0.0052 ft) = 0.1602

$(\varepsilon/\mathbf{D})_{\text{horiz.door.cracks}}$ = (0.000833 ft/0.0104 ft) = 0.0801

The Reynolds number $\mathbf{N}_{\mathbf{re}} = \mathrm{VD}_{\rho/\mu}$; V = exfiltration flow bulk velocity; $\rho/\mu$ = air density/dynamic viscosity ratio

**For standard air, $\rho/\mu \cong$ (0.075 lbm/ft$^3$) ÷ (0.017 centipoise × 0.000672 lbm/cP-ft-sec) =**

$\cong$ **(0.075/0.0000114) sec/ft$^2$** $\cong$ <u>**6,579 sec/ft$^2$**</u>; we have already derived values for "D"; that leaves us with a problem! What is the value of the exfiltration velocity **"V"?** Not to sweat; just another of the trial and error type of solutions, which seem to come up all the time in these doggoned fluids and heat transfer problems.

For a trial value of "V", we can recognize that velocity and flowrate through the exfiltration cracks are a functions of the fluid viscosity in turbulence, of the total pressure drop available to cause the flow, of the flow cross-sectional area, of the path geometry, and finally the path surface friction factor itself.

Take these factors one at a time:

- The viscosity of air is a function of temperature, and thus is fixed for the standard air temperature condition we assumed (i.e., 0.017 centipoise.)
- The amount of turbulence reflects in the magnitude of head loss coefficients which are $\mathbf{K}_{\mathbf{entry}}$, {f × (L/D)}, and $\mathrm{K}_{\mathrm{exit}}$ and by determining a trial value for them we will be in effect taking care of the turbulence factor.
- We have specified the total pressure loss as 0.05 in of water, so it is not variable.
- The flow area is nonvariable for the problem so we do not have to consider it at this time either.
- The path geometry consists of a flush entry, in which we have already noted that the loss coefficient $\mathbf{K}_{\mathbf{entry}} \cong$ **0.50,** a short straight "duct run" through the length of the crack, which is the **f** × **(L/D)** part, and the exit into the corridor plenum, for which by definition, $\mathbf{K}_{\mathbf{exit}}$ = exactly **1.00.**
- The path surface friction factors **(f)** therefore remain as true variables in our problem. As such, they are the "knobs we can twiddle" in our process of guessing a good starting value for "V". That is what we will do now; "twiddle the knobs" ("reason it out").
- Here is the way I like to "reason it out," although there are certainly other ways to go about it:

The minimum friction factor we can ever possibly find in *any* problem is ***<u>zero</u>***; if there were no duct wall friction at all, then the exfiltration flowrate would be at its maximum possible value; (for the constant "fixed" duct flow area, this means the bulk flow velocity, hence the crack exit velocity, would be maximized at a single resultant value.) This does <u>**not**</u> mean there is ***<u>no</u>*** head loss, it just means that we will see what happens *when the **<u>wall friction</u>*** head loss component of the total is let to be zero. We still have the <u>entrance and exit losses</u> to contend with; in fact, since the duct flow passages are straight and constant in cross-section, the entrance and exit losses are the only losses left to consider. (If there were additional path geometry factors such as duct turns, area shape or size changes, flow over obstructions or through screens, etc. then we would need to consider them too. We would assign suitable "k" coefficient values for them, and "keep on truckin'.")

When the wall friction factor is set to zero, the expression for total head loss of the exfiltration becomes:

**Total head loss = 0.05 in $H_2O$** = (Entry loss) + (Duct loss) + (Exit loss)

*f* is made = zero, so the Duct loss component becomes zero.

$= (\mathbf{K_{entry}} \times V^2/2g) + (\cancel{f})[L/D] \times V^2/2g) + (K_{exit} \times V^2/2g) =$
$(0.5 + 0 + 1.00) \times (V^2/2g)$;

The velocity head term, in inches of water gauge units for standard air, was given previously as:

$(\mathbf{V^2}/2g) = (V/4{,}005)^2$ when V is expressed in feet per minute (ft/min) units.

Therefore we can write: $(\mathbf{V_{max}}/4{,}005)^2 \times (0.5 + 1.00) = 0.05$ in $H_2O$, and $\mathbf{V_{max}} \cong$ **4,005 ✓ (0.05/1.50) ≅ 730 ft/min.** ≅ **(730/60)** or **<u>12.167 ft/sec</u>**

and this is the velocity at which *the artificial maximum possible exfiltration rate* will flow out of the various slot and door crack exit areas in our problem. Keep the artificiality of this velocity number in mind; we are simply finding a boundary limit solution at this time. Now calculate the Reynolds numbers based on this maximum possible velocity:

$(\mathbf{N_{re}})_{\text{wall slot}} = (V)(D)(\rho/\mu) = (12.167)(0.154)(6{,}579) = 12{,}327 = 1.23 \times 10^4$ {*turbulent*}

$(\mathbf{N_{re}})_{\text{vert. door crack}} = (12.167)(0.0052)(6{,}579) = 416 = 0.416 \times 10^3$ {*laminar*}

$(\mathbf{N_{re}})_{\text{horiz. door crack}} = (12.167)(0.0104)(6{,}579) = 832 = 0.832 \times 10^3$ {*laminar*}

In the following section, you will find a copy of Moody's diagram, courtesy of our friends at CRANE, from Page A-24 of their classic Technical Paper No. 410, which every mechanical, civil, and chemical engineer should certainly possess. We will use it to find the duct wall friction factor "f" for each type of exfiltration path in our problem, with the caveat once more that the velocity we used is a max, and the actual velocity will be a bit smaller. We will check the ramifications of this in our post-solution discussion.

- For the turbulent regime flow in the **<u>wall slots,</u>** we read the friction factor from the Moody diagram using our parameters $(\mathbf{N_{re}}) = 1.23 \times 10^4$ and **(ε/D) = 0.0054.**

I get a value for **"f"** of about **<u>0.0368</u>** from the Moody plot.

The door crack flows are laminar, and while we could squint at the chart, it is easier to simply calculate "f" from the laminar regime curve's equation, which is:

$$\mathbf{f = 64/N_{re}}$$

- **Vertical crack** f = 64/416 = **0.154**, and
- **Horizontal crack** f = 64/832 = **0.077**

## <u>Step #3:</u>

Calculate the exfiltration velocities and flowrates through each leakage path, and then the total exfiltration from the room. Here are the formulas:

- To calculate velocity of each path of type "i", denoted **"$\mathbf{V_{path\ i}}$"**:

$$0.05'' \ H_2O = [K_{entry} + (f)(L/D) + K_{exit}]_{\text{path "i"}} \times (\mathbf{V_{path\ "i"}}/4005)^2$$

- To obtain mass flow in *standard cubic feet per minute,* ***SCFM,*** of one path of type "i":

$$\mathbf{SCFM_i} = (\text{Face area}_i)(V_{\text{path } i})(\rho_{air} \text{ in Lbm/ft}^3/0.075)$$

- To calculate total exfiltration from the room:

$$\mathbf{SCFM_{total\ of\ room}} = \sum_{\mathbf{i=1}}^{3} (\text{quantity of path type})_i \times (\text{SCFM})_i$$

**<u>Path Type 1: Wall Slots:</u>**
$0.05 = [0.50 + (0.0368)(1 \text{ ft}/0.154 \text{ ft}) + 1.00] \times (V_1/4{,}005)^2$
$0.05 = [1.50 + 0.24] \times (V_1/4{,}005)^2$

$\mathbf{V_1} \cong 4{,}005\ \sqrt{}(0.05/1.74) \cong 679$ ft/min ≅ 11.3 ft/sec
*(vs. 730 ft/min trial value)*

$\mathbf{SCFM_1} \cong (1'' \times 12''/144 \text{ sq in per sq ft})(679 \text{ ft/min})(0.075/0.075) \cong 56.6$ SCFM

**<u>Path Type 2: Vertical Door Cracks</u>**:
$0.05 = [0.50 + (0.154)(1354 \text{ ft}/0.0052 \text{ ft}) + 1.00] \times (V_2/4{,}005)^2$
$0.05 = [1.50 + 4.01] \times (V_2/4{,}005)^2$

$\mathbf{V_2} \cong 4{,}005\ \sqrt{}(0.05/5.51) \cong 382$ ft/min ≅ 6.4 ft/sec *(vs. 730 ft/min trial value)*

$\mathbf{SCFM_2} \cong (0.0026 \text{ ft} \times 7 \text{ ft}^2)(382 \text{ ft/min})(0.075/0.075) \cong 7$ SCFM

**<u>Path Type 3: Horizontal Door Crack</u>**:
$0.05 = [0.50 + (0.077)(0.1354 \text{ ft}/0.0104 \text{ ft}) + 1.00] \times (V_2/4{,}005)^2$
$0.05 = [1.50 + 1.003] \times (V_3/4{,}005)^2$

$\mathbf{V_3} \cong 4{,}005\ \sqrt{}(0.05/2.503) \cong 566$ ft/min ≅ 9.4 ft/sec *(vs. 730 ft/min trial value)*

$\mathbf{SCFM_3} \cong (0.0052 \text{ ft} \times 6 \text{ ft}^2)(566 \text{ ft/min})(0.075/0.075) \cong 17.7$ SCFM

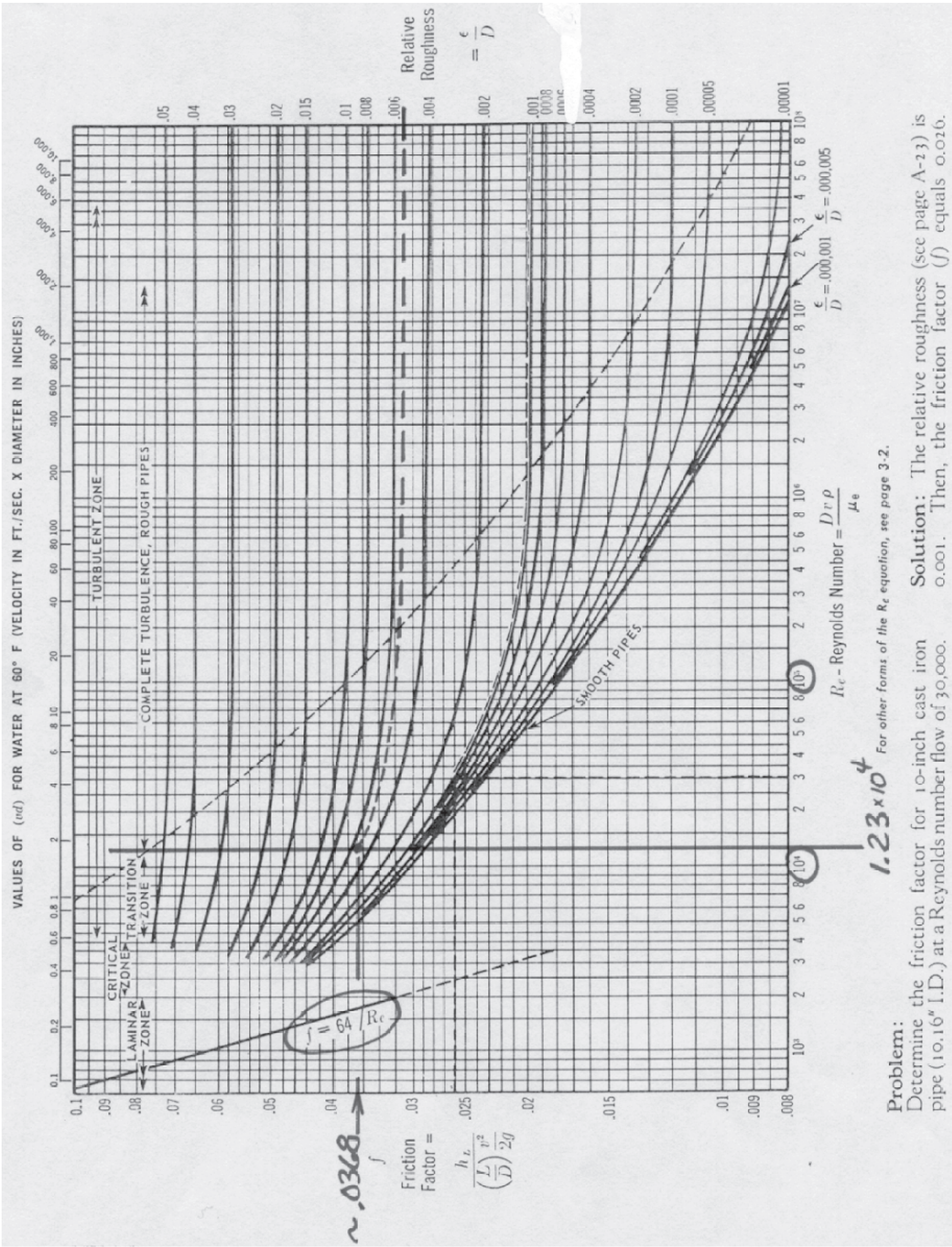
$\mathbf{SCFM_{total\ of\ room}} \cong (56.6 \text{ SCFM})(4 \text{ slots}) + (7 \text{ SCFM})(2 \text{ VC}) + (17.7 \text{ SCFM})(1\text{HC}) \cong 258$ SCFM

## Tabulated Results

| Exfiltration Path Totals | V, ft/min | A, $ft^2$ | SCFM |
|---|---|---|---|
| 1. Wall slots @ conduits | 679 | 0.333 | 226.4 |
| 2. Vertical door edge cracks | 382 | 0.036 | 14.0 |
| 3. Horiz. door floor crack | 566 | 0.031 | 17.7 |
| Room total | — | 0.400 | 258.1 |

## <u>Discussion of Results</u>

1. The wall slots dominate the exfiltration rate. If we wish to reduce the flowrate for whatever reason, we should consider blanking off some of the slot face area accordingly.
2. How accurate are these results? We assumed the velocity thru each path was at the theoretical maximum **730 ft/min,** and we found that 88% of the flow, i.e., wall slots, occurs at a calculated **679 ft/min, which is an error of only about 7.5%.** So our initial assumption was very good, and if we choose to make a second round of calculations to refine our results, using the 679, 382, and 566 ft/min velocity values as trial values for the second round, my guess is that the final difference will not amount to much, and that we will end up selecting the same exhaust fan and drive, regardless of whether we go with 258 SCFM as our basic room leakage exfiltration




FRICTION FACTORS FOR ANY TYPE OF COMMERCIAL PIPE[18] value, or a second (or third) more refined value. I will leave it as a practice exercise for you to make a second set of trial calcs and see if my guess is right.

## CONCLUSION REQUIRED EXHAUST FAN FLOWRATE @ THE DESIGN CONDITIONS

(Exhaust SCFM) = (Supply SCFM) – (Exfiltration SCFM)

**Exhaust SCFM = Supply SCFM – 260**
{answer}

## FINAL THOUGHTS

The "Supply SCFM" quantity is a separately calculated number, of course, which depends on cooling/heating/ventilation load demands and the particulars of the supply air system. In the example problem, what would we do if "Supply scfm" = 200? This would make the Exhaust SCFM a <u>negative</u> number equaling (200 – 260) = **<u>–60 SCFM.</u>**

Which means it would have to be an additional air supply quantity, not an exhaust flow, if nothing else were to be changed. And to avoid upsetting the desired room temperature and humidity, the additional supply air quantity would have to come through the same room air handling unit as an additional load, and the "coil leaving" conditions, reheat quantity, etc. adjusted accordingly for the new SCFMs.

Of course that would be a dumb approach. Instead, to maintain the desired room conditions *including the +0.05″ static pressure differential,* it would be much smarter to seal up those conduit pass-through slots in the wall. That would reduce the total exfiltration loss rate to only about 30 SCFM, via the combined door cracks, and keeping the 200 SCFM supply quantity as it was originally supposed to be, the revised "EXHAUST SCFM" would be (200 – 30) = **(+)170 SCFM;** the "positive" exhaust quantity meaning "flowing in the initially assumed direction", which of course was out the exhaust grille and exhaust fan to the outdoors. Our exhaust fan selection would be for a variable speed controlled centrifugal unit (or equivalent) which would operate continuously, comfortably and relatively efficiently, within a flow range centered on the nominal 170 SCFM design point. The **flow range** "±" and fan total static pressure head specification would be calculated by the engineer from the particulars for the room and mechanical system.

Oh, yes. I was about to forget the "door force" calculation. Well, that is the easy part. One door leaf must be openable at a time, and it is $3 \times 7$ ft or 21 $ft^2$ in area. So, the opening force for one door leaf is the pressure differential × leaf area =

= [ 21 $ft^2 \times 144\ in^2/ft^2 \times 0.05$ in $H_2O \times (0.4331/12)$ psi/ft $H_2O$] = **<u>5.46 lb force.</u>** Whether this is acceptably small enough or not, would need to be determined. Is there some kind of code for that? Probably. But I don't know what it is. I'm just the plumber.

CHAPTER

# 7

# Water Chiller Decoupled Primary—Secondary Loops

This topic is illustrated by the following 31 pages of reproduced visual aids hardcopy. It is part of a "standard" presentation I sometimes make to facility owners, general-purpose plant engineers, and central utility operators who have chilled water plant problems, or plan to make expansions.

Because of the intended audience, I tried hard in preparing the presentation to present what was needed for full qualitative understanding by the nontechnical attendees, while providing enough technical detail for an engineer to quickly grasp the principles involved.

Although the 31 pages are standalone in themselves, I wish to add a few additional points to further your own understanding of the salient points. You should study the 31 pages first, then these notes.

1. The **"Hot Tank-Cold Tank"** chilled water generation equipment configuration achieves hydraulic decoupling of the primary and secondary loops ***in open systems.*** The usual benefits of an open system appeal only to industrial users:
   a. Greatly improved maintenance facilitation of chilled water purity control; inspection and cleanout are easy, because the tank lids are easily designed for access. This is necessary in direct contact systems such as airwashers, and when contamination of various sorts can be expected. General-purpose HVAC system owners have no such needs, and would find the necessity for water cleanliness maintenance an unwelcome burden. They are better off with closed systems.
   b. The provison of adequately large decoupler tanks builds a thermal flywheel effect into the system during times of low load and rapidly changing load. In conjunction with proper controls, variations in chilled water supply temperature can be made arbitrarily small (within reason, of course; fractions of a degree are not out of the question). This can be vital in industrial applications, especially where precise control of airside humidity is a goal, but is unnecessary in commercial and institutional HVAC, straight comfort cooling applications.
2. For ***closed systems*** design, I refer you to the customer literature and design guides distributed by the chiller manufacturers, such as by Trane, Inc. However, in my own designs, I go to extra pains to insure complete hydraulic loop decoupling. I do that by:
   a. Designating two straight runs of pipe, one in each loop, as the ***low-velocity headers.*** I make them always at least one pipe size larger, and often two sizes larger, than normal sizing practice. I orient them one over the other, and pointing in perpendicular directions.
   b. For example, if the primary and secondary loop return headers are sized at 10-in diameter based on flowrate, then I will make the two oversized header decoupler portions 12-in or 14-in pipe size. This eliminates header pipe friction and dynamic losses from the hydraulic decoupling.
   c. If the secondary loop low velocity header is oriented north-south, then I orient the primary loop low velocity header east-west, and either directly above or below, having exactly the vertical separation equal to the side takeout of two standard pipe tees. The decoupler pipe is actually just that: a pair of cross-connected tees, same size as the low velocity headers. On each oversized header spool, the tee is positioned at the center span point. The result is a cross formed by the headers in the horizontal plane, joined at the center by the two joined tee side outlets which make the decoupler. The crossflow between loops is therefore in the vertical plane. And it does so at negligible velocity, ensuring hydraulic decoupling.
3. It is always best to set up a primary-secondary system such that ***on average,*** the <u>primary loop gallons per minute (GPM) exceeds the secondary loop flow.</u> If the secondary loop flow chronically exceeds the primary gpm, control of supply chilled water supply temperature at the secondary loop pump inlet will be more difficult. This is especially important in open systems.
   a. For short periods of operation at low primary loop gpm, say an hour or so, the system will work just fine regardless of flow balance. But if most of the time the secondary flow largely exceeds the primary, you should consider designing redistribution into the secondary loop piping arrangement, to reduce average total secondary loop flowrate. <u>*This will lead to load groupings that tend to increase the average chilled water temperature rise across the terminal units.*</u>
   b. In the same regard, open systems must have sufficient combined hot tank + cold tank mass volume to prevent short-cycling of the chillers. Short cycling results when tank volume is too small to yield adequate "thermal flywheel effect." The manufacturer should be consulted for determining minimum time periods between "chiller off-chiller on" so that you can calculate an appropriate tank volume. Then, simple chilled water temperature-based control logic will take care of business!

## CHILLED WATER SYSTEMS : *THE SECRETS REVEALED* !

Get 100% of your chilled water generating plant's maximum capacity …
… whenever you need it, for as long as you need it, at *any time*.

Get rapid, but *stable*, automatic response to all changes in the chilled water cooling load over the entire range of tonnage demanded, *whether it be near zero, or at full 100% of capacity, or any operating point in between* …

… always at minimum possible total operating cost, regardless of the magnitude of the cooling load and its inevitable patterns of change.

The system which can do this for you has been applied many times, globally, with an excellent record of success :

Its technology is both logical and straightforward, being based on the most fundamental principles of heat transfer and applied thermodynamics.

It requires only the most simple of easy-to-understand, easy-to-use controls.

Its most valuable features are built-into the very system design …

Reliability

Flexibility

Maintainability

Complete Understandability and Predictability of Operation !!

{1} CHILLED WATER SYSTEM PRESENTATION {1}

## APPLICABLE TO ANY CHILLED WATER PLANT :

… from monster-sized industrial plants having 20,000 tons or more of chilled water peak generating capacity ~

…. to more "average"-size plant systems doing both "process cooling" duties, requiring precise environmental control, and "comfort HVAC" duties, for the plant's office and personnel support areas, over a combined cooling load range of 20 to 10,000 tons ~

….. to small, specialized, skid-mounted industrial process cooling systems dedicated and tailored to specific, rapidly - changing chiller loads ranging from 2 to 20 tons ~

……… as well as being the system of choice for clients with specially difficult needs, such as :

Hospital Complexes,

University Campuses, and

Large Commercial Buildings, with typically large daily and seasonal load swings.

{2} CHILLED WATER SYSTEM PRESENTATION {2}

## A TRUE REMEDY FOR OVERLOADED PLANTS :

..... Often applied to "overloaded" existing chilled water generating plants by retrofit during a scheduled facility shutdown period, keeping existing chillers and cooling towers in place, and making use of as much existing piping as practical, via tie-in to new piping work during the shutdown. Much of the new piping can usually be shop-prefabrication arrangements Existing piping and controls hookups to terminal units are left as-is, wherever possible and prudent to do so.

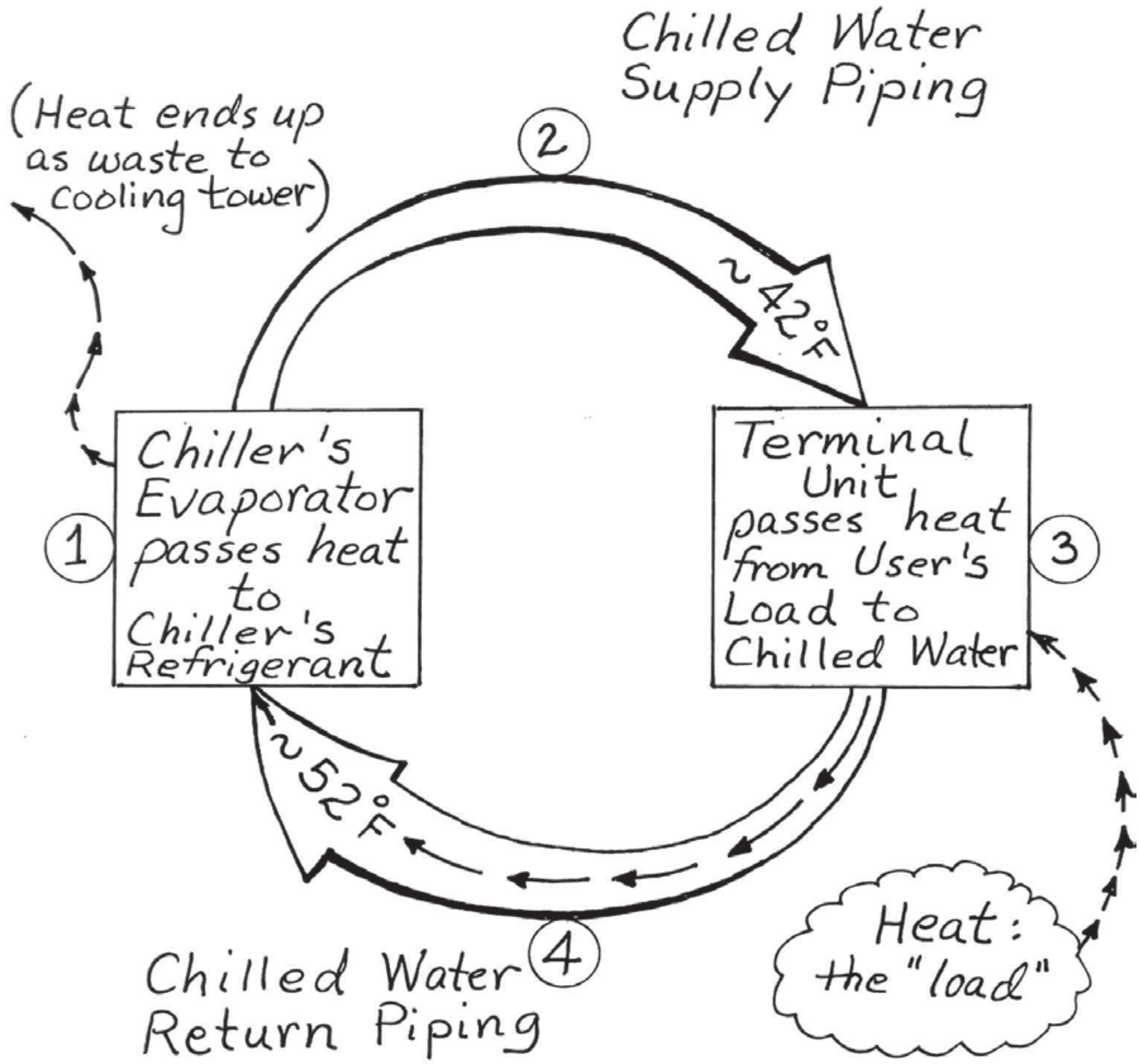
..... The retrofit usually results in the happy discovery that the reconfigured chilled water system has a surprisingly large surplus tonnage capacity, instead of the formerly-perceived shortage.

In later slides, we will provide the complete technical explanation of why this occurs.

{ 3 }　　CHILLED WATER SYSTEM PRESENTATION　　{ 3 }

***And now, the facts :*** NECESSARY FACT No. 1

***A chilled water system must continuously recirculate a stream of water through a sequence of four mechanical subsystems :***



**Facts, continued :** NECESSARY FACT No. 2

**A chilled water generator (chiller) provides its best performance --- i.e. maximum capacity and efficiency --- when the <u>water flow rate (CHW) through the chiller's evaporator is held strictly constant , at a certain specific value</u> which is determined by the chiller manufacturer to provide the desired range of chilled water temperatures. Here's how it works :**

| CHWR (Return) | EVAPORATOR | CHWS (Supply) |
|---|---|---|
| → | operating @ 100% | → |
| $T = 52\,^{\circ}F$ CONSTANT FLOW | full cooling capacity | $T = 42\,^{\circ}F$ CONSTANT FLOW |

Note: chiller designed for [$10\,^{\circ}F\ \Delta T$ :]
$(52^{\circ} \rightarrow 42^{\circ})$, $(50^{\circ} \rightarrow 40^{\circ})$, etc.

By 1$^{ST}$ Law of thermodynamics for the water :

- Rate of Heat Exchange, in BTU's per Hour

$$= (\text{mass flowrate} \times \text{specific heat} \times \Delta T)$$

$$= 500 \times (\text{Gallons / Minute}) \times \Delta T \;;$$

One Ton of Refrigeration $\equiv$ 12,000 BTU/Hr

$$\therefore \left\{ \begin{matrix} \text{GPM per} \\ \text{ton of} \\ \text{chilling} \end{matrix} \right\} = \frac{12{,}000\ \text{BTU/hr.} \times 1\ \text{ton}}{500 \times [10\,^{\circ}F\ \Delta T]} = \mathbf{2.40}\ \frac{\text{GPM}}{\text{ton}}$$

{6} CHILLED WATER SYSTEM PRESENTATION {6}

## ***Facts, continued :*** NECESSARY FACT No. 3

***By itself, the evaporator is very simple :***

- ***If the design engineer matches the chiller capacity (tonnage) to the demand (cooling load), and***
- ***if the load range is never less than about 50% or more than 100% of the chiller's capacity, and***
- ***if the CHW flow rate through the evaporator is held constant at the optimum specific value of 2.40 GPM per ton of chiller capacity,***

***then the chiller's factory-standard internal thermostatic load controller would automatically vary the cooling rate (amount of refrigeration) to match the user's load demand, maintaining the water temperature leaving the chiller (i.e. the CHW Supply Temp.) at a constant 42 °F.***

If there were no other factors , then no other controls would be necessary !

**But, of course, life's not that simple ; there are other factors to consider ...**

## *Facts, continued :* NECESSARY FACT No. 4

***Cooling demand varies over a wide range , depending on time of year, time of day, outdoor weather conditions, process conditions and interior loading factors in the facility being cooled. Typical range of demand for cooling tonnage is from near zero load to 100% load over the course of a year.***

***The way those variations are accommodated in chilled water systems is by having terminal unit thermostats, which are completely independent of the chillers and of each other, to make corresponding variations in CHW flow rate through each terminal unit's heat exchanger, as needed to satisfy the conditions of the moment.***

***Need to satisfy a demand increase? Then supply increased CHW flow rate (GPM) to the terminal unit, and vice-versa.***

***Physically, this is accomplished by modulating the size of the flow opening in a CHW flow control valve in each terminal unit's piping circuit; i.e. by having each thermostat "stroke" the chilled water supply flow control valve serving its terminal unit.***

***<u>Therefore, it is absolutely necessary that the chilled water flow rates through the user terminal units be permitted to vary randomly, anywhere from Zero GPM to 100% of Max Design GPM !!</u>***

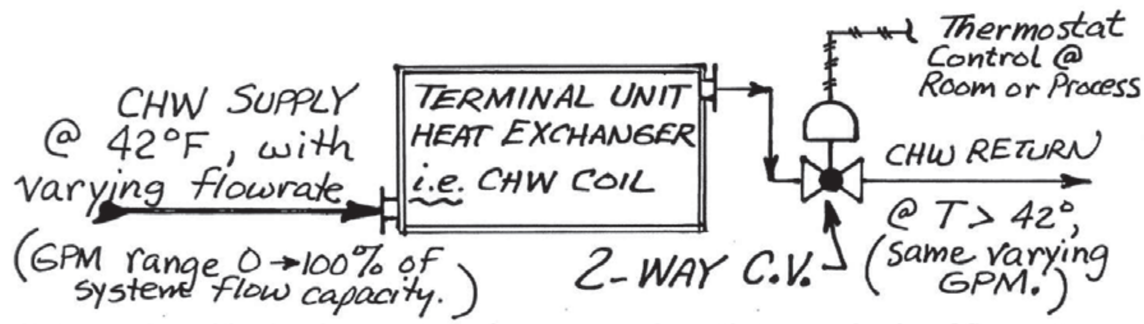
***TERMINAL UNIT FLOW RATE OF CHILLED WATER CANNOT BE HELD CONSTANT !!***

{ 8 } CHILLED WATER SYSTEM PRESENTATION { 8 }

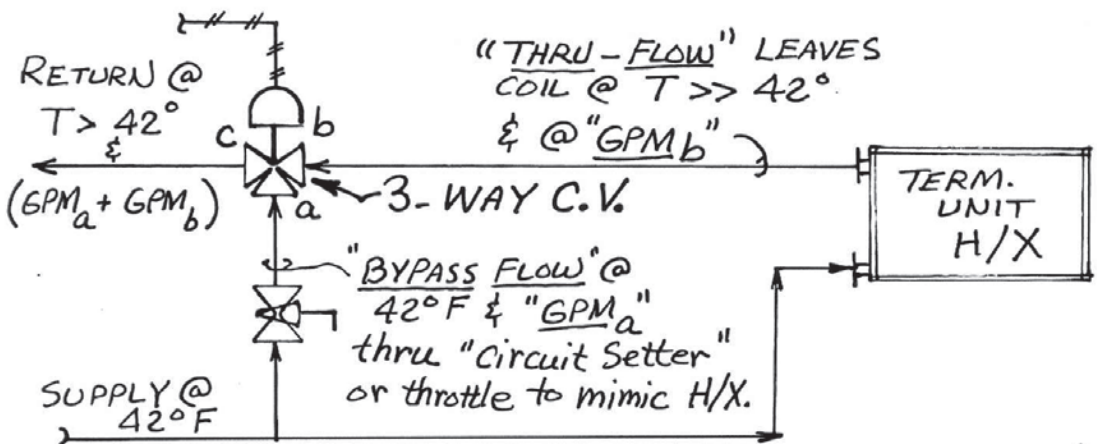
## ***Facts, continued :*** NECESSARY FACT No. 5

***Terminal unit flow control valves are usually one of two different types: TWO - WAY and THREE – WAY. Many systems use both types.***

Two-way valves definitely cause variable flow rates, as seen below:




Set up painstakingly, three-way valves can reduce the magnitude of flow rate fluctuations seen by the overall CHW system. But because the 3-way valve's ***pair*** of characteristics **($C_v$)** change with stem position, and by different amounts, they simply cannot provide ***constant*** system flow rate.




Values of ($GPM_a$ + $GPM_b$) are not "constant enough"!

## ***Facts, continued :*** NECESSARY FACT No. 6

***99 times out of 100, existing chilled water system problems stem from <u>one root cause</u> : namely , failure of the system designer to resolve the unavoidable conflict between :***

***the NEED for CONSTANT flow rate through the chiller evaporator, @ 2.40 GPM per ton of chiller's maximum design capacity ,***

***and***

***the NEED for VARIABLE flow rate through the terminal unit, @ whatever random value between 0 % and 100 % of maximum design flow the terminal unit thermostat forces upon the system at any given time.***

{ 10 } CHILLED WATER SYSTEM PRESENTATION { 10 }

***Facts, continued :*** NECESSARY FACT No. 7

***That root cause, the failure during original system design to resolve the flow conflict, is an HYDRAULIC DESIGN PROBLEM !***

***And that problem has an optimal "fix" , which is conversion to th type of system which we are about to disclose and discuss.***

***It involves making changes in the chilled water piping , plus changes in the chilled water pumping , plus a (simplifying) modification to the external chiller controls.***

{11} CHILLED WATER SYSTEM PRESENTATION {11}

## ***Facts, continued :*** NECESSARY FACT No. 8

***Historically, the problem has been perceived (wrongly) in one of two ways:***

***... as a "shortage of chilled water" ;*** usually "fixed" by installing additional chilled water pumps in parallel with the original pumps. This does not help, and system performance hardly changes (but the electricity bill does increase !)

***... or as "needing additional chiller capacity",*** with an attempt to fix by adding another chiller in parallel with the original chillers. This action not only does not help, but usually exacerbates the situation by making the system unstable in its operation.

Finally, the frustrated owner generally brings in a succession of HVAC controls people, one after the other, who tweak // change the controls endlessly and fruitlessly, being "fired" in turn by the owner as the resulting chilled water plant operation becomes more and more complex, unworkable, and unsatisfactory.

***Like we said, the root problem is in the basic original design of the piping system ; the failure to resolve the conflicting flow needs of chiller evaporators versus terminal unit heat exchangers.***

***All the other "fixes" were doomed from the start.***

## HOW THE CONFLICT IS RESOLVED : *The Optimal Solution*

The solution is usually called the "**PRIMARY – SECONDARY SYSTEM**."

It is actually a chilled water pumping & piping scheme , which breaks the conflicting parties – ***Chiller Evaporators vs. Terminal Units*** – into two separately-pumped piping loops. The loops allow flow of chilled water from one to the other ***without hydraulic coupling*** , meaning flow rate in one loop does not affect flow rate in the other, even though the two "separate" loops share the same physical volume of water.

This is what enables the **Chiller Evaporators** to receive the ***"constant, precise 2.40 GPM per ton"*** flow rates they need , while the **Terminal Units** receive ***the "randomly variable 0 – 100% of max design"*** flows which they must have.

The constant stream through a **chiller evaporator** is called a ***Primary Loop***.

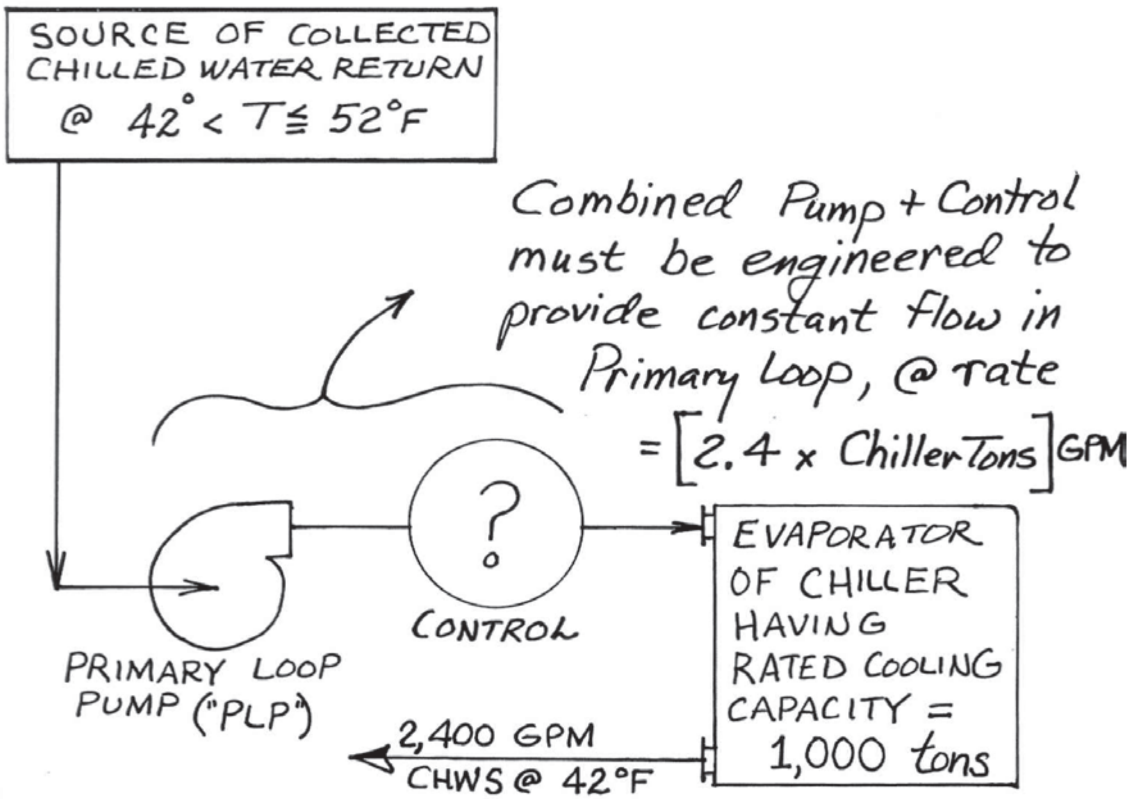
The ever-changing stream through the **terminal units** is a ***Secondary Loop***.

*Let us now review the technical basis and design features which make this system work.*

{13} CHILLED WATER SYSTEM PRESENTATION {13}

## CONSTANT EVAPORATOR GPM

### *The PRIMARY LOOP*



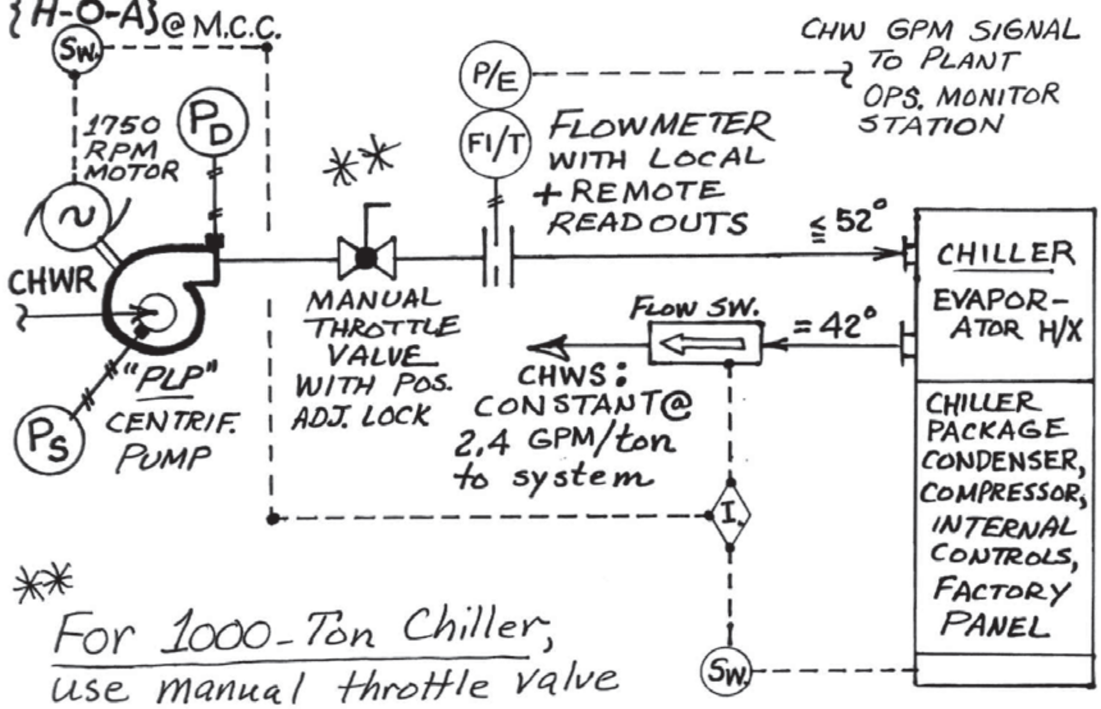

***For example , a chiller is rated at 1,000 tons capacity when generating chilled water at 42 °F, with the usual 10 °ΔT :***

- The primary loop flow must be **(2.40 x 1,000) = 2,400 GPM**.
- The chiller is able to cool a maximum flow rate of **2,400 GPM** from a **52 °F** return temperature to the **42 °F** supply temperature, which is set on the chiller's control panel.

{14} CHILLED WATER SYSTEM PRESENTATION {14}

## CONSTANT EVAPORATOR GPM

### *Configuring The PRIMARY LOOP*



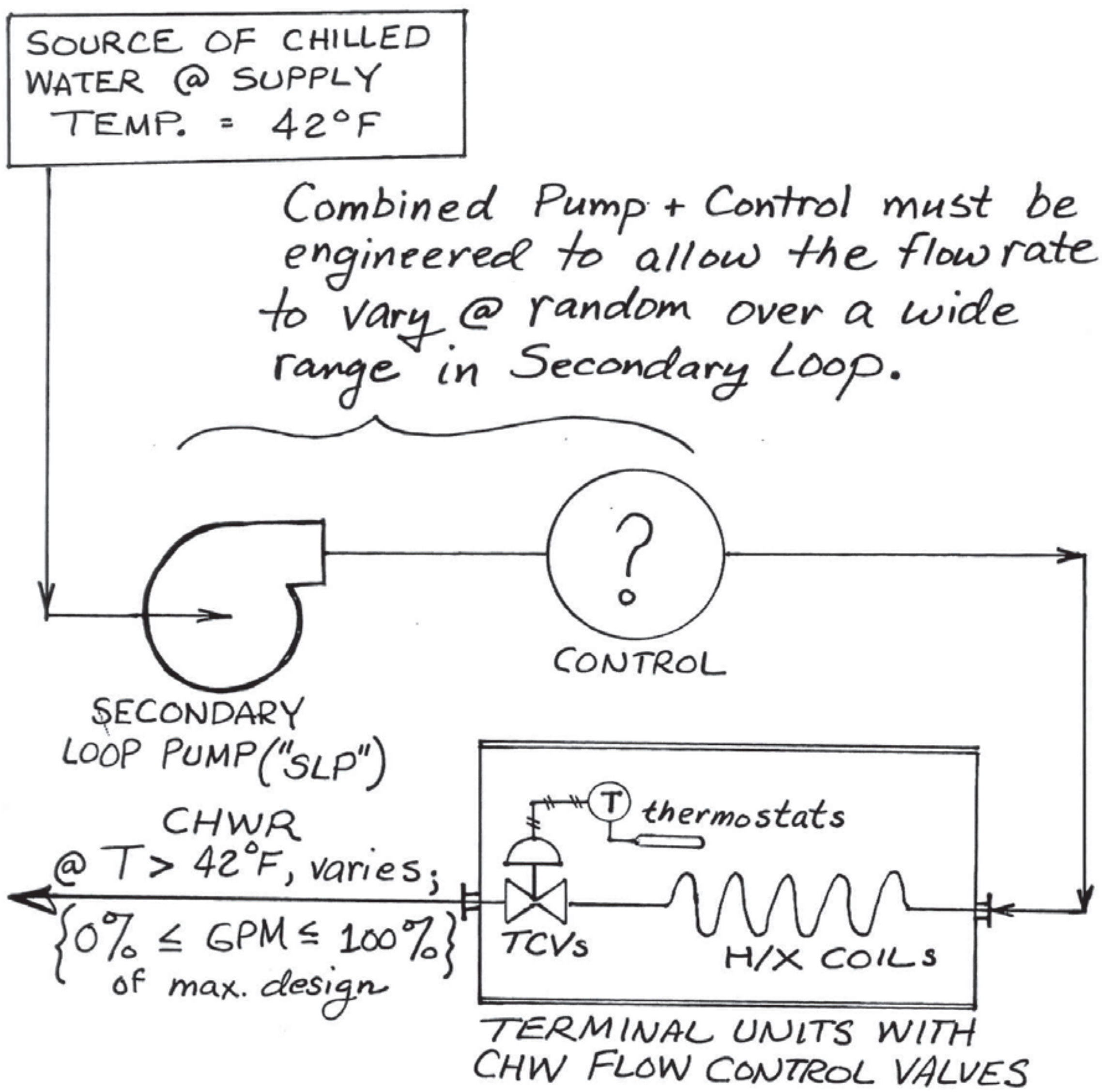

** For 1000-Ton Chiller, use manual throttle valve to set pump delivery @ exactly 2400 GPM.
Lock the throttle at that setting. Periodically monitor the flowrate, reset 2400 GPM adjust. as req'd. if it "drifts" over time.

The primary loop's hydraulic system design must ensure that both the pressure readings at the pump, $P_S$ and $P_D$, remain constant. In other words, the pump's total dynamic head, labelled "TDH" on most graphs of the pump's delivery curve (which equals **0.4331 x [$P_S$ - $P_D$]** with **$P_S$ & $P_D$** in psi units) does not change.

{15} CHILLED WATER SYSTEM PRESENTATION {15}

## VARIABLE TERMINAL UNIT GPM

### *The SECONDARY LOOP*




**PLEASE SEE THE DISCUSSION FOLLOWING ON PAGE {17}**

## VARIABLE TERMINAL UNIT GPM

### *The SECONDARY LOOP (continued)*

Examples, for a **1,000-ton chilled water plant producing 42°F supply water** :

@ 6:45 a.m., the total terminal unit cooling load demand = **583.3 tons** ;
We measure CHW return temperature = **49°F**, so **ΔT = (49 - 42) = 7°F** ;
The secondary loop GPM would be = **(500 x 583.3 x 7°F / 12,000) = 2,000 GPM**

@ 3:45 p.m., the total terminal unit cooling load demand = **939.5 tons** ;
We measure CHW return temperature = **50.2°F**, so **ΔT = (50.2 - 42) = 8.2°F** ;
The secondary loop GPM would be = **(500 x 939.5 x 8.2 / 12,000) = 2,750 GPM**

**We cannot directly calculate secondary loop GPM, because we cannot directly measure terminal unit demand (since "tonnage" is a rate of heat transfer, and like all types of energy transfer, cannot be measured directly but can only be deduced by calculation of other measured quantities.)**

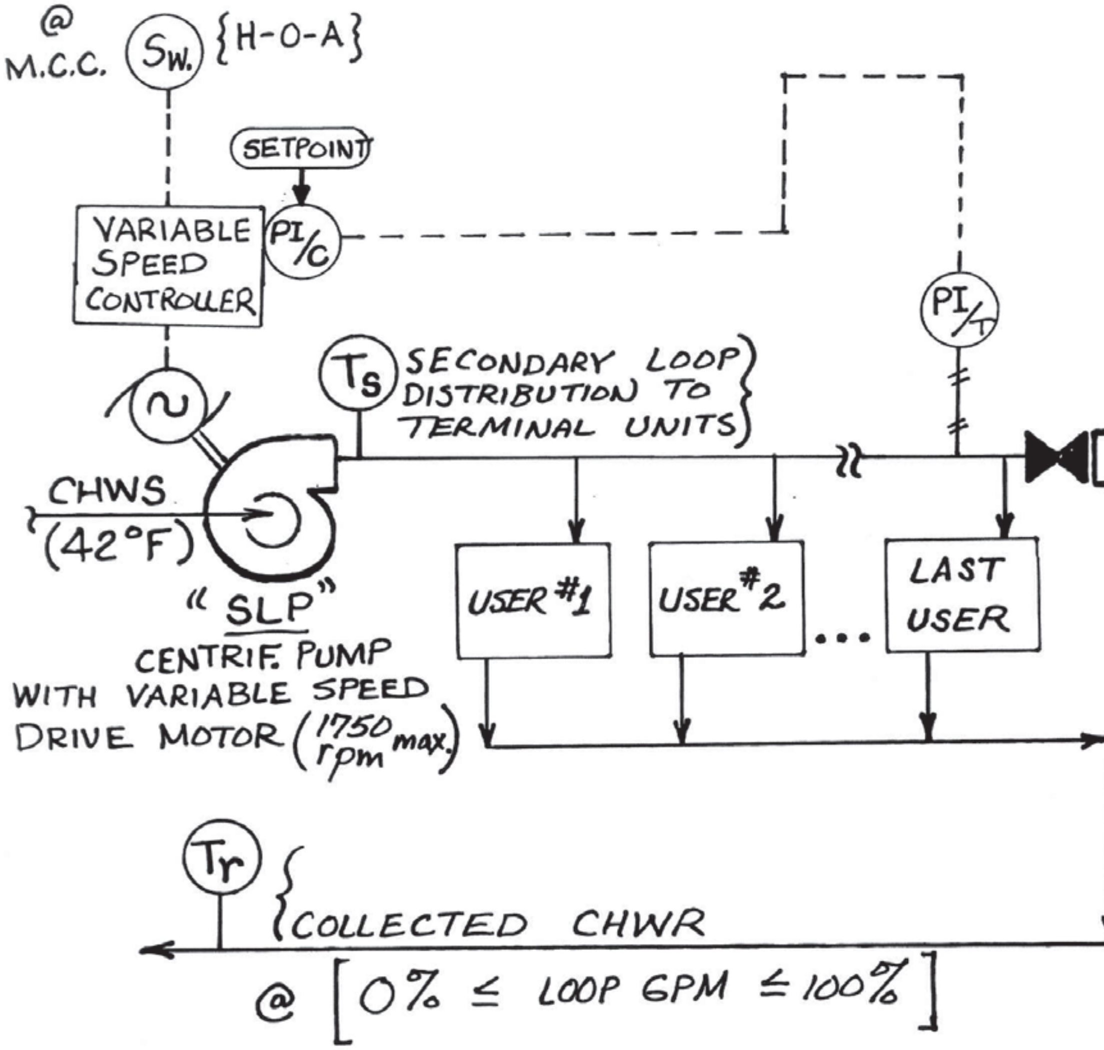
We could use a flow meter to directly measure the GPM if we wished ; then we could directly calculate the terminal unit demand (tonnage) at that instant time, by calculating it from our direct measurements of instantaneous GPM and **ΔT.**

**However, we do not need to know either the instantaneous secondary loop GPM or the instantaneous terminal unit cooling load demand ! The system design engineer needs to know approximately and conservatively their maximum annual peak values and range , obviously , to properly design the chilled water plant. Otherwise, the actual instantaneous values are of academic interest only. *THIS IS A KEY TO UNDERSTANDING HOW CHILLED WATER SYSTEMS ACTUALLY WORK.***

{17} CHILLED WATER SYSTEM PRESENTATION {17}

# VARIABLE TERMINAL UNIT GPM

## *Configuring The SECONDARY LOOP*




PLEASE SEE THE DISCUSSION FOLLOWING ON PAGE {19}

## VARIABLE TERMINAL UNIT GPM

### *Configuring The SECONDARY LOOP (continued)*

***Constant Pressure Control 'Stat Setpoint*** = the recorded steady value of static test pressure measured at the PI / T location after initial post-constructio system startup and flow balancing have been completed, with the secondary loop pump intentionally driven at 100% of its maximum speed while all termina unit chilled water control valves are intentionally held wide open. This determines the maximum possible total secondary loop chilled water flow rate.

At this measured pressure condition, each terminal unit can receive ***at least, if not more than, 100% of the max peak GPM it will ever require***. Th terminal unit thermostat can then take care of the task of setting its own individual flow, by moving the chilled water flow control valve "%-open" as required by the cooling load demand of the moment.

Although in actual operation the CHW flow rate will vary widely because c changes in the "%-open" positions of the terminal unit thermostatic control valves, the constant pressure controller will cause the secondary pump motor speed to vary up or down in response to the tendency of pressure change. Th results in the sensed pressure remaining always constant, at the setpoint valu

*Secondary Loop GPM* = **whatever flow rate automatically results from th secondary loop pump's driven speed (rpm) as required to keep the constant pressure control 'stat "nailed" to its setpoint , under the termin unit flow control valves' cumulative demand of the moment.**

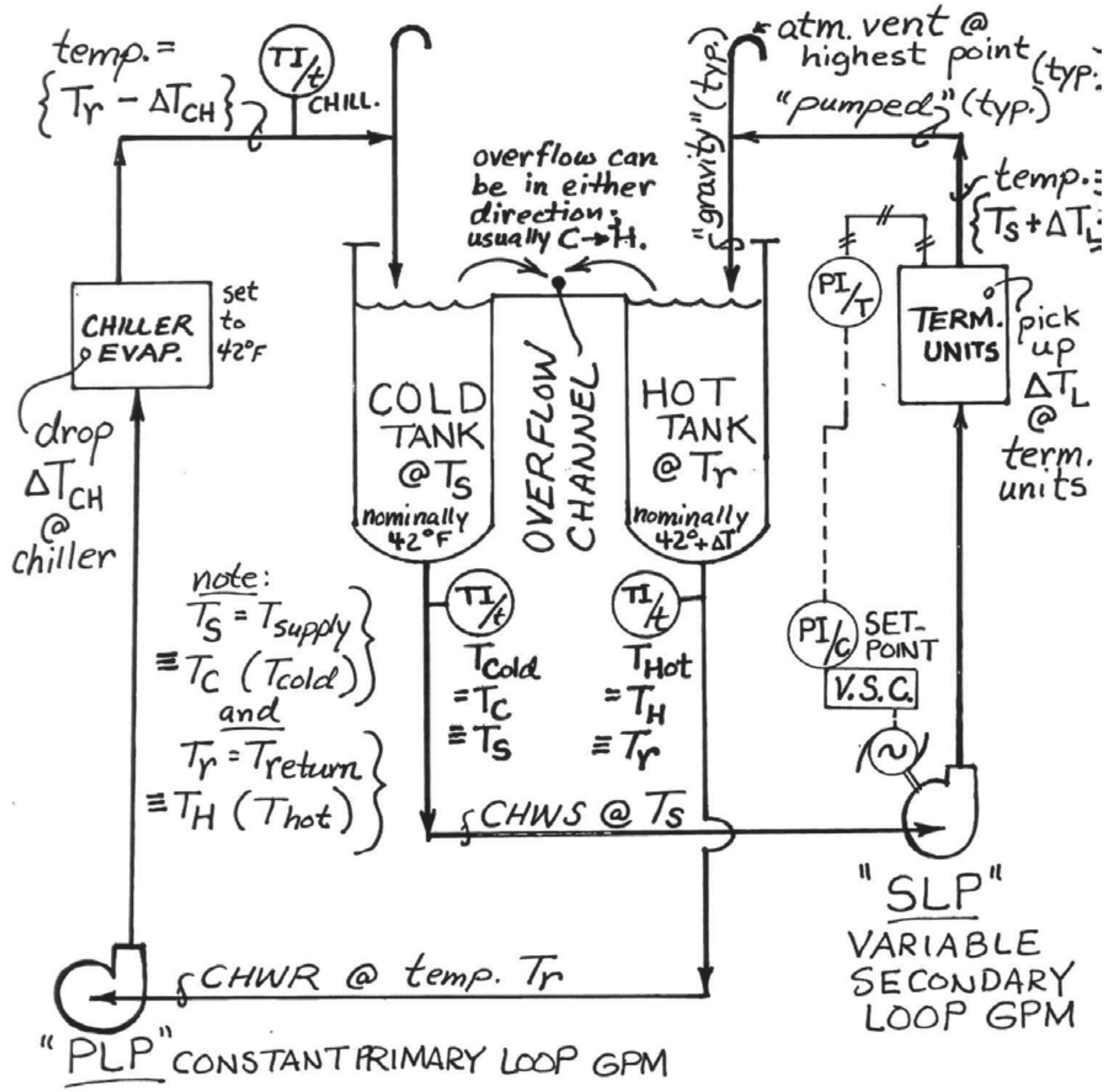
*CHWR Temperature Tr* = **whatever the total loop cooling load of the moment causes it to be ; remember the relationship from thermodynami**

$$\text{LOAD, tons} = \{\ 500 \times \text{GPM} \times (\text{Tr} - \text{Ts})\ /\ 12{,}000$$

{19} CHILLED WATER SYSTEM PRESENTATION {19}

## PRI. & SEC. LOOPS WORK TOGETHER VIA THE "DECOUPLER" :

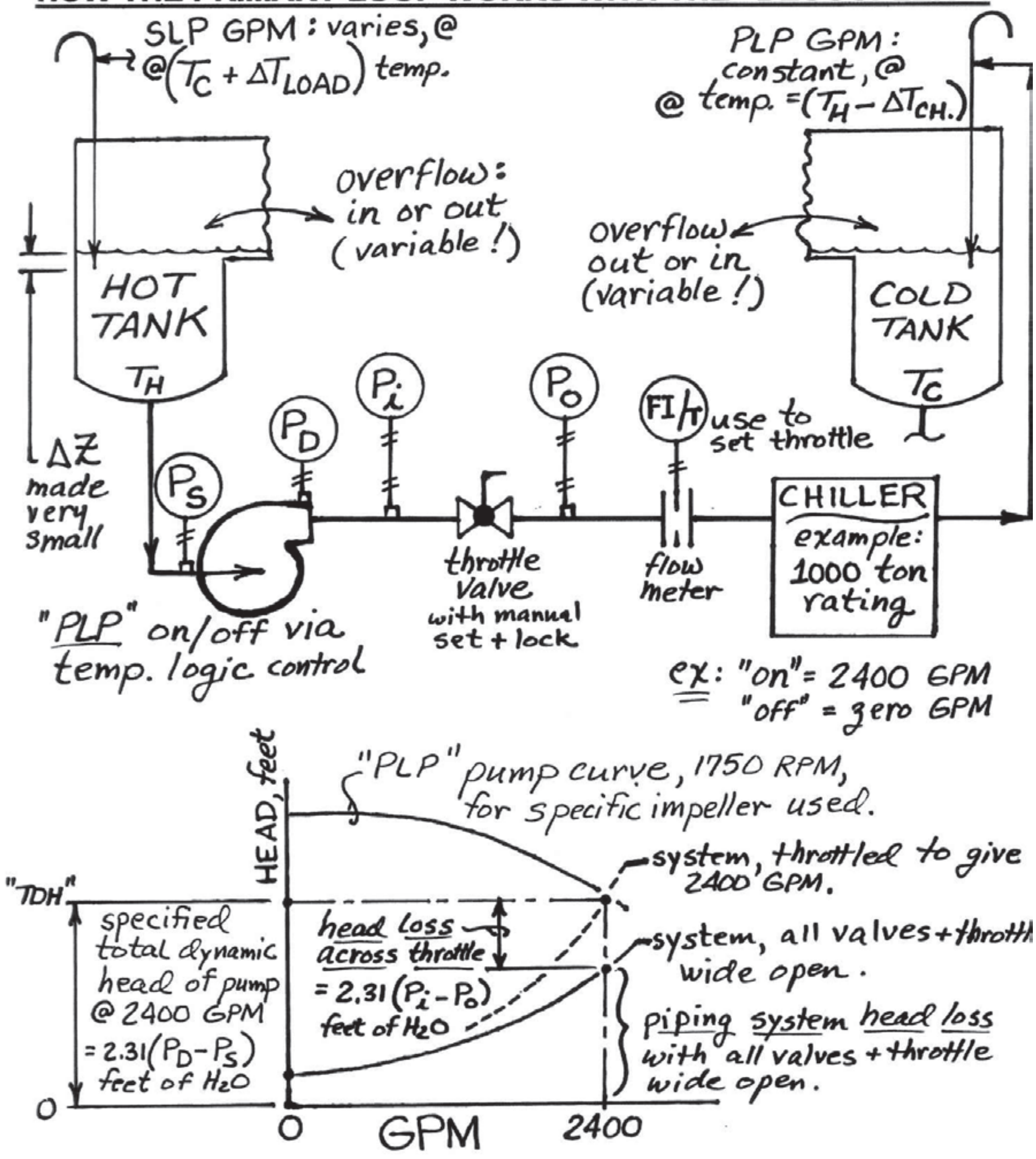
***THE OPTIMUM SOLUTION : the "HOT TANK - COLD TANK SYSTEM"***



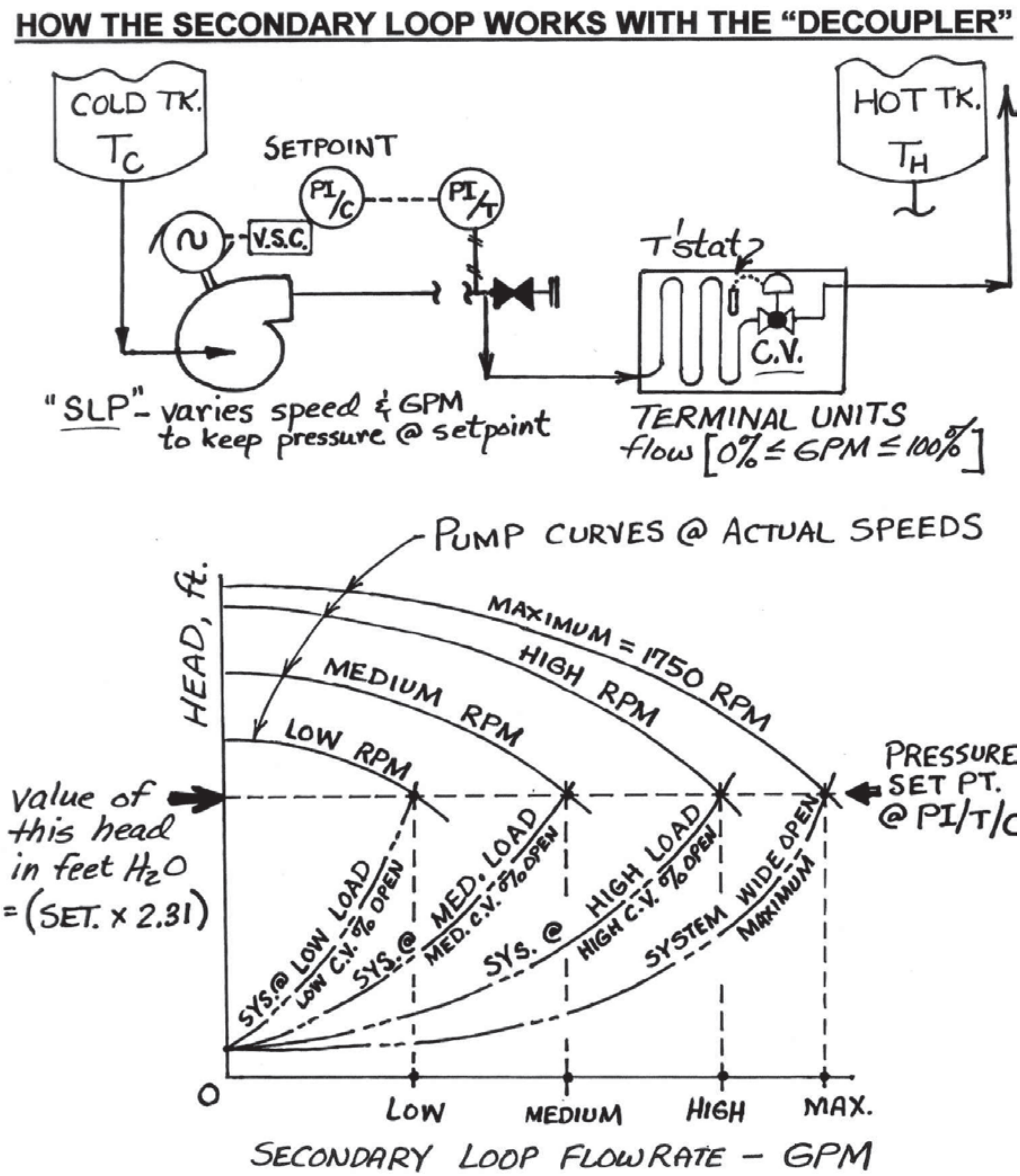

**PLEASE SEE THE EXPLANATION BEGINNING ON PAGE { 21 }**

{ 20 } CHILLED WATER SYSTEM PRESENTATION { 20 }

## HOW THE PRIMARY LOOP WORKS WITH THE "DECOUPLER" :



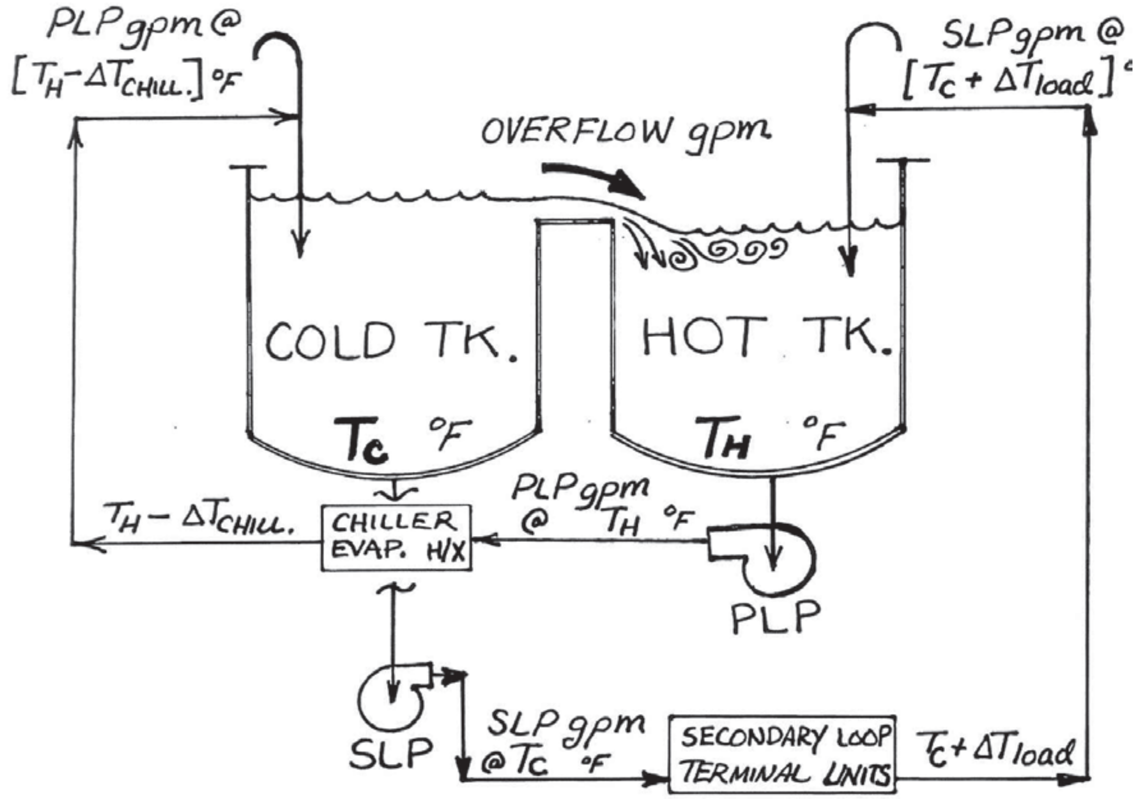

{21} CHILLED WATER SYSTEM PRESENTATION {21}




{22} CHILLED WATER SYSTEM PRESENTATION {22}

## HOW THE "DECOUPLING" TANKS WORK : example #1

**NOTE : System is in thermal & hydraulic equilibrium ; "PLP" and "SLP" are Primary and Secondary Loop Pumps, respectively. A nominally-rated 1,000 ton chiller is running , set at *42 °F*.**

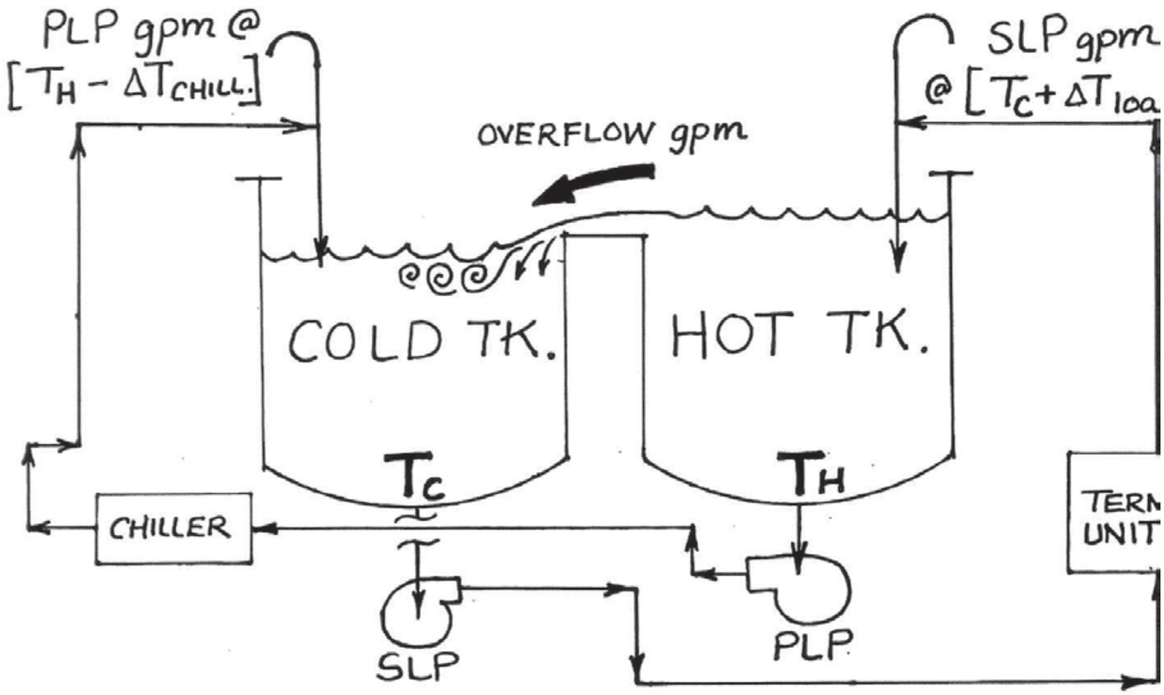



**GIVEN : Secondary Loop Load = 500 tons , SLPgpm = 800 GPM.**
***PLPgpm = (1,000 ton capacity x 2.4 GPM per ton) = 2,400 GPM.***
***$\Delta$T load = ( 500 tons x 12,000 / 500 x 800 GPM ) = 15.0 °F***
***$\Delta$T chiller = ( 500 tons x 12,000 / 500 x 2,400 GPM ) = 5.0 °F***
***Since the chiller is not overloaded, ( $T_H$ - $\Delta$T chiller ) = 42 °F , so***
***$T_H$ = (42 °F + 5.0 °F) = 47.0 °F***
***Overflow is ( 2,400 – 800 ) = 1,600 GPM , from Cold Tank to Hot Tank , so***
***$T_C$ = ($T_H$ - $\Delta$T chiller), and $T_C$ = ( 47.0 ° – 5.0 ° ) = 42.0 °F***

{ 23 } CHILLED WATER SYSTEM PRESENTATION { 23 }

## HOW THE "DECOUPLING" TANKS WORK : example #2

**NOTE : System is in thermal & hydraulic equilibrium ; "PLP" and "SLF are Primary and Secondary Loop Pumps, respectively. A nominally-rate 1,000 ton chiller is running , set at *42 °F*.**




**GIVEN : Secondary Loop Load = 950 tons , SLPgpm = 2,850 GPM.**
***PLPgpm = (1,000 ton capacity x 2.4 GPM per ton) = 2,400 GPM.***
***$\Delta$T load = ( 950 tons x 12,000 / 500 x 2,850 GPM ) = 8.0 °F***
***$\Delta$T chiller = ( 950 tons x 12,000 / 500 x 2,400 GPM ) = 9.5 °F***
***Since the chiller is not overloaded, ( $T_H$ - $\Delta$T chiller ) = 42 °F , so***
***$T_H$ = (42 °F + 9.5 °F) = 51.5 °F***
***Overflow is ( 2,850 – 2,400 ) = 450 GPM , from Hot Tank to Cold Tank , s***
***$T_H$ = ($T_C$ + $\Delta$T load), and $T_C$ = ( $T_H$ - $\Delta$T load ) = ( 51.5 ° – 8.0 ° ) = 43.5 °***

TO AVOID HAVING "***$T_C$*** " EVER TO EXCEED THE CHILLER TEMPERATU SETTING WHEN THE CHILLER IS NOT OVERLOADED , ***THE SYSTEM DESIGN ENGINEER*** ENSURES THAT THE PEAK POSSIBLE VALUE OF ***SLPgpm*** CANNOT EXCEED ***PLPgpm***. UNDER ANY SET OF POSSIBLE OPERATING CONDITIONS.

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## WIDE RANGE OF COOLING LOAD DEMAND ?

- If a chiller is overloaded, it will still remove its full capacity of heat from the chilled water primary loop stream , but its discharge temperature will rise above its setting. The amount of rise is roughly proportional to the amount o overload. Thermal equilibrium " will eventually halt the rise at a higher "terminal unit space temperature, if other conditions remain steady.

- However, each chiller has a certain minimum load it can handle without freezing up and suffering severe damage. The internal cooling rate (load) controller will modulate the chiller down to that minimum allowable loading, but no further. To protect against freezing, internal safety controls automatically trip the chiller off-line when its discharge temperature falls ver much below the setting. And that invokes a definite nuisance, requiring a manual reset switch to be pushed by a human operator.

So, it is best to break up the chilled water plant into two or more primary loops using multiple identical chillers, each with its own primary loop pump and pipin to & from the decoupler tanks.) The individual primary loops can be turned on and off as called for by variations in total plant demand , keeping the running chillers heavily loaded, where they enjoy their best efficiency, and avoiding the low-load nuisance trips. *An illustrative example might be* :

Annual Peak Plant Chilled Water Demand = **3,000 tons** ;
Annual Minimum Plant Demand = **50 tons** ;
Demand can be prolonged over several days at any point in the range from the minimum 50 tons to the maximum 3000 tons ;
The type of chiller being considered for purchase, can "turn down" to a practica minimum loading = 30% of full capacity ***without energy waste***.
***Solution*** : use 5 independent primary loops, consisting of this lineup:
- 3 identical large chillers at **1,000 tons capacity each** ;
- 1 medium-size chiller at **300-ton capacity** ;
- 1 small chiller of **100-ton capacity** .

**With this mixture, the plant will always enjoy maximum energy efficiency without low-load trips or overloads, regardless of demand ! But how would the chilled water plant be controlled to assure this ?? (next slide!)**

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## CONTROLLING THE CHILLED WATER PRIMARY LOOPS

***All we need to do for primary loop control is to have the optimal mixture of chillers in operation at any given time***. Everything else takes care of itself. By reading three key temperatures while knowing the identity of running chillers at any given time, ultra-simple calculations allow us to use simple "look-up table logic" to determine the optimal mix for plant control, as will be illustrated on the next chart below.

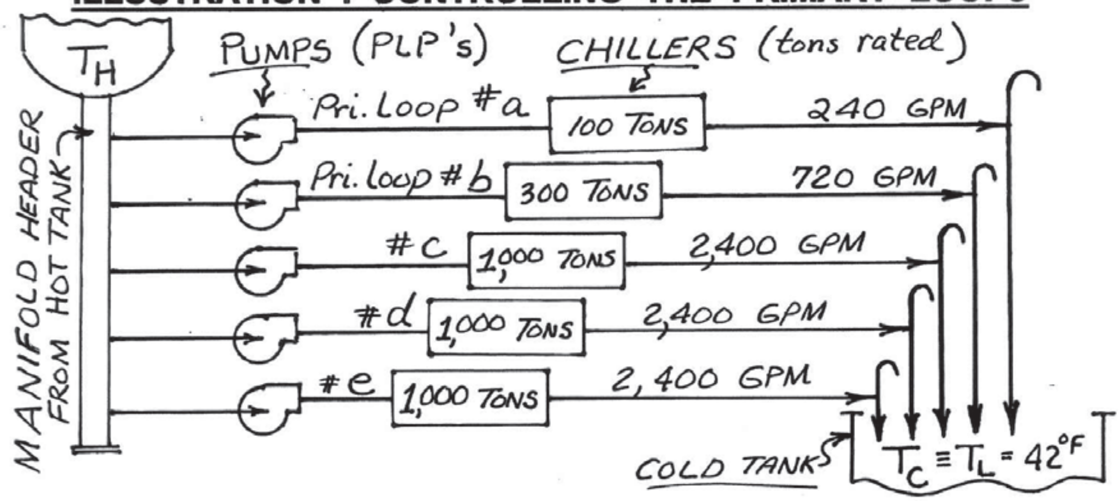
This is made possible ***only*** because each chiller evaporator, when running, always receives the constant, optimum 2.40 GPM of CHW flow per ton of the chiller's 100%-design capacity !!

Now, the basic rules of primary loop control (assuming no equipment malfunctions) are these :

1. If chiller discharge temp remains steadily above its setting, it is **overloaded** ; we will need to turn on an additional chiller.
2. If the chiller discharge temp falls below its setting, we have **underload** , and a low-temperature trip will occur if we don't turn off a chiller.
3. **Plant energy efficiency is improved dramatically by running as few chillers, each at as high a %-loading, as possible at all times.**

To illustrate these control rules, using the preceding example of a chiller plant mixture of 5 machines for a 50 – 3000 ton load range, please see the next chart.

## ILLUSTRATION : CONTROLLING THE PRIMARY LOOPS




***In all cases below, the chillers are set to deliver 42 °F CHW supply temp, and are doing so because they are neither overloaded nor underloaded.***

**CONSOLIDATED POINTS FROM CONTROLLER'S "LOOK-UP TABLE" :**

| **CONDITION:** | **MEASURED:** | | **CALCULATE:** | **RESULT FOR CONTROL:** | |
|---|---|---|---|---|---|
| **LOAD, tons** | **TH °F** | **TL °F** | **ΔT = (TH – TL)** | **RUN THESE CHILLERS:** | |
| 50 to 100 | 47 to 52 | 42 | 5 to 10 | # a {total CHW = 240 gpm} | |
| 101 to 300 | 45.4 to 52 | 42 | 3.37 to 10 | # b | {720 gpm} |
| 301 to 1000 | 45.0+ to 52 | 42 | 3.01 to 10 | # c | {2400 gpm} |
| 1001 to 1100 | 51.1 to 52 | 42 | 9.10 to 10 | Nos. (c + a) | {2640 gpm} |
| 1101 to 1300 | 50.5 to 52 | 42 | 8.47 to 10 | Nos. (c + b) | {3120 gpm} |
| 1301 to 2000 | 48.5+ to 52 | 42 | 6.51 to 10 | Nos. (c + d) | {4800 gpm} |
| 2001 to 2100 | 51.5+ to 52 | 42 | 9.53 to 10 | Nos. (c + d + a) | {5040 gpm} |
| 2101 to 2300 | 51.1+ to 52 | 42 | 9.14 to 10 | Nos. (c + d + b) | {5520 gpm} |
| 2301 to 3000 | 49.7 to 52 | 42 | 7.67 to 10 | Nos. (c + d + e) | {7200 gpm} |

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## ADDITIONAL IMPORTANT INFO :

- *The system design engineer sizes the combined Hot – Cold Tankage volume such that the largest installed chiller can safely be controlled ON – OFF, to meet the cooling load swings, without danger of "short-cycling". Besides "minimum time out", the engineer's sizing calculations must factor-in the allowable rise in chilled water temperatures while the cycling chiller is OFF.*

- *This allowance for temperature rise must be based on a careful study of a number of factors, primarily "user process-related" considerations such as cooling load cyclic data, dehumidification coil temperature requirements, process condenser performance as the chilled cooling water temperature changes, etc.*

- *For closer control of temperatures, larger tank volumes are employed. The increased volumes effectively spread out the thermal changes over time, so that the thermal inertia ("flywheel" effect) gives the control system a sufficiently-large "time constant" for excellent stabilization of the thermal transients. {Simpler commercial and institutional "comfort-cooling-only" applications can get by with only minimal thermal inertia, because of naturally slow-occurring load demand changes, and often can in fact use closed systems, in which decoupling of the primary and secondary loops is done with pipe manifolds in lieu of hot – cold tank pairs.}*

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## A WORD ABOUT COOLING TOWERS :

- *Cooling water flow through the chiller condenser should be set up and controlled exactly as is the evaporator's primary loop , and for much the same reasons .*
- *In standard chiller design , the condenser cooling water flow rate should be maintained constant at 3.00 GPM per ton of rated chiller capacity .*
- *Impressive economic benefits in first cost avoidance and annual power savings can be realized by special condenser design , however . For example , the TRANE Company , which leads the industry , can now furnish highly-efficient chillers which are designed for only 2.00 GPM per ton through their condensers !*
- *Further $$ savings accrue to maintaining as near the optimum cooling water temperature entering the chiller condensers as possible . This is usually accomplished by using specially engineered , thermostatically-controlled 3-way tower bypass / mixing valve arrangements and controls logic, along with the "constant-flow-thru-chiller" feature . This results in a cooling water set-up very analogous to the chilled water primary – secondary loops . Flow through the chiller condenser would always be constant , for maximum chiller capacity , but flow into the cooling tower would be variable according to load and weather , for maximum overall system economy .*
-

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## SUMMARY OF BENEFITS :

*1. Chillers always run as near full load as possible to match changing demand ; essential to yield maximum chiller operating efficiency.*

*2. Eliminates "gang pumping" altogether ! One pump per primary and secondary loop and per condenser , each properly sized for the job , and drawing no more than the minimum horsepower needed ; this yields minimum pump operating cost.*

*3. Optimal condenser water temperature control is a double winner : it insures maximum chiller operating efficiency , and by thermostatically governing variable speed control of cooling tower fans , insures minimum cooling tower operating cost.*

*4. Smooth , stable , and totally-predictable automatic operation of the chilled water plant is built-in to the design from the very beginning.*

*5. Huge annual reduction in kilowatt-hours demanded and consumed results in very attractive payback on investment , in new plants as well as in retrofitting existing installations.*

*6. Provides a way to guarantee the level of precision of chilled water temperature control over wide load ranges , needed in critical process applications , without wasteful "overkill" schemes.*

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## SUMMARY OF BENEFITS , PAGE TWO :

*7. This approach reduces maintenance requirements to the "bare bone", both in man-hours needed ( fewer pumps , motors , strainers , automated control valves , etc. ) and in level of technical skills required ( no instrumentation / controls / computer super-guru needed just to trouble-shoot , operate or maintain the chilled water generating plant. )*

*8. Easy for process engineer to analyze annual and/or periodic chiller and cooling tower system performance , simply by looking at recorded water temperatures and the equipment motors run-time log. No guesswork necessary to figure remaining capacity , impact of changes , or requirements for future upgrades and plant expansions .*

*9. The system of de-coupled loops makes future expansions simple to plan and execute , with no adverse affect on efficiency ; build-in adequate equipment space and piping provisions for planned expansions during the initial installation , and simply add more loops as needed in the future.*

***Thank you very much for your interest !***

*~ END ~*

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CHAPTER

# 8

# A COLLECTION OF HANDY HYDRAULIC FORMULAS

## Based on an Industry-Standard Reference for Pressure Drop Calculations, Incompressible Fluid Flow in Piping and Ducts—Crane Technical Paper No. 410

### Definitions and Units:

**$K_d$** = coefficient of pressure drop, looked up in tables for fittings and calculated from the ***Kd vs. f relationship*** for straight pipe and duct wall friction **(dimensionless. When the term $K_d$ is used, the value is based on the actual inside diameter for the local piece of pipe.**

**K = without a subscript means that the value is based on the largest pipe inside diameter in the analyzed system. See $d_s$, $d_L$ and b, below.**

**f =** Fanning's friction factor, defined in the ***Darcy-Weisbach*** *pipe/duct fluid flow equation,* calculated from the ***Colebrook*** *equation* or looked up on the familiar ***Moody*** diagram **(dimensionless).**

**L** = length of pipe or duct run, ft

**d** = pipe or duct inside diameter, in

**$d_s$** = smaller pipe inside diameter, in

**$d_L$** = larger pipe inside diameter, in

**$d_{MAX}$** = largest pipe ID in the entire system, in; See the discussion for usage of this term.

$\beta$ **= ratio $d_S$**/$d_L \leqq$ 1.00 (dimensionless)

**D** = pipe or duct inside diameter, ft

**Q** = flowrate, gallons per minute, gpm, GPM

**$C_V$** = valve flow coefficient **(dimensionless),** defined by the following relationship:

"For water at 60°F flowing across the valve with a pressure loss of 1.00 psi, the flowrate in GPM is numerically equal to the valve's value of $C_V$."

**$\Delta$P** = pressure loss, psi; $lb_f/in_2$

$\rho$ = fluid weight density, = **62.37** $lbf/ft_3$ for water at 60°F; $lb_f/ft^3$

**$h_L$** = head loss; **ft of fluid flowing**

**V** = fluid bulk velocity; ft/sec

**g** = gravitational acceleration constant used when weight density $\rho$ is used as given above, i.e., **g ≡ 32.2 ft/sec²**.

**$N_{Re}$** = Reynolds number for flow based on inside diameter or hydraulic radius of pipe or duct **(dimensionless);** **$N_{Re}$**=$VD\rho/\mu$, where the viscosity $\mu$ is expressed in **Lb/ft-sec.** Note: it makes no difference whether we express the pound units in defining $\mu$ and $\rho$ as pounds-mass (lbm) or pounds-force (lbf) since we are working with the fluid's mass density at the earth's surface, where local "g" = universal "$g_c$" = numerically ***32.174;*** **that is, our value of (g/gc) = 1.00; a mass of 1 lbm weighs 1 lbf.**

## FOREWORD

Crane Technical Paper No. 410 is the reference most used by facility and utility design engineers, plant engineers, and piping/HVAC consultants for mechanical (pressure) piping systems analysis.

Commercial software for PCs is available for performing the number-crunching for most parts of the Crane reference. I wrote my own program back in the early 1980s, combining the Crane differential pressure – liquid flowrate relationships and fitting loss data, Bernoulli's equation for solving system curve analysis, Colebrook's equation for calculating friction factor f, and miscellaneous adjunct utilities. I mention this because part of this topic includes a formula derivation made by myself, and it references *"LIQUIDFLOW,"* which is the name I gave my program. I no longer make that program available to others since it runs under an *MS-DOS* shell only, which few people understand any longer. Modern commercial software, such as commercially available applications of the Crane manual, is written for *MS-Windows.*

***I believe strongly that it is dangerous and unprofessional to use engineering software without having the ability to check its output via hand calculations.*** To that end, in this topic I have summarized some of the most useful relationships from Crane Technical Paper No. 410, plus the additional one derived by myself (equivalent loss coefficient for flow across parallel pipe branches.) These are noted or illustrated where helpful.

### Handy Hydraulic Formulas

1. ***Relationship between friction factor f and local loss coefficient $K_d$:***

1. $K_d = f \times (L/D)$

## Example:

Find the **local loss coefficients** $K_d$ for **60°F water** flowing at **7.00 ft sec** through **100 ft** of **nominal 8-in pipe size Schedule 40** steel pipe, as well as for the *same flowrate* going through **100 ft** of **nominal 10-in Schedule 40** steel pipe, both pipes having **average** interior surface roughness, i.e., for **e = 0.0002 feet as taken from the Moody diagram table of roughness data.**

## Solution:

We first must solve for **f** and **(L/D).** To find **f** we *could use Colebrook's equation* (nearly impossible without the right computer program) or we can, and shall, use the ubiquitous ***Moody diagram.*** A copy of the Moody diagram is included here. Your own college text on fluid mechanics also contains this diagram in some form. Note that values for pipe wall roughness **e** are tabulated on the diagram.

a. For an 8-in Schedule 40 pipe:

ε = 0.0002 ft avg. steel pipe
**V** = 7.00 ft sec given
**D** = 7.981 in/12 = 0.66508 ft
water @ 60°F, $\rho = 62.37$ lbm/ft$^3$
water @ 60°F, $\mu$=0.000672 × 1.13 centipoise
= 0.00076 lbm/ft-sec
$N_{Re}$= VDρ/μ =(7)(0.66508)(62.37)/(0.00076)
= 382,062 = **3.82 × 10$^5$**
ε/D = 0.0002 ft/0.66508 ft = **0.0003**
From Moody, **read f = 0.0165**
**L/D** = 100 ft/0.66508 ft = 150.4
$K_d$ = f × (L/D) = (0.0165)(150.4) = **2.482**

b. For 10-in Schedule 40 pipe:

**e =** 0.0002 ft avg steel pipe
**V** = (A × V) for 8 in. pipe/(A for 10 in. pipe) =
= { (8 in.)2 ÷ (10 in.)2} × 7.00 = 4.48 ft/sec
**D** = 10.02 in/12 = 0.835 ft
water @ 60°F, ρ = 62.37 lbm/ft$^3$
water @ 60°F, μ = 0.000672 × 1.13 centipoise
= 0.00076 lbm/ft-sec
$N_{Re}$=VDρ/μ = (4.48)(0.835)(62.37)/(0.00076)
= 306,992 = **3.07 × 10$^5$**
ε/D = 0.0002 feet/0.835 ft = **0.00024**
From Moody, **read f = 0.0165 also.**
**L/D** = 100 ft/0.835 ft = 119.8
**Kd** = f × (L/D) = (0.0165)(119.8) = **1.976**

2. ***To get the overall loss factor for a series of pipes and fittings of various pipe sizes, based on the largest pipe diameter in the series:***

a. $\beta$ **= ratio** $d_s/d_L \leqq 1.00$

**Example:** for the two pipes in the previous **example,** we find;

(1) **For an 8-in Schedule 40 pipe:**

β= 7.981 in/10.02 in = **0.7965** for the 8-in pipe, and of course

(2) **For 10-in Schedule 40 pipe**:

β= **1.00** for the 10-in diameter (largest pipe in the system.)

b. $\mathbf{K = K_d/(\beta)^4}$

Sticking with the two pipes of the previous example, we find;

(1) **For 8-in Schedule 40 pipe:**

**Overall K for 8-in pipe size based upon max pipe size ≡ $K_{10}$**
= $K_{d8}$ in / $(\beta_8)^4$ =
= 2.482 / $(0.7965)^4$ = **6.167**

(2) **For 10-in Schedule 40 pipe:**

**Overall K for 10-in pipe size based upon maximum pipe size ≡ $K_{10}$**
= $K_{d10}$ in / $(\beta_{10})^4$ = 1.976 × 1.00 = **1.976**

3. ***To calculate the head loss $h_L$*** and the pressure drop ΔP vs. local loss coefficient $K_d$ and *vs. overall loss coefficient K:*
$h_L$ = head loss; feet of fluid flowing

**a.** $h_L = 0.00259\ KQ^2/d^4$
**ΔP** = pressure loss, psi; lbf/in$^2$
**b.** $\Delta P = 0.00001799\ K\ \rho Q^2/d^4$

When used with local loss coefficient **"$K_d$"**, you must also use **"$d_{LOCAL}$"** for the pipe diameter in the denominator of Eqs. (3a) and (3b), that is, **use "$d_s$"**, the smaller pipe inside diameter.

When used with overall loss coefficient **"K",** you must also use **"$d_{MAX}$"** for the pipe diameter in the denominator of eqs. (3a) and (3b), the largest pipe diameter in the series, upon which "K" based, i.e., use **"$d_L$"**, the larger pipe inside diameter.

Finally, we are getting to the point of demonstrating the value of the overall-coefficient "K". As we shall see, it makes calculation of points on the system curve very easy.

***You have two choices for calculating $h_L$*** and ΔP for a series of piping elements with eqs. (3a) and (3b): You can:

(1) Calculate the local loss for each local element, using **"$K_d$"**, and then sum all the individual local losses to obtain the overall system loss. Or, you can
(2) Calculate the overall system loss for the entire series of elements using **"$K_{total}$"**, which is found from **Eq. (3c),** below:

**c.** $K_{total} = \Sigma_{i=1}^{i=n} (K)_i$

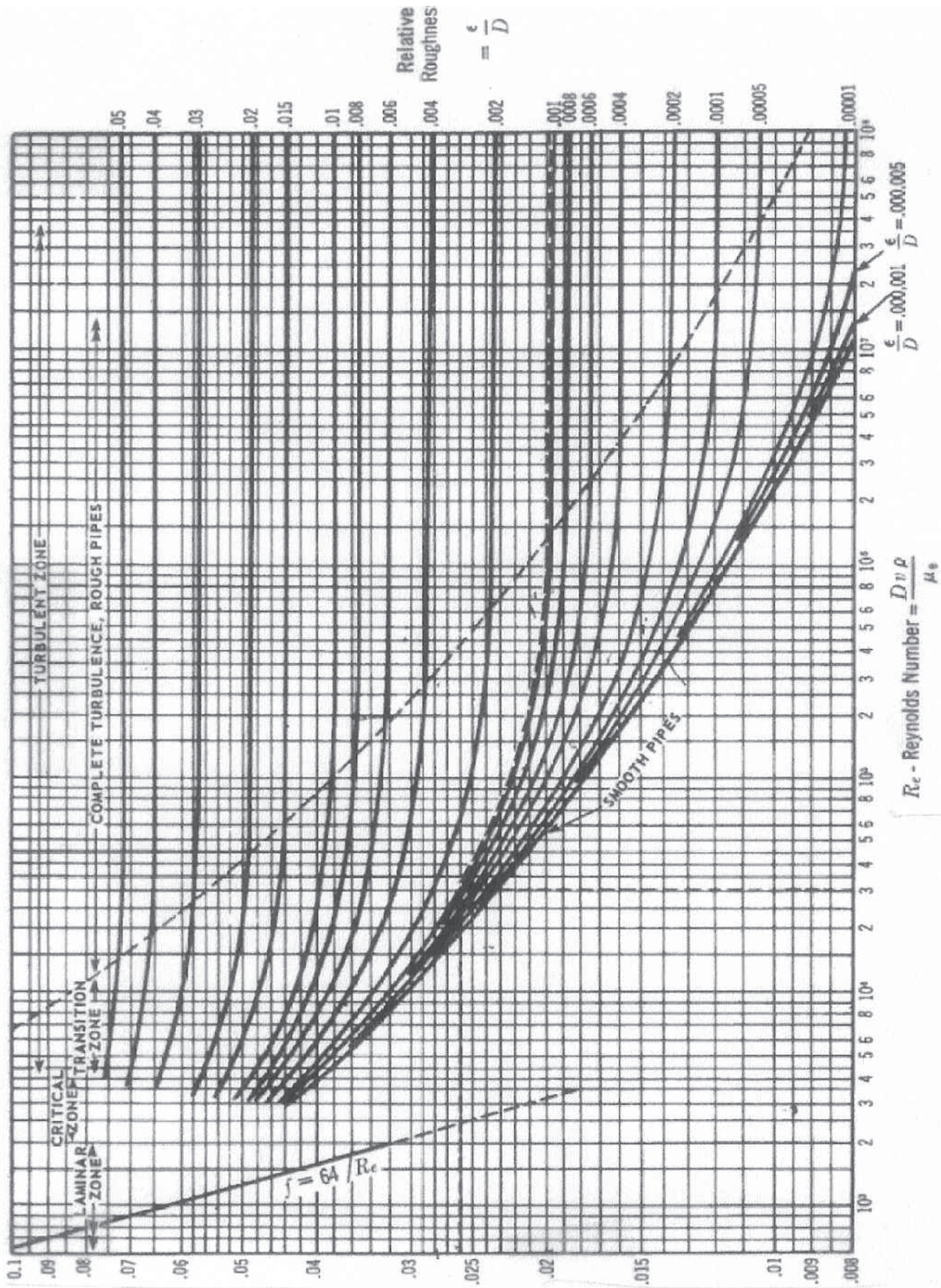
**In eq. (3c) the term $(K)_i$** is simply the *i*th piping element's loss coefficient "K" based on the largest pipe size, and *n* = number of individual elements in the series.

## Example

To demonstrate, let's continue with the two pipes of the **previous example,** only with the addition of a gradual enlargement fitting (welded standard reducer) connecting the two pipe sizes. Flow is from the smaller pipe into the larger pipe through the fitting. Use both methods to find the overall head loss and pressure drop for the system.

## Solution

*First we must find* $K_d$ and **K** for the **gradual enlargement,** reducer fitting. For that we use our Crane Technical Paper Appendix A, page A-26, Formula 3 (@ end of Chapter 8.) It says:




MOODY DIAGRAM

$\epsilon$, in feet

| Kind of Pipe or Lining (New) | Values of ε in ft Range | Design Value |
|---|---|---|
| Brass | .000005 | .000005 |
| Copper | .000005 | .000005 |
| Concrete | .001-.01 | .004 |
| Cast Iron — uncoated | .0004-.002 | .0008 |
| " " — asphalt dipped | .0002-.0006 | .0004 |
| " " — cement lined | .000008 | .000008 |
| " " — bituminous lined | .000008 | .000008 |
| " " — centrifugally spun | .00001 | .00001 |
| Galvanized Iron | .0002-.0008 | .0005 |
| Wrought Iron | .0001-.0003 | .0002 |
| Comm. & Welded Steel | .0001-.0003 | .0002 |
| Riveted Steel | .003-.03 | .006 |
| Transite | .000008 | .000008 |
| Wood Stave | .0006-.003 | .002 |

Colebrook's Eq.

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{\epsilon}{3.76D} + \frac{2.51}{\sqrt{f}\,N_{Re}}\right)$$

Fits these curves for all $N_{Re} \geq 4000$. $\left\{\text{For } N_{Re} < 2000, \text{ laminar flow}; f_{laminar} = \frac{64}{N_{Re}}\right\}$

$Kd = [2.6 \sin(\Phi/2)(1-\beta^2)^2] / (\beta^4)$

If you calculate the approximate value of ($\Phi/2$) for a standard 10 in × 8 in reducer fitting, you will find it is about 7 or 8 degrees. We already found $\beta$ for the 8 in pipe based on 10 in maximum size is: $\beta_8$ in = 0.7965. Substituting to find $K_d$ for the fitting, we get

$\mathbf{K_d} = \{2.6 \sin(7°)(1 - 0.7965)^2\} / \{/0.7965^4\}$
$= \{0.31686\,(1 - 0.63441)^2\} / \{0.40248\}$
= **0.10522;** and so the overall coefficient for the reducer fitting is
$\mathbf{K = K_d}/(\beta)^4 = 0.10522 / 0.40248 = \mathbf{0.26}$

*We also must calculate **"Q"**, the flowrate in gallons per minute. It is:*

**Q = AV =** $\{ [\pi(7.981'')^2/(4)(144)\text{ sq ft}] \times (7.00\text{ ft/sec}) \times (60\text{ sec/min}) \times (7.4805\text{ gal/ft}^3)\} =$
= **1,091** gpm

**Now for the head loss and pressure drop calculations:**

(1). <u>**For the 8-in Schedule 40 pipe segement:**</u>
$h_L = 0.00259\,(\mathbf{K_d})Q^2/d_s^4$
$= 0.00259\,(\mathbf{2.482})\,(1{,}091^2)/(\mathbf{7.981\text{ in}})^4 =$
**= 1.89 ft of water head loss**
also, $h_L = 0.00259\,(\mathbf{K})Q^2/d_L^4$
$= 0.00259\,(\mathbf{6.167})\,(1{,}091^2)/\mathbf{10.02\text{ in}^4} =$
**= 1.89 ft of water head loss**
$\Delta P = 0.00001799\,K\,\rho Q^2 \div d_L^4$
$= 0.00001799(\mathbf{6.167})(62.37)(1{,}091^2)/(10.02\text{ in})^4$
**= 0.82 psi permanent pressure drop**

(2) <u>**For the 8-in × 10-in reducer:**</u>
$h_L = 0.00259\,(\mathbf{K_d})Q^2/d_s^4$
$= 0.00259\,(\mathbf{0.10522})(1{,}091^2)/(\mathbf{7.981\text{ in}})^4 =$
**= 0.08 ft of water head loss;**
also, $h_L = 0.00259\,(\mathbf{K})Q^2 \div d_L^4$
$= 0.00259\,(\mathbf{0.26})(1{,}091^2)/(\mathbf{10.02\text{ in}})^4 =$
**= 0.08 ft of water head loss**
$\Delta P = 0.00001799\,K\,\rho Q^2 \div d_L^4$
$= 0.00001799(0.26)(62.37)(1{,}091^2)/(10.02\text{ in})^4$
**= 0.035 psi permanent pressure drop**

(3) <u>**For the 10-in Schedule 40 pipe segment:**</u>
$h_L = 0.00259\,(K_d)Q^2/d_s^4$
$= 0.00259\,(K)Q^2/d_L^4$
$= 0.00259(1.976)(1{,}091^2)/(10.02\text{ in})^4 =$
**= 0.60 ft of water head loss**
$\Delta P = 0.00001799\,K\,\rho Q^2/d_L^4$
$= 0.00001799(1.976)(62.37)(1{,}091^2)/(10.02\text{ in})^4$
**= 0.26 psi permanent pressure drop**

(4) <u>**For the entire system:**</u>
The first method is to sum up all the local losses. That way, we obtain:

system $h_L$ = (1.89 ft + 0.08 ft + 0.60 ft) =
**= 2.57 ft of water**

<u>The preferred way is using overall K:</u>

$\mathbf{K_{total}} = \Sigma_{i=1}^{i=n}(K)_i$
= (6.167 + 0.26 + 1.976) = **8.403;**

and $h_L = 0.00259\,(K)Q^2 \div d_L^4 =$
$= 0.00259(8.403)(1{,}091^2)/(10.02\text{ in})^4 =$
**= 2.57 ft of water**

Note that we only have to make one head loss calculation when we use "K" as opposed to one for each piping element when we use "$K_d$". Since we have to find each value of "$K_d$" anyway, using overall $\mathbf{K_{total}}$ to find the system head loss saves a bunch of work!

Of course, if for some reason you need to calculate the drop across a single element, you can still use the "$K_d$" relationship for that particular element.

A typical use of these formulas is to find the points for the system curve plot, which graphs overall system head loss against system flowrate. This is easily done. The term $\mathbf{d_L}$ is a constant. $K_{total}$ is also a constant for the system of piping (**well, *almost*** constant, per final note below). Since this leaves the flowrate "Q" as the only remaining variable in Eqs. (3a) and (3b) we have:

$\mathbf{h_{L1}} = 0.00259\,(K_{total})(Q_1)^2/d_L^4$
$\mathbf{h_{L2}} = 0.00259\,(K_{total})(Q_2)^2/d_L^4$
**therefore,** $(\mathbf{h_{L1}}/h_{L2}) = (Q_1/Q_2)^2$

This shows that once we have the head loss $\mathbf{h}_{L1}$ for a given flowrate $\mathbf{Q_1}$, then to find the corresponding head loss $\mathbf{h}_{L2}$ for a new flowrate $\mathbf{Q_2}$ we simply use:

**d.** $\mathbf{h_{L2}} = (h_{L1}) \times (Q_2/Q_1)^2$

My usual lazy-man approach is to find the system head loss for the nominal "design" value of flowrate, and then to calculate losses for 1/4, 2/4, 3/4, 1.00, 5/4, 6/4, 7/4 and 2.00 times the design flowrate. This gives me eight data points to plot on a copy of my pump curve, for the tried-and-true system curve-pump curve overlay method of analyzing the system hydraulics.

## Example

Calculate eight data points for the system friction head loss versus flowrate graph via Eq. (3d) using the **previous system example.**

## Solution

Starting with the previous finding system head loss $h_L$ = 2.57 ft of water at the design flowrate of 1,091 gpm, construct a table or spreadsheet as follows:

| Point # | Fraction | gpm | System $h_L$, ft |
|---|---|---|---|
| 1 | (1/4) | 272.75 | (1/16 × 2.57 = 0.16 ft |
| 2 | (2/4) | 545.50 | (4/16) × 2.57 = 0.64 ft |
| 3 | (3/4) | 818.25 | (9/16) × 2.57 = 1.45 ft |
| 4 | (4/4) | 1091.00 | (16/16) × 2.57 = 2.57 ft |
| 5 | (5/4) | 1363.75 | (25/16) × 2.57 = 4.02 ft |
| 6 | (6/4) | 1635.50 | (36/16) × 2.57 = 5.78 ft |
| 7 | (7/4) | 1909.25 | (49/16) × 2.57 = 7.87 ft |
| 8 | (8/4) | 2182.00 | (64/16) × 2.57 = 10.28 ft |

***A final note:*** $K_{total}$ is truly constant only when the friction factor "f" is constant, which *is* the case wherever the values of "**e**/D" and **"$N_{Re}$"** conspire to yield a flow regime which is completely and fully turbulent everywhere along the entire pipe length. A look at the Moody diagram shows that in the lower values portion of the turbulent-flow range of Reynolds number, however, friction factor has an inverse relationship with velocity. Meaning that "f" is only approximately constant for varying flowrates.

The deviation is pretty small in the range of Reynolds number in which we normally find ourselves *(typically on the order of $10^5$)*.

So, this nonlinearity is negligible in turbulent flow except at very low percentages of the design flowrate, so it causes us no real problem in our analysis. It does illustrate that the analysis will be more accurate for small bore rough pipes at normal velocities, than for large smooth pipes at normal velocities. If in doubt, do a few spot check calcs, and if the flow regime is too near the "curvy" portion of the friction factor plot at the left-hand side of the turbulent flow portion of the Moody diagram, then don't use Eq. (3d) if the analysis is ultracritical; plot the curve point by point instead.

*And of course if the flow regime is laminar, Reynolds number less than 4,000, the relationship of friction factor is with velocity ratio to the first power, not with velocity ratio squared.* We are not normally concerned with laminar flow in pipes, which is an engineering design topic beyond the intended scope of difficulty for this book. Laminar flow design problems should be placed under the care of bona fide fluid mechanics experts.

4. ***To calculate the local head loss coefficient*** $\mathbf{K_d}$ ***from the valve characteristic Cv and vice versa.***

**Cv =** a measure of the head loss (permanent pressure drop) across a valve, flowmeter, or other restrictive device. For a control valve, you can obtain tables from the manufacturer listing the **Cv** value for various per cent openings of the valve flowpath as governed by the valve plug position. For an isolation valve which is normally either 100% open or 100% closed, and not used as a throttling device, the value given for its **Cv** is understood to be for the 100% wide open condition.

Recalling the correct units,
**Q =** flowrate, gpm
$\Delta$**P** = pressure loss, psi; Lbf/in$^2$
$\rho$ = weight density in Lbf/ft$^3$
**d** = pipe inside diameter, in

a. $K_d = (891)(d)^4 / (Cv)^2$

b. $C_v = (29.85)(d)^2/\sqrt{(Kd)}$

c. $Q = (7.8994)\ (\sqrt{(\Delta P/\rho)})$

d. $\Delta P = (\rho/62.4)\ (Q/Cv)^2$

These relationships are a great help when one needs to incorporate valves or other devices characterized by **Cv** into a system head loss analysis.

Among other attachments to the end of this chapter, one sheet you may find very interesting is a copy of the three-way converging flow control valve "Design YS", 1989 Fisher Controls International, Inc., Catalog 10. It shows the variation of **Cv** with percent opening as well as the relationship between the three valve ports as one direction through the valve first lags, then dominates, flow through the other direction. This is great data for the engineer who is faced with analyzing a parallel flow network. ***Speaking of parallel networks:***

5. ***To calculate the equivalent combined head loss coefficient*** $\mathbf{K_{equiv}}$ ***for total flow across a pair of parallel flowpaths*** **(loop):**

Parallel flowpaths occur when a single flow stream splits through a pipe tee or lateral branch connection. If the two streams rejoin, we call the combined network a ***"loop";*** the ***loop*** begins with the splitter fitting (diverging tee), continues simultaneously with flow through both branch pipes and their valves and fittings, and includes as a terminal element the flow-rejoining mixer fitting (converging tee.)

I always complained about unnecessary difficulty in analyzing parallel hydraulic flow networks. After all, the electrical engineers and every physics student receives the simple formula for parallel resistances in an AC or DC electric power circuit; why not the same for us poor mechanical slobs? Well, some years ago I quit bemoaning the lack of a published formula for converting a pipe loop into a single equivalent resistance coefficient "K". I sat down and derived it for myself, and include it for your consideration as Eq. (5a) below. As far as I can tell, it is valid within its framework of assumptions, which are as stated in the derivation. I have included a reproduction of that hand analysis as follows and hope some rainy day you will take the time to check it for me.

$\mathbf{K_1}$ and $\mathbf{K_2}$ are the overall loss coefficients for **pipe branch #1** and **pipe branch #2,** respectively, and are to be found by calculation as already discussed in this topic.

$\mathbf{K_{equiv}}$ = head loss coefficient for the combined loop: both tees and the two branch paths connected between them.

**a.** $\mathbf{K_{equiv}} = (K_1)(K_2)/\{\checkmark(K_1)+\checkmark(K_2)\}^2$

This expression bears only superficial resemblance to the equivalent resistance of parallel resistors in an electric flow circuit, which as I recall is $\mathbf{R_T} = (R_1)(R_2)/\{R_1 + R_2\}$ but is intended for use in a perfectly analogous way. The question is not of analogous behavior of fluids and direct electrical currents, but one of the range of applicability over which the fluid mechanics version applies. This turns out to be rather complex, as we shall see. Therefore do not use Eq. (5a) until the rest of this chapter has been carefully studied and understood.

Note that to use **Eq. (5a)** you must first find the value of $\mathbf{K_{d1}}$ **and** $K_{d2}$ for the two series of flow elements as usual, then convert them to overall coefficients $\mathbf{K_1}$ and $K_2$.

## TO FIND EQUIVALENT HYDRAULIC LOSS COEFFICIENT FOR FLOW THROUGH A PARALLEL LOOP:

### First, the Results (Refer to Diagram p. 101)

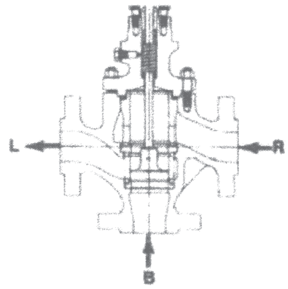
The head loss of the total flow from node "A" to node "B" is denoted "$HL_{AB}$". Its dimensional units are ***"feet of liquid flowing."*** Flow ***"Q"*** is in ***gallons per minute*** (gpm) and pipe diameter ***"d"*** is in ***inches.***

$$\mathbf{HL_{AB}} = [0.00259\ (K_{equiv})(Q_{total})^2] \div (d_T)^4$$

where $\mathbf{K_{equiv}} = (K_1)(K_2) \div [(\checkmark K_1 + \checkmark K_2)^2]$

$\mathbf{K_1}$ and $\mathbf{K_2}$ are the flow resistances, which are the pipe friction head loss coefficients based upon the largest pipe diameter **"$\mathbf{d_T}$"** as calculated by the usual method, that is, ***Crane Technical Paper No. 410*** and the hydraulic piping computer programs which are based on that method:

**K ≡ f (L/d); f ≡ Colebrook's equation friction factor; head loss ≡ K($\mathbf{V^2}$/2g).**




# Design YS

## Converging Flow

## Flow Coefficients

For additional body information see *Bulletin 51.1:YD*

### Converging Flow

Linear Characteristic

| Coefficients | Body Size, Inch | Port Diameter, Inch | Total Travel, Inch | Flow Path(1) | Valve Opening—Percent of Total Travel: 0 (Plug Down) | 10 | 20 | 30 | 40 | 50 | 60 | 70 | 80 | 90 | 100 (Plug Up) | $K_m$(2) and $C_1$ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| $C_v$ (Liquid) | 1/2 | 1-5/16 | 3/4 | R to L | 8.43 | 7.37 | 6.62 | 6.11 | 5.66 | 5.20 | 4.73 | 4.25 | 3.65 | 2.48 | 0 | 0.63 |
| | | | | B to L | 0 | 1.74 | 2.44 | 3.16 | 3.61 | 4.10 | 4.95 | 5.85 | 6.38 | 7.07 | 8.42 | 0.65 |
| | 3/4 | 1-5/16 | 3/4 | R to L | 11.8 | 10.4 | 9.37 | 8.71 | 8.07 | 7.25 | 6.27 | 5.32 | 4.40 | 2.78 | 0 | 0.62 |
| | | | | B to L | 0 | 2.65 | 3.72 | 4.87 | 6.30 | 7.43 | 8.29 | 9.16 | 9.97 | 11.2 | 12.5 | 0.59 |
| | 1 | 1-5/16 | 3/4 | R to L | 18.4 | 16.5 | 15.3 | 13.8 | 12.0 | 9.84 | 7.63 | 5.37 | 3.38 | 1.64 | 0 | 0.87 |
| | | | | B to L | 0 | 2.11 | 3.03 | 4.45 | 6.15 | 8.18 | 10.5 | 13.0 | 15.7 | 18.3 | 20.5 | 0.77 |
| | 1-1/2 | 1-5/16 | 3/4 | R to L | 20.6 | 18.0 | 16.2 | 14.3 | 12.4 | 9.90 | 7.70 | 5.42 | 3.30 | 1.65 | 0 | 0.82 |
| | | | | B to L | 0 | 2.60 | 4.90 | 6.90 | 9.40 | 12.6 | 15.4 | 18.8 | 22.0 | 24.5 | 25.1 | 0.68 |
| | 2 | 2-5/16 | 1-1/8 | R to L | 66.1 | 63.6 | 61.9 | 59.3 | 63.7 | 47.0 | 37.7 | 27.6 | 17.1 | 6.68 | 0 | 0.84 |
| | | | | B to L | 0 | 3.85 | 5.48 | 9.16 | 14.6 | 21.5 | 30.7 | 42.4 | 56.3 | 71.9 | 85.6 | 0.73 |
| | 2-1/2 | 2-5/16 | 1-1/8 | R to L | 72.8 | 70.0 | 68.0 | 65.3 | 59.2 | 51.7 | 41.5 | 30.3 | 18.8 | 7.35 | 0 | 0.80 |
| | | | | B to L | 0 | 4.24 | 6.03 | 10.1 | 16.0 | 23.7 | 33.8 | 46.7 | 62.0 | 79.0 | 94.2 | 0.70 |
| | 3 | 3-7/16 | 1-1/2 | R to L | 140 | 131 | 121 | 111 | 99.3 | 85.3 | 68.3 | 46.1 | 23.9 | 9.07 | 0 | 0.74 |
| | | | | B to L | 0 | 14.1 | 27.4 | 40.3 | 53.8 | 68.3 | 87.1 | 111 | 138 | 164 | 185 | 0.69 |
| | 4 | 4-3/8 | 2 | R to L | 234 | 231 | 225 | 216 | 200 | 175 | 140 | 103 | 65.2 | 30.0 | 0 | 0.75 |
| | | | | B to L | 0 | 2.81 | 11.9 | 25.0 | 43.1 | 69.0 | 106 | 149 | 200 | 256 | 312 | 0.70 |
| | 6 | 7 | 2 | R to L | 413 | 386 | 363 | 331 | 296 | 252 | 207 | 157 | 102 | 49.5 | 0 | 0.76 |
| | | | | B to L | 0 | 38.4 | 70.1 | 110 | 156 | 208 | 262 | 324 | 393 | 473 | 556 | 0.70 |
| $C_g$ (Gas) | 1/2 | 1-5/16 | 3/4 | R to L | 243 | 203 | 177 | 159 | 146 | 136 | 126 | 114 | 98.9 | 70.1 | 0 | 28.8 |
| | | | | B to L | 0 | 67.3 | 88.4 | 109 | 126 | 137 | 148 | 160 | 172 | 198 | 244 | 29.0 |
| | 3/4 | 1-5/16 | 3/4 | R to L | 336 | 296 | 262 | 237 | 217 | 195 | 170 | 144 | 116 | 74.9 | 0 | 28.5 |
| | | | | B to L | 0 | 77.4 | 106 | 137 | 168 | 187 | 209 | 242 | 266 | 294 | 352 | 28.2 |
| | 1 | 1-5/16 | 3/4 | R to L | 608 | 538 | 499 | 451 | 394 | 328 | 256 | 180 | 116 | 60.3 | 0 | 33.0 |
| | | | | B to L | 0 | 65.5 | 103 | 148 | 202 | 263 | 337 | 418 | 497 | 564 | 642 | 31.3 |
| | 1-1/2 | 1-5/16 | 3/4 | R to L | 640 | 565 | 525 | 475 | 415 | 345 | 269 | 190 | 122 | 63.5 | 0 | 31.1 |
| | | | | B to L | 0 | 70.0 | 110 | 158 | 216 | 282 | 360 | 447 | 532 | 603 | 687 | 27.3 |
| | 2 | 2-5/16 | 1-1/8 | R to L | 2300 | 2220 | 2140 | 2020 | 1840 | 1620 | 1300 | 945 | 582 | 246 | 0 | 34.8 |
| | | | | B to L | 0 | 147 | 206 | 328 | 513 | 748 | 1060 | 1450 | 1880 | 2330 | 2670 | 31.2 |
| | 2-1/2 | 2-5/16 | 1-1/8 | R to L | 2420 | 2330 | 2250 | 2120 | 1930 | 1700 | 1370 | 990 | 612 | 258 | 0 | 33.2 |
| | | | | B to L | 0 | 154 | 216 | 344 | 533 | 784 | 1110 | 1520 | 1970 | 2450 | 2800 | 29.7 |
| | 3 | 3-7/16 | 1-1/2 | R to L | 4730 | 4430 | 4090 | 3750 | 3360 | 2880 | 2310 | 1560 | 807 | 306 | 0 | 33.8 |
| | | | | B to L | 0 | 434 | 843 | 1240 | 1660 | 2100 | 2680 | 3420 | 4250 | 5050 | 5700 | 30.8 |
| | 4 | 4-3/8 | 2 | R to L | 7980 | 7720 | 7540 | 7150 | 6560 | 5730 | 4660 | 3340 | 1930 | 916 | 0 | 34.1 |
| | | | | B to L | 0 | 145 | 443 | 897 | 1490 | 2350 | 3620 | 5040 | 6670 | 8310 | 9590 | 30.7 |
| | 6 | 7 | 2 | R to L | 14,500 | 13,500 | 12,400 | 11,200 | 9900 | 8220 | 6450 | 4600 | 2850 | 1350 | 0 | 35.1 |
| | | | | B to L | 0 | 1330 | 2490 | 3860 | 5360 | 7060 | 8820 | 10,900 | 13,100 | 15,300 | 17,300 | 31.1 |
| $C_s$ (Steam) | 1/2 | 1-5/16 | 3/4 | R to L | 12.1 | 10.2 | 8.85 | 7.95 | 7.30 | 6.80 | 6.30 | 5.70 | 4.95 | 3.51 | 0 | 28.8 |
| | | | | B to L | 0 | 3.37 | 4.42 | 5.45 | 6.30 | 6.85 | 7.40 | 8.00 | 8.60 | 9.90 | 12.2 | 29.0 |
| | 3/4 | 1-5/16 | 3/4 | R to L | 16.8 | 14.8 | 13.1 | 11.9 | 10.9 | 9.75 | 8.50 | 7.20 | 5.80 | 3.75 | 0 | 28.5 |
| | | | | B to L | 0 | 3.87 | 5.30 | 6.85 | 8.40 | 9.35 | 10.5 | 12.1 | 13.3 | 14.7 | 17.6 | 28.2 |
| | 1 | 1-5/16 | 3/4 | R to L | 30.4 | 26.9 | 25.0 | 22.6 | 19.7 | 16.4 | 12.8 | 9.00 | 5.80 | 3.02 | 0 | 33.0 |
| | | | | B to L | 0 | 3.28 | 5.15 | 7.40 | 10.1 | 13.2 | 16.9 | 20.9 | 24.9 | 28.2 | 32.1 | 31.3 |
| | 1-1/2 | 1-5/16 | 3/4 | R to L | 32.0 | 28.2 | 26.2 | 23.8 | 20.8 | 17.2 | 13.5 | 9.50 | 6.10 | 3.18 | 0 | 31.1 |
| | | | | B to L | 0 | 3.50 | 5.50 | 7.90 | 10.8 | 14.1 | 18.0 | 22.4 | 26.6 | 30.2 | 34.3 | 27.3 |
| | 2 | 2-5/16 | 1-1/8 | R to L | 115 | 111 | 107 | 101 | 92.0 | 81.0 | 65.0 | 47.3 | 29.1 | 12.3 | 0 | 34.8 |
| | | | | B to L | 0 | 7.35 | 10.3 | 16.4 | 25.7 | 37.4 | 53.0 | 72.5 | 94.0 | 117 | 134 | 31.2 |
| | 2-1/2 | 2-5/16 | 1-1/8 | R to L | 121 | 116 | 113 | 106 | 96.5 | 85.0 | 68.5 | 49.5 | 30.6 | 12.9 | 0 | 33.2 |
| | | | | B to L | 0 | 7.70 | 10.8 | 17.2 | 26.6 | 39.1 | 55.5 | 76.0 | 98.5 | 123 | 140 | 29.7 |
| | 3 | 3-7/16 | 1-1/2 | R to L | 236 | 222 | 205 | 188 | 168 | 144 | 116 | 78.0 | 41.0 | 15.3 | 0 | 33.8 |
| | | | | B to L | 0 | 21.7 | 42.2 | 62.0 | 83.0 | 105 | 134 | 171 | 213 | 253 | 285 | 30.8 |
| | 4 | 4-3/8 | 2 | R to L | 399 | 386 | 377 | 358 | 328 | 287 | 233 | 167 | 96.5 | 45.8 | 0 | 34.1 |
| | | | | B to L | 0 | 7.25 | 22.2 | 44.9 | 74.5 | 118 | 181 | 252 | 334 | 416 | 480 | 30.7 |
| | 6 | 7 | 2 | R to L | 725 | 675 | 620 | 560 | 495 | 411 | 323 | 230 | 143 | 67.5 | 0 | 35.1 |
| | | | | B to L | 0 | 66.5 | 125 | 193 | 268 | 353 | 441 | 545 | 655 | 765 | 865 | 31.1 |

1. The end connections are identified on the body.
2. This column lists the $K_m$ values for the $C_v$ coefficients and the $C_1$ values for the $C_g$ and $C_s$ coefficients at maximum flow.



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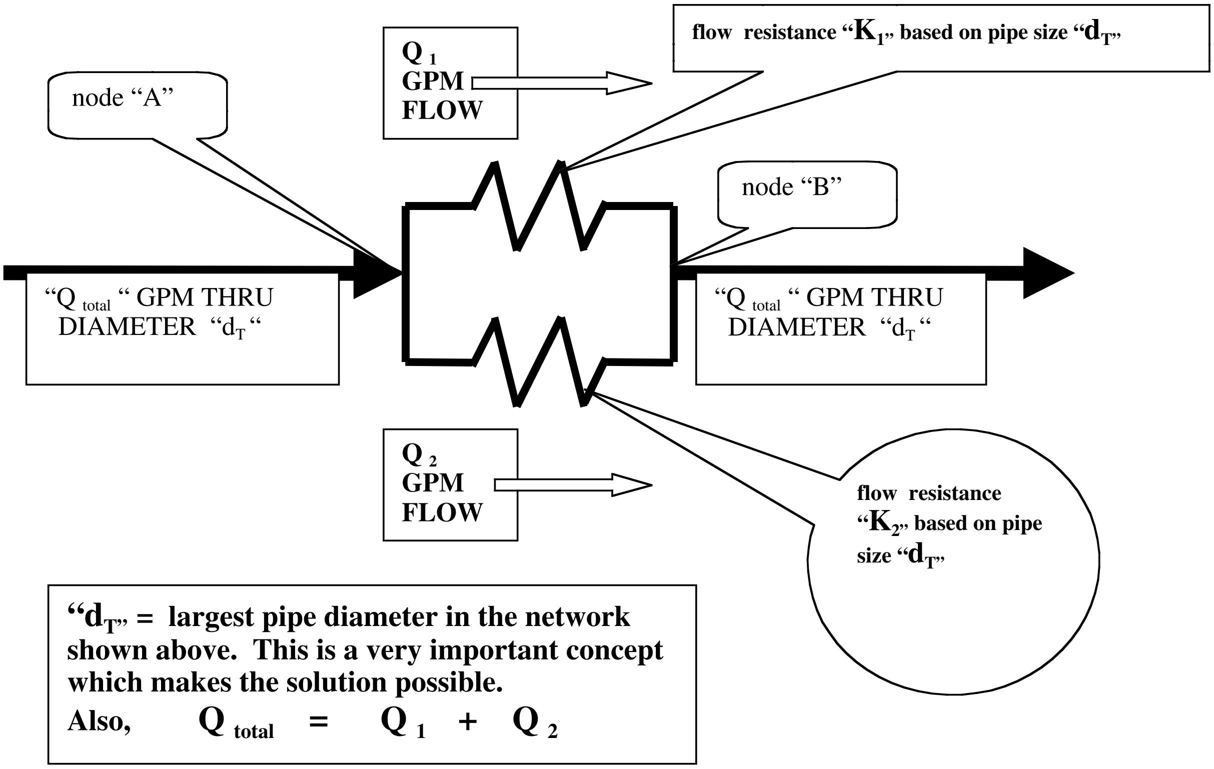
Catalog 10

FLOW COEFFICIENTS




**TO FIND EQUIVALENT HYDRAULIC LOSS COEFFICIENT FOR FLOW THROUGH A PARALLEL LOOP**

***The following pages define the problem, derive the results, and narrate the procedure.***

## General Statement of Problem (Incompressible Liquids)

A pipeline carrying a known flowrate encounters a loop (flow splits into two branch pipelines, which rejoin into a single pipe somewhere downstream). Simultaneous flow occurs through the two branches, both moving in the direction *away from* the initial branch-off point, node "A" on the previous diagram, to rejoin once more at node "B". We call flow through this loop geometry "parallel flow."

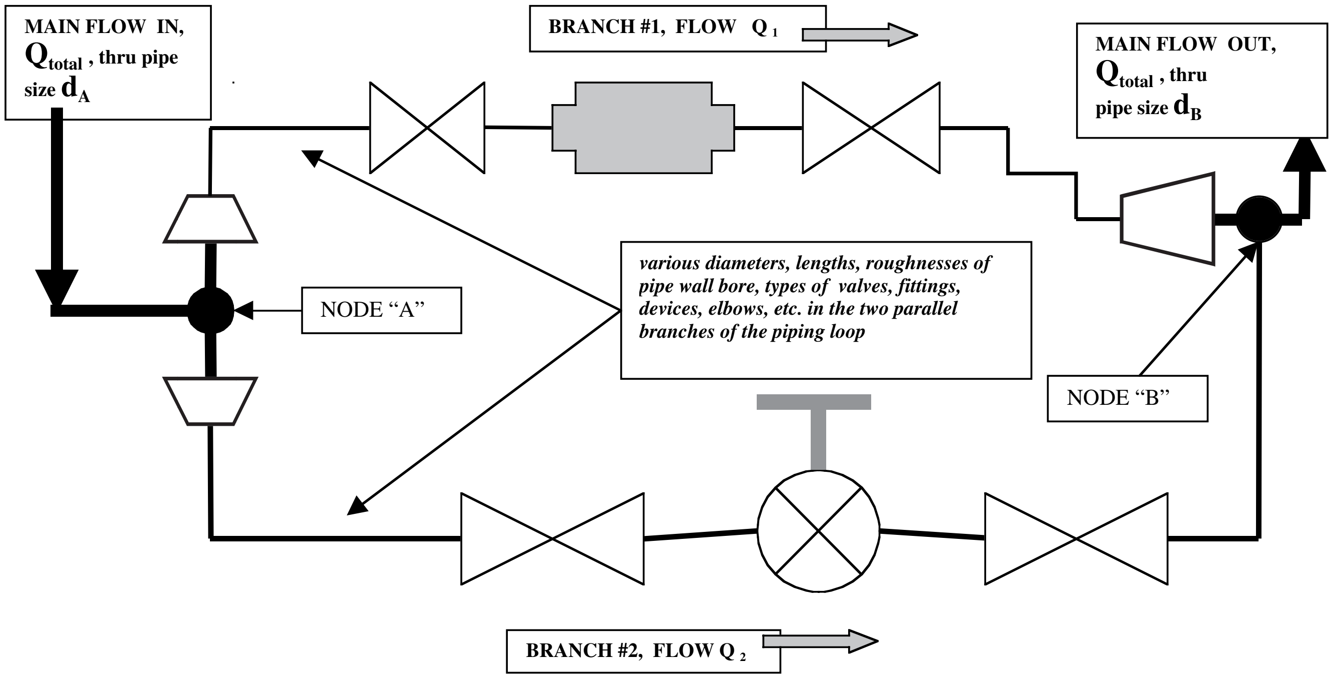
Our task is to analyze the hydraulics of the flows through the loop, as part of the overall pipeline system flow analysis, without performing a bunch of trial-and-error calculations in which we have to make a series of assumptions of the individual branch flowrate quantities.

It would be nice and easy, if we knew the numerical flowrate values through the parallel branches before we begin the solution, but, unfortunately, *we do not.* Therefore we must answer the question, ***"What is the equivalent resistance to the combined flows through the parallel loop, in terms of the piping elements that compose the two branches?"***

A further condition on our analysis is that, to permit use of the available computer program for hydraulic calculations and enable accurate checking, we must utilize the standard Darcy-Weisbach formulas and terminology.

## Our response

Upon reflection, we see that the loop is made up of elements of various different pipe sizes, none of which are *necessarily* of the same diameter and wall thickness schedule as the single main header which leads into node "A" and continues on past node "B." Obviously, the overall analysis would quite simple if the parallel branches were a single pipe of the same size instead of being a conglomeration of different-sized bits and pieces. Is there a way to express the loop as a single entity, of the same pipe size as the rest of the system, which we could simply plug as another element into the input framework of our computer hydraulics program? It turns out that the answer is "maybe." Let's expand our "system sketch" a bit:




## Framework of Assumptions Necessary for Solution:

1. The fluid flowing is either a **Newtonian liquid** or an **ideal gas** not near its condensing phase change condition. If the fluid is an **ideal gas,** the flow regime **MUST be incompressible.** (If in doubt about compressible vs. incompressible, I always calculate the flowing Mach number as a guide. The Mach number must be ***low***, for the following reason:
   ***a.*** In his text "The Dynamics and Thermodynamics of Compressible Fluid Flow," Vol. I, published by the Ronald Press, New York, 1958, author Shapiro in Article 3.1 derives proof that at low Mach numbers, gas flow can be approximately described by the incompressible flow relationships, with a small error. The derivation is based on pressure variations in isentropic flow. It turns out that the error is on the order of: ***error ~ 0.25 × (Mach no.)²***. **{For example, an incompressible flow approximation of an ideal gas flowing at Mach 0.20** ~ (0.25) (0.2 × 0.2) ~ 0.01, meaning the error ~ approximately 0.01 × 100 = 1.00%.

   **For engineering accuracy, then. I would not use the incompressible flow approximation for steam flow (or any condensable vapor), or for an ideal gas flow if the Mach no. ≥ 0.20.**
2. Friction factors are per the Darcy-Weisbach relation/ Colebrook equation.
3. Flowrate is steady and constant (any rate changes are given time to dampen out.)
4. The flow regime is fully turbulent **(Reynolds number quite large; very questionable!)**
5. **If $d_A \neq d_B$, then the maximum pipe size shall be the larger of either $d_A$ or $d_B$.** We will always use the designation **"$d_M$"** to denote the largest pipe size in the analyzed network, upon which the loss coefficients are always, and must be, based as the reference diameter.
6. The simplifying assumption is necessary that the head losses of divergence, from the tee which is node "A" to branch #1, and likewise from the tee Node "A" to branch #2, are equal; also, that the head losses of convergence, from branch #1 to the tee Node "B" and from Branch #2 to tee node "B" are equal. For computation in real examples, we will use the handbook factors for "flow through the side of tee reduced (or increased) by half", and in doing so will realize that the branch flows $Q_1$ and $Q_2$ are not necessarily equal, and probably are *not* equal. Making the assumptions just stated is required to enable a closed-form solution. So our analytical journey must begin with at least two questionable assumptions (this tee-split loss factor and the really big Reynolds number.)
7. We are solving only for dynamic/ frictional head losses in a loop, and therefore must prohibit the addition, or removal, of heat or work or gravitational potential energy to, or from, the fluid stream between Nodes "A" and "B". ***We are doing a fluid-mechanical derivation, not a full-fledged thermodynamic analysis.***

## Analysis:

For flow through a series of resistance elements, including straight pipe viscous friction loss in smooth and rough pipe runs, and dynamic losses due to gradually converging/diverging pipe walls (reducers), abrupt area changes (bushings and stub connections), flow past obstructions (through valves, etc.), directional turns (tees, elbows, and miters), flow-changing fittings (tees and branch fittings as well as field branch connections), and combinations of all these, the empirical Darcy-Weisbach formula is:

**1.** $h_L = f\,(L/d)\,(V^2/2_g)$
$h_L$ = head loss, in *feet of the flowing fluid;*

**f =** *dimensionless* friction factor, found from Colebrook's equation (Moody diagram), a function of pipe diameter, pipe wall bore roughness, and Reynolds number of the flow;

**L =** equivalent length of straight pipe causing the same head loss as the actual piping element at the same bulk velocity in the same (referenced) pipe diameter, in *feet;* <u>in straight pipe runs this is just the actual straight length,</u> and for everything else this is truly an <u>*equivalent*</u> length and is determined by series of <u>empirical laboratory tests</u> as given in Crane Technical Paper no. 410.

**d =** <u>actual</u> circular inside diameter of the equivalent pipe, in ft;

* **V =** bulk fluid velocity based on assumed uniform velocity profile, in ft/sec;

**g =** local acceleration due to gravity, in feet/sec$^2$;

**2. * $V = Q/A = 4Q/\pi d^2$**

**Q =** constant volumetric flowrate through the pipe, in Ft$^3$/sec

Substituting Eq. (2) into Eq. (1) yields $h_L = f(L/d)(1/2g)(4Q/\pi d^2)^2$, and results in

**3.** $h_L = (8fL/\pi^2 gd^5)(Q^2)$

Now compare this result with the formula used for head loss $h_L$ in the always-recommended *Crane Technical Paper* No. 410 and associated computer programs:

**4.** $h_L = (0.00259)(K)(GPM^2)/d_M{}^4$

**K =** *dimensionless* loss coefficient of the pipe element series resistance;

$d_M$ = largest pipe size (max inside diameter) in the analyzed network, in *inches;*

**GPM =** constant volumetric flowrate through the pipe, in gpm;

Now you all pay attention, doggone it! It's going to get a little tricky from here in, mainly because of a bunch of new highly specific terms and parameters you may not be used to using every day. To start with, we must introduce the Crane Technical paper no. 410 loss coefficient "$K_d$":

**5.** $K_d \equiv f_T (L/d)$; "$K_d$" is used for all analyzed piping system element resistances

<u>*except*</u> for straight runs of pipe at constant diameter, which use the friction factor "f" and not "$f_T$", and in which "L" is a measured straight length and not an empirical equivalent. In other words, $K_d$ signals the use of some element other than a straight run of pipe; that is, a fitting, such as an elbow, or a device, such as a valve or filter or strainer, or a change in pipe size, such as a concentric or eccentric reducer. It will also be used for splitting and joining flow streams via tees and branches. Remember, it is going to describe *a cause of dynamic head loss other than straight pipe wall friction.*

The factor **"(L/d)" in this expresion** is the "equivalent length of the fitting to the fitting's connecting pipe size actual inside diameter" ratio of the pipe fitting or element other than straight pipe. And the factor "$F_T$" is also a special animal; it is the <u>fully turbulent friction factor,</u> found from Colebrook's equation (Moody diagram), and is a function of pipe diameter and pipe wall bore roughness, but is <u>completely independent of the Reynolds number</u> of the flow. For any given value of the measured *wall roughness to inside diameter ratio* ***"∈/D",*** if the flow regime is definable at all, the friction factor *"f"* decreases in a curvilinear manner with increasing value of the Reynolds number beyond Nre = 4,000. This signals increasing departure from smooth laminar flow. At sufficiently large values of Reynolds number, the flow becomes fully turbulent across the entire velocity profile in the pipe, and at that point the friction factor becomes and remains a constant. The Moody diagram shows a downward-curving dotted line of demarcation for all "rough" pipes, beyond which *"f"* remains constant, and drops no further. This envelope marks the zone of "fully developed turbulent flow", where the constant "$f_T$" for friction factor applies to the given ratio of ***(∈/D).***

*A couple of aside comments are in order here. First, we work mostly with pipe materials in which the absolute roughness* ***"∈"*** is pretty well known, defined, and tubulated in the hydraulics references and texts, at least when relatively new; all bets are off if the pipe is 175 years old, or you are trying to pump hydrofluoric acid through carbon steel pipes! Next, when obtaining values which bear on pipe diameter and roughness, you will usually find that for initial system design engineering purposes, you can use a value of **"∈/D"** corresponding to "20-year-old rough" steel pipe as the basis for a second computer calc of pressure drop, paired with an initial calc based on "clean new steel pipe", and thereby obtain a satisfactorily safe envelope of operating heads for selecting your pumps and drive motors and "ranging" those throttle control valves.

Recognizing that "$f_T$" is independent of velocity is very important. It means that the fitting loss **coefficient** can be computed for an assumed flowrate and used as a <u>constant,</u> as long as the flow regime is fully turbulent, which we assumed to be the case in point. This does not mean the pressure drop associated with the fitting is independent of velocity; pressure drop still increases with increasing velocity head (V squared!) It just means that the fitting pressure drop equals a <u>*certain empirical number of velocity heads*</u>, regardless of velocity, thanks to the constant loss coefficient in the fully turbulent flow regime assumed in this analysis.

<u>To express the fitting or device's loss coefficient **"K",**</u> which is initially calculated for the particular pipe fitting on the basis of its actual connecting pipe size in the network, in terms to be useful to us on the basis of our analysis, meaning in terms of the analyzed network's largest pipe size "$d_M$":

a. $K \equiv K_d/(\beta^4)$; note carefully the fourth power here!

<u>The definition of the all-important parameter **"β"**</u> is :

b. $\beta \equiv d/d_m \leqq 1.00$

In other words, we defined relationship **(5)** $K_d = f_T(L/d)$ above with **"L/d"** being the tabulated formulated empirical "number of equivalent pipe diameters" of the pipe fitting, in terms of the friction in that length of rough pipe of diameter "d", and with **"d"** being <u>the fitting's actual connecting pipe size in the network.</u> We must take special care to keep these Ks and d's straight, or our analysis will get hopelessly bollixed up!

The utility of (5a) is it obtains for us a single flowrate versus loss coefficient relationship which is independent of known changes in the actual pipe diameters of all the fittings , valves, etc. and of the initially unknown actual flowrates as we progress along the series of elements of the piping flow path. Which we have to be able to do, since we start out not knowing the individual flowrates in the loop's two branches! The fact that $f_T$ is a constant for everything but straight pipe runs makes it possible to at least attempt this analysis.

Thus far, we have accepted the equivalence of two separate equations, having different dimensional units between them, one of which we derived and the other of which we looked up in a reference text (Crane Technical Paper No. 410.) These equations, given above, are:

(3) $\mathbf{h_L} = (8fL/\pi^2 gd^5)(Q^2)$,

**and**

(4) $\mathbf{h_L} = (0.00259)(K)(GPM^2)/dm^4$

I don't know about you, but I would feel a darn sight more comfortable if I could prove to myself that these two relationships are truly equivalent. So that is what I will examine next. You can skip this part if you want, but I wish you wouldn't because it needs to be checked!

To start with, all of Eq. **(3)** is in units of feet and seconds. But Eq. **(4)** mixes <u>feet</u> (head loss $\mathbf{h_L}$), <u>inches</u> (the max pipe diameter $\mathbf{d_M}$), gallons, and minutes (flowrate **gpm**).

Get Eq. **(4)** on the same dimensional basis as Eq. **(3)**:

- Convert GPM *gallons/minute* to Q $ft^3$/sec.
  <u>Q</u> = GPM gal/min × (1 min/60 sec) × (1 $ft^3$/7.4805 gal) = <u>(GPM/448.83) $ft^3$/sec</u> and ***<u>GPM = ***(448.83 Q) gal/min</u>
- convert $d_M$ *inches* to d *feet:*
  <u>$d_M$</u> = d *feet* × (12in/1 ft = <u>(12 d) ft</u> and ***<u>d = ***($d_M$/12)</u>

Substitute the ***GPM and ***d terms into Eq. (4), and obtain the following:

$h_L = (0.00259)(K)(GPM^2)/d_M^4 =$
$= (0.00259)(K)(448.83\ Q)^2/(12d)^4 =$
$= [(0.00259)(448.83)^2/(12)^4] \times [KQ^2/d^4] =$
$= \ldots \ldots [0.025161616 \times KQ^2/d^4]\ldots$

Juggling Eq. (3) appropriately, $\mathbf{h_L} = (8fL/\pi^2 gd^5)(Q^2) = \mathbf{(8/\pi^2 gd^4)(fL/d)(Q^2)}$;

Now recalling that **K ≡ f(L/d), (here setting the value of β ≡ $\mathbf{d/d_M}$** ≡ 1.00), we can rewrite our modified Eq. (3) a bit more, obtaining this:

$h_L = (8/\pi^2 gd^4)(fL/d)(Q^2) =$
$= (8/\pi^2 gd^4)(K)(Q^2) =$
$= (8/\pi^2 g) \times (K)(Q^2)/(d^4) =$
$= \ldots [(8/\pi^2\ g) \times KQ^2/d^4]\ldots;$

therefore we have reduced the question of equivalence of Eqs. (3) and (4) to the question of equivalence of the two constants <u>**0.025161616**</u> and <u>**$8/\pi^2$ g,**</u> which should be easy enough. If we plug the usual "engineering" values of **3.14** and **32.2** into the term <u>**$8/\pi^2$g**</u> in Eq. (3), we get **(8.00)/[(3.14)$^2$** × 32.2] = <u>*0.025198507*</u> which compares very nicely with the crane-derived constant ***0.025161616*** in Eq. (4). The percent difference based on the latter figure is (0.025198507 – 0.025161616/0.025161616) × 100% = <u>***0.1466%.***</u> Pretty small. If we plug the usual "slightly more refined engineering" values of **3.141592654** and **32.174** into the term <u>**$8/\pi^2$g**</u> in Eq. (3), we get <u>***0.025193307,***</u> which compares even better (percent difference only <u>***0.126%.***</u> And all this without even diddling with the numerical conversions of gallons to cubic feet!)

SO! <u>**Can we agree**</u> to say that, for engineering purposes, we can round off all these constants to a value of ***0.0252*** and declare Eqs. (3) and (4) equivalent? After all, considering the incredibly advanced age of Crane original paper No. 410, not to mention that of author MechMentor himself, the original constant ***0.00259*** which Crane used was probably computed with a <u>slide rule</u> (a 10-in "K&E" brand, no doubt; all the Mech. Eng. students used the 10-in "K&E"; only the Chem. Eng. students used the "Post" brand 10-inchers. And who trusts ChemE's to make hydraulic calculations anyhow?) And <u>*nobody*</u>, not even the professors, could afford one of those 12- or 14-inch slide rules which were supposed to be good all the way out to four significant figures!!

Well, I say <u>**we can!**</u> And it is my vote that counts!! So without further whining and groveling on anyone's part, let us continue the analysis!! And it is with our old friend, the Principle of Continuity, that we do just that. Continuity tells us that, if none of the fluid leaks out of a crack in the piping or from a crappy old dried-out valve stem seal, then:

**6.** $\mathbf{(Q_{total})}_{node\ \text{"A"}} = Q_1 + Q_2 = (Q_{total})_{node\ \text{"B"}}$

And by definition, the head loss from node A to node B, **"$\mathbf{HL_{AB}}$"** (per the schematic diagram) can have ***one and only one value*** at any given instant, determined by the physics of the system, and which in a steady flow condition will be constant, numerically unique, and independent of which branch we take through the loop, branch #1 or branch #2, In simpler terms,

**7.** $\mathbf{HL_{AB} = HL_1 = HL_2}$

Using the Crane format, Eq. (4), for the flows through the parallel branches of the loop, we can also write:

(6a) $\mathbf{GPM_2} = GPM_{total} - GPM_1$
(4a) $\mathbf{HL_1} = 0.00259\ K_1\ (GPM_1)^2/dm^4$
(4b) $\mathbf{HL_2} = 0.00259\ K_2\ (GPM_2)^2/d_M^4$; plugging these into Eq. (7), then we obtain:

**8.** $\mathbf{K_1\ (GPM_1)^2 = K_2\ (GPM_2)^2}$

Now, we assign a trial or "design" value to $\mathbf{GPM_{total}}$, and we can independently calculate approximate values of $\mathbf{K_1}$ and $\mathbf{K_2}$ from the elemental constituency and geometry of the pipe network, by summing known values of the serial element loss coefficients in each branch for assumed values of $\mathbf{GPM_1}$ and $\mathbf{GPM_2}$.

<u>***What we are seeking is this:***</u> a value of an equivalent loss coefficient **"$\mathbf{K_{equiv}}$"** such that we can directly calculate:

**9.** $\mathbf{HL_{AB} = 0.00259\ (K_{equiv})(GPM_{total})^2/d_M^4}$

The next step is to solve Eqs. (4a) and (4b), using Eq. (7), for an expression of $\mathbf{(GPM_1 + GPM_2)}$.

$(GPM_1)^2 = (HL_1\ d_M^4)/0.00259\ K_1 = (HL_{AB}\ d_M^4)/0.00259\ K_1;$
**define** $\boldsymbol{\varphi \equiv d_M^4/0.00259;}$
$(GPM_1)^2 = \varphi\ HL_{AB}/K_1$ ; $GPM_1 = \sqrt{\varphi HL_{AB}/K_1}$ and
$(GPM_2)^2 = \varphi\ HL_{AB}/K_2$ ; $GPM_2 = \sqrt{\varphi HL_{AB}/K_2}$ ; therefore

**10.** $\mathbf{(GPM_1 + GPM_2) = \{\sqrt{\varphi HL_{AB}/K_1} + \sqrt{\varphi HL_{AB}/K_2}\ \} \equiv GPM_{total}}$

Now substitute Eq. (10) into Eq. (9) and prepare for a little algebraic humdrum.

$\mathbf{HL_{AB}} = (0.00259/d_M^4)(K_{equiv})\ \{\sqrt{\varphi HL_{AB}/K_1} + \sqrt{\varphi HL_{AB}/K_2}\ \}^2$

Since by our shorthand designation ***(0.00259/$d_M^4$)*** = 1/ φ, we obtain:

$HL_{AB} = (K_{equiv.}/\ \varphi)\ (\varphi\ HL_{AB}/K_1 + 2\sqrt{\varphi\ HL_{AB}/K_1}\ \sqrt{\varphi\ HL_{AB}/K_2} + \varphi\ HL_{AB}/K_2)$

$HL_{AB} = (K_{equiv.}/\ \varphi)\ [(\varphi\ HL_{AB})(1/K_1 + 1/K_2) + 2\sqrt{(\varphi\ HL_{AB})^2/K_1K_2}]$

get common denominator for first term in brackets and simplify the radical expression:

$HL_{AB} = (K_{equiv.}/\ \varphi)\ [(\varphi\ HL_{AB})(K_1 + K_2/K_1K_2) + 2\ \varphi\ HL_{AB}\sqrt{(1/K_1K_2)}]$

$HL_{AB} = (K_{equiv.}/\ \varphi)(\varphi\ HL_{AB})(K_1 + K_2/K_1K_2) + 2\ \varphi\ HL_{AB}\ (K_{equiv}/\varphi)\sqrt{(1/K_1K_2)}]$; divide through by $HL_{AB}$ **and note that** $\varphi$ **cancels out:**

$$1 = (K_{equiv})(K_1 + K_2/K_1K_2) + 2(K_{equiv})\sqrt{(1/\ K_1K_2)}$$

$$1 = (K_{equiv})\ [(K_1 + K_2/K_1K_2) + 2\ \sqrt{(1/K_1K_2)}]$$

$1 = (K_{equiv})\ [K_1 + K_2/K_1K_2 + 2\sqrt{(1/K_1K_2)}]$; now get a common denominator again:

$$1/(K_{equiv}) = [(K_1 + K_2) + 2\ K_1K_2\ \sqrt{\ (1/K_1K_2)}]/[K_1K_2]$$

$$1/(K_{equiv}) = (K_1 + K_2) + 2\sqrt{(K_1K_2)^2/(K_1K_2)}]/[K_1K_2]$$

**Therefore by inverting both sides of the expression we obtain the desired result:**

$(\mathbf{K_{equiv.}}) = (K_1K_2) \div (K_1 + K_2 + 2\sqrt{(K_1K_2)})$, *which reduces to our **final expressions:***

*FOR FLOW ACROSS THE PIPE LOOP DEFINED BY NODES A AND B:*

**{11}** $\mathbf{(K_{equiv.}) = K_1K_2 \div (\sqrt{K_1} + \sqrt{K_2})^2}$

**and**

**{12}** $\mathbf{HL_{AB} = 0.00259\ (K_{equiv})\ (GPM_{total})^2 \div d_M{}^4}$

It seems interesting, to me at least, to compare result (11) with the expression for equivalent resistance of parallel resistors in an electrical direct current circuit, which yields:

| | | |
|---|---|---|
| $[\mathbf{R_{equiv.}} = R_1R_2 \div (R_1 + R_2)]$ | *versus* | $[(K_{equiv.}) = K_1K_2 \div (\sqrt{K_1} + \sqrt{K_2})^2]$ |
| **Direct electrical current** | | **Fluid mechanical** |

## Examine the Results (11) and (12) for Sensibility:

Let us say that a single series resistance flowpath having a total head loss coefficient **equal to 100** is replaced by a pipe loop, made up of two tees as in our flow schematic herein, Nodes "A" and "B", separated by a pair of parallel pipe branches similar to those in our schematic. Let the loss coefficient of each branch, including the pair of "side of tee" exit and entrance head loss factors, also be equal to **exactly 100.** The loop's equivalent loss coefficient, by Eq. (11), would be:

$$(\mathbf{K_{equiv}}) = K_1K_2 \div (\sqrt{K_1} + \sqrt{K_2})^2 = 100 \times 100 \div (\sqrt{100} + \sqrt{100})^2 = \mathbf{10{,}000/400 = \underline{25}}$$

For the equal-resistance branches, the flowrate through each of the two branches is obviously equal to exactly 1/2 of the total flowrate leading into and exiting from the loop. This is mathematically provable as well as intuitively true. Now, since no other factors change (both "GPM" and "$d_M$" remain unchanged) then the resulting head loss across the loop per Eq. (12) would be exactly 1/4 of that across the original full-sized single series resistance. That is, the ratio $\mathbf{K_{loop\ config}}/K_{original\ series\ config} = 25/100 = 0.25 = 1/4$.

This makes sense to me, because the underlying physics of fully turbulent liquid flow is that the dynamic (frictional flow) head loss is directly proportional to velocity squared, therefore also to the flowrate squared, since $Q = A \times V$, and area A of the equivalent loop is the same as that of the original single-series resistance path, being based upon the same inside diameter as the larger pipe size entering the "entrance tee" at upstream end of Node "A" and leaving the "exit tee" at the downstream end of Node "B", namely, diameter "$\mathbf{d_M}$". Remember, when we analyze a "real" parallel-branch loop by the method just derived herein, what we are doing is mathematically replacing the "actual" pair of tees and the "actual" branch pair of pipelines with an imaginary hypothetical *single* pipeline segment carrying the sum total of the two branch flows and which does not contain those tees, but which instead consists of an imaginary single equivalent pipeline segment of diameter "$\mathbf{d_M}$" and loss coefficient $(\mathbf{K_{equiv}})$ found by Eq. (11). When we plug "$\mathbf{GPM_{total}}$" into the head loss relationship using $\mathbf{K_{equiv}}$ and $\mathbf{d_M}$ as our parameters, we obtain exactly the same head loss as we would find between Nodes "A" and "B" in either of the two "actual" branches carrying their "actual" flowrates. The only difference is this; we have gained the ability to simplify our analysis by getting rid of the loop and its attendant trial-and-error branch flowrate-iteration solution. And this is quite a gain, not only for the head loss calculation at design flowrate, but for the Bernoulli's equation system scaling analysis which follows it and which is necessary to do a proper job of engineering the pump and driver selection/specification!

So it makes sense to me. But I still am counting on you, reader, to let me know if the analysis is wrong. So check it out!

## Before Applying the Results to Your Work, Please Consider the Following Qualifications:

Earlier, we stressed the importance of the constancy of loss coefficients for valves, fittings, etc. (everything but straight runs of pipe) by virtue of their being defined as some numerical multiplier times the "fully turbulent wall friction factor" which is tabulated in Crane 410 as a constant for a pipe of the same nominal diameter as the fitting or valve. This feature makes the loss *coefficient* of a fitting independent of "GPM" (not the actual *loss,* of course), and thus enables the derivation of the loop's equivalent resistance expression in terms of the component branch values $K_1$ and $K_2$. However, it does not eliminate the fact that the *friction factor* and hence the *head loss per foot of pipe* in a straight run of pipe *does* vary with the numerical value of the flow's **Reynolds number,** $\mathbf{VD\rho/\mu}$. And the various fittings, valves, elbows, etc. are usually connected by runs of straight pipe in the real world!

If the Reynolds number is less than that required for fully developed turbulent flow, for a given pipe roughness/diameter ratio as we discussed earlier, then the *loss coefficient* for the straight run of pipe *will* change with flowrate to some extent (velocity will change though diameter remains constant.) As a result, if the flow in a particular problem contains some straight pipe runs as well as fittings, and is characterized by low-velocity, low-Reynolds number turbulent flow, or even worse by transitional regime flow, then the preceding analysis can be no better than "approximate" because it is based on the assumption of friction factor independence from velocity in the pipe. And if the flow regime is laminar, signaled by a Reynolds number < 4,000, the preceding loop analysis is no good at all.

The question is, is this a problem? And if so, how can it be evaluated?

My answer, as usual in our "nothing comes easy" engineering profession, is *"It depends." "Well of course it depends, MechMentor, you old idiot!"* "you say? *"We need to know what the heck it depends upon!!"* Fair enough. It depends on where the operating point falls on the Moody diagram. Let's use a couple of real pipes as examples to illustrate this.

Let's use a 1-in pipe and an 8-in pipe, both of standard wall thickness, which for them is Schedule 40, and both made of welded steel material. The small table in the bottom right-hand corner of the copies of Moody's diagram included in this chapter tells us to use an average value of 0.0002 ft for the "new pipe" roughness of this material. We look up the inside diameters of 1 in and 8 in Schedule 40 pipes as 1.049 in and 7.981 in, respectively. So we calculate roughness/diameter ratios respectively of (12 × 0.0002)/1.049 and (12 × 0.0002/7.981) respectively, which resolve numerically to 0.00229 for the small 1-in pipe and 0.00030 for the larger 8-in one.

On the Moody chart, we find that if the relative roughness is 0.0023, the 1 in-pipe will be transporting fully turbulent flow if the Reynolds number exceeds about $\mathbf{5 \times 10^5}$ (500,000.) So the 1-in pipe's friction loss factor **"f"** read from the Moody chart will be a constant, and about ≈ **0.024** for all flows having Reynolds numbers of $\mathbf{5 \times 10^5}$ and greater. For relative roughness of 0.0003, we see that the 8-in pipe friction factor **"f"** will be a constant ≈ **0.015** for all flows having Reynolds numbers of $\mathbf{5 \times 10^6}$ (5,000,000) and greater. (Crane Technical Paper No. 410 lists the values of $\mathbf{f_T}$ for clean 1-in nominal pipe size steel pipe as **0.023** and as **0.014** for the 8-in pipe. I won't quibble over the numerical differences.)

Now here is the part we are interested in; what sort of minimum velocity and flowrate do the pipes need, in order to have fully turbulent flow, and thus a constant friction factor? To answer, we simply look up values for density of water at 60°F., which is 62.37 lbm/ft$^3$, and for the dynamic viscosity, which is about 1.13 centipoise × conversion constant 0.000672 = 0.00076 lbm/ft-sec and plug them into the expression for Reynolds number. Using the $\mathbf{5 \times 10^5}$ for our minimum velocity basis in the small pipe, this yields:

$$\mathbf{V]}_{\text{1-in water pipe}} = (5 \times 10^5 \times 0.00076)/(1.049\text{ in}/12\text{ in per ft})(62.37) = \underline{69.7\text{ ft/sec!}}$$

**I hope your reaction to this result is something similar to:**

***"Wow! Hokey Smokes!! 69.7 feet per second?!?! How can that be?!*** **We seldom go above 3 or 4 feet per second velocity in 1-in pipe, which is around 8 to 10 gpm flowrate, because if we do, the pressure drop gets tremendously large and we get all sorts of water hammer bangs when we shut the pump off!! No, Sir! Not on my watch, we don't go 69.7 feet per second!!"**

If that was not your reaction, please go back and read the first part of this chapter again. A velocity of 70 feet per second for any liquid in any pipe is way out of bounds. In tiny pipes such as this 1-inch example, I personally try to keep velocity no higher than maybe 3 feet per second, or flowing at about 8 gpm maximum in a 1-in pipe. And for the very reasons stated in (hopefully) your own reaction above.

But before I finish this long-winded sermon, we need to repeat what we just did, this time for the larger 8-in pipe size. When we do, we obtain the following:

$$\mathbf{V.}_{\text{8-in water pipe}} = (5 \times 10^6 \times 0.00076)/(7.981\text{ in}/12\text{ in per ft})(62.37) = \underline{91.6\text{ ft/sec!!!}}$$

**This is even worse!** Old MechMentor rants and raves about setting a velocity limit around 7 or so ft/sec maximum in larger pipes. And that limitation is not so much because of high pressure drop due to friction as it is to dangerous levels of kinetic energy in potential water hammers, and to a lesser degree to the large motor size/high power consumption in the pump drive motors brought on by the higher speeds. A velocity of even ***9*** feet per second is out of the question as far as I am concerned, and ***90*** feet per second is totally ridiculous for any actual design application.

## Now What Have We Learned?

- We learned that friction factor in straight pipes is not independent of flowrate and velocity in any real-world piping system unless we force it to run at velocities far beyond the permissible practical safe limits. The friction factor goes up as the velocity goes down, and if we try to use fully-turbulent flow assumptions in low-velocity piping, we will grossly underestimate the head losses. Which is a bad thing.
- We cannot safely push liquids fast enough in usable pipe sizes to achieve fully developed turbulent flow. Because of this fact, changes in flowrate will not be completely negligible in friction factor and combined loss coefficient calculations. They *will* affect the accuracy of our hydraulic calculations.
- So if we try to apply our painfully derived expression for parallel loop equivalent networks verbatim, in a loop which contains straight piping as well as the various valves and fittings, we are not going to be very accurate. We could reasonably expect somewhat better accuracy when operating at higher velocities. But we have no reason to take comfort from this, because we are very unlikely to operate very far from 2 or 3 feet per second minimum to 7 or 8 ft/sec maximum in any properly engineered pressure-flow systems, regardless of pipe size or fluid chemistry.
- We come a bit closer to the truth with our assumptions in very small, very rough pipes, than we do with more usual normal-sized non-corroded pipes. But, big deal! Who sets out to use teeny-tiny rough-as-a-cob pipes anyway?! Nobody I ever saw! Why would we ever?
- So we must accept that if we try to play electrical engineer, with neat networks of constant-resistance elements which can be network-simplified into simple lumped series by simple algebra, with nice, neat, simple, once-through, closed-form potential-flow solutions, then we are likely to get burned in the process.
- The question is just how badly we get burned. Singed, scorched, or toasted? Or maybe just warmed up a little bit while we scramble to check our predictive calculations against field-measured startup results at system commissioning time?

We probably need to look just a bit further before we shut down this discussion, since the use of parallel branch loops is not going away in practice, and neither is our responsibility for the hydraulic calculations. Our whining feels good right now, but in the long run it helps nothing.

So here's the bottom line. We can start by recognizing that "constant friction factor" is probably not a really bad assumption, usually, because most systems requiring precise flowrate control (hence exact knowledge of flowrate) are going to be operated at or very near a single design value. Even more, they will usually be brute-forced to do so by an automated flow loop controller of some sort. So it just falls on us to put in enough pump, with enough motor, to take care of relatively minor hydraulic uncertainties, comfortably, without gross overkill.

*(We can actually be much worse off if our system design is overkilled, that is, with the pump being way too big and powerful for the actual installed system, resulting in having to operate the system in a severely throttled-down condition, than if we were to*

*accidentally "underkill" it just a little bit through underestimation of system head losses. We put throttles and balance valves in the system for just that purpose, and you must plan on the flow balances and throttles being partly closed at all times accordingly in your system curve calculations. Then, if you underestimated the head loss at design flow a little, you can always open the valves a bit. The reverse situation is not so easily handled. I know that someday, if you continue mechanical engineering and have not already seen this paradoxical fact demonstrated, then you will see it sooner or later. It happens all the time. Don't you be one of the ones it happens to! Make crusty old MechMentor proud of you, sons & daughters; arrgh!)*

Next, when you do encounter a piping loop, your analysis well be considerably more accurate if you first calculate the serial branch fitting losses $\mathbf{K_2}$ and $\mathbf{K_1}$ first, then use this relationship when guessing at the flow through each branch: $\mathbf{K_2}/K_1 \sim (GPM_1/GPM_2)^2$. Having a more nearly accurate pair of values for $\mathbf{GPM_1}$ and $GPM_2$ to begin with, will make your estimate of straight run Reynolds numbers, friction factors, and hence the combined head losses a lot more accurate (as compared to guessing a 50/50 branch flow distribution and trucking onward without ever looking back).

Finally. FINALLY, I SAY. When things are really critical, and you have to be as right as you can possibly be, then you can use your Crane 410-based computer program with the following trial and error procedure to get very, very close.

Step 1. Estimate $GPM_1$ and $GPM_2$ as suggested above.
Step 2. Use that estimate to compute $K_2$ and $K_1$.
Step 3. find the head loss across the loop by:
$HL_{a-b} = 0.00259\,[(K_2)\,(K_1)/(\sqrt{K_1} + \sqrt{K_2})^2]\,(GPM_1 + GPM_2)^2 \div d_M{}^4$
Step 4. use the following expression to find $GPM_1$

$$HL_{a-b} = 0.00259\,(K_1)\,(GPM_1)^2 \div d_M{}^4$$

and then find $GPM_2 = GPM_{total} - GPM_1$
Step 5. Recompute $K_2$ and $K_1$ based on the new values of $GPM_1$ and $GPM_2$
Step 6. Repeat Step 3 using the new values of $K_2$ and $K_1$ to find the new value of $HL_{a-b}$.
Step 7. repeat Step 4 and then compare $GPM_1$ and $GPM_2$ with previous values. Reiterate this series of steps until it converges to your desired level of accuracy.

Whoof.

"K" FACTOR TABLE—SHEET 1 of 4

# Representative Resistance Coefficients (K) for Valves and Fittings

*("K" is based on use of schedule pipe as listed on page 2-10)*

## PIPE FRICTION DATA FOR CLEAN COMMERCIAL STEEL PIPE WITH FLOW IN ZONE OF COMPLETE TURBULENCE

| Nominal Size | ½″ | ¾″ | 1″ | 1¼″ | 1½″ | 2″ | 2½, 3″ | 4″ | 5″ | 6″ | 8-10″ | 12-16″ | 18-24″ |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Friction Factor ($f_T$) | .027 | .025 | .023 | .022 | .021 | .019 | .018 | .017 | .016 | .015 | .014 | .013 | .012 |

## FORMULAS FOR CALCULATING "K" FACTORS FOR VALVES AND FITTINGS WITH REDUCED PORT

- **Formula 1**

$$K_2 = \frac{0.8 \sin\frac{\theta}{2}(1-\beta^2)}{\beta^4}$$

- **Formula 2**

$$K_2 = \frac{0.5\,(1-\beta^2)\sqrt{\sin\frac{\theta}{2}}}{\beta^4}$$

- **Formula 3**

$$K_2 = \frac{2.6 \sin\frac{\theta}{2}(1-\beta^2)^2}{\beta^4}$$

- **Formula 4**

$$K_2 = \frac{(1-\beta^2)^2}{\beta^4}$$

- **Formula 5**

$$K_2 = \frac{K_1}{\beta^4} + \text{Formula 1} + \text{Formula 3}$$

$$K_2 = \frac{K_1 + \sin\frac{\theta}{2}\left[0.8\,(1-\beta^2) + 2.6\,(1-\beta^2)^2\right]}{\beta^4}$$

- **Formula 6**

$$K_2 = \frac{K_1}{\beta^4} + \text{Formula 2} + \text{Formula 4}$$

$$K_2 = \frac{K_1 + 0.5\sqrt{\sin\frac{\theta}{2}}\,(1-\beta^2) + (1-\beta^2)^2}{\beta^4}$$

- **Formula 7**

$$K_2 = \frac{K_1}{\beta^4} + \beta\,(\text{Formula 2} + \text{Formula 4}) \text{ when } \theta = 180^\circ$$

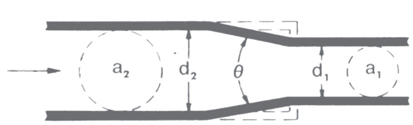
$$K_2 = \frac{K_1 + \beta\left[0.5\,(1-\beta^2) + (1-\beta^2)^2\right]}{\beta^4}$$

$$\beta = \frac{d_1}{d_2}$$

$$\beta^2 = \left(\frac{d_1}{d_2}\right)^2 = \frac{a_1}{a_2}$$

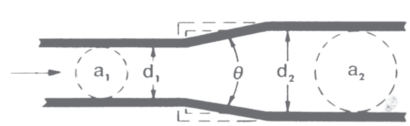
Subscript 1 defines dimensions and coefficients with reference to the smaller diameter.
Subscript 2 refers to the larger diameter.

## SUDDEN AND GRADUAL CONTRACTION




If: $\theta \leq 45^\circ$ . . . . . . . . . $K_2$ = Formula 1
$45^\circ < \theta \leq 180^\circ$ . . . $K_2$ = Formula 2

## SUDDEN AND GRADUAL ENLARGEMENT




If: $\theta \leq 45^\circ$ . . . . . . . . . $K_2$ = Formula 3
$45^\circ < \theta \leq 180^\circ$ . . . $K_2$ = Formula 4

"K" FACTOR TABLE—SHEET 2 of 4

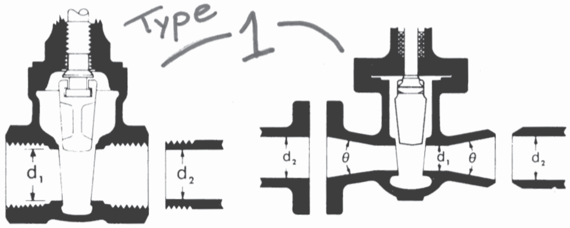
# Representative Resistance Coefficients (K) for Valves and Fittings

*(for formulas and friction data, see page A-26)*

*("K" is based on use of schedule pipe as listed on page 2-10)*

## GATE VALVES
### Wedge Disc, Double Disc, or Plug Type

Type 1



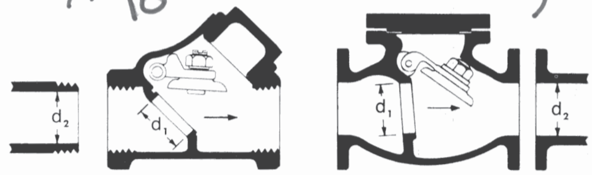

If: $\beta = 1,\ \theta = 0 \ldots\ldots\ldots\ldots K_1 = 8\,f_T$

$\beta < 1$ and $\theta \leq 45° \ldots\ldots K_2 =$ Formula 5

$\beta < 1$ and $45° < \theta \leq 180° \ldots K_2 =$ Formula 6

## SWING CHECK VALVES

Type 10 Type 11




$K = 100\,f_T$

Minimum pipe velocity (fps) for full disc lift $= 35\sqrt{\bar{V}}$

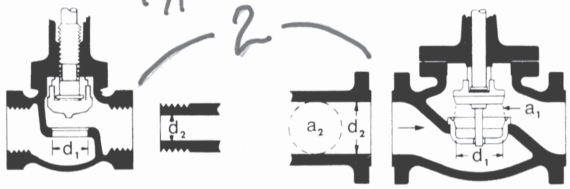
$K = 50\,f_T$

Minimum pipe velocity (fps) for full disc lift $= \cancel{48}$ 60 $\sqrt{\bar{V}}$

or $100\sqrt{\bar{V}}$ for U.L. listed valve

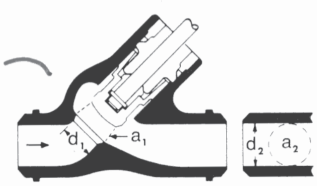
## GLOBE AND ANGLE VALVES

Type 2



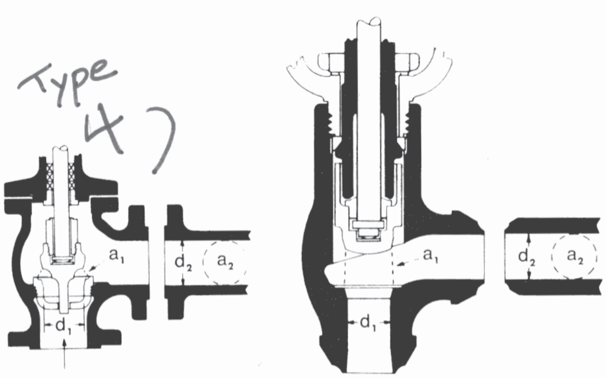

If: $\beta = 1 \ldots K_1 = 340\,f_T$

Type 3




If: $\beta = 1 \ldots K_1 = 55\,f_T$

Type 4

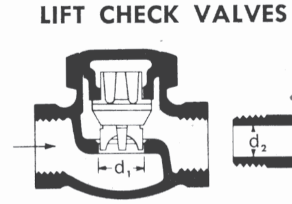



If: $\beta = 1 \ldots K_1 = 150\,f_T$ If: $\beta = 1 \ldots K_1 = 55\,f_T$

All globe and angle valves, whether reduced seat or throttled,
If: $\beta < 1 \ldots K_2 =$ Formula 7

## LIFT CHECK VALVES

Type 12

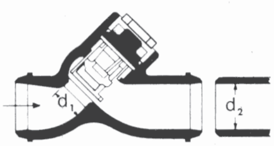



If: $\beta = 1 \ldots K_1 = 600\,f_T$

$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity (fps) for full disc lift $= 40\,\beta^2\sqrt{\bar{V}}$

Type 13

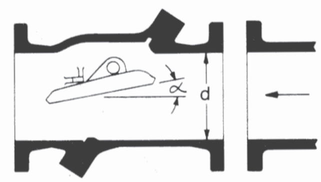



If: $\beta = 1 \ldots K_1 = 55\,f_T$

$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity (fps) for full disc lift $= 140\,\beta^2\sqrt{\bar{V}}$

## TILTING DISC CHECK VALVES




| | $\alpha = 5°$ | $\alpha = 15°$ |
|---|---|---|
| Sizes 2 to 8"... $K =$ | $40\,f_T$ | $120\,f_T$ |
| Sizes 10 to 14"... $K =$ | $30\,f_T$ | $90\,f_T$ |
| Sizes 16 to 48"... $K =$ | $20\,f_T$ | $60\,f_T$ |
| Minimum pipe velocity (fps) for full disc lift = | $80\sqrt{\bar{V}}$ | $30\sqrt{\bar{V}}$ |

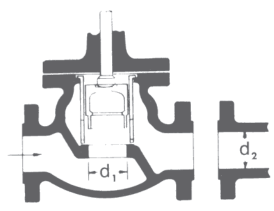
"K" FACTOR TABLE—SHEET 3 of 4

# Representative Resistance Coefficients (K) for Valves and Fittings

*(for formulas and friction data, see page A-26)*

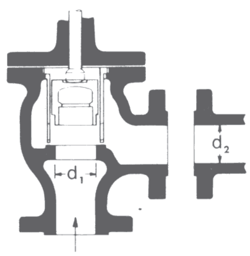
*("K" is based on use of scheduled pipe as listed on page 2-10)*

## STOP-CHECK VALVES (Globe and Angle Types)



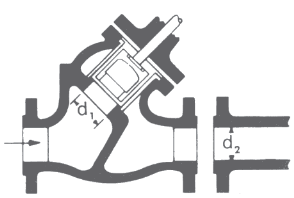

If:
$\beta = 1 \ldots K_1 = 400\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity for full disc lift
$= 55\, \beta^2 \sqrt{\bar{V}}$

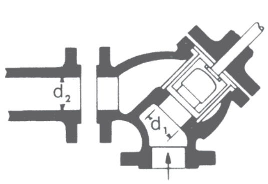



If:
$\beta = 1 \ldots K_1 = 200\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity for full disc lift
$= 75\, \beta^2 \sqrt{\bar{V}}$

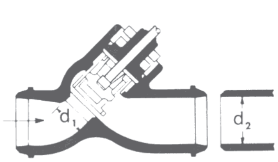



If:
$\beta = 1 \ldots K_1 = 300\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

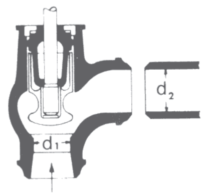



If:
$\beta = 1 \ldots K_1 = 350\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity (fps) for full disc lift
$= 60\, \beta^2 \sqrt{\bar{V}}$




If:
$\beta = 1 \ldots K_1 = 55\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

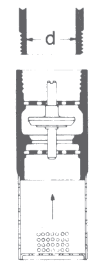



If:
$\beta = 1 \ldots K_1 = 55\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity (fps) for full disc lift
$= 140\, \beta^2 \sqrt{\bar{V}}$

## FOOT VALVES WITH STRAINER

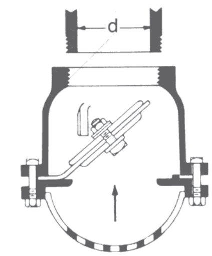
### Poppet Disc




$K = 420 f_T$

Minimum pipe velocity (fps) for full disc lift
$= 15 \sqrt{\bar{V}}$

### Hinged Disc



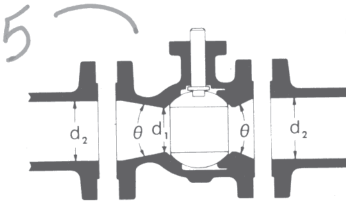

$K = 75 f_T$

Minimum pipe velocity (fps) for full disc lift
$= 35 \sqrt{\bar{V}}$

## BALL VALVES

TYPE 5

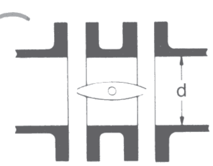
LIQUID-FLOW PROGRAM USES THIS




If: $\beta = 1,\ \theta = 0 \ldots\ldots\ldots\ldots K_1 = 3\, f_T$
$\beta < 1$ and $\theta \leq 45° \ldots\ldots\ldots K_2 =$ Formula 5
$\beta < 1$ and $45° < \theta \leq 180° \ldots K_2 =$ Formula 6

## BUTTERFLY VALVES

Type 6




61 – Sizes 2 to 8″ ... $K = 45\, f_T$
62 – Sizes 10 to 14″ ... $K = 35\, f_T$
63 – Sizes 16 to 24″ ... $K = 25\, f_T$

"K" FACTOR TABLE—SHEET 4 of 4

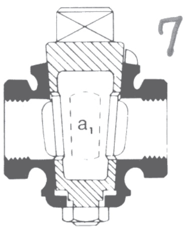
# Representative Resistance Coefficients (K) for Valves and Fittings

*(for formulas and friction data, see page A-26)*

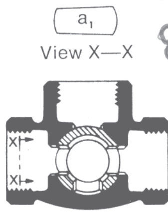
*("K" is based on use of schedule pipe as listed on page 2-10)*

## PLUG VALVES AND COCKS

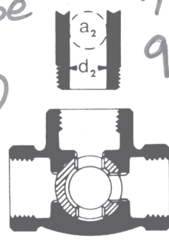
**Straight-Way** Type 7)




**3-Way** Type 8)




Type 9)



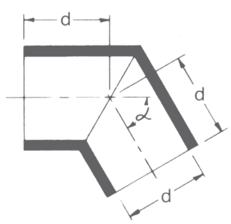

If: $\beta = 1$, $K_1 = 18\,f_T$

If: $\beta = 1$, $K_1 = 30\,f_T$

If: $\beta = 1$, $K_1 = 90\,f_T$

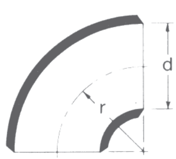
If: $\beta < 1 \ldots K_2 =$ Formula 6

## MITRE BENDS




| α | K |
|---|---|
| 1 0° | 2 $f_T$ |
| 15° | 4 $f_T$ |
| 30° | 8 $f_T$ |
| 45° | 15 $f_T$ |
| 60° | 25 $f_T$ |
| 75° | 40 $f_T$ |
| 90° | 60 $f_T$ |

## 90° PIPE BENDS AND FLANGED OR BUTT-WELDING 90° ELBOWS




| r/d | KR | r/d | KR |
|---|---|---|---|
| 1 | 20 $f_T$ | 8 | 24 $f_T$ |
| 1.5 | 14 $f_T$ | 10 | 30 $f_T$ |
| 2 | 12 $f_T$ | 12 | 34 $f_T$ |
| 3 | 12 $f_T$ | 14 | 38 $f_T$ |
| 4 | 14 $f_T$ | 16 | 42 $f_T$ |
| 6 | 17 $f_T$ | 20 | 50 $f_T$ |

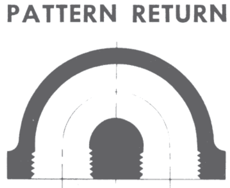
The resistance coefficient, $K_B$, for pipe bends other than 90° may be determined as follows:

$$K_B = (n - 1)\left(0.25\,\pi\, f_T \frac{r}{d} + 0.5\,K\mathrm{R}\right) + K\mathrm{R}$$

$n$ = number of 90° bends i.e. bend angle ÷ 90°

$K$ = resistance coefficient for one 90° bend (per table)

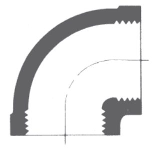
## CLOSE PATTERN RETURN BENDS



$K = 50\,f_T$

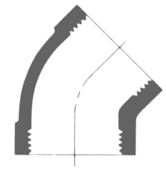
## STANDARD ELBOWS

90°



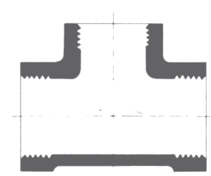
$K = 30\,f_T$

45°



$K = 16\,f_T$

## STANDARD TEES

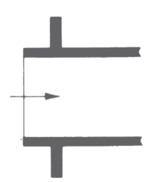


Flow thru run . . . . . . . $K = 20\,f_T$

Flow thru branch . . . . $K = 60\,f_T$

## PIPE ENTRANCE

**Inward Projecting**

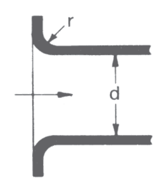


$K = 0.78$

| r/d | K |
|---|---|
| 0.00* | 0.5 |
| 0.02 | 0.28 |
| 0.04 | 0.24 |
| 0.06 | 0.15 |
| 0.10 | 0.09 |
| 0.15 & up | 0.04 |

*Sharp-edged

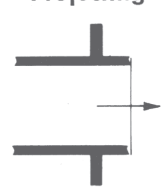
**Flush**




For $K$, see table

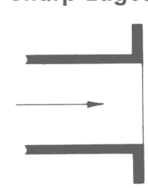
## PIPE EXIT

**Projecting**



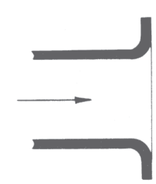
$K = 1.0$

**Sharp-Edged**



$K = 1.0$

**Rounded**



$K = 1.0$

CHAPTER

# 9

# WATER CHILLERS IN TURNDOWN

## Calculating Time Required for a Chiller to Draw Down the Temperature of an Insulated Closed Water Reservoir to a Relatively Low Value (Near Chiller Minimum Discharge Ability)

Modern packaged water chillers come equipped with a variety of talented microprocessor and/or PC-based internal controllers and system controls, which can be interfaced with building automation systems and other networks quite easily.

These controls, in conjunction with modern, throttleable-by-inlet-vane-or-other-means refrigerant compressors (or their equivalent) such as the high-efficiency centrifugals, scroll-types and screw machines, make something not only possible but easy and relatively efficient, which was previously difficult and risky: operation at a heat transfer rate much ***smaller*** than the chiller's maximum capacity.

Commonly called "turndown," or "part-load operation", the ability of a chiller to run continuously at only a small fraction of its design capacity is very important to the system designer. Precision HVAC systems often need to maintain tight control of room air dry bulb temperature and humidity. The HVAC system control must be continuous and simultaneous, able to adapt rapidly and effectively to changes over a wide range of air-cooling and moisture-removal loadings.

Such systems can be very effectively and economically designed around variable air volume (VAV) airside mechanical units, utilizing chilled water generating plants of the primary-secondary loop type for the refrigeration source (see more on this topic in chapters 7 and 10 of this book). In these systems, the chilled water flowrate through the chiller evaporator is maintained at a constant, preset optimum value. A typical value would be a flowrate of **2.4 gpm per ton** (12,000 Btu/Hr) of chiller capacity.

For example, a 200-ton chiller would constantly recirculate a water flow of **exactly** (200 × **2.4**) = 480 gpm through its evaporator (the primary chilled water loop pump and its flowrate setter control are dedicated to this task.) This would yield a maximum water temperature drop of **10°F** across the evaporator whenever the external load on the chiller was 200 tons (2,400,000 Btu/hr or more.

But when the external cooling load was less than 200 tons, the chiller's internal control system would act to prevent the chilled water temperature from dropping below the desired thermostatic setting, which might be as low as 38°F or as high as maybe 44°F. To do this, it unloads the compressor in a manner that reduces the amount of heat picked up from the water by the refrigerant in the chiller evaporator. Otherwise the refrigerant temperature would begin to drop below its intended range, which if unattended could lead to big problems!

It may be instructive to demonstrate what we just discussed with some numbers. This will illustrate what happens when the external load, as reflected by the warmed-up chilled water return temperature entering the evaporator (i.e., the ***EWT***) varies from high to low.

### Example

***200-ton chiller*** *@ constant 480 gpm water (primary loop) flowrate pumped through evaporator, with chiller thermostat set to produce constant 38°F* water temperature leaving evaporator (i.e., the **LWT**):

| | Load, Tons of Refrigeration | | | |
|---|---|---|---|---|
| | **>200** | **200** | **100** | **50** |
| ***% Load*** | ≧100% | 100% | 50% | 25% |
| ***Turndown*** | None | None | 2:1 | 4:1 |
| ***EWT, °F*** | ≧50°F | 48°F | 43°F | 40.5°F |
| ***LWT, °F*** | ≧40°F | 38°F | 38°F | 38°F |
| ***Water gpm*** | 480 | 480 | 480 | 480 |
| ***(EWT - LWT)*** | 10°F | 10°F | 5°F | 2.5°F |

During part-load operation, the chiller must protect itself against abnormally low refrigerant temperatures which might freeze-up the compressor and cause significant damage. The chiller can and will trip itself offline for its own safety's sake, if continued operation below a safe minimum percent load is forced upon it by the HVAC controls. Once tripped-off by the chiller's factory-engineered internal safety instrumentation and control package, the chiller must remain de-energized for a safe length of downtime; see your chiller factory rep for a list of reasons why this is so, and how long a particular chiller must remain tripped off before restarting.

When the partial refrigeration load is within the particular chiller's programmed safe range of operation, the chiller can operate continuously. For example, assume a 200-ton screw compressor with available 20:1 turndown ratio; theoretically, the chiller could operate continuously at a heat removal rate in its evaporator of only (200/20) = 10 tons, or 120,000 Btu/hr which is only 5% of its design capacity. (Whether you would wish to operate at 5% for a long time is another question; we are only discussing possibilities here.)

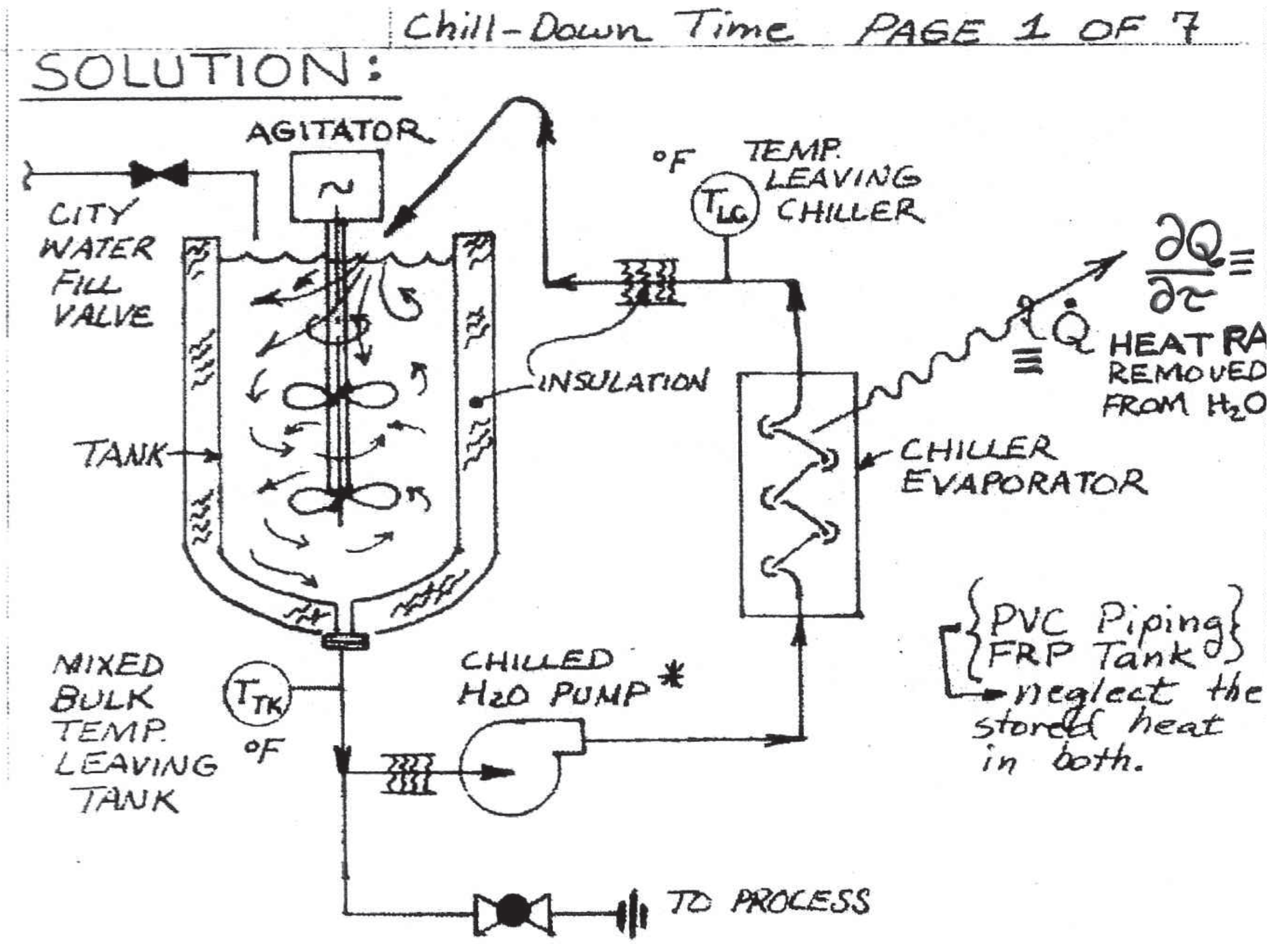
To accomplish this turndown operation, a typical chiller controller package would probably include an internal standalone microprocessor with PID (proportional-integral-derivative) logic program. Among other tasks, the controller would act upon the chiller load-control mechanism, in response to the chilled water discharge thermostatic setpoint and the sensed water temperature leaving the evaporator, to adjust the refrigeration rate produced at the compressor to match the external cooling load, with the intended result of maintaining a constant leaving water temperature **(LWT)** at the evaporator. If it failed to do this, the temperature would rise or fall in proportion to the mismatch.)

Now for a practical application of this knowledge which someday you may be called upon to demonstrate on the job.

Once, I was asked this question regarding chiller sizing to do the following chilled water job: (process, not HVAC)

*"A 10,000 gallon water volume will start off at 85°F. The tank and piping are plastic and are thermally well insulated. Before feeding the water into a batch process, the water is to be cooled to 40°F by recirculating it through a 100-ton screw-type chiller. The chiller was specified to produce 38°F at full load. How long will it take the 100-ton chiller to do the job?"*

The illustrated solution is self-explanatory. It is worked out in terms of an unspecified chiller tonnage of maximum capacity = $Q'_{CAP}$ tons. So the original question, "how many hours will it take" can be answered by plugging-in the chiller "nameplate" tonnage of your choice in place of the variable $Q'_{CAP}$".




*Pump delivers 2.40 gpm per ton of chiller capacity. At full load, the chilled water $\Delta T$ across the chiller evaporator equals:

$$\Delta T = (T_{TK} - T_{LC})^{o}F = 1 \text{ ton} \times (12{,}000 \text{ BTU/Hr.-ton}) \div (500 \times 2.4 \text{ gpm/ton}) = 10\ ^{o}F$$

let $\tau \equiv$ elapsed time (in Hours) to reduce TTK by the specified number of degrees;

let $Q \equiv$ heat (in BTU's) contained in the total 10,000 gallon water volume;

let $\{\partial \mathbf{Q}/\partial \tau\} \equiv$ rate of heat removal from the water in the chiller evaporator (in BTU/Hr.)

let $Q'_{CAP} \equiv$ full –load capacity of the chiller in tons ( as usual, 1 ton $\equiv$ 12,000 BTU/Hr.)

The goal is to start with a 10,000 gallon charge of water at 85°F, recirculate through the chiller until $\mathbf{T_{TK}}$ = 40°F. The question is *"For a chiller of given nameplate capacity "$Q'_{CAP}$"* tons, how long will it take for this to happen?"

## Assumptions

1. The chiller develops its full capacity when the value of temperature leaving the chiller evaporator $\mathbf{T_{LC}}$ = **38°F.** The chiller will unload (reduce its refrigeration rate automatically) proportionally such that $\mathbf{T_{LC}}$ stays constant and **never drops below 38°F.**
2. Water flowrate through the evaporator remains constant at **2.40 gpm per ton of full-load capacity** (assumed same as its "nameplate" or rated capacity) regardless of actual load at all times.
3. This gives us a **temperature drop across the evaporator ΔT = 10.00°F at full load.**
4. **When $\mathbf{T_{TK}}$** ≧ 48°F, the chiller can develop full tonnage, and $(\mathbf{T_{TK}} - T_{LC}) = \Delta T$ = 10.00°F.

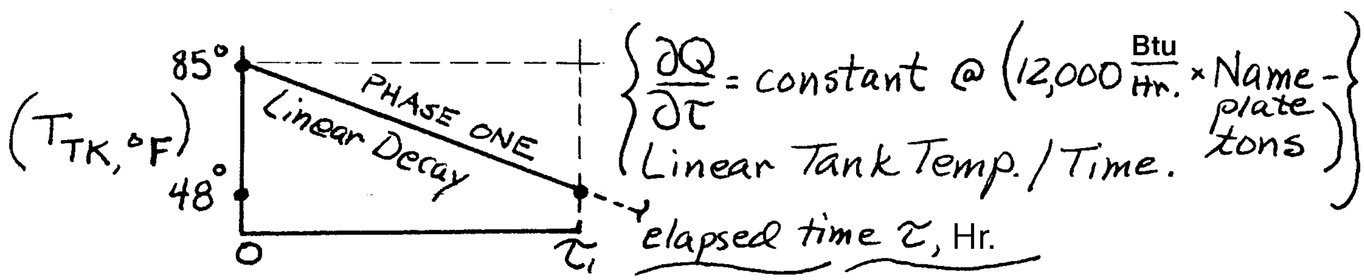
   **Example: when $\mathbf{T_{TK}}$** = 85°F, then
   $\mathbf{T_{LC}} = (T_{TK} - \Delta T)$ = (85°F – 10°F) = 75°F

   **Example: when $\mathbf{T_{TK}}$** = 48°F, then
   $\mathbf{T_{LC}} = (T_{TK} - \Delta T)$ = (48°F – 10°F) = 38°F
5. **When $\mathbf{T_{TK}}$** < 48°F, the chiller will automatically unload itself in a linear fashion, such that $\mathbf{T_{LC}} = (T_{TK} - \Delta T)$ = 38°F:

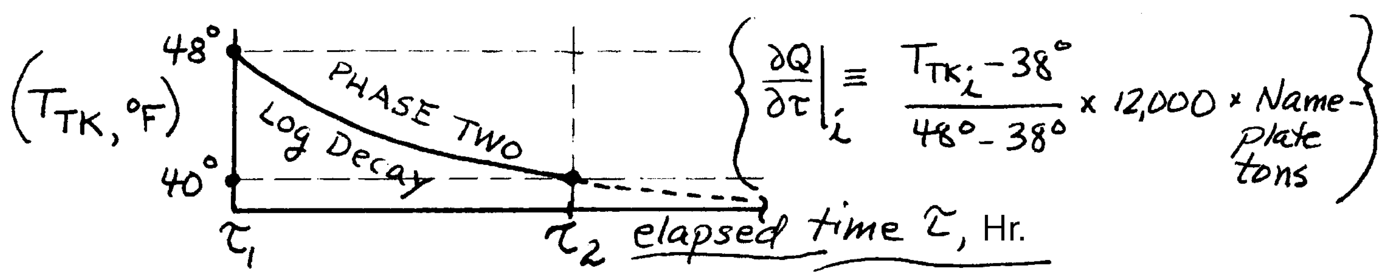
   **Example: when $\mathbf{T_{TK}}$** = 47°F, then
   **ΔT = (47°–38°) = 9°F**.
   $\mathbf{T_{LC}} = (T_{TK} - \Delta T)$ = (47° – 9°) = 38°F and the chiller's % load = (9°/10°) × 100% = 90% load

   **Example: when $\mathbf{T_{TK}}$** = 40°F, then
   **ΔT = (40°F – 38°F) = 2°F**
   $\mathbf{T_{LC}} = (T_{TK} - \Delta T)$ = (40°F – 2°F) = 38°F, and the chiller's % load = (2°F/10°F) × 100% = 20% load

The result should be two phases of evaporator heat transfer. Phase One will be when the tank temp. $T_{TK}$ is equal to or greater than 48°F, placing the chiller at full load; the plot should show a linear decrease rate of $T_{TK}$ with time.




Phase Two is @ $T_{TK}$ < 48 °F., when chiller load is < 100%, with a declining Log plot. We expect a log decay because the refrigeration rate is forcibly reduced by its controller in direct linear proportion to ΔT as ΔT falls below 10°F.



## Calculations

Find Elapsed Time″ $\tau_{2''}$ in terms of the chiller's full-rated name-plate capacity "name-plate tons", which we will designate as $Q'_{CAP}$, for the total water volume 10,000 gallons to be cooled from the initial tank temperature $T_{TK}$ = 85°F. @ time $\tau = 0$ to the desired final 40°F @ time $\tau = \tau_2$.

- Total sensible heat transfer requirement $\equiv Q_{tot} = m_{tot} \times Cp \times \Delta T_{TK\ total}$ =

  = [10,000 gal × 8.34 lbm/gal × 1.00 Btu/Lbm-°F × (85° – 40°) F] = 3,753,000 Btu

- Phase One: linear temperature decrease with elapsed time.

Define $Q_1 \equiv$ sensible heat transferred in Phase One, which after starting the chiller takes the tank from 85°F down to an intermediate 48°F water storage temperature in the linear fashion.

For the linear rate of temperature fall, the elapsed time is the amount of heat removed during that time interval divided by the refrigeration capacity being applied at full load condition. The amount of heat removed in a linear removal rate is just the fraction of $Q_{tot}$ contained between the linear chiller action limits, which we defined as from 85°F. → 48°F. Our arbitrary temperature lower bound for zero heat content = 40°F.

$$Q_1 = [(85 - 48)/(85 - 40)] \times 3{,}753{,}000 \text{ Btu} = 3{,}085{,}800 \text{ Btu}$$

The elapsed time for phase one is that heat quantity divided by the 100% cooling rate capacity, which we express here as $\mathbf{Q'_{CAP}}$;

$$\Delta\tau_{0\_1}\Big|_{\tau=\tau_0}^{\tau=\tau_1} = 3{,}085{,}800 \text{ Btu}/(12{,}000 \times Q'_{CAP}) \text{ Btu/hr} = \{257.15/\mathbf{Q'_{CAP}}\}$$

*{Remember, $Q'_{CAP}$ is* in "tons".*}*

- Phase Two: logarithmic temperature decrease with elapsed time.

Define $Q_2 \equiv$ sensible heat transferred in phase two, which takes the tank from down from the intermediate 48°F water storage temperature to the final 40°F, where we shut the chiller off.

This time, the instantaneous rate of heat removal in the chiller evaporator is continuously decreased as the chiller's built-in microprocessor load-controller gradually removes load from the compressor using PID logic. It does that by sensing entering water temperature and decreasing load setting as the temperature falls, keeping the evaporator water discharge temperature constant at 38°F.

So we need an expression for the instantaneous heat removal rate at any given instant of time during this phase. We will then be able to use it in an overall expression like the one used above for Phase One only which can be integrated with respect to time to give us the value of the final elapsed time period $\Delta\tau_{1\_2}$, which is what we are seeking. We will add it to $\Delta\tau_{0\_1}$ for the total answer.

The instantaneous heat removal rate at time "$\tau$" = $\partial\mathbf{Q}/\partial\tau$ on interval from τ1 through τ2 is:

$$\partial\mathbf{Q}/\partial\tau\Big|_{\tau=\tau_1}^{\tau=\tau_2} = \mathbf{[(T_{TK_\tau} - 38)/(48 - 38)]} \times (12{,}000 \times Q'_{CAP})$$

*note: {Btu/hr cooling rate}*

Define $J \equiv (12{,}000 \times Q'_{CAP})/(48 - 38) = 12{,}000\ Q'_{CAP}$

$$\frac{\partial Q}{\partial \tau} = J(T_{TK} - 38); \text{ separate the variables}$$

$$\frac{\partial Q}{(T_{TK} - 38)} = J\partial\tau$$

Since $Q \equiv mCp\,(T_{TK} - 40)$,

$\partial Q \equiv mCp\ \partial T_{TK} = (10{,}000 \text{ x } 8.34 \text{ x } 1.00)\partial T_{TK}$

$\partial Q = 83{,}400\ \partial T_{TK}$

Substitute into our equation and obtain:

$$\frac{83{,}400\ \partial T_{TK}}{(T_{TK} - 38)} = J\partial\tau = 1{,}200\ Q'_{CAP}(\partial\tau);$$

Divide both sides by J and integrate:

$$\frac{69.5}{Q'_{CAP}} \int_{48}^{40} \frac{\partial T_{TK}}{(-38 + T_{TK})} = \int_{\tau_1}^{\tau_2}$$

My tired old eyes found a solution for this integral form, hidden away in my ancient, musty-smelling, spineless-covered dog-eared copy of the Mathematical Tables from the *Handbook of Chemistry and Physics,* 11th ed., Chemical Rubber Publishing Company, 1959, page 256, form #29, ***which verily doth saith:***

$$\int \mathbf{dx/(a + bx) = (1/b)} \log_e (a + bx)$$

In our example, a = (–38) and b = (+1). Substitute these values into that solution and obtain for a value of the integral part of the expression developed thus far:

a = –38, b = 1;

$$\frac{69.5}{Q'_{CAP}} \int_{48}^{40} \frac{\partial T_{TK}}{(-38 + T_{TK})} =$$

$$= \frac{69.5}{Q'_{CAP}} \left[\left(\frac{1}{1}\right)\log_e(-38 + T_{TK})\right]_{48}^{40} =$$

$$= \frac{69.5}{Q'_{CAP}} \left[\log_e(-38 + 40) - \log_e(-38 + 48)\right] =$$

$$= \frac{69.5}{Q'_{CAP}} [0.693147 - 2.302585] =$$

$$= \frac{-111.856}{Q'_{CAP}} = \int_{\tau_1}^{\tau_2} \partial\tau = (\tau_2 - \tau_1);$$

Nevermind the negative 111.856, because the term $Q'_{CAP}$ itself is negative (heat loss.)

$$(\tau_2 - \tau_1) = \frac{111.856}{Q'_{CAP}} \text{ hours.}$$

Earlier, our result for the phase one (linear decrease) elapsed time interval was found as:

$$\Delta\tau_{0\_1}\Big|_{\tau=\tau_0}^{\tau=\tau_1} = 257.15/Q'_{CAP}$$

Therefore we have found that the elapsed time interval required to cool the 10,000 gallons of water from 85°F to 40°F =

$$(\tau_2 - \tau_1) = [(257.15/Q'_{CAP}) + (111.856/Q'_{CAP})] \text{ (hr)}$$

{*remember, $Q'_{CAP}$* is in "tons".}

Now for sample results for the 100-ton chiller sizes.

Sample #1: Let $Q'_{cap}$ the small chiller nameplate capacity be = 100 tons of refrigeration.

The elapsed time = (257.15/100) + (111.856/100) = 2.57 + 1.12 = 3.69 hours total.

CHAPTER

# 10

# BASICS OF HYDRAULIC LOOPS

*Chapter 8 in this book discusses the most widely used of the accepted techniques for calculating head losses, pressure drops, and flowrate capacities in commercial/industrial fluid piping systems. (Crane Technical Paper No. 410 basis.) In this chapter, it is assumed that all of chapter 8 has been assimilated by the reader. It is to be understood that reference to Chapter 8 is intended whenever "typical hydraulic calculations" are mentioned or prescribed.*

Chapter 7 covers decoupled hydraulic loop systems for distribution of chilled water in certain industrial HVAC designs. The basic "primary – secondary" approach of chapter 7 applies to a lot of HVAC bulk fluid-moving applications besides chilled water. Hydronic heating hot water, cooling tower condenser water, and heavy-oil recirculation piping loops for boiler fuel handling installations are three that come to mind immediately.

But the decoupled loops of Chapter 7 apply to many other fields of application than just HVAC. And far more common than the "hot tank – cold tank open industrial systems" of Chapter 7 are fully-closed, fully-pressurized systems which have <u>no capacitor tanks at all.</u>

In this topical chapter, we will examine some of these other types of applications, note similarities and differences, and discuss the physical components of such systems from an engineering design point of view.

We need to discuss the "why" as well as the "how," especially when we come to system controls approaches. When we are through, I hope to have laid out for your future study a roadmap to all the pertinent engineering basics of hydraulic utility loops; from the basic concepts, through P&ID development, some typical component sizing & selection criteria, a few vital working details, the more frequent control philosophies and proven workable control schemes.

And at the risk of making the discussion too rudimentary, I will at least mention the reasoning behind many of the commonplace components we use, and try to identify the more insidious system problems as well as some obvious pitfalls to be avoided. *I do not wish to lose the attention of the more experienced reader, but neither do I wish to create mystery or perpetuate myth for the novice engineer trying to survive his or her rookie year in that first engineering job.*

## BASIC CONCEPTS OF HYDRAULIC DISTRIBUTION LOOPS

**Please refer to Figure 10-1.** It reduces the distribution loop idea to its simplest form.

### Basics of the Primary Loop

The basic Primary Loop elements are:

- a conditioning machine,
- the primary pump,
- some piping connecting them, and
- a pair of nodal points shared with the secondary loop.

### Conditioning Machine

The **machine** and the <u>*conditioning*</u> it provides to the primary loop liquid can be:

- a **chiller;** <u>*(removes heat to maintain constant cold loop water temperature)*</u>
- an **ice storage bank;** <u>*(removes heat to maintain constant cold loop chilled glycol/water mixture temperature)*</u>
- a **hydronic hot water generator** or "hot water boiler" (**note:** no actual phase change allowed, all fluid remains liquid throughout the loop) <u>*(adds heat to maintain constant hot loop water temperature)*</u>
- a **process heat exchanger** <u>*(adds or removes heat to maintain constant loop fluid temperature or temp. range)*</u>
- a **filter** device or system <u>*(to remove foreign particulate matter)*</u>
- a **chemical treatment** system <u>*(to maintain constant pH, solution strength, etc.)*</u>
- an **accumulator,** such as a heated #5/#6/Bunker C fuel oil vessel <u>*(adds heat and motion to maintain viscosity limits and prevent density gradients so that secondary loop pump can function properly)*</u>
- some other device which acts to control one or more physical/chemical properties within specified limits.

### Conditioning Machine Similarities

Most primary loop conditioning machines are, by nature and design, able to operate at <u>peak efficiency</u> only if the <u>primary loop flowrate is held constant within rather tight limits</u>. This is especially true for optimum operation of refrigeration machines (chillers).

### Primary Pump

The primary pump's job is to provide a steady source of conditioned fluid ready to supply to the secondary loop, where something else will use part or all of it for some purpose on an

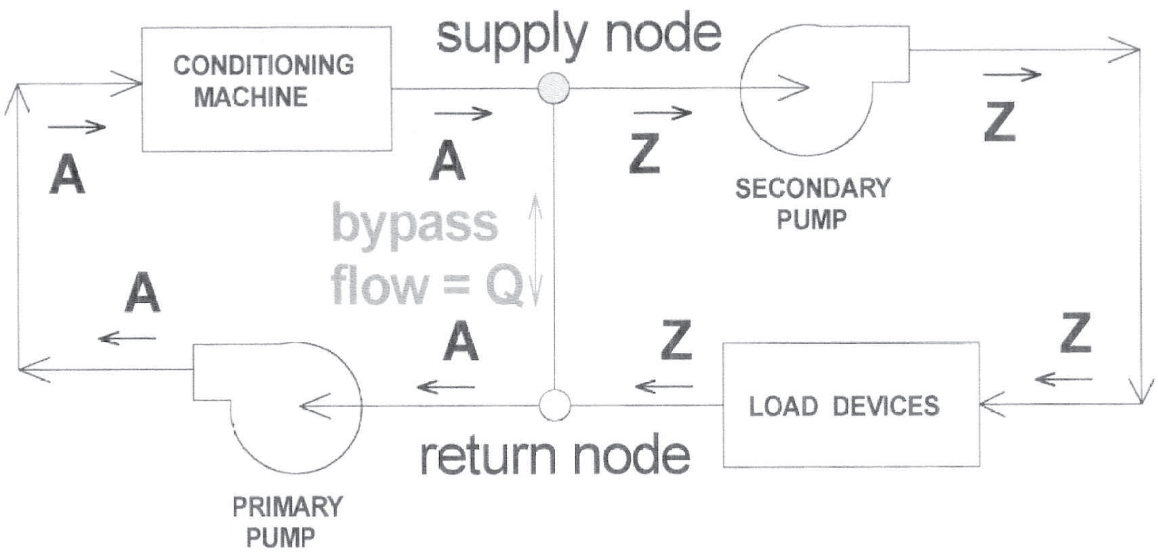



**FIGURE 10-1 OVERALL HYDRAULIC DISTRIBUTION SYSTEM. PRIMARY LOOP PREPARES FLUID CONDITION FOR USE BY THE LOAD DEVICES, AND SUPPLIES IT TO SECONDARY LOOP PUMP INLET HEADER. SECONDARY LOOP CIRCULATES THRU THE LOADS, THEN RETURNS USED FLUID TO PRIMARY PUMP INLET HEADER FOR RECONDITIONING.**

as-needed demand basis. It does that by causing continuous recirculation through the active conditioning machine, at a constant specified volumetric flowrate which is designed to achieve maximum efficiency of energy consumption and best performance of the conditioning machine.

The life cycle cost of electrical power consumption by the conditioning machine and pump will be key to project success. I recommend going for maximum quality and energy efficiency in both items, as the first-cost differential between an excellent piece of equipment and a cheap piece of crap will quickly and completely disappear from any long-term economic analysis, but crappy performance will stick to your reputation forever. No fooling!!

There may be, and in fact usually are, several primary loops in parallel. Parallel loops permit block load switching among a bank of smaller-sized machines when the secondary loop demand varies over a wide range over time. Use of multiple parallel loops enables the individual capacity controls on each separate conditioning machine to perform with precision within its range of capability, maintaining a high value of overall plant energy efficiency.

This is an important feature, and will be discussed later. There definitely are "right" ways and "wrong" ways to configure parallel pumped loops!

## Primary Loop Piping

The amount of frictional head loss in the piping loop should be minimized, because the loop will typically run at 100% of its design flowrate for many hours per year. I suggest that an accurate, simple 20-year owning and operating cost analysis will justify the pipe sizing decision, since the initial piping installation cost, predictable system lifetime, maintenance & replacement costs and the pump motor electrical power consumption are the only economic factors involved. You will probably find that a larger-pipe, lower-velocity primary loop size will be the ticket. In any case, keep the "old pipe condition" head loss per hundred feet of pipe figure at a value smaller than "ten feet of loss", per the reproduced tables of Cameron's Hydraulic Data included in Chapter #2, "Gravity Flow of Liquids in Pipes."

Of course, the piping material should be selected for the operating conditions such that it will last the intended service lifetime and that corrosion and erosion of the pipe wall will not drive the friction and pumping costs too high over time.

## Shared Nodal Points

These are the "supply node" and "return node" shown in Figure 10-1.

Design of these nodal points, believe it or not, is one of the most significant success factors involved with these systems. They are what guarantees hydraulic decoupling of the primary loop from the secondary.

What they are, physically, depends on the system design. We will discuss them following our secondary loop description.

## Secondary Loop Basics

The basic secondary loop elements are:

- one or more load devices, the "users";
- the secondary pump;
- the piping connecting them, and
- a pair of nodal points shared with the primary loop.

## Load Device: the User

In an HVAC application, by definition, the heat transfer load is highly variable. The load device will be an HVAC terminal unit of some sort, such as an air handling unit coil, or a coil in a variable air volume (VAV) box, or a coil in a duct or in a unit heater or a convector. In any of these, except maybe in VAV boxes, a common (and in my opinion the preferred) method of load capacity control is by varying the liquid media flowrate through the coil. (Terminal VAV box control schemes are airside-based, and those boxes incorporating tempering coils of some sort will have their own best peculiar control logic.)

Anyway, concerning HVAC user devices, the key words regarding the secondary loop scheme are *variable* and *liquid flowrate*.

This is also true for a lot of other types of users. Secondary loops are intended to accommodate a wide range of demand of liquid flowrate, from zero to 100% of capacity, but always being supplied the same specified condition (such as a constant chilled water supply temperature).

## Load Device Similarities

We conclude that most secondary loop load devices are designed to operate over a wide range of hydraulic flowrate demands, and load capacity control is achieved by varying the secondary loop flowrate as required to meet the instantaneous net demand.

## Secondary Pump and Loop Piping

The secondary pump's job is to provide a reliable source of conditioned fluid directly to the user load devices, at whatever flowrate the load device may demand at any time. It does that by causing continuous recirculation at *some* rate, which may vary from zero to 100% of the combined peak demands of all the loads, through the secondary loop piping. The secondary piping is a hydraulic circuit consisting of one or more parallel supply headers, a branch *out to* and *back from* each of the load devices, and one or more return headers.

The exact manner in which the secondary loop ***header*** flowrate is made to vary and is controlled to match the always-changing user volumetric demand is one of the characteristics that identifies the type of system. There are several ways in which this may be done successfully. We will discuss them a bit later.

Likewise, each ***user load device branch circuit*** will typically contain an automated flow control of some sort. The branch receives conditioned liquid from the secondary loop supply header, puts it to the device inlet, picks it up at the device outlet, puts it through the flow controller, and finally delivers the used fluid back to the secondary loop return header.

## Shared Nodal Points

Once again, these are the "supply node" and "return node" shown in Figure 10-1.

The **supply node** joins the conditioning machine's fluid discharge to the secondary pump suction. Thus a joining pathway from primary loop to secondary loop is created. The conditioned fluid supply flow direction is as shown by the arrows on Figure 10-1, from "A" through the supply node to "Z".

The **return node** joins the user load devices' combined fluid discharge to the primary pump suction. Thus a joining pathway from secondary loop back to the primary loop is created. The used fluid flow direction, returning for reconditioning, is as shown by the arrows on figure 10-1 from "Z" through the return node to "A".

We have completed a tour around both loops, primary and secondary. This is the combined circuit that serves the basic function of the hydraulic system.

"But," I hope you will ask, "how can we keep the conditioned flow **A** at its strictly maintained constant value, and yet still achieve a wide range of variability in the user demand flow **Z** when the two loops are physically connected at the **nodes**?"

Glad you asked!! Now we can discuss the all-important hydraulic decoupler and the necessary bypass flow which it must carry.

## Decoupling Flow Basics

The primary pump will be set up to deliver a certain, specified, measured volumetric flowrate. Later on we will look at the necessary hardware to make that happen.

Likewise, the secondary pump will be set up to deliver a continuously variable flowrate, and we will look at that too.

Obviously, it would be a very rare event for the two separate loop flowrates to be equal. In fact, if they are ever equal it is a random accident. A flowrate mismatch between primary and secondary loop must exist. For that purpose, we must install an independent flowpath between the two loops, which will permit the surplus flow to bypass the regular circuit and allow the loop flows to be radically different from each other, and yet somehow have no effect of one loop's flow on the other. A neat trick indeed, the heart of the system's concept.

We do that by connecting the two nodes directly with a special bypass path. We will see later how this can be designed, as there are several ways to go about it. But first, let's derive the flowrate relationships which must exist within this network.

Referring once more to Figure 10-1, we see we can apply Kirchoff's law at the two nodes, each of which is a three-way junction because of the bypass path.

## Derive Rules For Flow

**Define flow into a node ≡ (+) positive.**

**Define flow out of a node ≡ (–) negative.**

**Define the node as a physical volume incapable of change in the amount of fluid stored within itself.**

By Kirchoff's law, if we apply these definitions consistently, we conclude that the net mass flowrate at a node must be exactly zero. (What comes in must go out and vice versa. No change in stored mass of fluid in the node can happen ; "conservation of mass" governs all. And since liquid is nearly incompressible, we can use volumetric flowrates instead of mass flow without tangible error.

So, using the letters **A** for primary loop flowrate, **Z** for secondary, and **Q** for bypass **per Figure 10-1** we obtain:

- **@ Supply node, A – Z ± Q = 0**
- **@ Return Node, Z – A ± Q = 0**
- If A ≡ Z then Q ≡ 0
- **If A > Z then direction of Q Is from supply to return.**
- **If Z > A then direction of Q is from return to supply.**

## Open-system Decoupler Design

1. In an **open system,** the decoupler flowpath is the **gravity overflow pipe or chute** which connects the top of the **hot (return) tank** and the top of the **cold (supply) tank.** This is

explained in great detail in Chapter 7 **Chiller Primary-Secondary Loops.**

a. In an open system, referred to in Figure 10-1, the supply node physically is the cold tank of Chapter 7.
    (1) The section of piping from conditioning machine to supply node in open systems is physically the discharge pipe which runs from the chiller evaporator chilled water outlet to the top inlet of the Cold Tank.
    (2) The section of piping from the supply node to the secondary pump suction in open systems is the pipe connecting the cold tank bottom outlet nozzle with the suction inlet flange of the secondary pump.
    (3) The gravity overflow pipe or open channel chute connecting the high water elevation points of the open system cold tank and hot tank fulfills the function of the bypass connecting the two nodes of Figure 10-1.
b. In an open system, referred to in Figure 10-1, the return node physically is the hot tank of Chapter 7.
    (1) The section of piping from load devices to return node in the open system is physically the collected return main from terminal units to the top inlet of the hot tank.
    (2) The section of piping from the return node to the primary pump suction in open systems is the pipe connecting the Hot Tank bottom outlet nozzle with the suction inlet flange of the primary pump.

**For the open system gravity overflow chute or pipe to work properly, there are some very important guidelines to observe**:

First, the overflow pipe or chute must be designed for a level horizontal installation; there is zero pipe or open channel invert slope gradient to work with. Why? Because the bypass flow must be able to go in either direction with equal ease. This means that the elevation head available for causing the bypass flow to occur, by gravity alone, must be created by allowing the water to "pile up" in the top of the tank which is tending to overflow. So the overflow pipe-sizing vs. flowrate calculation is made in the same way you would design open-system open-channel flow over a weir. The overflow pipe or chute is actually just a horizontally extended plain stub-ended nozzle with "*weir-type*" discharge, from either end as required. The liquid level builds up deeper in the overflowing tank and channel in direct proportion to the increase in bypass flowrate. You can find the calculation procedure in your elementary fluid mechanics reference text for the exact cross-sectional shape design you utilize for the channel's construction.

The overflow is the intended bypass stream which occurs due to the designed imbalance of flows inside the primary and secondary loops. Now, if a very deep pile-up occurs in the overflowed tank (i.e., if the weir channel is made too small in cross-section) a deficit in normal tank level will occur in the other. The result? The inlet net positive suction head (NPSH) Available to both pumps will be different from normal. This means that the pumps will be operating outside their normal setup conditions; the pump fed by the overflowing tank will pump too much, and the pump receiving its suction from the underflowing tank will pump too little, if no other changes occur. A standard pump hydraulic analysis utilizing Bernoulli's Equation with an overlay of system resistance head loss curve on the pump total dynamic head (TDH) curve illustrates this phenomenon. Thus, the net result of undersizing the overflow pipe or channel is that the normal flowrate in the primary loop will no longer be constant but will be affected, changed proportionally *(hydraulicaly coupled)* to changes of flowrate in the secondary loop. This is exactly what we do NOT want to happen! It would defeat the whole purpose of the system, which would eventually go out of control.

**So the overflow channel pipe size or flow cross-sectional area must be *big!*** How big? Make it big enough to carry the entire flowrate of the primary loop, or the 100%-of-maximum peak design flowrate of the entire secondary loop, whichever is larger, with no more than a 6 or 8-in-water depth in the channel.

**Also, and in addition to this 6- or 8-in maximum depth requirement, if your overflow channel is to be a piece of round pipe, size the pipe such that it cannot ever flow more than about 70% full. This to avoid wave formation, slug flow, and resulting dangerous dynamic resonance effects that might physically wreck the system.**

(Because of these two joint requirements, you will probably find that several large pipes in parallel would be necessary for the bypass, and that the physical diameters of the hot and cold tanks might have to be increased beyond initial planned size, just to fit the overflow pipes in! ***This is definitely a drawback of the open system, probably the ugliest practical feature of what is otherwise a beautiful hydraulic system.*** However, instead of tank side nozzles and round pipes for the overflow path, you can substitute a flat-bottomed, square-sided open channel being only, say, 8-in deep on the sides, for a maximum 6-in flow depths. The cross-sectional width can then equal the diameter of the tanks. ***This is a drawback too, because the structure of a wide flat channel must be built strong enough to remain rigid, without detectable deflection in bending regardless of water depth flowing through it.*** This requires some additional engineering as well as construction cost to guarantee, but may be a preferable choice to big fat tanks with multiple overflows!)

In smaller industrial point-of-use hydraulic loop systems, the loop maximum flowrate is not very large in the first place, and the size of the required overflow channel is not a problem. For example, if the loop demand is no more than 20 or 30 gpm, you might end up with a pair of 500- or 600-gallon tanks connected by a single 6- or 8-in diameter overflow bypass pipe. Not an ugly deal, especially. No real problem. But if the flow is really big, then the whole open system thing gets pretty unwieldy! That is why large capacity central systems are rarely built on the open system model and the tankless closed system is used instead.

I think the biggest *open hot tank – cold tank* system I ever engineered was a retrofit job which involved a 2,000 ton industrial chilled water system; the resulting 4,800 gpm loops called for huge tankage, and the tanks had to go up on top of an existing steel roof which was none too bountiful in remaining excess strength capacity! Once the structural problems with roof steel, tank configuration, and overflow chute design were overcome, the system worked fine, but it was **big,** and **ugly,** with ***no place to hide!*** As you drive past that plant entrance today, that huge old tank system just ***looms*** like a pair of big steel clouds with a drawbridge between them, crouching over the roof on stubby column legs, *looking like hell to come*! If I had it to do over again, I would try to convince the owner to go with a closed system instead of the open-tank type. That's for sure!

## Closed-System Decoupler Design

In a **closed system,** there are no tanks. The decoupler flowpath is no more than a pipe that cross-connects the header pipes exactly as indicated in Figure 10-2. And the nodes are nothing more than

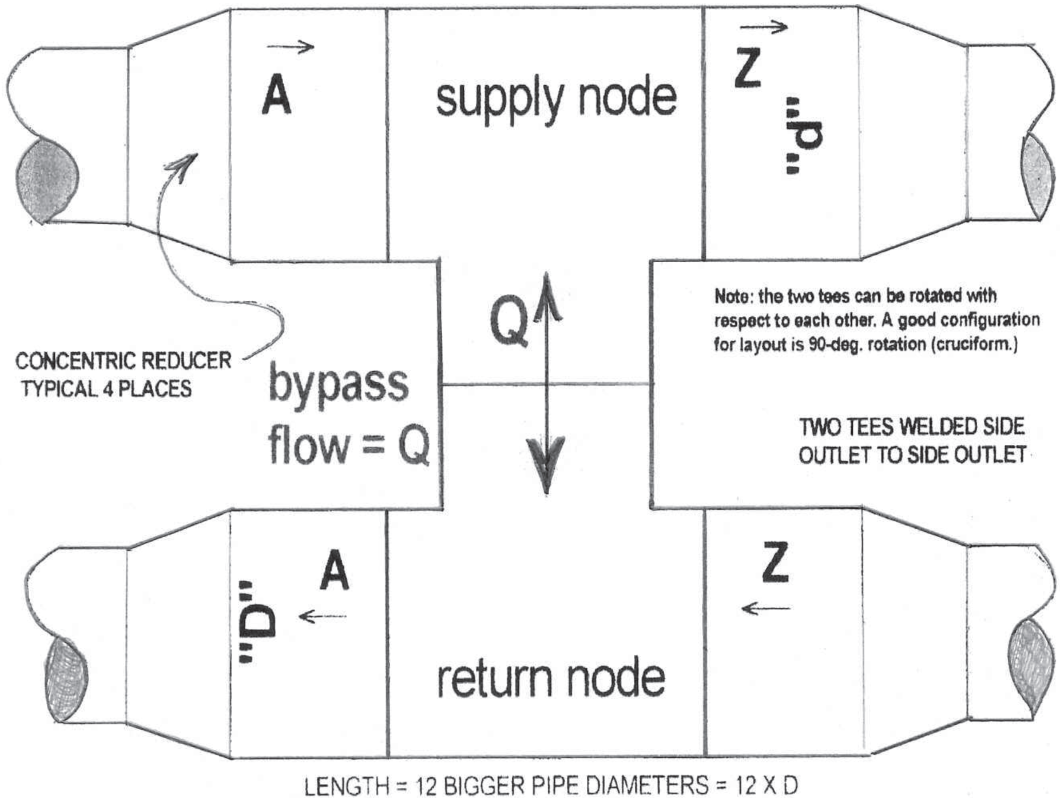



**FIGURE 10-2 DECOUPLING THE CLOSED SYSTEM COMPARE TO FIGURE 10-1**

pipe tees. The whole system flows full all the time, and the only things with air in them are the hydronic system expansion tanks.

Even though this is much simpler to design than the open system, we still have to find a way to hydraulically decouple the primary loop from the secondary. The simplest, cheapest, and most effective way to set up a bypass which will reliably do that is as follows. **Please refer to the illustration in Figure 10-2.**

Decoupling will occur in the arrangement shown in Figure 10-2, if the velocity of flow in the region of the two piping loop's nodal tees, and in the bypass itself, is low enough to trivialize the head losses in the portion of piping depicted in Figure 10-2. <u>And by that I mean if the pressure drop is low enough in the bypass arrangement to be *insignificant*</u> to either of the system pumps, then any ***change*** in that pressure drop due to changes in loop flowrate ***will have no coupling effect.*** As far as the two loops can tell, the pressure in the bypass connector always stays about the same, *regardless* of the percentage of bypass flow, 0–100%.

Let's use some representative numbers to get a better feeling for this relationship.

## Illustration of Decoupler Pipe Sizing

It makes no difference to the sizing procedure in which direction the bypass flow occurs.

First we will find the head loss in the decoupler bypass path for a minimal bypass flowrate "Q", and later we will find it for a maximum bypass "Q". Then we will compare the two figures in light of the primary loop pump head curve and the primary loop piping system head loss, to see if hydraulic decoupling is accomplished or not.

## Minimal Value of "Q"

Let's assume a primary loop flowrate <u>"A" = 2,400 gpm</u> and <u>"Z" = 2,200 gpm</u> through the secondary loop. This would be typical for a 1,000 ton water chiller as the conditioning machine, and a bunch of cooling coils at near-peak demand drawing a total of 2,200 gpm. (*Note:* chilled water **$\Delta$T = 10°F** is a typical value across the chiller evaporator, and we like to have just a little more capacity in the primary loop, both in flowrate and in conditioning, than the peak demand of the secondary loop.

So if the user load **$\Delta$T** were also $\simeq$ **10°F** and the secondary loop flow demand = 2,200 gpm, the total instantaneous load cooling demand would be:

demand = (500 × 2,200 gpm × 10°F)/12,000 = <u>917 tons</u> (91.7% of chiller capacity.)

In this case the bypass flowrate <u>"Q" = (2,400 − 2,200) = 200 gpm,</u> **from** the supply node **to** the return node. This would be about the minimum bypass flowrate the system would normally see. Now look once more at Figures 10-1 and 10-2; *we shall see how this minimal bypass flowrate affects flow in the primary loop:*

In sizing the primary and secondary loop headers, we apply the usual criteria for maximum liquid velocity in large pipe sizes:

$\mathbf{V_{max}} \leqq 7.0$ ft/sec, with a head loss $HL_{100ft\ of\ pipe}$ in old pipe $\leqq$ 10.0 ft of water (@ old pipe roughness, C = 100)

**So we make main supply and return header diameters "d" = 12-in size.**

In sizing the two loop pumps, my own practice is to place the dynamic head loss of the following flowpath segment upon the primary pump:

***On Figure 10-2 let the primary pump furnish the dynamic head loss of the flowpath flow "A" to Supply Node tee, flow "Q" out through side of tee, Q thru Bypass, Q into side of return node tee, and flowrate A out the return node tee outlet and back toward the primary pump suction.***

The approximate summation of elemental head losses through that path, using the Crane Technical Paper no. 410 techniques of Chapter 8, and assuming "D" is only one pipe size larger than "d", namely, 14-in size, is as follows:

## Bypass Segment Piping Head Loss on Primary Pump ≃

$$\simeq \Sigma\, \mathbf{HL_i}\ i = a \text{ through } i = f, \text{ where}$$

**a.** Concentric increase from d = 12 in to D = 14 in @ flow = A; (from primary pump to supply node)
**b.** Friction loss thru pipe run "six D" long @ flow = A;
**c.** Out thru side of supply node tee @ flow = Q (this is the bypass stream);
**d.** Reenter thru side of return node tee @ flow = Q;
**e.** Friction loss thru pipe run "6 D" long @ flow = A;
**f.** Concentric decrease from D = 14 in to d = 12 in at flow = A (heading back to primary pump suction)

## Segment (a) Concentric Increase:

*(a)* $\mathbf{K = 2.6 \sin (\upsilon/2)\ (1 - \beta^2)^2/\beta^4}$;
the reducer sidewall half angle $(\upsilon/2)$ is approximately arctan [(1/2)(14 in – 12 in)/(13 in)] ≃ arctan (0.077) ≃ 4.4 degrees;
Diameter reduction ratio beta is:

$$\beta = (12\text{ in}/14\text{ in}) = 0.857$$

$$\beta^2 = 0.735$$

$$\beta^4 = 0.539$$

Loss coefficient K by the Crane formula is
$\mathbf{K_{(a)}}$ = [2.6 sin (4.4 deg.)] [(1 – 0.735)$^2$] ÷ 0.539 =
= (2.6) (0.07672) (0.07023)/0.539 =
= 0.026;
The velocity of 2,400 gpm in the 14 in pipe is 5.578 ft/sec and the velocity head VH =
=$V^2/2g$ = (5.578)$^2$(2)(32.2) = 0.483 ft water; So the head loss in segment (a), the concentric increase from d to D =
= $\mathbf{K_{(a)}}$ × VH =
= (0.026)(0.483) = ***0.013*** ft water column.

## *Segment (b) Run length six x D:*

*(b)* Head loss of 2,400 gpm of water thru a run of length 6 × 14 in = 84 in of 14 in pipe in old rough condition = (84 in/12) ft of pipe × (1.305/100) ft of water loss per hundred feet of pipe run = ***0.091*** ft of water column;

## *Segment (c) Out side of supply node tee:*

*(c)* $\mathbf{K = 60\ f_T}$; for 14-in. steel pipe, Crane Technical Paper No. 410 lists the fully developed turbulent friction factor $\mathbf{f_T}$ = 0.013; so the loss coefficient out side of tee = (60)(0.013) = **0.78;** The velocity of 200 gpm in the 14 in pipe is 0.467 ft/sec and the velocity head V H =
= $V^2/2g$ =
= (0.467)$^2$/(2)(32.2) = 0.003 ft water;
So the head loss in segment (c), the flow out the side of the supply node tee =
$\mathbf{K_{(c.)}}$ × V.H. =
= (0.78)(0.003) = ***0.002*** ft water column.

## *Segment (d) In side of Return Node Tee*

*(d)* by usual Crane 410-type procedures, this is identically same as (c.), $\mathbf{K = 60\ f_T}$; remember, we are working with approximate, order-of-magnitude precision when we use these empirical loss coefficients.

So the head loss in segment (d), the flow in the side of the return node tee = $\mathbf{K_{(c)}}$ × VH = (0.78)(0.003) = ***0.002*** ft water column.

## *Segment (e) Run length six x D:*

*(e)* this is identically same as (b), so head loss = ***0.091*** ft of water column.

## *Segment (f) Concentric Decrease:*

*(f)* $\mathbf{K = 0.8 \sin (\upsilon/2)(1 - \beta^2)}\ /\beta^4$; as in segment (a), the reducer sidewall half angle $(\upsilon/2)$ is approximately 4.4 degrees; likewise, diameter reduction ratio BETA is:

$$\beta = (12\text{ in}/14\text{ in}) = 0.857$$

$$\beta^2 = 0.735$$

$$\beta^4 = 0.539$$

Loss coefficient K by the Crane formula is
$\mathbf{K_{(f.)}}$ = [0.8 sin(4.4 deg)][(1 – 0.735)] ÷ 0.539 =
= (0.8)(.07672) (0.265)/0.539 = 0.030;
as in segment (a.) the velocity head VH =
=$V^2/2g$ = (5.578)$^2$/(2)(32.2)= 0.483 ft water; So the head loss in segment (f), the concentric decrease from D to d =
= $\mathbf{K_{(f)}}$ × VH =
= (0.030)(0.483) = ***0.015*** ft water column.
Adding them up we obtain the bypass segment piping head loss carried by the primary pump, ≃ Σ $\mathbf{HL_i}$, i = a through i = f

$$\simeq \{0.013 + 0.091 + 0.002 + 0.002 + 0.091 + 0.015\}$$

≃ **0.214 ft water column**

**(0.214 ft × 12 in/ft = 2.5 in of water)**

## Maximum Value of "Q"

Maximum bypass occurs when the secondary loop flow is at its minimum, while primary loop flow remains, as ever, constant for the conditioning machine's optimum performance.

Since our example involves an HVAC application, and chilled water is the fluid, we can safely assume that the minimum load demand will be only about ***20%*** of peak capacity, which we said for our example would be 2200 gpm. So the bypass flowrate "Q" = (2,400) – (0.20)(2,200) = 1,960 gpm, still **from** the supply node **to** the return node in its direction.

Now that ***"20%"*** is just a good ballpark guess, not a magic number by any stretch of the imagination. So-o-o-o, being the typical engineer, shrewd but lazy, I will round up this odd value for "Q" (1,960 gpm) to a ***nice, round 2,000 gpm!!!*** for which the values of velocity, velocity head, and friction loss per hundred feet of pipe are all tabulated in my and your Cameron's Hydraulic Tables. This saves my having to do a bunch of calculations and interpolations, which I am sure you appreciate. And the conclusions of our analysis will be the same as if we had used the nastier number 1,960. So here we go again, with **Q = 2,000 gpm !!**

***Segments (a) + (f), Concentric Increase and Decrease of Pipe Size:***

**(a)** and **(f)** both remain the same as before, because nothing has changed. These losses are based on the velocity head for flowrate "A", which remains 2,400 gpm, and on the fixed geometry of the 14 in × 12 in concentric reducer for loss coefficients. So the loss for the increaser segment remains = **0.013** ft water column, and **0.015** ft for the reducer, total 0.013 + 0.015 = **0.028** ft water column.

***Segments (b) + (e), two times run length of (6 × D) ft:***

**(b),(e)** likewise are unchanged, so their combined value remains 0.091 + 0.091 = **0.182** ft of water column.

***Segments (c) + (d), 2 times Side of tee:***

**Aha! These two *do*** change!

***(c)* K = 60 $f_T$** as before, and since $f_T$ is just a function of pipe size, which has not changed, K remains = (60) (0.013) = **0.78;**

But Q is now 2,000 gpm, and we find the velocity head for 2,000 gpm in a standard weight 14-in steel pipe, according to our trusty Cameron's data, is **0.34** ft of water. So for the two side of tee flows, our head loss = 2 × 0.78 × 0.34 ft = **0.530 ft** of water. So easy!

Adding them up for the max bypass case, we obtain the bypass segment piping head loss carried by the primary pump ≃

$\simeq$ **{ 0.028 + 0.182 + 0.530 }**

$\simeq$ **0.740 ft. water column**

**(0.740 ft × 12 in ft = 8.9 in of water.)**

**The change in bypass flowrate then has a coupling effect of (0.740 – 0.214) =0.526 ft = *0.53 ft of water*** (this is the maximum possible coupling interaction, the actual is less as we will see.)

## Analyze Effect of Changing "Q" on Primary Pump Operation: Are the Two Loops Decoupled?

Now we have to apply some judgment. We have determined that during a season of operation the primary loop pump, if somehow it could continue to deliver its design flow of 2,400 gpm constantly, regardless of changing flow conditions in the bypass leg, would suffer a change in its total dynamic head equal to (0.740 – 0.214) = 0.526 ft water column. Of course the pump head cannot change without a change in flow unless the pump rpm or the impeller diameter are modified accordingly. But our system is set up without speed control, and there is no automated throttle valve in the primary loop which could respond to bypass changes and force the loop flow to remain constant. On the contrary; our goal is to make a simple, stable stand-alone primary loop without automated controls, which will keep constant gpm without coupling effects.

***So the question is, "Does a system head loss change of as much as 0.53 ft of water column cause too large a change in primary pump delivery flowrate?"*** If it does, then our loops are "coupled", and if not, they are considered successfully decoupled.

(It will be clear that practical design restrictions will prevent our ever building a totally 100% decoupled loop arrangement, but 100% decoupling is overkill. I contend that a primary loop flow change of around 1% or 2%, or thereabouts, will never be noticed in the field in the vast majority of applications. If a primary loop variation of less than that affects the process, then a more complicated system—an active automated primary loop flow control subsystem—would be required. This does not happen in facility HVAC and plant utility applications.)

## Analytical Procedure, Pump Coupling

Please refer to *both* versions of **Figure 10-3** at this time. One version is a reproduced overlay of the system curves on an actual pump curve (Bell & Gossett size 10 × 12 × 12M HSC-series horizontal split case centrifugal pump.) The other version is a CAD sketch I made, to blow up the intersections region of the system and pump curves so that it can be seen and understood more easily. When I refer to "Figure 10-3," I mean both versions taken together.

The procedure is:

- Select a primary loop pump for maximum efficiency. Our example runs at 85%.
- Reproduce a copy of the catalog pump curve and use it as the basis of Figure 10-3.
- Mark the selection point on the sheet. For our example, I have assumed that the primary loop system head loss, **when the secondary loop flow is at its maximum, and the bypass flow is at its minimum (Q = 200 gpm), at A = 2,400 gpm is 31 ft of water,** about right for a chiller application. In a real application of course you must make the actual loop piping head loss calculation and use the actual loss in lieu of the 31 ft assumed here. *(For the loop system curve calculation and plotting, use the Cυ of the loop manual throttle valve wide open, then add a realistic amount of throttle loss on top of the system curve to get the operating loop head loss.)* The pump impeller selected must be that impeller whose curve runs through the intersection point of design gpm (2,400) and operating loop head loss (assumed 31 ft here, which includes our throttle allowance.)
- Calculate points for, and plot, the system curve points on top of the pump curve grid. Use the scaling laws:

  1. velocity proportional to liquid flowrate in the given pipe size;
  2. flowrate ratio therefore equal to velocity ratio; ($GPM_1$/$GPM_2$) = ($V_1$/$V_2$).
  3. piping system head loss (use ***HL*** for abbreviation) is directly proportional to the velocity head, i.e., to the term ($V^2$).
  4. ratio of change in pipe head loss HL for a change in flowrate GPM is therefore proportional to the ratio of change in velocity squared = proportional to ratio of change of flowrate squared.

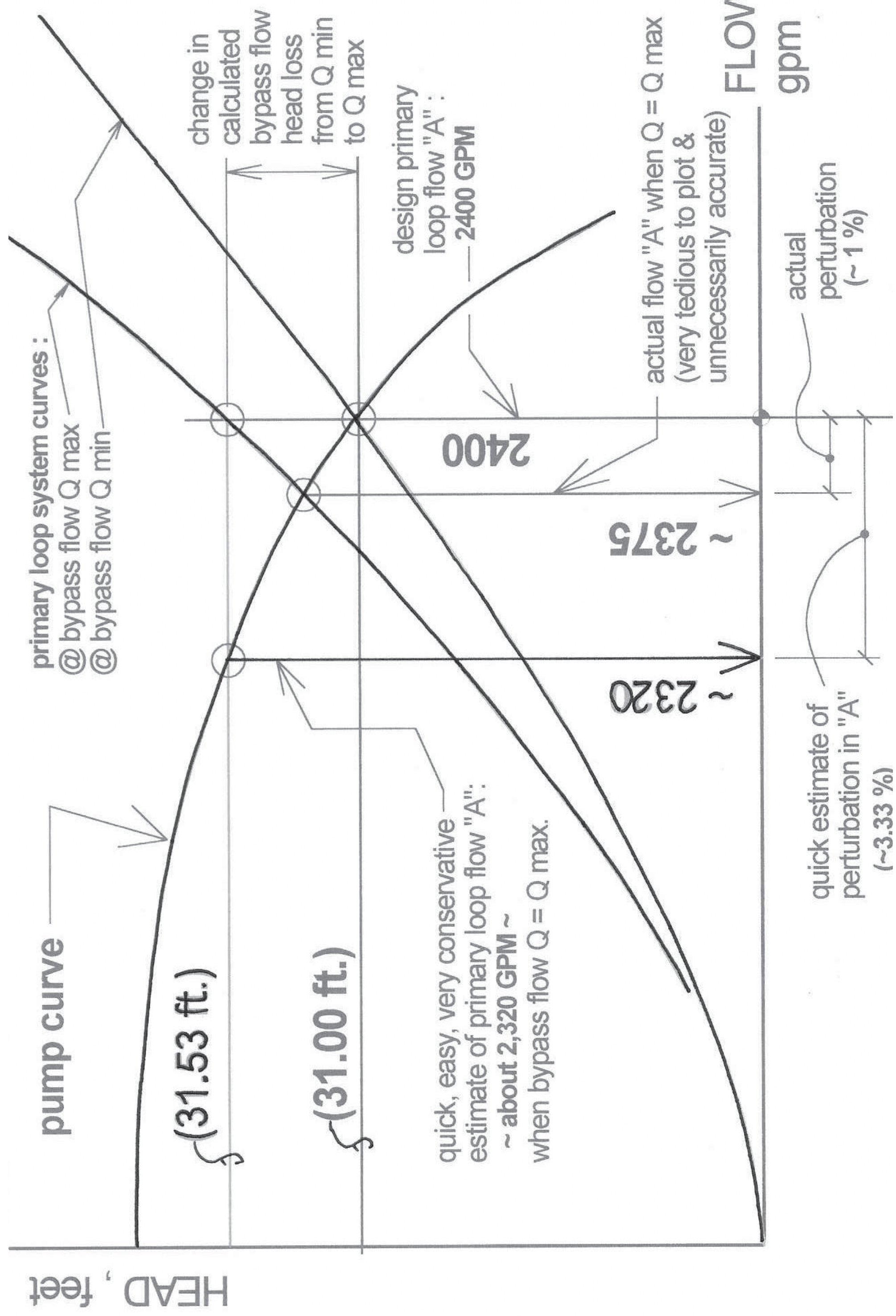



**FIGURE 10-3A DECOUPLING THE CLOSED SYSTEM: VERIFICATION BY PUMP CURVE – SYSTEM CURVE OVERLAY FOR MAX AND MIN BYPASS FLOW MODES**

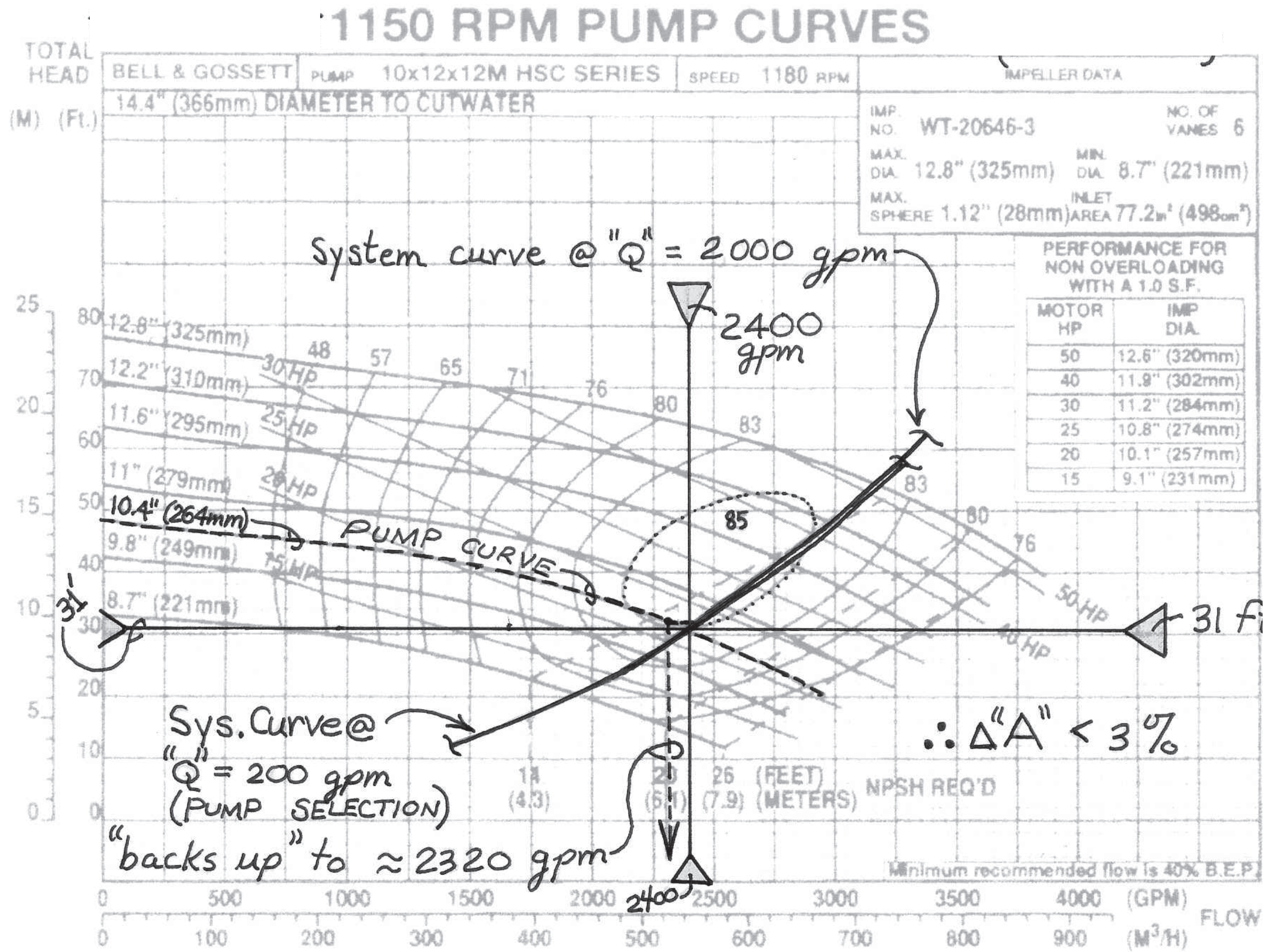



**FIGURE 10-3 1150 RPM PUMP CURVES**

Therefore:

$$\mathbf{HL_2} = HL_1 \times (GPM_2/\ GPM_1)^2$$

Some points on the system HL curve are calculated as follows:

*HL @ gpm of 0 = 0*
*HL @ gpm of 2,400 = 31.0 ft. (loop calculations)*
*HL @ gpm of 1,500 =* $(1{,}500/2{,}400)^2 \times 31.0 =$
= 12.1 ft
*HL @ gpm of 2,000 =* $(2{,}000/2{,}400)^2 \times 31.0 =$
= 21.5 ft
*HL @ gpm of 3,000 =* $(3{,}000/2{,}400)^2 \times 31.0 =$
= 48.4 ft., etc.

- From the multiple intersection point of design gpm, operating system head loss curve and selected pump impeller curve, go vertically upward and place a point on the design gpm line which intersects a head value equal to the design head plus the amount of perturbation caused by the variation in bypass flow (which in our example was found to equal about 0.53 ft of water column). Therefore we place the point 2,400 gpm @ (31.00+0.53) = 31.53 ft on our plot.
- From 2,400 gpm @ 31.53 ft, draw horizontally leftward until you intersect the pump curve. Read the flow there; in our example, Figure 10-3, it is about 2,320 gpm. This is quick, easy and very conservative. It places an upper bound on the coupling effect of not more than

**(2,400 – 2,320) × 100%/2,400 = 3.33%**

- That is to say, the effect of a full range of variation in secondary loop flow will cause a variation of no more than 3.33% in primary flowrate "A". But remember, that is just a quick conservative estimate. What actually happens is this: when the bypass flow "Q" climbs from initial minimum 200 gpm to final maximum 2,000 gpm, the primary loop system curve undergoes a change : an increase. The primary loop curve for "bypass flow Q max (plotted in green in Figure 10-3) actually passes through the point "2,400 @ 31.53". This makes it lie above and to the left of the design curve plotted in red, the Q min curve. Notice that because the pump curve slopes downward as we move to the right, the green Q max curve and pump curve actually intersect to the **right** of the quickly-estimated 2,320 gpm flowrate! This would be the actual operating point of our simple system at Qmax, and the flow "A" would be considerably higher than the conservative 2,320 gpm estimate. I would guess that the actual coupling effect in our

example is on the order of 1%. This is certainly a weak coupling at its very worst, and not enough to be concerned about.
- Therefore conclude that the two loops are successfully decoupled when the piping arrangement of Figure 10-2 used, with the reducers and enlarged connection tees being one pipe size larger ("D") than the properly sized primary loop header ("d").

## Closing Comments:

- *Should we make the connecting tees and reducers two pipe sizes larger than the loop header, for even more decoupling?* No, it would be uneconomic overkill, unless the loop pipe was undersized to start with, which is a big no-no.
- *Why not just use line size all the way, and save the money of increasing at the bypass?* Well maybe you can do that, on a specific case-by-case basis. Do the calculations and plotting for yourself and see. However, and this is just my personal opinion based on observing a bunch of real systems in operation, you run the risk of getting more than the calculated coupling when the velocity of bypass flow is too high. Flow in and out the side of tees is not very stable, and the instability can be disruptive if velocity is too high. Our back-to-back tees arrangement (Figure 10-2) would tend to exacerbate the instability, because the vortices shed by the first tee can enter the second, driving the head loss higher than calculated as 2x a single tee outlet. Crane's loss coefficients are based on uniform straight flow upstream and downstream of single standard components in isolation. Two or more fittings back to back can cause more loss than just the sum of individual calculated losses per fitting.

This exercise also shows why a centrifugal pump which exhibits a good bit of "droop" in its flow vs. head curve tends to be stable for control purposes, while a pump with a nearly flat curve may be too sensitive to throttle changes for some applications, and may border on being unstable when used that way.

For stability, a small perturbation in system resistance should not cause a large change in pump flowrate. A nice linear proportional change in gpm for a given head change is what we usually seek, and is what makes accurate throttle valve control possible.

Now for a closely related phenomenon which will probably test you at some point in your mechanical engineering career: the use of multiple centrifugal pumps in a parallel network of piping, in lieu of the correctly designed primary loop/secondary loop with decoupling.

Before the separate loops concept was put into wide use, it was common to hook a pair of pumps in parallel, between a pair of headers (one suction header and one discharge) and then to pipe the discharge header to the chiller evaporator. After leaving the chiller, the chilled water supply header ran to the load it served, split up into branches for multiple cooling unit coils, recombined into a single return header and terminated back at the pump suction header, thus completing the whole circuit in one giant loop.

The intent of having two pumps installed was for one to be active, with the other turned off but immediately ready if needed in case of failure of the first pump. In other words, at all times one pump would serve as a standby unit, and the other would supply 100% of the system flowrate.

Then two things would typically occur.

First, one or two terminal units would not provide enough cooling, due in fact to poor hydraulic system flow balance, but blamed upon the chiller or pump or both. "Not enough chilled water" was the oft-heard lament. The engineer's reply was to run the standby pump simultaneously with the active pump. This increased system flow by about one third, and usually brought a bit of improvement, albeit due to brute force.

Second, later on, the conditioned space in the building increased in area and volume, picking up more heat from outdoors, and inherited additional lighting, people and other sources of internal cooling load. You know the result, of course.

Now the plant engineer faced a dilemma. New chillers are very expensive, and installing one is a big deal. On the other hand, installing a third and fourth pump, parallel to the original pair, is not so hard and not nearly as expensive, especially if one or two "spare pumps of about the right size" (figured in motor horsepower, not delivery and head) just happened to be available in the plant somewhere. So he installed the odd pair by extending (not enlarging) the header pipes, and now had four unmatched pumps in parallel between two undersized headers. Of course, the results were not good.

Not very good at all. In fact, there not only was no improvement, things got worse! The complaints doubled, and tripled. The engineer got fired.

His replacement won approval to add a second chiller, which he did. He added it by extending the supply manifold. He also added two new pumps to go with the new chiller (now there were six pumps in parallel, because he added the new pumps in the same way as the new chiller: he simply extended the headers.) Now completely bottlenecked hydraulically, the chilled water system had passed the point of no return; no one would dare suggest tearing it all down and starting over with separate circuits and adequately sized pipe! No no no! No, the lousy system just lay there, getting worse with time, the evil inheritance of the seemingly endless stream of plant engineers and maintenance supervisors and replacement plant managers who drifted through the plant as the years rolled on. Once in a while, another chiller and a few more pumps might be added, but they just never seemed to work out right.

In fact, when three or more "identical" centrifugal pumps are piped in parallel, their suctions and discharges being tee'd into a single common suction header manifold and a single common discharge header manifold, and run simultaneously, the result seen by the hydraulic system is a combined pump delivery which has a nearly flat horizontal head vs. flow characteristic curve. The system behaves as if there were only one pump, having a flat delivery "curve", that is, a straight horizontal line.

It has to be this way. If each pump delivers 100 gpm at 90 ft of total head, then the three pumps create a single pump curve having a flow of 300 gpm at 90 ft head. If each delivers 150 gpm at 89 ft, then the second point on the combined curve is … 450 gpm at 89 ft of head. One foot of system friction loss head changes the system delivery by 150 gpm, which is a change of (450 – 300)/300 or 50% flow decrease for a (90 – 89)/90 or 1.11% increase in head!!!!! Does that seem like a good controls deal to you???? The multipump head value is essentially constant and equal to the zero gpm shutoff head for any one of three or more identical parallel pumps; regardless of gpm, the system head remains constant. (Actually, the ***correct interpretation of the setup is that the combined flow through the parallel pump arrangement is indeterminate and cannot be either calculated, predicted, or controlled.)***

It never ceases to amaze me that so many big existing central hydraulic systems—chilled water and cooling tower water—are built in just this way. I have personally witnessed closed single-loop chilled water systems in plants with as many as eight active centrifugal pumps in the 100 horsepower class hooked up in parallel between a pair of header manifolds no larger than 14-in pipe size! If you ever have the misfortune of being assigned to debottleneck such a screwed-up system, you will find out just what I mean. Good luck to you brother! It cannot be done! One has to scrap the existing system and build an entire new one, with independent primary-secondary loop piping for each pump-chiller circuit, which is ***EXTREMELY COSTLY. No engineer I am aware of has ever had the guts to approach the plant manager with a straight-faced proposal to do just that! Hell, no! It would be job suicide!***

In fact, as we saw earlier, the usual engineer's solution to the managers' complaint of "inadequate chilled water" is to add yet another chiller and another couple of pumps in parallel with the malfunctioning mob of paralleled pumps and chillers which already exists. By the time the managers find out that the new, bigger system has even more operating problems and less capacity than the previous version, the engineer will have served out his tour of duty in plant maintenance/utilities and will have moved on to another staff assignment (or another job altogether!) ***And the lousy system, improperly designed in the first place, just gets lousier with time.***

Oh, well. This situation makes fees for consultants like me. Silver lining in every cloud.

And, oh yes, by the way: those parallel pump setups: how often do you imagine that any two "matching identical" pumps have exactly the same flow-head delivery characteristics, just as they come straight from the factory? Not very often! To make them truly equal, careful custom machining of the impellers would have to be done.

So when multiple pumps are set up in parallel, any pump having a sufficiently lower shutoff head than the average will not be able to force itself on line. Its impeller will spin without producing flow, pressure pulsations may be produced, water-hammering the system, and self-destruction of the "weak" pump will result sooner or later. The more pumps there are in parallel, the worse the problem becomes.

The correct engineering solution is to separate the gang of flow circuits by having each pump draw from a big, low-velocity suction manifold, and discharge into its own pipe. In other words, to be DECOUPLED from the others. And the way to do that in an existing closed system is to make the suction (return) manifold a couple of pipe sizes larger than "normal," and to NOT tie the pump discharges back into a single discharge (supply) manifold, but to run new piping for each supply circuit independently to its own chiller.

***Figure 10-3b illustrates the foregoing ideas in extremely simplified form.***

- Now that the basic loop concepts have been defined and illustrated, it is time to get on with the system components and their control.

## BASIC COMPONENTS AND CONTROL OF PRIMARY LOOPS

**Please refer to Figure 10-4.** It is a schematic of a complete primary loop in a closed system, just like the one previously illustrated. We will stick with the water chiller example, since it is perfect for illustration and is familiar to most plant engineers, facility/utility designers, and consulting engineers. It reduces the primary loop idea to its simplest form, which is the best form.

### Primary Loop Basic Components

The basic primary loop components of a 200-ton 480-gpm single-chiller single-pump system, as numbered on Figure 10-4, are described in the following list. As an aid in component and pipe sizing, each component size for the schematic example is given in parentheses. Pertinent notes are included.

I recommend using black low-carbon steel piping for closed hydronic chilled and hot water and chilled glycol systems. I recommend that the ASME B31.3 Process Piping Code be used in large system construction (over 100 tons of chiller capacity). I have no direct knowledge of the B31.9 Code, however, and know of no one who uses it.

Joints at piping spool piece connections should be made per ASME B31.3 by butt welding, and ASME B16.5 and B16.9 flanges and fittings should be used. I prefer ASME B31.3 socket welding (using ASME B16.11 fittings) for the small stuff (1-1/2 in pipe size and smaller) as opposed to screwed joint fittings and couplings, because the screwed joints leak and you cannot seal weld the threaded malleable iron couplings.

Standard pipe wall weight would be appropriate for the illustrated system. ASTM A-106 Grade B is the best piping for this duty, followed by A-53 Grade B and then the same two in Grade A. Some owners use API 5-L for cost savings. It is a listed material and okay to use. But under no circumstances use any A-120 galvanized steel screw-joint piping or fittings in pressure-pipe duty. It is suitable for use as fence posts, but not as piping.

### Figure 10-4 Component Listing

1. **Return node**, as indicated on Figures 10-1 and 10-2. This is a standard welding tee ***(8 in pipe size.)*** The **bypass** is the short vertical connection joining the two tees (1) and (22).
2. **Inline air separator** ***(6 in pipe size).*** The line size for 480 gpm is 6 in. You should smell a rat if when sizing the air separator you find its nozzle sizes to be different from, especially *smaller* than, your hydronic loop piping line size. The separator needs low velocity to work correctly.
3. **Flexible connector** (6 in pipe size). ***Typical note for all items of this type:*** Specifically mfg. for this duty; flanged ends to match pipe flanges; flexible hose encased in protective stainless wire mesh wrap. Specify applicable pressure-temperature rating for worst case duty.
4. **Isolation valve** (6 in pipe size). ***Typical note for all items of this type:*** Also called a maintenance or block valve; either wide open or fully closed. We want the minimum pressure drop when wide open, and bubble-tight shutoff when closed. Gate valves are ideal, but are too costly in large pipe sizes. So we tend to use full-port ball valves in smaller sizes and butterfly valves in larger sizes. Lug-type butterflies are worth the extra money.
5. **Concentric reducer** (6 in × ?? pipe size). If item (6) has a 6-in pipe size inlet, this reducer is not required. If item {6} inlet is 4-in pipe size then this reducer is a 6 in × 4 in size. If item (6) inlet is smaller than 4-in pipe size, then a bad mistake in pipe sizing calculations or pump selection has been made somewhere along the line!
6. **Suction diffuser**. Typically, the correctly sized 1,750 rpm or 1,100 rpm centrifugal pump will have the same suction inlet size as your line pipe size (6 in in our example) and a pump discharge one size smaller. I would expect that a typical primary loop chiller pump for our example would be a

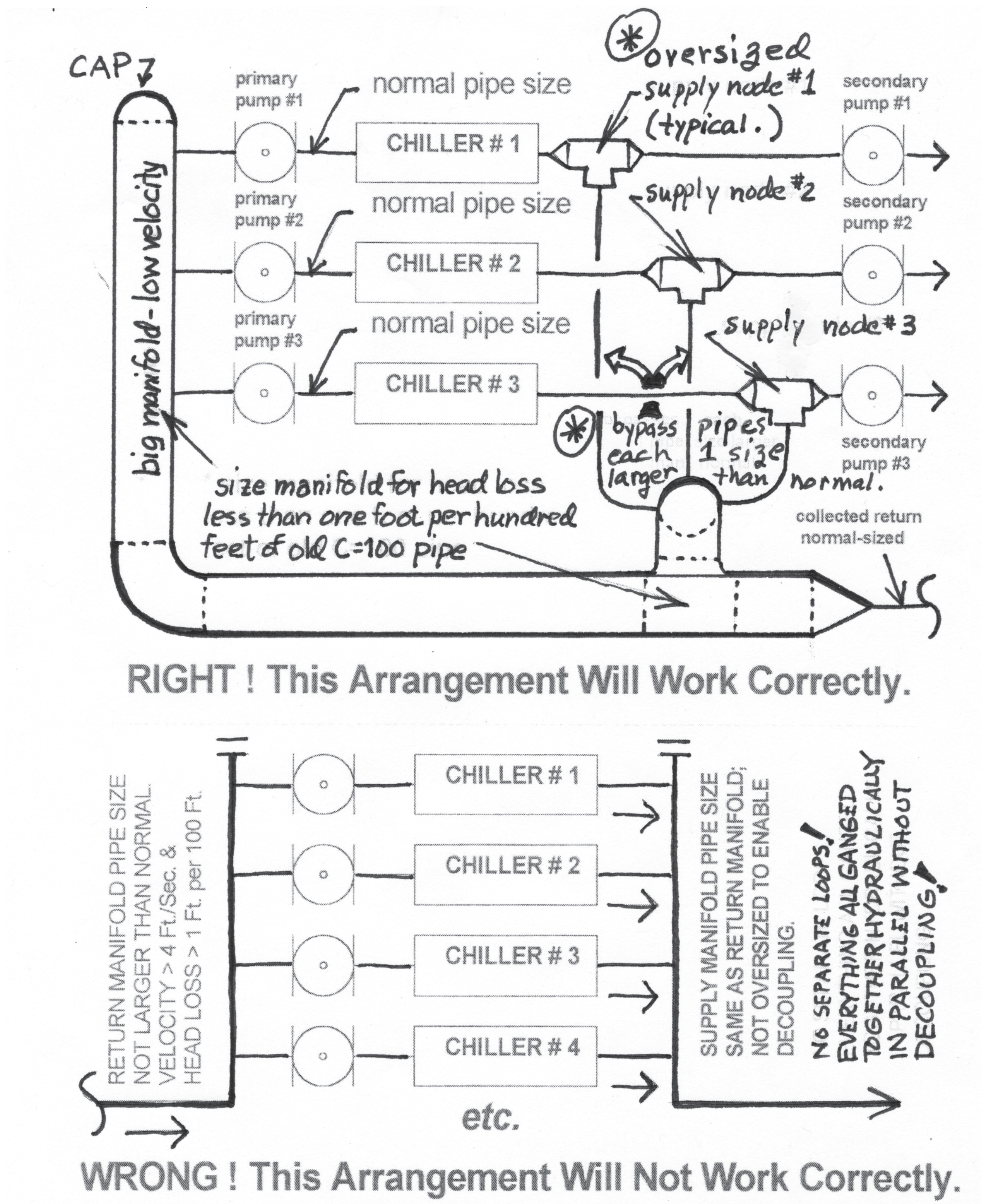



**FIGURE 10-3B PARALLEL PUMPS *RIGHT & WRONG***

6 in suction size by 4 in discharge size 1,750 rpm pump. In that case the suction diffuser, item (6), would be a 6 in inlet × 6 in outlet size, and the reducer item (5) would not be required. Most 3,500 rpm pumps will have smaller connection sizes for the same flowrate. In any case, the inlet of the suction diffuser matches the pipeline size, and the outlet of the suction diffuser matches the pump suction flange size. Suction diffusers having same-sized inlets & outlets, as well

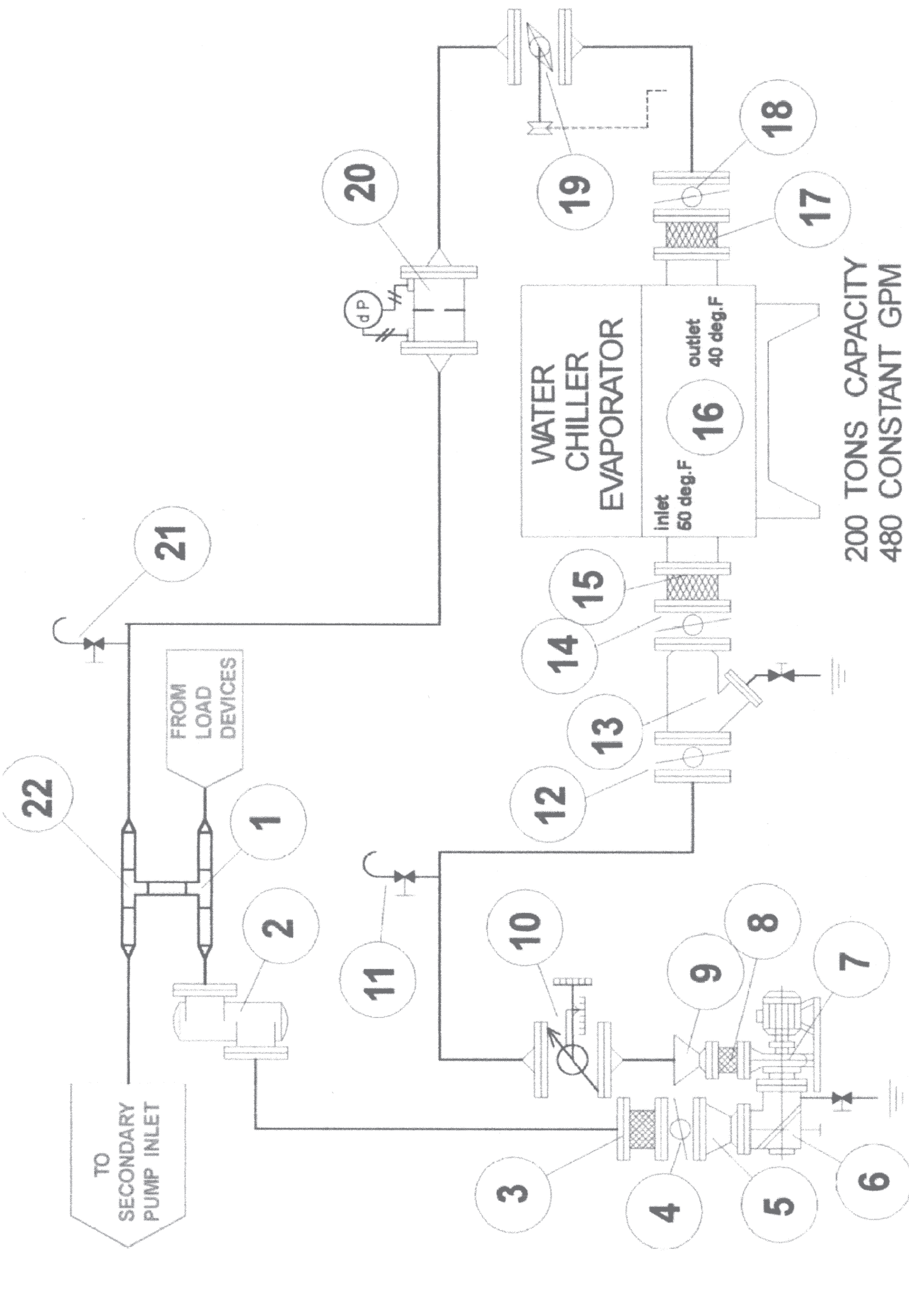



FIGURE 10-4 PRIMARY LOOP SCHEMATIC

as outlets one pipe size smaller than the inlet, are routinely available from the hydronic equipment manufacturers (Bell & Gossett, Taco, Armstrong, etc.).

The suction diffuser is a very important piece of equipment. It straightens out the flow into the pump inlet (rotation degrades pump performance). So it permits a compact piping arrangement while providing "clean hydrodynamics" into the pump inlet. It also has a large-mesh trash collector and bottom drain fitting. It may include a startup strainer with a fine mesh, but this must be removed after initial startup period.

7. **Primary loop pump** (probably a 6 in inlet flange × 4 in discharge flange pump size). And probably a 1,750 rpm base-mounted end-suction or horizontal split case type pump, for the flowrate assumed in our example. Really large flowrates may result in 1100 rpm pumps as the most efficient, and these would have line-sized connections. However, 3500 rpm pumps may have to be used in special circumstances (high head low flow). The 3,500 rpm pumps are typically less efficient, noisier, shorter-lived, and have small nozzles. A 3,500 rpm centrifugal sized to deliver 480 gpm might be a 4 in × 3 in pump, or even a 4 in × 2-1/2 in pump.
8. **Flexible connector** (4 in pipe size, i.e., one that matches the pump discharge).
9. **Concentric reducer** (6 in × 4 in pipe size).
10. **Triple duty valve** (6 in pipe size). A special-purpose device which could also be called a nonreturn valve. It takes the place of three separate valves. It contains a spring-loaded check valve feature to prevent reverse flow. It will manually shut off leak-tight against its seat. However, it is a globe-plug pattern valve, designed for throttling duty. Its manual stem operator is equipped with an infinitely adjustable screw thread and lock feature with a memory stop indicator, which is very useful in setting the valve in intermediate "throttled" position. Use it for "rough trim" of the primary loop flow, to get the pump approximately on its design flow point [See also item (19)].
11. **Manual air vent** (1/2 in pipe size.) A soft-seated 1/2 in bronze gate or globe valve is typical.
12. **Isolation Valve** (6 in pipe size.) The same as Item (4).
13. **Strainer with 1/4 in blowdown piped to drain** (6 in pipe size.) This strainer matches system temperature and pressure rating.
14. **Isolation valve** (6 in pipe size). The same as Item (4).
15. **Flexible connector** (6 in pipe size).
16. **Chiller evaporator** (6 in pipe size assumed; otherwise install reducers and flanges—or couplings if there are grooved joint connectors on the chiller—as required to match the chiller connection size).
17. **Flexible connector** (6 in pipe size).
18. **Isolation valve** (6 in pipe size). The same as item (4).
19. **Primary loop manual flowrate control valve** (6 in pipe size). Use, it for "final fine trim" of the primary loop flow, to obtain the exact design flowrate [see also Item (10)]. In smaller sizes use a globe plug control valve with published $C_V$ table. A threaded rising stem globe with regular valve twist handle is okay in smaller sizes. In larger sizes, globe valve cost may be prohibitive, and a quarter-turn ball or butterfly will have to be used instead. Select one with published $C_V$ table, worm gear operator, and chain fall or hand wheel actuator. (Necessary because a big quarter-turn valve under pressure can slam open, and in that event a simple rod handle can easily break a man's arm.) An operator lock is a good idea too, to maintain the valve setting once adjusted properly during the final balance.
20. **Flowmeter used with (19).** (Body size as required for the selected meter. For an orifice plate, the orifice flanges would be line-sized—6 in. If different from line-size, install between concentric reducers.) Do not fail to install adequate straight lengths of pipe upstream and downstream of the meter body to satisfy the manufacturer's requirements for obtaining best accuracy of the instrument. Do not fail to ensure calibration of the differential pressure readout.
21. **Manual air vent** (1/2 in pipe size). A soft-seated 1/2 in bronze gate or globe valve is typical.
22. **Supply node** (as indicated in Figures 10-1 and 10-2). This is a standard welding tee ***(8 in pipe size.)*** The **bypass** is the short vertical connection joining the two tees **(22)** and **(1).**

## Secondary Loop Pipe Sizing and Arrangements

It is best to size the main supply and return secondary loop headers piping and the branch piping to and from the terminal units under the same criteria as was used for the primary loop. You will be okay if, for the maximum design flowrate in each section of piping, you keep the "old pipe condition" head loss per hundred feet of pipe figure "in the green" at 10 or less, per the reproduced tables of Cameron's Hydraulic Data included in Chapter 2, Gravity Flow of Liquids in Pipes.

It is natural to ask "Why not size pipe smaller in the secondary loop, which is intended to have widely variable flow in it? Will not the secondary loop flowrate demanded by the terminal load equipment be a lot less less than 100% of max design for much of the time?"

The answer is, "Because we must have a calculation basis, an accurately predictable system frictional head loss at 100% of design flow, in order to correctly size the pump and all the control valves. And the system must be capable of running continuously at 100% of maximum, without generating fluid noise, or creating water hammer on shutdown, or causing internal erosion due to high velocity." As we will see shortly, we have a trick up our sleeves to save energy when running at part load—variable speed pumping!

Finally, and this is crucial to success, the secondary loop must be designed and built as a true "REVERSE RETURN" hydraulic circuit. Any other type of parallel flow arrangement involves laddering which will not tend to self-balance but hydraulically will short circuit and starve all but the first few terminal units.

**Figure 10-5** shows how a reverse return circuit is done. The discharge branch from the first terminal unit served by the supply Header, becomes the beginning of the Return Header. The return header and supply header are routed closely beside each other, side by side, following the same twists and turns. Flow in both headers must have the same direction. If the two headers are routed side by side running due North – south, then flow inside them must be in the same direction: either both flowing southward, or both flowing northward. Each place at which another terminal unit branch in the sequence of fluid service takes off from the supply header, then the return branch is brought back to tee into the return header at the same relative point. (Both headers are resized with reducer fittings up or down as required to keep within the velocity and head loss limits specified earlier, of course.) If you will faithfully ensure that the Reverse Return circuit is built just so, then the flow controls

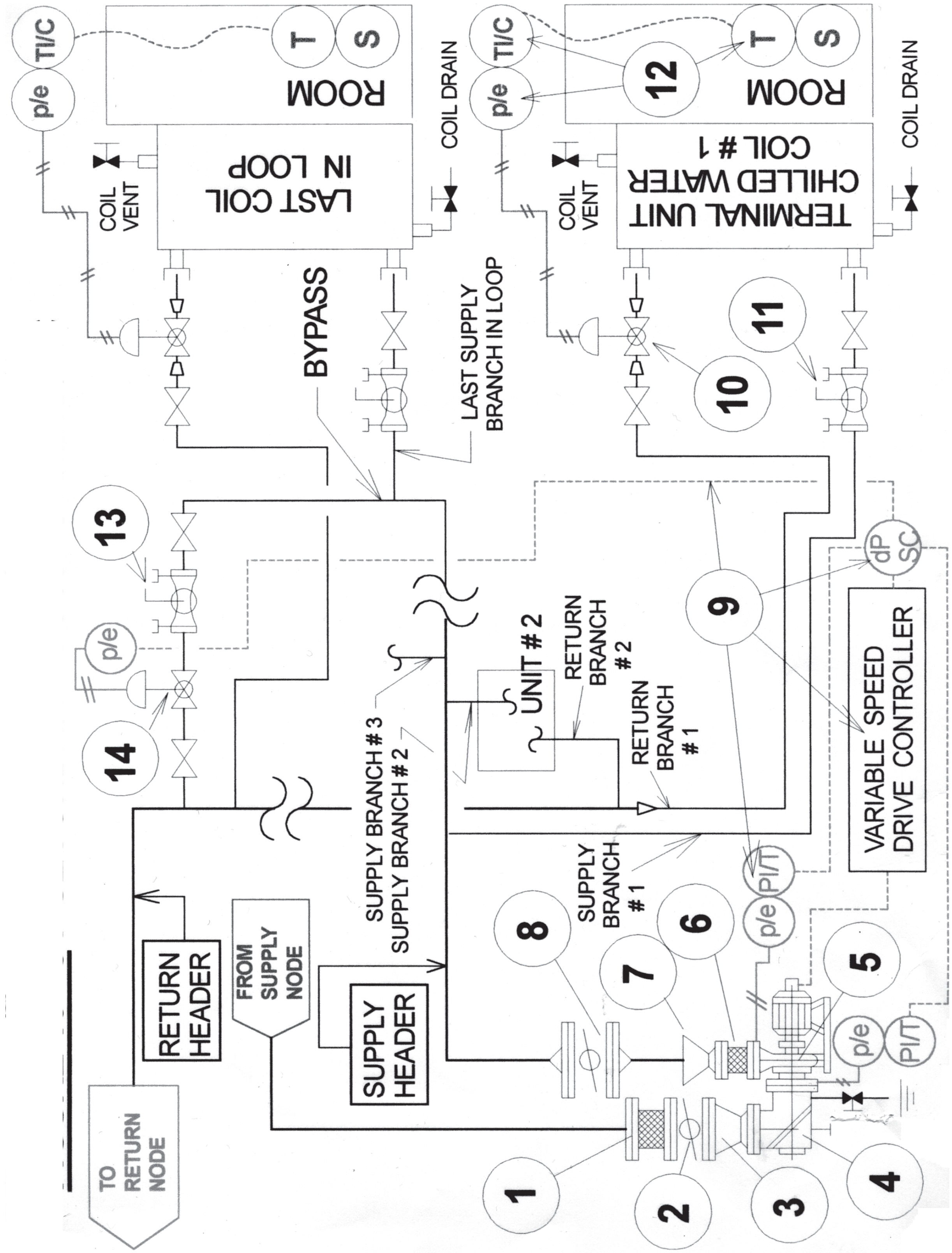



FIGURE 10-5 SECONDARY LOOP SCHEMATIC

and pump controls will work properly, and your reward will be a nice, well-behaved, and totally predictable hydraulic system. I simply cannot emphasize this point too strongly. Once the piping is built and installed, you will be stuck with the result, so the design (reverse return) must be made correctly, and the piping must be installed as designed!

## Secondary Loop Basic Components

The basic secondary loop components of a chilled water system are the secondary loop pump, its controls, the load devices, their controls, and the secondary loop piping network.

We continue our illustration with the item-numbered schematic, (Figure 10-5). In our example system, the load devices are chilled water cooling coils in constant air volume terminal air handlers. In each terminal unit, a filtered supply fan would deliver a constant flow of mixed outside and return air across the coil, which would cool and dehumidify the air, making it "supply air." Ductwork would deliver the air to the conditioned spaces, or rooms, where it would pick up heat and moisture, part would then be exhausted to outdoors and the remainder would return to the unit inlet and mix with more fresh outside air, completing the circuit.

This is about as simple as they get, which is why I chose it for illustration. We could have selected more modern and effective variable air volume (VAV) terminal units, in which our "load devices" would be the chilled water coils in the primary air handler(s), and obtained exactly the same discussion insofar as the chilled water loop hydraulics are concerned. In a process heat transfer application, the terminals could be shell and tube heat exchangers, with the same results.

## Figure 10-5 Component Listing

1. **Flexible connector** (line sized). ***Typical note for all items of this type:*** Specifically mfg. for this duty; flanged ends to match pipe flanges; flexible hose encased in protective stainless wire mesh wrap. Specify applicable pressure-temperature rating for worst case duty.
2. **Isolation valve** (line sized). We want the minimum pressure drop when wide open, and bubble-tight shutoff when closed. Gate valves are ideal, but are too costly in large pipe sizes. So we tend to use full-port ball valves in smaller sizes and butterfly valves in larger sizes. Lug-type butterflies are worth the extra money.
3. **Concentric reducer.** If item (4) has a line-sized inlet, this reducer is not required.
4. **Suction diffuser.** Typically, the correctly sized 1,750 rpm or 1,100 rpm centrifugal pump will have the same suction inlet size as your line pipe and a pump discharge one size smaller. I would expect that a typical secondary loop pump for our example would be a 1,750 rpm type. However, in a low flowrate high head loss application, it might be a 3,500 rpm pump. Most 3,500 rpm pumps will have smaller connection sizes for the same flowrate. In any case, the inlet of the suction diffuser matches the pipeline size, and the outlet of the suction diffuser matches the pump suction flange size. Suction diffusers having same-sized inlets and outlets, as well as outlets one pipe size smaller than the inlet, are routinely available from the hydronic equipment manufacturers (Bell & Gossett, Taco, Armstrong, etc.).

   The suction diffuser is a very important piece of equipment. It straightens out the flow into the pump inlet (rotation degrades pump performance). So it permits a compact piping arrangement while providing "clean hydrodynamics" into the pump inlet. It also has a large-mesh trash collector and bottom drain fitting. It may include a startup strainer with a fine mesh, but this must be removed after initial startup period.
5. **Secondary loop pump,** probably a 1,750 rpm base-mounted end-suction or horizontal split case centrifugal type pump. Really large flowrates with low head requirements may result in 1,100 rpm pumps as being the most efficient, but 3,500 rpm pumps may have to be used in special circumstances (high head low flow.) The 3,500 rpm pumps are typically less efficient, noisier, shorter-lived and have small nozzles
6. **Flexible connector** (matches the pump discharge size).
7. **Concentric reducer** (increases back to line size).
8. **Isolation valve** (line sized). *Do not install a check valve, because we are going to create low flowrates periodically. Check valves chatter when flow velocity falls off, because the velocity pressure becomes too low to nail the valve wide open; the return spring will set up oscillations which in turn can cause severe water hammer.*
9. **Pump control system**. This is illustrated as a simplified schematic of a variable speed pump drive motor, solid state variable speed drive controller, differential pressure controller with field adjustable setpoint, a pair of pressure indicator/transmitters and pressure-to-electric transducers set up to read the differential between pump discharge pressure and pump suction pressure [which is the total dynamic head, (TDH) of the pump], and an adjunct electronic control output and p-to-e transducer to operate the end-of-line bypass valve, item (14).

   We will discuss this system in sufficient detail when the secondary loop control schemes are discussed. For now, suffice it to say the control maintains a field-determined optimum pump TDH as the loop flowrate varies under independent control. It does that by continually monitoring the pump TDH and modulating the pump rpm as required to keep the TDH at its magic setpoint.
10. **Terminal unit load control valve**. (Typically, but not always, the load control valve will work best when its body is one pipe size smaller than the branch line. That is why pipe reducer fittings are shown before and after the control valve.) This engineered-for-purpose, thermostatically operated pneumatically-actuated globe plug valve is the heart of the system. Its characteristic must be carefully selected. If you don't know exactly how on your first design attempt, please get an **expert** control valve vendor to help you, **not a hack,** and learn from him or her how to correctly select the valve type, the specific model number, its features, its trim, its materials of construction, and its exactly correct size for the application. If you have 20 different terminal units in the loop, you may well have 20 different control valves to make them work right.

    What we want to accomplish is for the pressure drop across each control valve when wide open to be significant, and equally significant to operation of the system, and for motion of the valve plug to control flow predictably over a gpm range from 10% of maximum flow

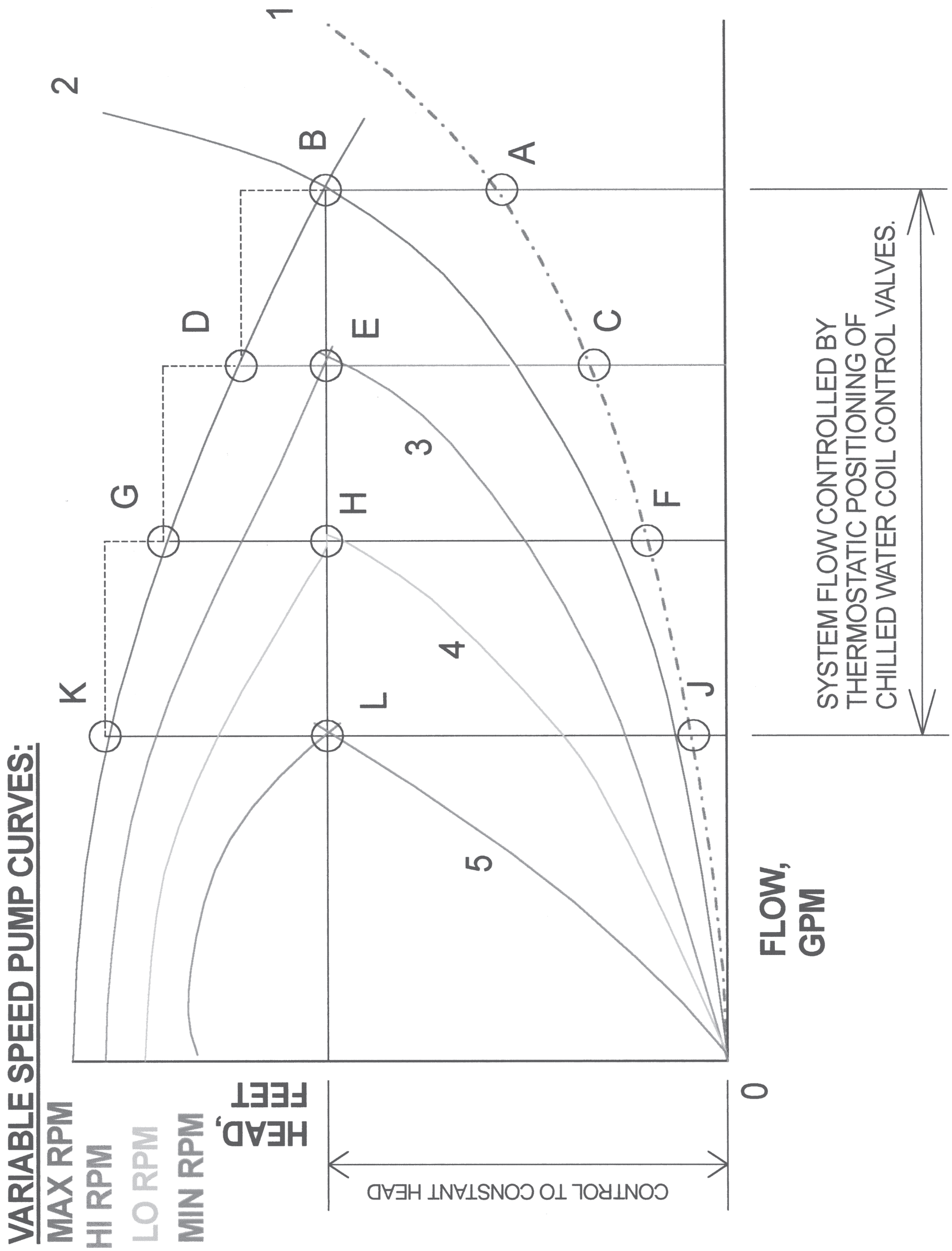



**FIGURE 10-6 SECONDARY LOOP PUMP SPEED CONTROL**

to 100% of full design maximum flow, while moving from about 30% open at 10% flow to 100% of flow being just reached when the wide open position is achieved by the valve's stroke control actuator. **This will not be possible if you select an off-brand valve with unknown table of Cv versus % open values. Good control valves and their systems are never cheap, but you get exactly what you pay for. If you want excellent performance, pay the price and obtain excellent control valve equipment. As a vital added benefit, you will find that the manufacturers of topline controls do not allow hacks to remain on their payrolls. This can save your butt!**

We will discuss control valve action and its integration into the overall system in sufficient detail when the secondary loop control schemes are discussed.

11. **Flow Balance Valve (circuit setter)** (branch line sized). An engineered specialty item, consisting of manually operated, infinitely adjustable throttle valve with calibrated differential pressure taps on both sides of the valve plug, which permit direct reading of volumetric flowrate with the factory-furnished readout device connected (calibrated manometer or $\Delta P$ gauge.) During the flow-balancing exercise, they are to be manually adjusted to the point that simultaneously throughout the entire secondary loop, each terminal unit receives the scheduled design maximum gpm.

    *We need this manual device in each terminal unit branch of a reverse return piping circuit to make a flow balance possible. If these devices, or their equivalent are not installed, even if the circuit is a properly designed reverse return type, there will be no way to balance the system hydraulically; the system will not operate correctly if not balanced with an adequate degree of precision. The equivalent to a circuit setter valve is a manual globe plug or needle valve in series with a separate flowmeter, installed as flowmeters must be (long straight lengths of pipe upstream and downstream of the flowmeter element.) In most HVAC applications, the extra expense is not warranted and the one-piece circuit setter, although not a highly precise piece of hardware in the sense a top-line control valve is, will be adequate. In process applications (non-HVAC) needing precise control, however, the separate throttle valve and flowmeter in each terminal unit branch will most probably be necessary for sufficiently precise terminal unit chilled water flow balance (either that, or a more expensive terminal unit flow controller altogether).*

12. **Terminal unit flow control loop**. To control our example constant air volume HVAC unit, a room setpoint thermostatic unit with dry bulb temperature sensor, the electronic temperature indicator/controller, and the necessary hardware and transducer to convert the electronic control signal to the pneumatic positioning signal at the actuator of the terminal load flow control valve, item (10).

    The key feature of this scheme is total independence of the quantity of cooling done by each separate terminal air handling unit.

    You can visualize the actual unit controls working best by considering the summation of volumes of building space served by the same air handling unit as a single thermodynamic control volume in steady state equilibrium, isolated from all the other volumes, receiving a certain amount of sensible heat and moisture (latent heat) across its surface, receiving a fixed flowrate of cold dry air plus cold water vapor from the terminal unit, and returning that same dry air volume to the unit at higher temperature and moisture content.

    A thermal and mass balance is assumed for the steady state. The space is assumed held at a uniform sensible temperature; actually, the space thermal condition is represented by the temperature and moisture content of the return air stream as it enters the return air duct to the unit.

    The cooling load, which is the amount of heat and moisture which must be removed constantly by the terminal unit to keep the control volume of air at thermal equilibrium, is determined by the thermostat set temperature and environmental factors. Lower settings cause increased heat transfer into the system from the surroundings. Likewise, increased energy transfer across the control surface, such as increased electrical power being converted to heat at lighting fixtures and motors inside the volume, also raise the cooling load. The moisture content of the fresh air fraction and the internal moisture release govern the latent load (see a good text on air conditioning, such as the *ASHRAE Fundamentals Volume* for the details if interested.)

    Again, the control volume's cooling load is an independent quantity of heat plus mass (water vapor) transfer. The controlled variable is the *psychrometric statepoint of the return airstream.* That which does the controlling is the ***supply airstream, by absorbing the excess heat and moisture on its journey through the space to the return duct.***

    Obviously, only three airstream parameters can be controlled by the terminal air handling unit:

- **Mass flow of dry supply air** into the control volume (in our example, this is a mechanically fixed constant value which we cannot change, but in VAV systems it is a major control variable);
- **Sensible (dry bulb) temperature of the supply air,** and
- **Latent heat content (wet bulb temperature) of the supply air.**

The last two parameters can be controlled in our constant airflow CFM example by three mechanical quantities:

- **The physical cooling coil itself:** surface area of tubes and fins and its inherent overall heat transfer characteristics (a design variable only; controllable only by initial coil selection and installation into the terminal unit, of course.)
- **The temperature of the chilled water entering the cooling coil** (in our example and in nearly all existing systems, this parameter is a design parameter only and is set up at the chiller controls and left alone (invariable) once successful system operation has been obtained.)
- **The chilled water flowrate through the coil.** Aha! This is the variable we control! So the success or failure of the terminal unit to control the space temperature and humidity in our example system depends solely upon the proper action of the load flow control valve, item (10)! **What this means is just this:**

1. *The room thermostat setting and the current room cooling load act together to decide exactly what the % open position* of the control valve Item (10) will be at any given time. **The valve plug position is wherever it must be to maintain thermal equilibrium inside the room (air space) control volume.** The terminal unit control loop accomplishes this physically by mechanically setting the position

of the valve stem (which sets the % open location of the valve plug with respect to the fully-closed position on its seat). Hence the position of the valve plug (percentage of valve stem full stroke) indicates the cooling load directly, and nearly proportionately, as determined by the valve's characteristic profile.

2. *The actual % open value of the control valve at a given time determines the head loss (total pressure drop) across each entire terminal piping branch, from outlet of its tee at the supply header to inlet of its tee at the return header, at that same time.* **Thus the chilled water flowrates and head losses across each terminal unit branch are independent of each other, and are determined solely by the cooling load on each terminal unit.**
3. *It is the duty of the secondary loop pump to furnish the sum of individual terminal branch gpm demands at all times. The pressure (pump TDH) at which the flow is delivered is of no concern to proper functioning of the system, that of temperature control in the room air volumes. As long as the supply header has enough pressure to create the needed branch flowrates, any excess pressure will simply be wasted across the control valve.* ***However, the pump TDH IS of concern to the system thermodynamic efficiency. Power = Flowrate × TDH × a constant. We cannot change the flowrate, because it is the direct function through which the cooling load demand is satisfied, all other parameters being fixed at the time of construction of the system and purposely made specific permanent constants by the initial system settings, startup, and balance actions. However, we can do things to reduce the head loss, and we will.***

Now it is time to study the pump curves, system curves, pump variable speed drive philosophy, and the pump controls in detail. Our goal is to provide the required flow range at minimum head loss while under perfect control at all times.

## Secondary Loop Pump Flowrate and Total Dynamic Head Control Systems

***Figure 10-6 must be studied carefully and repeatedly in order to appreciate the following narrative.***

In the last couple of pages we saw that the main controls of the secondary loop hydraulic system are the individual thermostatic control loops, acting through the coil flow control valves. These control loops respond to changes in terminal unit return air temperature and nothing else. If more cooling is needed, the return air temperature will rise above the thermostat setpoint and the system responds by creating increased chilled water flow through the coils, and vice versa. Nothing could be simpler.

To facilitate that wide range of flowrate demands smoothly and efficiently, we use a centrifugal pump. Its characteristics are to produce a smooth continuous curve of unique operating points on a plot of total pump head, total dynamic head (TDH), on the ordinate, versus pump volumetric flowrate, gpm, on the abscissa.

When tested in the laboratory, or in the field, a specific centrifugal pump consisting of casing, specific impeller diameter, shaft and seals assembly, installed in a simple piping loop containing a fixed resistance (pipe and fittings and wide-open isolation valves plus a load device such as a cooling coil), and a variable resistance (an infinitely adjustable throttle valve), will create a unique GPM-TDH delivery curve for each specific constant rotational speed, rpm, at which it is driven.

**Figure 10-6** depicts four members of the family of curves for one such pump impeller diameter. They are labeled **"MAX RPM, HI RPM, LO RPM** and **MIN RPM."** As would be expected, for a given flowrate the pump generates significantly more total head in the flowstream at high impeller RPMs than at low RPMs. The TDH is just a measure of the velocity with which the liquid is slung off the impeller onto the diffuser casing, which slows it down so that the velocity pressure is partly converted to static pressure. Of course the TDH is measured as a *total pressure* rise from the planes of the pump suction flange to the pump discharge flange, through the pump, and converted to head in feet of liquid flowing. (For water, multiply the total pressure rise in psi by the constant 2.31 to obtain TDH in feet of water units.)

Another feature of the pump curve family is that at high RPM's the pump will deliver a greater maximum flowrate too. So the family of curves is "nested" with the highest RPM at the top and lowest at the bottom, per **Figure 10-6.**

So, high RPM means big flows generated at high pump heads, which translates to high fluid energy which needs big power input to the motor. All characteristics decrease with falling RPM: less flow capacity, lower pressure, less power input.

Now, if we didn't care about power consumption, the only pump control we would need would be the motor starter. The pump would deliver the amount of flow determined by the point at which its delivery curve intersects the piping system resistance head loss curve. When the system resistance went up because of thermostats reducing the per cent opening of the terminal unit flow control valves, the pump would respond by *decreasing* the flowrate delivered, while *increasing* the TDH it generated. In familiar parlance, the pumps would ***"back up on its curve,"*** that is, move **up**ward on the head-ordinate and **back** to the left on the flowrate-abscissa.

But we do care about wasting power. And so today we make an investment in a variable-speed drive unit for the pump motor, and reduce the pump's driven speed when the flow demand drops off. We generate the same GPM at significantly less TDH that way, at great energy savings which amortize the investment quickly. (When I was a brand new ME graduate, a transistor was an expensive novelty. Microchips and workable digital microprocessors or computers had not yet been invented. To control the speed of an alternating current motor directly, electronically, was simply not an option. Shaft speed was varied mechanically, by hydraulic transmissions or variable belt sheaves and clutches, at considerable expense. So a variable-speed water pump would have been out of the question.)

Today, solid state motor controls and drives are cheap, reliable, and commonplace. Which is very very good.)

Now examine **Figure 10-6** while we discuss how the variable speed secondary loop pump controls should work. It's really quite logical and straightforward. Did I ever mention that simpler is better?

## During the design phase, we make our total head loss calculation for the secondary loop piping at 100% of the design flowrate. This will be the rating point for the secondary loop pump.

Now remember, the secondary loop is a reverse return circuit with a number of parallel flowpaths to choose from, when we make our calculation. While we could (painfully) reduce the whole loop to a single hydraulic resistance, using a network program of some sort or by utilizing the method I offer in Chapter 8 of this book, it is usually not necessary to do that much work. By quick hand order-of-magnitude calculations we can identify the path of greatest resistance, the one with the highest pressure loss with all control valves wide open, and use it as the "terminal branch" in our mathematical model of the piping system. The system becomes simply:

1. The supply header, from secondary loop pump discharge to the point of exit thru tee to the selected terminal branch;
2. the "highest loss" terminal branch in its entirety, including the flow control valve at its 100% wide open point;
3. The return header from the selected terminal branch rejoining tee all the way back to the pump suction flange.

**For accuracy we should always make the calculation with the actual numerical value of wide-open $C_v$** of the control valve we intend to use in construction.

**If you do not know that value, then the system is already out of control, even before it is "designed."**

**To have control of the system, you must have complete foreknowledge of the control valve's characteristics during design.** The pump selection must be made for the correct 100% design point, and since about half the piping system head loss is the drop across the control valve, if you do not know what that drop is, you have no idea what the pump rating point must be!

Having made the single rating-point flow-head loss calculation, we can quickly calculate a table of values for 6 or 8 points on the system head loss curve, covering a range from zero to 200% of the design flowrate. Armed with that table, we can make our pump selection perfectly. (We cover the "how-to" of head loss calculation and system curve point plotting in a lot of detail in Chapter 8.)

- **The pump we select needs the capacity to deliver the rating point, which once again is defined as the 100%-of-design-flowrate and the matching calculated total head loss, when running at about 95% of its maximum rpm. For an 1,800 rpm pump, select the impeller that will deliver the rating point at about 1,710 rpm on the pump drive. This gives us an operating margin to work within, and to cover uncertainties in the system head loss calculations. The pump should deliver the rating point at or near its region of best pump efficiency for this speed. A competent pump vendor can help you obtain the necessary pump data to verify the capacity.**

Now examine **Figure 10-6** again. See the family of system loss curves; they begin at the lower left-hand point (0,0) and rise from left to right.

The lowest-down dotted dark-red curve labeled **O-J-F-C-A-1** is a plot of the secondary loop fixed resistance only; this is the head loss due to pipe friction, fittings, the cooling coil and all the inline devices and wide open isolation valves. This curve remains the same regardless of what the control valve and pump are doing.

The next curve up is the really important one: it is labeled **O-B-2.** It is the sum of fixed resistance plus the drop across the **wide-open flow control valve, item (10)** Figure 10-5. In other words, curve **O-B-2** is obtained by adding the head loss across the wide-open control valve at each value of GPM, to the fixed resistance curve **O-J-F-C-A-1.** If we make no change to any part of the secondary loop, including the postion of the control valve plug, then curve **O-B-2** always characterizes the loop hydraulically.

There is a family of increasingly higher-resistance system curves to the left of **O-B-2** as well. They are **labeled 0-3-E, 0-4-H,** and **0-5-L. They represent the system with increasingly smaller % openings of the control valve.** For example, curve **0-3-E** might represent the secondary loop with control valve 90% open, curve **0-4-H** with the valve 80% open, and **0-5-L** with it 70% open.

Also on Figure 10-6, of course, is the set of four pump curves representing the RPM range available to us, as already discussed. In practice, it is your job to plot the additional pump curves (the ones at less than maximum rpm) on a clear copy of the pump curve taken from the manufacturer's published catalog. This is done by using the pump affinity laws with the published (100% maximum rpm) curve. Then, plot the system curves on the same sheet. It helps to enlarge the scale of the copy, and to print it dark so that the grid lines and axis tick marks show up. This done, we can analyze the system operation.

- **It is absolutely required that the range of pump flowrates available via drive speed control must satisfy the full range of summed terminal unit flowrates demanded by the system. The overlay of system curves on the pump curves (Figure 10-6), is the tool that enables verification of this requirement.**

*I think the best way to complete the explanation of the control system analysis is to describe what must happen during the initial system startup and flow balancing exercise. It is during this phase that the various control setpoints are established.* So let us imagine that our new primary-secondary chilled water system is installed and cleaned out, ready for setup and balancing. We shall begin now.

- We begin with the Figure 10-5 system fully charged with water, and with all trapped air pockets completely eliminated by virtue of the high point vents in cooling coils and piping network. The secondary loop pump is off.
- Next, we set the pump speed controller to drive the impeller at 95% of full speed. As we noted above, for an 1,800 rpm pump we would set the variable speed drive (VSD) controller to deliver $1{,}800 \times 0.95 = 1{,}710$ rpm. For the time being the VSD controller, Item (9), should be disabled and the VSD manual control will be used for balancing. When all is done with the setup and flow balance, we will enable the automatic feedback control loop to operate the VSD speed setting.
- Next, set each terminal branch flow control valve Item (10) (the actuator) in the locked-wide-open position. It is vital to proper operation of the system that all terminal unit flow control valves be 100% wide open during system flow balancing.
- Now, start the secondary loop pump.
- Then, flow-balance all of the terminal units to their scheduled design-maximum gpms. **This is done incrementally by manually adjusting the circuit setters, item (11) in Figure 10-5.**

  It is highly advisable to have this work carefully done by certified hydraulic balance contractors using competently trained and experienced personnel, calibrated instrumentation, and well-established procedures. If this is not done correctly and accurately, the system will never work correctly.

  By the way, if your head loss calculation was short of the actual loss, you will find out now. If any of the circuit setters have to be set wide open and the desired flow still has not been attained, it will be necessary to add another 1.0% to the motor rpm and try again. If you really screwed up the calculations and at 100% rpm still have not attained desired flow through all circuit setters, you will be forced either to overspeed the pump, which is not good, or replace the impeller with a larger-diameter one, which is a huge embarrassment and costs a few bucks, but is the right thing to do. (P.S. Don't forget to install a new drive motor to be non-overloading with the bigger impeller if this is necessary: check with your pump technical rep to be sure.)

*And if the impeller is already the maximum diameter for the pump casing, then a new, bigger pump is in order, and also probably a job with a new firm will be necessary before long for the poor design engineer who shorted the head loss calculation. Don't short your system head loss calculation.*

- Okay. The system is balanced, all the terminal unit control valves are wide open, and all circuit setters are reporting 100% of their scheduled design flowrates. Now, without changing anything whatsoever, read the differential pressure across the pump, and convert it to head in feet (psi × 2.31). This is the TDH for the system at 100% flowrate. Next tally up the sum of flowrates that were measured during balance; the sum is the actual 100% maximum flowrate for the system. Plot this point on your certified factory pump curve, which was furnished by the pump factory according to your design specification. This will show you where you are. **It is shown as point B on Figure 10-6.**

  And write down the four figures of merit for your file: the final actual RPM, the 100% flowrate GPM, the TDH, and the pump differential pressure measurement in psi. **These are the "magic numbers."**
- Finally! Energize the feedback control loop Item 9 for the pump VSD Controller, and set the differential pressure setpoint to be exactly and precisely that which you just measured. Now, no matter what happens out in the secondary loop, the pump will automatically adjust its RPM such that the "magic" TDH is maintained constant.
- Reenable the pneumatic actuators for each of the terminal unit control valves, so that the space temperature thermostats have control of them, and you are done!!!

*Well, almost.*

We still have to see what happens when the thermostats begin to close the control valves. **And we have to cover that last pair of items in Figure 10-5 which so far have been ignored: (13) and (14).**

- When the system is up and running and under full automatic control, most of the time the secondary loop pump gpm will be less than the 100% design maximum because the cooling load is usually less than the peak. So the pump will be running at less than 100% rpm. Lets see what happens as peak summer changes to fall and finally to winter; follow along on **Figure 10-6.**

At a reduced load, let us assume the thermostats have forced the system flow to be that which goes with **point E** on **Figure 10-6.** The TDH of the pump and the total head loss of the piping system remain exactly the same as when we started; **points B and E** lie on the same horizontal line of constant head, thanks to our VSD Controller.

Notice that in this condition, the pump speed is on the **"HI rpm"** curve, and it intersects **the 0-3-E** system loss curve at the current GPM operating point. The head loss through fixed piping resistance is that at **Point C,** and the additional head loss across the partly closed control valves is the vertical distance on Figure 10-6 between **point C** and **point E.**

**If we didn't have the VSD Controller, the flowrate would be the same, but the TDH would be at Point D instead of E. The additional loss would be created by the control valve being sent to a more-closed position than is the case with pump speed reduction.**

**So we are saving power indeed by reducing rpm, by reducing the head by the amount measured between points D and E while still delivering exactly the same thermostatically required chilled water flowrate to the system.**

And so it goes. Eventually there will come a point where demand is so low that the pump gpm drops off too much and the controls may become erratic. We gain nothing by being out of control. So what we do is set up the end-of-the-header bypass control valve, which is **item (14) on Figure 10-5 and up to now has been fully closed,** to come open at the pump drive rpm which corresponds with flow point **J-L-K** on **Figure 10-6.** This is the flowrate where the "MIN RPM" pump curve and system curve 0-5-L intersect.

You determine where this point should be by field experience with the system after it has been balanced. It will be a fairly small flowrate, maybe only 10–15% of the maximum design gpm. Maybe a bit more. Typically I size the bypass pipe for about 7 ft/sec flow velocity at 15% of the max secondary loop pump GPM, and use the **circuit setter Item # 13** to trim as required. This is not brain surgery; we just want enough flow to remain under proper VSD control over the entire range of system demand.

*OK. Now we really are done with this topic!!!*

CHAPTER
11

# ESTIMATING THERMAL BLACK–BODY TEMPERATURE OF THE NIGHT SKY FOR RADIATION HEAT TRANSFER CALCULATIONS

My reference for the following equations is an excellent engineering text for practicing engineers entitled **Heat Transfer Professional Version** by Lindon C. Thomas, 1st edition, Prentice-Hall, Englewood Cliffs, New Jersey, 1993. The equations themselves are based upon work by Duffie and Beckman, published in ***Solar Energy Thermal Processes,*** John Wiley & Sons, New York, 1974.

I have found this data quite handy for making accurate estimates of the thermal equilibrium temperature of outdoors surfaces exposed to the night sky. For example, will an exposed surface temperature drop below the dew point during the night? Is it possible that a surface might drop below the freezing point even though the surrounding air temperature is above freezing?

First we shall see the equations, with a bit of explanation. Then we shall demonstrate their use via a copy of the hand calculated solution of a simple illustrative problem. The example problem answer checks the solution to the same problem published as Example 5-27, pp. 322–323 of Thomas's text.

## SKY TEMPERATURE EQUATIONS

$\mathbf{T_{sky}}$, °R = ($T_{amb}$, °F + $\psi$)

($\mathbf{T_{sky}}$)= absolute temperature of the clear night sky (cloudless) in degrees Rankine (zero degrees Fahrenheit equals 459.7° Rankine)

($\mathbf{T_{amb}}$) = current measured thermometer temperature of the ambient air surrounding the radiating surface, degrees Fahrenheit.

$\psi$ = 424°R during winter-type atmospheric conditions: cold, clear, and dry, relatively low atmospheric dew point.

$\psi$ = 449°R during summer-type atmospheric conditions: warmer, clear, and moist, relatively high dew point

For example, on a clear **summer** night when the ambient temperature is **70°F,** the value of $\psi$ **= 449°R** and the night sky blackbody temperature is **(70 + 449) = 519°R. {519°R = 59°F}**

For another, on a clear **winter** night when the ambient temperature is **20°F,** the value of $\psi$ **= 424°R** and the night sky blackbody temperature is **(20 + 424) = 444°R. {444°R = –16°F}**

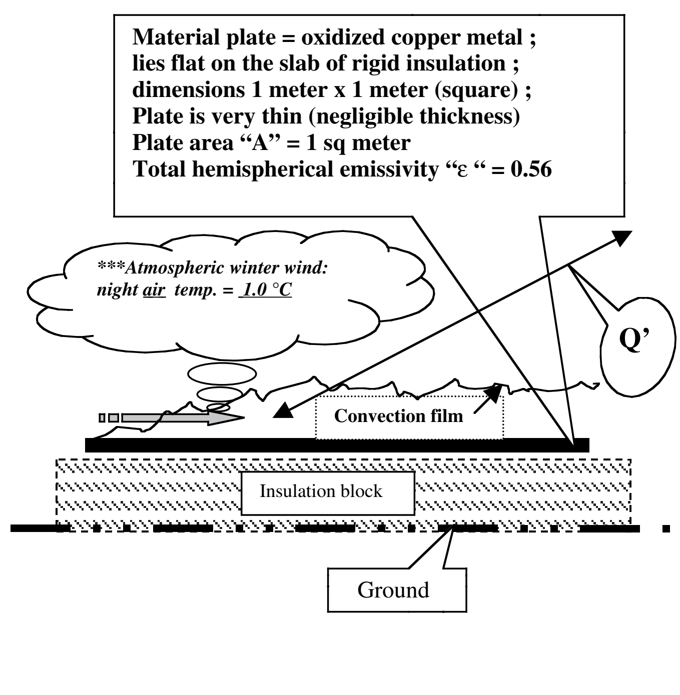
Obviously, the amount of water vapor present in the atmosphere affects the radiant heat balance.

If there were no atmosphere, the night sky temperature would be nearly absolute zero.

During daytime, there is a solar radiation heat flux falling from the sky upon the surface, for which you must go to ASHRAE Guide for data, or some other solar heat gain data source.

On a cloudy night, the radiation effect is much smaller and can usually be neglected for ordinary engineering calculations. Only conduction and atmospheric convection would come into play on a cloudy night, and an unheated exposed surface (no net conduction or radiation from nearby objects) will come to equilibrium at essentially the ambient air temperature.

Now let's see the example problem.



The long slanting arrow marked "Q′" is the combined convection and radiation heat transfer to/from the copper plate. The free convection film coefficient was calculated by the flat plate formula to be $\mathbf{h_C}$ = 10 watts/sq meter – deg. C. and is a result of an assumed average velocity of the night wind across the top of the plate.

Our quest is to estimate the equilibrium temperature of the copper plate in the conditions shown, in degrees Fahrenheit please.

Here are our assumptions:

1. There is no heat transfer between the copper plate and the ground.
2. The net heat gain from the airstream to the copper plate is fully accounted for in the free convection film coefficient "$\mathbf{h_C}$".
3. The net heat loss from the plate to the night sky is fully accounted for by assuming "grey body" radiation: which means that the radiation quantities are equal during equilibrium: $\alpha_P = \in_P = 0.56$ where $\alpha_P$ is the infrared absorption coefficient and of course $\in_P$ = total hemispherical emissivity of the material (tabular values of $\alpha_P, \in_P$).
4. Use $\mathbf{Q'_C}$ as the symbol for convective heat gain and $\mathbf{Q'_R}$ as the radiant heat loss symbol. These are in units of thermal energy per unit of time.
5. Use "T" for the appropriate temperatures: $\mathbf{T_P}$ for the copper plate, $\mathbf{T_{air}}$ for the night wind and $\mathbf{T_{sky}}$ for the effective black-body temperature of the night sky.

We solve for the thermal equilibrium condition, which requires that:

**Net Q′ gained ≡ Net Q′ lost**

Q′ GAINED = convection $Q'_C$ =

$Q'_C = h_C A(T_{air} - T_P)$

Q′ lost = radiation $Q'_R$ =

$Q'_R = \in_P A\ ({}_{plate}F_{sky})\ \sigma\ (T_P^4 - T_{sky}^4)$

$h_C = (10\ \text{W/m}^2 - \text{C})(3.413\ \text{Btu/hr/W}) \times (1.0\ \text{m}^2/10.765\ \text{ft}^2)(1.0°\text{C}/1.8°\text{F})$ =

= 1.761 Btu/hr – ft$^2$– °F

$T_{air}$ = 1°C × (1.8/1)(°F/°C) + 32° = 33.8°F

= (33.8 + 459.7) = 493.5°R.

The plate area "A" cancels out.

The geometric shape factor (${}_{plate}F_{sky}$) is the decimal fraction of the area seen by the copper plate which consists of night sky above the plate. Since our simplified model assumes the plate sees nothing but sky, then F = 1.00; if other objects were within the plate surface's field of view, they would exchange radiation with the plate also, and if the objects' temperatures were different from the plate, then they would change the outcome accordingly. We will use the value (${}_{plate}F_{sky}$) = 1.00.

Finally, "σ" is the Stefan-Boltzmann constant for black-body radiation exchange. We can use it because we are in thermal equilibrium (grey body assumption) and its value is

$$\boldsymbol{\sigma = 0.1714 \times 10^{-8}}\ \text{Btu/hr} - \text{ft}^2 - °\text{R}^4$$

Now use the appropriate relationship for night sky winter-type atmosphere radiation which was given on the first page of this chapter as:

$$\mathbf{T_{sky}},\ °R = (T_{amb},\ °F + \psi);$$

$\boldsymbol{\psi = 424}$ **°R** during winter-type atmospheric conditions: cold, clear, and dry, relatively low atmospheric dewpoint

$\mathbf{T_{sky}}$, °R = (33.8°F + 424°R) = 457.8°R; and this equals 457.8 – 459.7 = –1.9 °F. We must use the absolute temperature value (457.8) in the calculations.

Substitute values into the heat balance:

$h_C A(T_{air} - T_P) = \in_P A\ ({}_{plate}\ F\ {}_{sky})\ \sigma\ (T_P^4 - T_{sky}^4)$ and solve for $T_P$.

(1.761 Btu/hr – ft$^2$ – °F) (493.5 °R – $T_P$)
= (0.56)(1.00) (0.1714 btu/hr – ft$^2$ – °R$^4$)
× [$(T_P/100)^4 - (457.5/100)^4$];
(1.761)(493.5 – $T_P$) =
=(0.56)(0.1714)[$(T_P/100)^4 - (4.575)^4$];
This reduces to a simple equation,

$$(\mathbf{T_P}/100)^4 = 9{,}492.24 - 18.3468(T_P/100)$$

which we can solve by hand using trial-and-error iterations. We could also solve it using the dandy engineering mathematical computation program TK Solver, which has been one of my favorite calculation tools for a long time. You may not own a copy of TK Solver, so I will demonstrate the iterative solution.

Trial #1: Assume $T_P$ = 33°F = 492.7 °R : Obtain value for left side – right side : $(492.7/100)^4$ = LS = 589.29 ; 9492.24 – 18.3468 (4.927) = RS = 452.77; Then LS – RS = 589.29 – 452.77= (+)136.52.

Trial #2: Assume $T_P$ = 20°F = 479.7°R : Obtain value for left side – right side : $(479.7/100)^4$ = LS = 528.19 ; 9492.24 – 18.3468(4.797) = RS = 691; Then LS – RS = 528.19 – 691.28= (–)163.09.

By the usual interpolation method,

$\mathbf{T_P}$ **= (20 + 7.1) = 27.1°F. = –2.7°C** vs. $T_{air}$ = 33.8°F = 1.0°C

CHAPTER

12

# HANDY FACTS REGARDING TYPES OF THERMAL INSULATION

The good name of Mr. George Antaki, P.E. was summoned in Chapter 1 when steam hammer was under discussion. As I closed out my planning for this book, I ran across a neat little package concerning pipe (and duct and equipment) insulation which was excerpted from Mr. Antaki's book of ASME Professional Development Series lectures. It consists of the following reprinted pages.

It may help clarify some of the *qualitative* confusion factors a consulting plant or A/E engineer may face when insulation material tradeoffs are made. As always, be sure to also obtain the specific *quantitative* conductivity versus density data from the material's responsible vendor or manufacturer.

## THERMAL INSULATION

### Material Selection

There are two principal types of insulation

(1) A closed cell insulation. This material conserves energy, and also has a tortuous path that stops ambient vapors from penetrating and condensing on a cold pipe wall. It is used with sealed insulation jacket.
(2) An open cell insulation: This material conserves energy but is fibrous. It let's moisture out from hot pipes. It is used with overlap insulation jacket

**EXAMPLES OF INSULATION**

| Fluid Temperature (°F) | Insulation |
|---|---|
| –400 to –50 | Cellular glass |
| –50 to 35 | Cellular glass |
| | Phenolic foam |
| 35 to 180 | Elastomeric foam |
| | Fiberglass |
| | Cellular glass |
| | Phenolic foam |
| 180 to 250 | Fiberglass |
| | Cellular glass |
| | Phenolic foam |
| 250 to 650 | Calcium silicate |
| | Fiberglass |
| | Mineral wool |
| 650 to 1200 | Mineral wool |
| | Calcium silicate |

**INSULATION MATERIAL STANDARDS**

| Material | ASTM |
|---|---|
| Calcium silicate | C 533 |
| Cellular glass | C 552 |
| Elastomeric foam | C 534 |
| Fiberglass and Mineral wool | C 553, C 547 |
| Expanded perlite | C 610 |
| Phenolic foam | C 1126 |
| Aluminum jacket | C 921 |
| Stainless steel jacket | C 921 |

Characteristics of insulating materials

| Material (roughly by increasing price) | Density Lb/ft$^3$ | Notes |
|---|---|---|
| Elastomenc foam | 6 | Water resistant. |
| | | Can ignite with electricity. |
| | | Develops heavy smoke. |
| Fiberglass | 3.5 | Absorbs leaks (gets wet & soggy!). |
| | | Acoustic insulator is not rigid. |
| | | Broad range of temperatures. |
| Calcium silicate | 15 | Strong, can be stepped on is rigid. |
| | | Fire resistant. |
| Mineral wool | 8 | Acoustic insulator not rigid. |
| *Phenolic foam* | 2.2 | Not used on stainless steel or copper. |
| | | Water resistant. |
| | | Does not easily ignite. |
| Cellular glass | 8.5 | Needs to be thick. |
| | | Water resistant. |
| | | Does not easily ignite. |

## Insulation Thickness

To select insulation thickness, consult the vendor catalog. As an example, the insulation thickness (inches) for a fiber glass insulation for various pipe sizes and temperatures is

| Pipe size | 150°F | 250°F | 350°F | 450°F | 550°F |
|---|---|---|---|---|---|
| 1 | 1 | 1.5 | 2 | 2.5 | 3 |
| 6″ | 2 | 3 | 4 | 4 | 4.5 |
| 10″ | 2 | 3.5 | 4 | 4 | 5 |
| 14″ | 2 | 3.5 | 4 | 4 | 5 |

The thickness for a calcium silicate insulation would be very similar.

## Efficiency of Insulation

In one case at the hanger of a 12″ insulated hot process fluid operating at 900°F the measured temperature profile is

| Location | Temperature (°F) |
|---|---|
| Pipe wall (under insulation) | 900 |
| Pipe clamp (under insulation) | 860 |
| Hanger rod – clamp bolt (just outside insulation) | 130 |
| Hanger rod 15″ above top of pipe wall | 90 |

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