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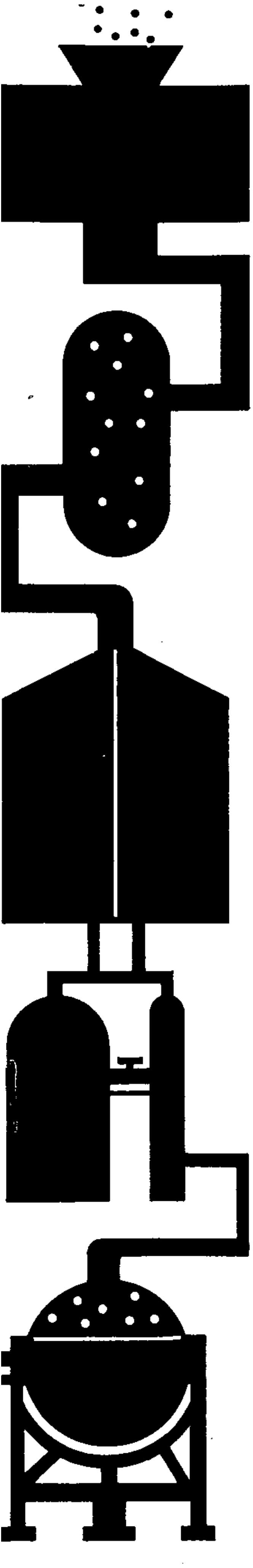

Calculation & Shortcut DESKBOOK

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CALCULATION & SHORTCUT DESKBOOK

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QUICK CONVERSION OF NO_x AND CO EMISSION RATES

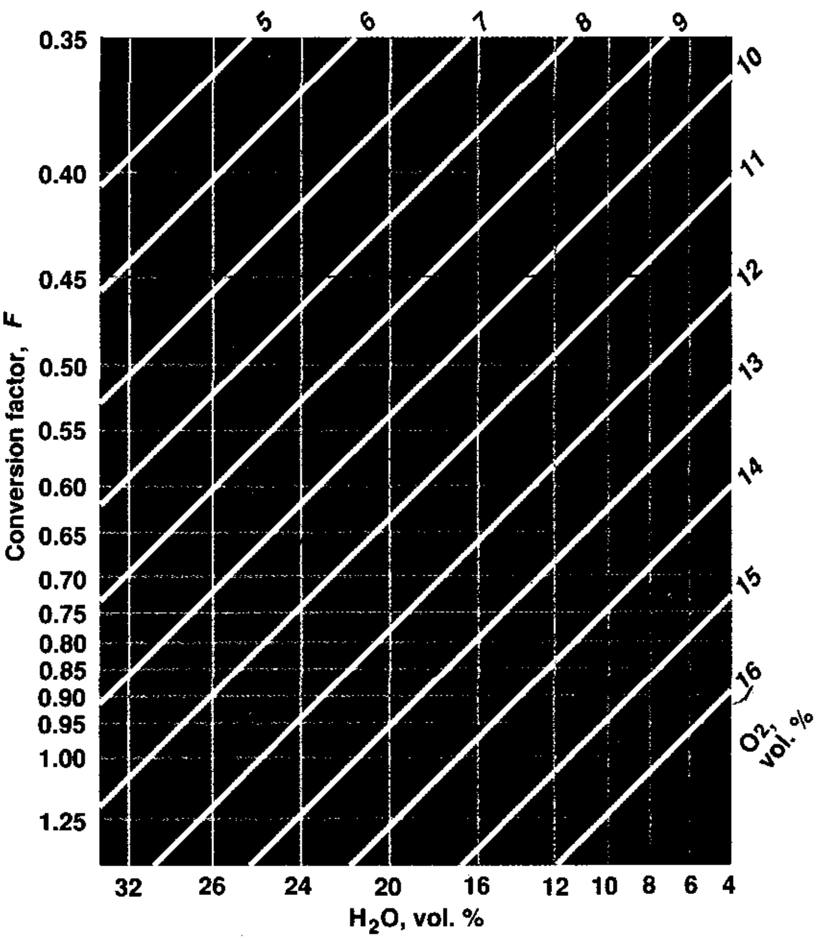
V. Ganapathy,
Abco Industries, Inc., Abilene, Tex.

Monitoring combustion emission products — NO_x and CO in particular — is increasingly *important, due to tightening* regulations. Gas turbine cogeneration or combined-cycle plants, for example, produce both of these emissions, and regulations pertaining to them are very strict. Because of this situation, it is desirable to have a quick and easy way of converting emission data from mass flowrates (lb/h) to ppm(v), dry basis, which is the usual way that they *are reported to regulatory* bodies. An oxygen concentration of 15%

Nomenclature

V = volume or volumetric ratio; V_n, volumetric ratio of NO_x (ppm); V_c, volumetric ratio of CO (ppm)
w = flowrate of a constituent gas in the exhaust stream, lb/h
W = total exhaust stream flowrate, lb/h
F = conversion factor
MW = molecular weight of exhaust stream, lbs
$\%H_2O$ = percent volume of water in the wet exhaust stream
$\%O_2$ = percent volume of oxygen in the exhaust stream

EMISSION-DATA CONVERSION CHART



The intersections of the vertical and diagonal lines yield conversion factors for calculating the volumetric ratios (in ppm) of CO and NO_x

is the usual reference in industry.

With the enclosed chart it is possible to accomplish this purpose. The chart provides a conversion factor, F, which allows both the lb/h-to-ppm(v) conversion and the reverse.

Underlying theory

If w, in units of lb/h, is the flowrate of NO_x (usually reported as NO_2) in a turbine exhaust stream of W lb/h flowrate, the following equation gives the volumetric ratio on a dry basis. (The molecular weight of NO_2 is 46.)

$$V = [(w/46)/(W/MW)] \times 100/(100 - \%H_2O) \qquad (1)$$

In this equation, $\%H_2O$ is the volume of water vapor in the exhaust stream, and MW is the molecular weight of the gases. V_n must further be corrected for the 15% oxygen, dry basis, as follows:

$$V_n\,[\text{in ppm(v)}] = \frac{10^6 V(21 - 15)}{21 - [100/(100 - \%H_2O)] \times \%O_2} \qquad (2)$$

In this case, $\%O_2$ is the percent volume of oxygen in the wet exhaust gas.

Similarly, the CO emission rate in ppm(v), 15% O_2, dry basis, is:

$$V_c\,[\text{in ppm(v)}] = 1.642\,V_n \qquad (3)$$

Figure 1 solves for the concentrations of both NO_x and CO; the conversion factor, F, is obtained from it as a function of $\%H_2O$ and $\%O_2$. A molecular weight of 28.2 was used for the construction of the chart. Then,

$$V_n\,[\text{in ppm(v)}] = (w_n/W) \times 10^6 \times F \qquad (4)$$

$$V_c\,[\text{in ppm(v)}] = 1.642 \times (w_c/W) \times 10^6 \times F \qquad (5)$$

Sample problem

Determine the NO_x and CO concentrations, in ppm(v), 15%, dry basis, if 25 lb/h of NO_x and 15 lb/h of CO are present in 550,000 lb/h of turbine exhaust gas. The percent volume of H_2O is 10, and the percent volume of O_2 is 11 in the wet exhaust gas.

Solution: From the Figure, F at 10% H_2O and 11% O_2 is 0.46. Then, V_n equals $(25/550{,}000) \times 10^6 \times 0.46 = 21$. Also, V_c equals $1.642 \times (15/550{,}000) \times 10^6 \times 0.46 = 20.6$. ■

When gas streams need to be incinerated, estimating flammability can be difficult. Here are a table and a procedure to help.

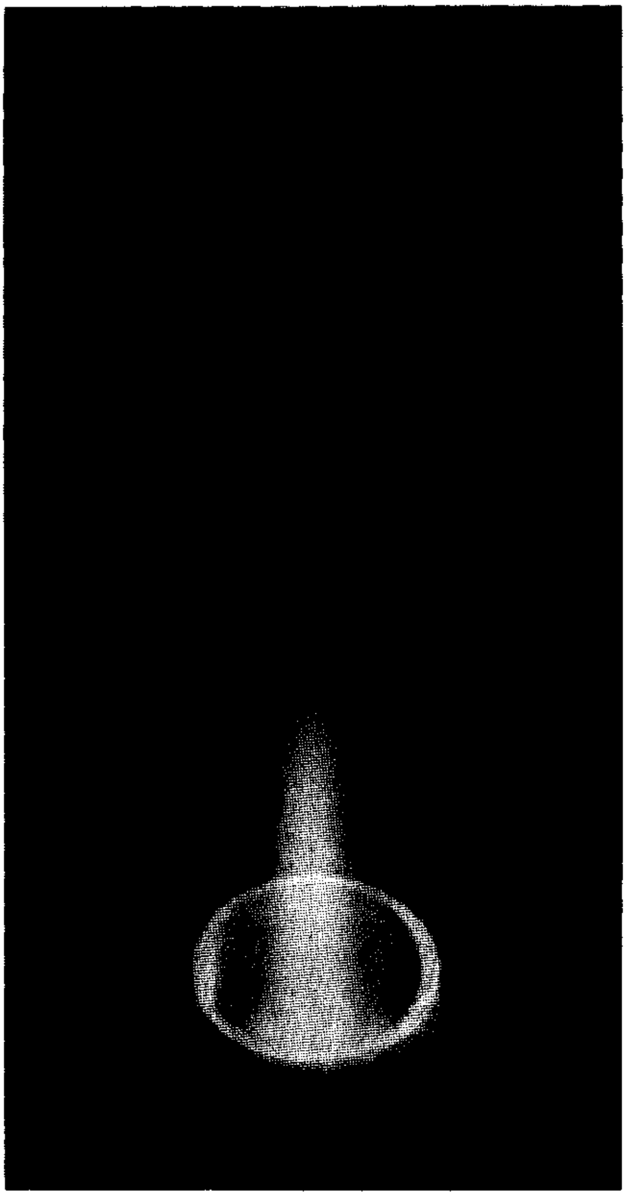
Flammability of low-Btu gases

John J. Clawson
Engineering Consultant

Every engineer whose responsibilities include furnaces, fuels, incinerators or processes involving offgases is occasionally confronted with an offgas or a waste stream that needs to be incinerated. Such streams are sometimes so low in their combustible content that one cannot even be sure that they will burn; in such cases, there is frequently a voice in the background requesting an immediate estimate of the amount and value of primary fuel required to incinerate the gas stream.

Estimating flamability

Accurately estimating the flammability of low-Btu gases has always been a problem. On such occasions, we pull a handbook off the shelf that contains a "Flammability Table," examine the data for several of the most common compounds in the mix, notice that differences between compounds seem radical, then start searching for a common denominator. If some of the items on the list "crack" with an endothermic reaction, the problem becomes even more sizable. If you're unlucky enough to own two handbooks with flammability tables, you may end up wishing that you were on vacation, because references don't always seem to provide the same answers. This article will examine this confusion and attempt to provide a more satisfactory solution.



For constituents undergoing cracking reactions that absorb heat, the heating value of the fuel is:

$((\text{lb carbon/lb fuel} \times 14{,}093\ \text{Btu/lb}) + (\text{lb hydrogen/lb fuel} \times 51{,}623\ \text{Btu/lb})) - \text{heat of formation}$

Thus the low heat value (LHV) of methane is:

$((0.7487 \times 14{,}093) + (0.2513 \times 51{,}623)) - 2{,}005.93 = 21{,}518.4.$

The maximum fluegas quantity or minimum flammability condition for methane is:

lb fluegas/lb fuel $= 21{,}518/(\text{carbon fraction} \times 745) = 38.58$ lb, where the carbon fraction represents active combustion carbon. Oxidized or partially oxidized carbon should be treated as inert in this equation.

For constituents undergoing cracking reactions that release heat, the heating value of the fuel is carbon energy, plus hydrogen energy, plus the heat of formation. Thus, the low heat value of acetylene is:

$(0.9226 \times 14{,}093) + (0.0774 \times 51{,}623) + 3{,}744.25 = 20{,}742,$

and the maximum fluegas quantity, or minimum flammability, for acetylene is:

lb fluegas/lb fuel $= 20{,}742/(745 \times 0.9226) = 30.18$ lb/lb.

Endothermic cracking

In the case of methane and other gases that absorb heat in the cracking process, oxidation will not occur until after the cracking reaction has taken place. Then, if the fluegas enthalpy is, or exceeds, 150 Btu/lb, the hydrogen oxidizes, triggering the carbon reaction. If, after the carbon has been oxidized, the fluegas stream enthalpy equals or exceeds 745 Btu per pound of free carbon, the combustion reaction is self-sustaining. Recognizing that most flammability charts are +/- at least 2.5% of the minimum quantity, this criterion can be easily confirmed within a reasonable tolerance.

Table — Minimum conditions for the combustion of fuel gas

Component	C fraction	LHV, Btu/lb	Maximum pounds fluegas/lb fuel	Min. % by volume	Minimum stoichiometric flame temperature, °F
Methane	0.7487	21,520	38.58	4.56	1,956
Ethane	0.7989	20,432	34.33	2.80	2,128
Propane	0.8171	19,944	32.76	2.02	2,175
n-Butane	0.8266	19,680	31.96	1.58	2,199
Iso-Butane	0.8266	19,629	31.87	1.58	2,200
n-Pentane	0.8323	19,517	31.47	1.29	2,215
Iso-Pentane	0.8323	19,478	31.41	1.30	2,214
Neo-Pentane	0.8323	19,396	31.28	1.30	2,214
n-Hexane	0.8362	19,403	31.15	1.10	2,224
Ethylene	0.8563	20,295	31.81	3.23	2,285
Propylene	0.8563	19,691	30.88	2.24	2,281
n-Butene	0.8563	19,496	30.56	1.71	2,281
iso-Butene	0.8563	19,382	30.38	1.72	2,280
n-Pentene	0.8563	19,363	30.36	1.38	2,280
Benzene	0.9226	17,480	25.40	1.49	2,464
Toluene	0.9125	17,620	25.90	1.24	2,434
Xylene	0.9050	17,760	26.34	1.06	2,411
Acetylene	0.9226	20,776	30.18	3.66	2,478
Naphthalene	0.9371	16,708	23.93	0.97	2,502
Carbon monoxide	---	4347	9.00	10.05	1,706

Note: The air-stream and fuel-stream temperatures are assumed to be 72°F for all calculations.

Exothermic cracking

Those constituents that release energy in the cracking process do not appear to obey the same rules as those that do not. Values in the published literature suggest that those compounds oxidize at a much lower fluegas enthalpy level. It should be remembered that the published flammability charts are basically a safety tool intended to prevent explosions. Their purpose is not to define the minimum fraction of fuel required for complete combustion.

Published flammability levels identify the concentration at which the cracking reaction and hydrogen oxidation begin to occur. A complete oxidation reaction, including the free carbon, occurs at the same enthalpy level stated for the other gases.

Because all components react in the same manner, mixtures also react in like manner, and the minimum combustion-flammability-level of a mixture is:

Max. lb fluegas = (LHV/lb fuel/(carbon fraction × 745).)

The important consideration is the definition of the C fraction. CO_2 is not combustible carbon and should be treated as an inert. CO is not combustible carbon but an entity unto itself. CO will burn when the fluegas enthalpy per pound of CO reaches 550 Btu/lb CO; however, any hydrocarbons in the mixture will be only partially oxidized at best.

Combustion flammability table

The table of combustion flammabilities, accompanying this article, is not intended to replace flammability charts that were designed for safety considerations. Rather, it is a tool for the combustion engineer who is seeking to achieve complete combustion.

The following comments are appropriate in understanding or using the chart:

1. All fuel components are in the vapor state at 72°F. If fuel is in a liquid state, the heat of vaporization must be subtracted from the low heating value of the component.

2. The values for carbon, hydrogen and the heat of formation are taken from a well-known chemical-engineering handbook; however, if one attempts to calculate low heat values from published component values, the low heat values do not always agree with published low heat values in the same handbook. These discrepancies have been reconciled by assuming that LHV values are correct and then adjusting the heat of formation.

3. A loss of energy from the reaction prior to completion of the reaction (i.e., radiation transfer to cold walls, a heat sink, etc.) can prevent the completion of the reaction if conditions are borderline. In those cases, the energy input must be increased to compensate for the loss.

4. The procedure described in this article establishes fluegas enthalpy as the criterion for a complete combustion reaction. Temperature is a result, not a criterion. For example, a fluegas stream resulting from the combustion of CO gas containing a significant amount of water vapor will burn at a temperature well below 1,706°F.

5. The minimum enthalpy value required to maintain combustion can be partially provided by preheating the combustion-air stream or the fuel stream, or both. A preheat enthalpy treatment for either or both streams is, for example, very common when using low-Btu gases for process heating purposes. In such cases, the minimum enthalpy limit can be used to define the minimal supplemental fuel requirement.

Roy V. Hughson, Editor

The author

John J. Clawson, 317 Warwick Road, Ponca City, OK 74601, is an engineering consultant specializing in heat transfer, fired heaters and boilers. Previously, he had been a contract consultant (fired heaters and boilers) to Exxon Research & Engineering Co., senior consultant (heat transfer) to Conoco Inc., senior staff engineer for Arthur G. McKee Co. His education includes a B.S. from Ohio University, the U.S. Steel Combustion Training Program at Carnegie Institute of Technology, and advanced-heat-transfer studies at Akron University. He has also served as a technical consultant to the U.S. Dept. of Energy, is a former chairman of the American Petroleum Institute subcommittee on fired heaters, and is the author and distributor of computer engineering packages on combustion and heat-transfer equipment.

Properties of mixtures via parametric analogy

Matthew Patrick and Kurt F. Kaupisch*

Determining physical-property data is critical in designing chemical processes. For binary mixtures, it is often difficult to find values of physical properties over the range of temperatures and concentrations that will occur throughout a process. This is so even for mixtures of some fairly common compounds.

Quick estimates of properties are needed during the early stages of design. This can often be done, and with reasonable accuracy, by employing what is known as the method of the parametric analogy. This method is based on ideas presented earlier by one of the authors [*1*].

Parametric-analogy method

Assume that we are dealing with a two-component, mutually soluble system. Also assume that we have a curve of a property vs. concentration at one temperature. The curve can be plotted from published data.

Now, assume that, for both compounds, there is no deviant behavior owing to changes in either temperature or *concentration*.

Then, for other temperatures, the shape of the curve will remain essentially constant. The only difference will be in its relative position as temperature varies.

To determine the temperature-dependent positions, use data found in the literature for pure components. That is, the endpoints of these new, shifted curves will be fixed by knowing the properties of the pure components at each temperature.

Example

To illustrate and clarify this method, we shall determine the surface tension of ethanol-water solutions at various temperatures:

Fig. 1 [*2*] presents the variation of surface tension with concentration, at 25°C.

For pure water and pure ethanol, the variation of surface tension with temperature appears in Fig. 2 [*3*]. These two graphs fix the endpoints for the curves developed in Fig. 3. The shape of the curves remains essentially the same; only the endpoints are different.

This method can be used for determining values of many other physical properties, including heat capacity, enthalpy, boiling-point rise and viscosity.

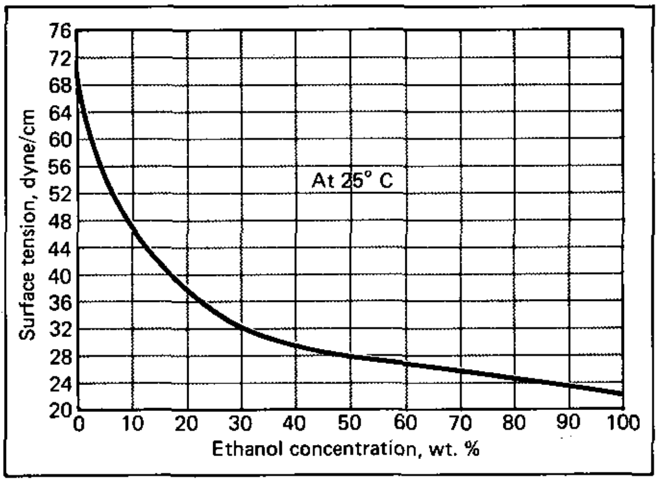


Figure 1 — By using parametric analogy, this graph of surface tension can be extrapolated to other temperatures

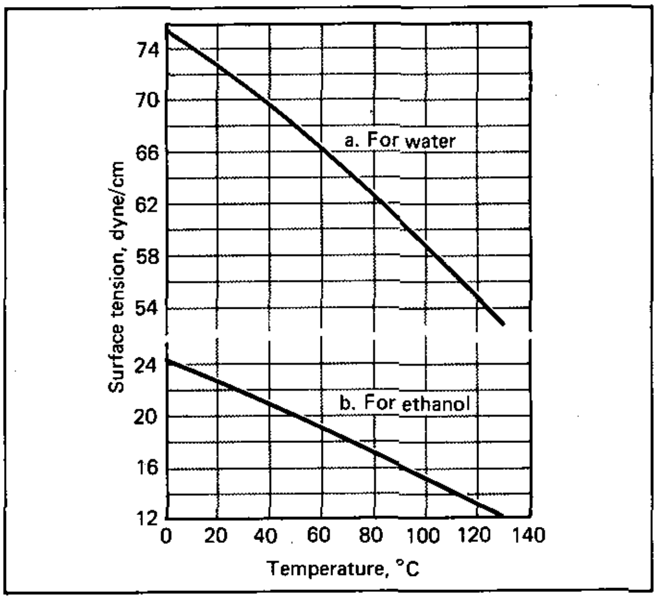


Figure 2 — Plots of the surface tension of water and ethanol are used in Fig. 3 to determine endpoints

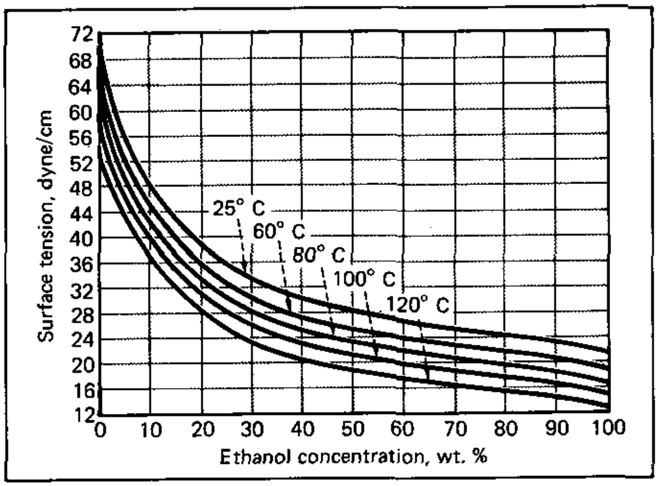


Figure 3 — This prediction of surface tension is accurate enough to use for preliminary design calculations

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2. Commercial Solvents Corp., "Industrial Ethyl Alcohol," p. 96, New York, 1967.
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*Belcan Corp., Engineering Div., 10200 Anderson Way, Cincinnati, OH 45242

Mentally converting from °F to °C

Kazem Abhary*

Here is a simple and accurate method to accomplish this conversion without use of a calculator or paper and pencil. Taking the standard formula:

$$C = (F - 32)/1.8 \tag{1}$$

and substituting 2 for 1.8, the formula becomes:

$$C' = (F - 32)/2 = F/2 - 16 \tag{2}$$

The error in using this approximation is:

$$(C - C')/C' = 0.11 = 11\% \tag{3}$$

This error can be decreased to 1% by using:

$$C'' = C' + 0.1\,C' \tag{4}$$

To illustrate this method, we will convert from 86°F to °C:

1. Divide °F by 2: 86/2 = 43.
2. Subtract 16: 43 - 16 = 27.
3. Add 10%: 27 + 2.7 = 29.7°C.

An exact conversion yields 30.0°C. If the temperature is an odd number, add 1 to it to make it even and follow the above procedure. In this case, adding 1 to the temperature will, to some degree, cancel out the 1% error, making the result more accurate. As an example, convert 93°F to °C:

1. 93 + 1 = 94.
2. 94/2 = 47.
3. 47 - 16 = 31.
4. 31 + 3.1 = 34.1°C.

A more-exact calculation yields 33.9°C.

*Mechanical Engineering Dept., Faculty of Engineering, University of Tehran, Tehran, 14174, Iran

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Estimate relative volatility quickly

M. P. Wagle*

A plant engineer often needs an estimate of the relative volatility of two components. Here is a simple expression that relates the average relative volatility to the normal boiling points and the latent heats of vaporization of the two components, in the temperature range of their boiling points:

$$\alpha = \exp\left[0.25164 \times \left(\frac{1}{T_{b1}} - \frac{1}{T_{b2}}\right) \times (L_1 + L_2)\right] \tag{1}$$

where:

α = relative volatility between the two components in the temperature range T_{b1} to T_{b2}

T_{b1} = normal boiling point of Component 1, K

T_{b2} = normal boiling point of Component 2, K

L_1 = latent heat of vaporization for Component 1 at T_{b1}, kcal/kmole

L_2 = latent heat of vaporization for Component 2 at T_{b2}, kcal/kmole.

If a compound's latent heat is not known, it can be estimated from the normal boiling points and molecular weight, M, using the following equations[†]:

For alcohols and acids:

$$L = T_b\left[19.388 + 3.1269 \log T_b - 6.1589\,\frac{T_b}{M} + 0.035021\,\frac{T_b^2}{M} - (5.1056 \times 10^{-5})\left(\frac{T_b^3}{M}\right)\right] \tag{2}$$

For other polar compounds:

$$L = T_b\left[10.604 + 3.664 \log T_b + 0.09354\,\frac{T_b}{M} + (1.035 \times 10^{-3})\left(\frac{T_b^2}{M}\right) - (1.345 \times 10^{-6})\left(\frac{T_b^3}{M}\right)\right] \tag{3}$$

For hydrocarbons:

$$L = T_b\left\{13.91 + 3.27 \log M + 1.55\,\frac{[T_b - (263M)^{0.581}]^{1.037}}{M}\right\} \tag{4}$$

Example

The average relative volatility of benzene and toluene can be determined using the following data: T_{bb} = 353.3 K, T_{bt} = 383.8 K, L_b = 7,352 kcal/kmole, and L_t = 7,930 kcal/kmole (where the subscripts b and t denote benzene and toluene, respectively). Substituting these values into Eq. (4) above, we find that:

$$\alpha_{bt} = \exp\left[0.25164 \times \left(\frac{1}{353.3} - \frac{1}{383.8}\right) \times (7{,}352 + 7{,}930)\right] = 2.375$$

This compares with a value of 2.421 for α determined using vapor-pressure/temperature charts.

*Curaçao Plein 14, 3193 LK Hoogvliet, the Netherlands.

[†] Reid, R. C., et al., "The Properties of Gases and Liquids," 3rd ed., McGraw-Hill, New York, 1977, p. 214.

Calculate kinematic viscosity of liquid water

A.E. Bastawissi*

Now you can calculate the Reynolds number for flowing water without looking up its viscosity.

An accurate and simple correlation to calculate the kinematic viscosity of liquid-water (ν) is expressed in terms of temperature as:

$$\nu = 0.168\ T^{-0.88} \quad \mathrm{cm^2/s} \quad 25^\circ\mathrm{C} < T < 250^\circ\mathrm{C} \quad (1)$$

where, T = liquid water temperature, °C.

Eq. (1) provides the required kinematic viscosity of liquid water in a one-step calculation at any temperature between 25 and 250°C.

The accuracy of this correlation lies within ±4%, as shown in the table below.

Also, Eq. (1) finds applications in fluid mechanics and heat transfer (e.g. for internal water flow) in calculating the Reynolds number, $N_{\mathrm{Re}} = DU/\nu$, which can be written as:

$$N_{\mathrm{Re}} = 5.95\ DUT^{0.88} \quad (2)$$

Hence, N_{Re} can be calculated on knowing tube diameter D, cm, and measuring flow velocity, U, cm/s, and water temperature (T°,C), without referring to tables to get kinematic viscosity, ν.

Replacement formula

This formula can replace the more complicated one that has been used before [*2,3*]:

$$\frac{1}{\mu} = 2.1482[(T - 8.435) + \sqrt{8{,}078.4 + (T - 8.435)^2}] - 120 \quad (3)$$

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4. Holman, J.P., "Heat Transfer," 4th ed., McGraw-Hill, Kogakusha, LTD, 1976.

The College of Engineering, Mansourah University, El-Mansourah, Egypt

T°,C	30	40	60	80	100	120	140	160	180	200	220
$\nu_1 \times 10^3$	8.42	6.54	4.58	3.55	2.92	2.49	2.17	1.93	1.74	1.58	1.46
$\nu_2 \times 10^3$	8.05	6.59	4.78	3.65	2.95	2.52	2.17	1.91	1.73	1.58	1.48
ν_1/ν_2	1.04	0.99	0.96	0.97	0.99	0.99	1.00	1.01	1.01	1.00	0.99

ν_1 = kinematic viscosity of liquid water from Eq. (1), cm²/s.
ν_2 = kinematic viscosity of liquid water from the literature [*1–4*], cm²/s.

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*Faculty of Engineering, Mansourah University, El-Mansourah, Egypt

Improved Z charts

Peter E. Liley*

In many calculations of the specific quantitity, volume, volumetric flowrate, mass flowrate, etc., of gases, the compressibility factor, Z, is taken as unity. However, conditions are often such that this is not a permissible approximation.

In these cases, Z can be found by: (1) directly looking up actual values in tables; (2) using equations of state; and (3) employing a generalized compressibility-factor chart. The first method requires an extensive library and is time-consuming; the second requires a significant degree of sophistication on the part of the user. Thus, generalized charts find wide use.

Improved charts

Possibly the most widely employed charts are those of Nelson and Obert [*1,2*]. These contain graphs of Z as a function of reduced pressure, P_r, and reduced temperature, T_r, for $0 \le P_r \le 0.1$ and $0.6 \le T_r \le 2.0$ as a small-scale (1×1 in.) inset; for $0 \le P_r \le 1$, $0.6 \le T_r \le 5.0$ and for larger ranges. No saturation-boundary curves are given, and the small-scale inset is difficult to use. In general applications, the author has found the need for a low-pressure/low-temperature chart of "readable accuracy" with an indicated saturation boundary. Also, some of the Nelson/Obert data may not be accurate.

Large-scale chart

In preparing a large-scale chart (see Figs. 1 and 2), argon was chosen as the reference gas. It has a simple molecular

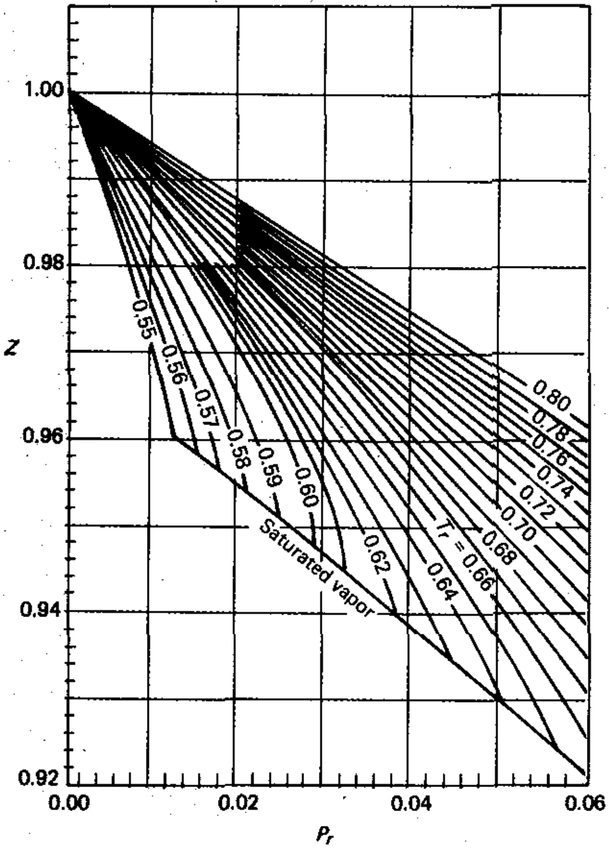

Figure 1 — First large-scale Z chart

Table — Compressibility factor for saturated vapor

As a function of reduced temperature

T_r	P_r	Z
0.55	0.0125	0.9610
0.56	0.0152	0.9590
0.57	0.0180	0.9567
0.58	0.0214	0.9532
0.59*	0.0248	0.9503
0.60	0.0288	0.9469
0.61	0.0322	0.9432
0.62	0.0385	0.9391
0.63	0.0442	0.9343
0.64	0.0506	0.9292
0.65	0.0569	0.9240
0.66	0.0649	0.9186
0.67	0.0728	0.9131

T_r	P_r	Z
0.68	0.0814	0.9064
0.69	0.0911	0.9002
0.70	0.1010	0.8929
0.71	0.1121	0.8850
0.72	0.1243	0.8779
0.73	0.1372	0.8704
0.74	0.1512	0.8611
0.75	0.1662	0.8518
0.76	0.1821	0.8423
0.77	0.1992	0.8321
0.78	0.2174	0.8218
0.79	0.2367	0.8111
0.80	0.2572	0.8003

As a function of reduced pressure

P_r	T_r	Z
0.02	0.5780	0.9543
0.03	0.6027	0.9458
0.04	0.6228	0.9378
0.05	0.6392	0.9298
0.06	0.6536	0.9220
0.07	0.6666	0.9149
0.08	0.6783	0.9077
0.09	0.6887	0.9011
0.10	0.6987	0.8938
0.11	0.7078	0.8867
0.12	0.7160	0.8807
0.13	0.7239	0.8747
0.14	0.7313	0.8686
0.15	0.7386	0.8622

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*School of Mechanical Engineering, Purdue University, West Lafayette, IN 47907

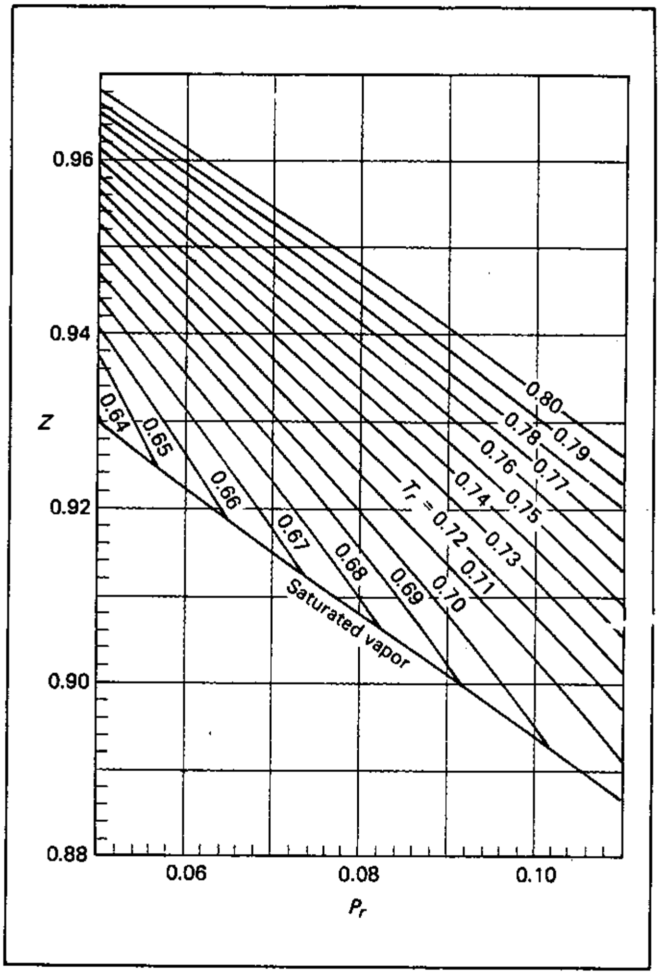

Figure 2 — Second large-scale Z chart

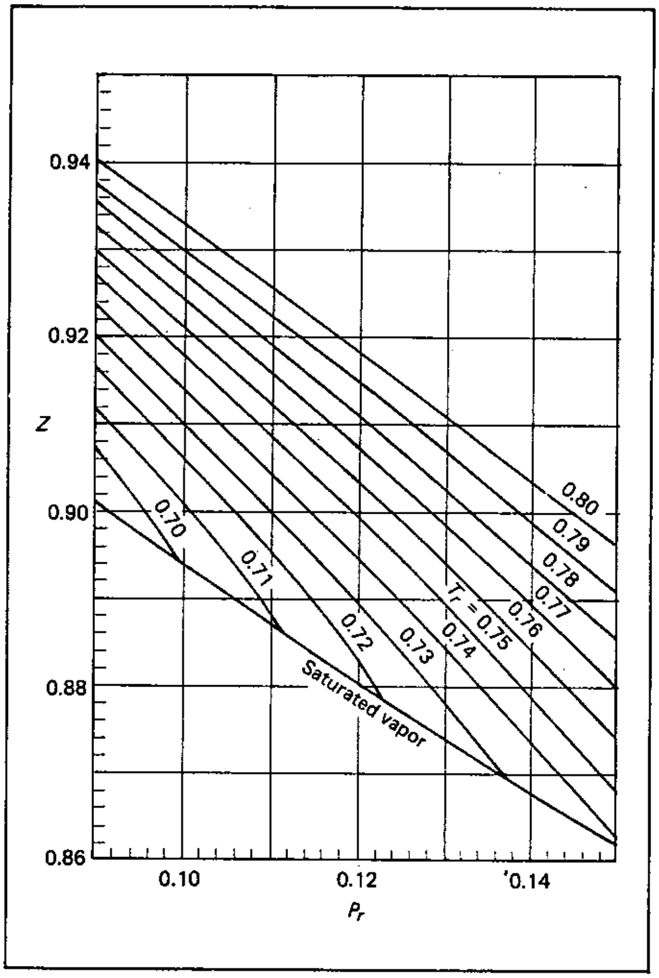

Figure 3 — Third large-scale Z chart

structure and is not subject to appreciable quantum effects. Also, it is a gas that has been well-studied; two relatively recent tabulations of its properties have appeared — the IUPAC tables [3] and the Russian standards tables [4]. Partly because the Rabinovich analysis (which uses generalized coordinates) has been so successful in representing the properties of air and partly because the Russian tables were available to the author, these tables were chosen for our purposes.

By repeated plotting and cross-plotting of the values from Ref. [4] as functions of pressure, temperature, reduced pressure and reduced temperature, the author has been able to prepare a table of Z as a function of P_r ($0 \leq P_r \leq 0.15$) and T_r ($0.55 \leq T_r \leq 0.80$), and for Z for the saturated vapor as a functions of P_r and T_r. The table shown contains values for saturated vapor.

(It also contains data for the extended chart, which is discussed below.)

With the possible exception of values of $T_r < 0.62$., the uncertainty in Z is felt to be ± 0.0005. The large-scale graph has been split into three sections, Figs. 1–3.

For reduced pressures over 0.15, the Nelson and Obert chart for $0 \leq P_r \leq 1$ still appears to be quite accurate, except near the critical point and the saturated region.

Extended Z chart

Fig. 4 presents the extended version of the chart. The small inset is omitted, since the large-scale charts contain that information.

A problem with the original chart was that no explanation was given for the curve marked "Deviation > 1%." To solve this and other problems, argon was chosen as the reference gas once again. By analyzing the extensive saturation-temperature tables [4], the author was able to add lines for liquid and vapor to the original diagram. It was found that the curve marked "Deviation > 1%" fell very close to the curve deduced for saturated vapor.

Little change to the superheat region was found to be

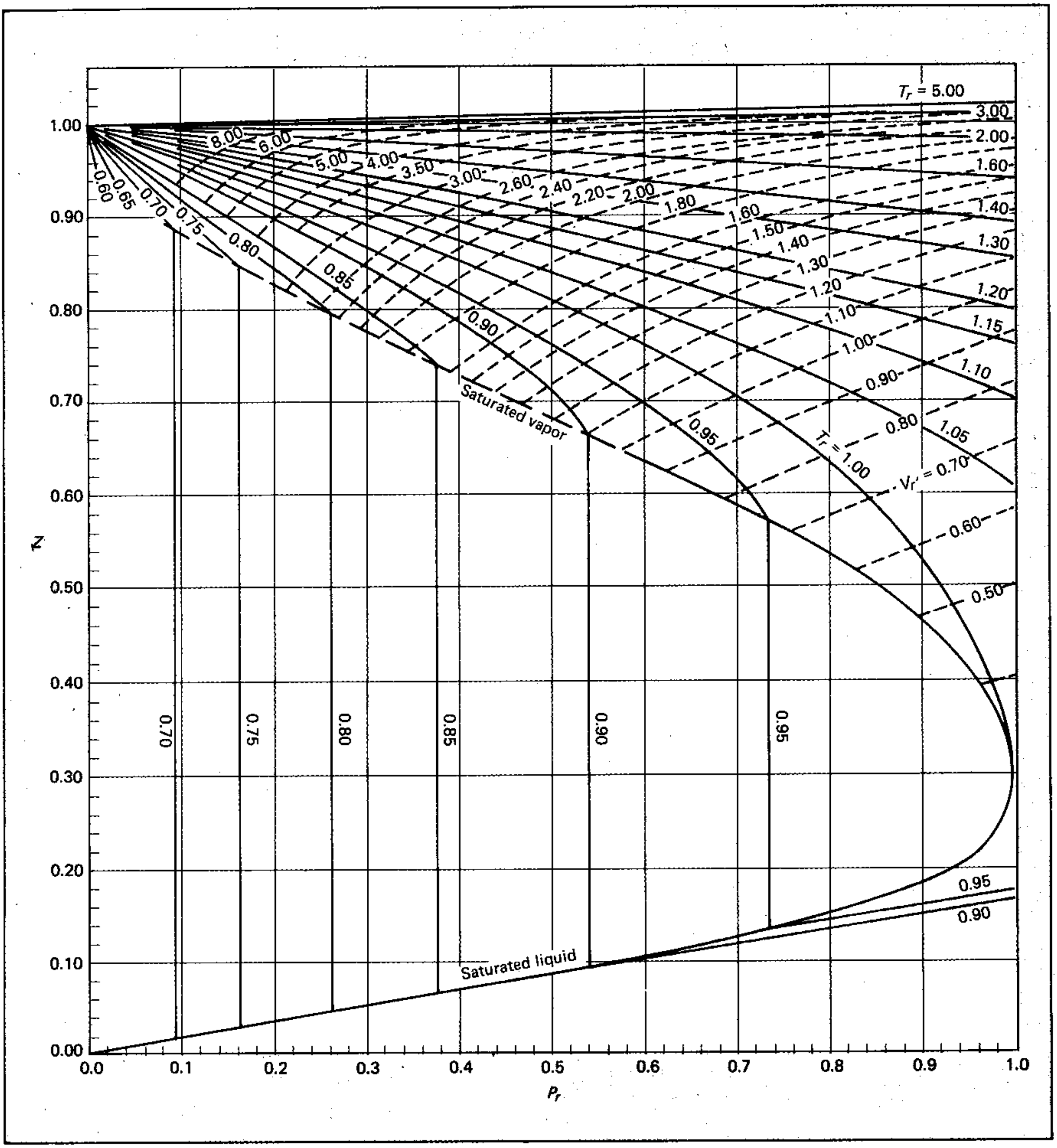

Figure 4 — Extended Z chart

required except near the critical point, $T_r > 0.7$, v_r' (critical reduced volume) < 0.75. Note that critical-region anomalies are not included on the chart.

Values of the compressibility factor for the saturated vapor appear in the table.

The author is indebted to Prof. E. F. Obert for allowing him to reuse the original chart, which was amended as indicated previously.

References

1. Nelson, L. C., and Obert, E. F., Generalized PVT Properties of Gases, *Trans. A.S.M.E.*, Vol. 76, No. 10, pp. 1057-1066, Oct. 1954.
2. Nelson, L. C., and Obert, E. F., Laws of Corresponding States, *A.I.Ch.E. J.*, Vol. 1, No. 1, pp. 74-77, March 1955.
3. Angus, S., and Armstrong, B., IUPAC Chemical Data Series No. 5, "International Thermodynamic Tables of the Fluid State: Argon," 1971, Butterworths, London, 1972.
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Nomograph for moist-air properties

*P. Gandhidasan**

☐ When engineers need to evaluate the values of moist-air properties, they generally refer to psychrometric charts or tables.

However, reading a chart can be difficult and time-consuming, and using tables often requires double interpolation.

The nomograph presented here can be used to estimate several moist-air properties at a glance, if any two pieces of information are known. Standard atmospheric pressure is assumed.

To use the figure, simply connect the two points representing the known quantities, and read the values of the other properties directly from the nomograph.

Example

An air sample has a dry-bulb temperature of 95°F (35°C) and a wet-bulb temperature of 80°F (26.7°C). Determine the relative humidity, the dew-point temperature, the humidity ratio and the vapor pressure.

Solution: On the nomograph, connect $t = 95$°F on the center axis with $t^* = 80$°F on both t^* axes. Read the following approximate property values: $RH = 53\%$; $t_d = 75$°F (24°C); $W = 130$ grains of moisture per pound of dry air (0.0186 kg moisture per kg dry air); and $P = 0.87$ in. Hg (22.1 mm Hg)

Nomenclature

P = Vapor pressure, in. Hg
RH = Relative humidity, %
t = Dry-bulb temperature, °F
t^* = Wet-bulb temperature, °F
t_d = Dew-point temperature, °F
W = Humidity ratio, grains/lb dry air

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*Texas A&M University, Mechanical Engineering Dept., College Station, TX 77843-3123.

Calculate thermal conductivity for unassociated liquids

Clayton P. Kerr*

Here is an equation for calculating the thermal conductivity of many unassociated industrial liquids. It is better than many other methods because it is more accurate and, more importantly, the input data are easy to obtain: parameters such as density, critical temperature, critical pressure, boiling point, molecular weight. It is suitable for either polar or nonpolar liquids, but cannot be used for associated liquids such as water, alcohols, or organic acids.

For spherical unassociated liquids (either polar or nonpolar), the dimensionless thermal conductivity, λ^*, is a function of dimensionless temperature, T^*, density, ρ^*, and dipole moment, μ^*[*1*]. Here, the shape limitation is removed by using the Pitzer accentric factor, ω [*2*].

The Pitzer acentric factor is a property of pure fluids and has been widely tabulated (for example, see Ref. [*2*]). It can be estimated in several ways, one of which [*3*] is given in the Nomenclature.

A regression analysis was performed on the properties of 25 substances, and the following equation was found to yield the best fit (standard error of 4.3%) of the data:

$$\lambda = \frac{T_b^{1/2}\rho^{2/3}}{M^{7/6}}\left[-0.01956 + (9.249 \times 10^{-3}\omega - 8.157 \times 10^{-4})\frac{T_b}{T} + (0.3738 - 0.1746\omega)\frac{T_c\rho}{P_cM} + (1.04\,\omega - 1.537)\left(\frac{T_c\rho}{P_cM}\right)^2 - 1.490\,\frac{\mu^2\rho}{T_bM}\right]$$

Note that this is a *dimensional* equation, so that the parameters used must have the units shown in the Nomenclature listing.

References

1. Hirschfelder, J. O., et al., "Molecular Theory of Gases and Liquids," Wiley, New York, 1967, p. 618.
2. Reid, R. M., et al., "The Properties of Gases and Liquids," 3rd ed., McGraw-Hill, New York, 1977, pp. 19–21, 629–665.
3. Edmister, W. C., Applied Hydrocarbon Thermodynamics, *Pet. Refiner*, Vol. 37, No. 4, April 1958, pp. 173–179.

Nomenclature

M	Molecular weight, g/g-mole
P_c	Critical pressure, atm
T	Temperature, K
T_b	Normal boiling point, K
T_c	Critical temperature, K
T^*	Dimensionless temperature
λ	Thermal conductivity, cal/(s)(cm)(K)
λ^*	Dimensionless thermal conductivity
μ	Dipole moment, debyes
μ^*	Dimensionless dipole moment
ρ	Density, g/cm^3
ρ^*	Dimensionless density
ω	Pitzer acentric factor, dimensionless
k	Boltzmann's constant $k = 3.29943 \times 10^{-24}$ cal/K
N	Avogadro's number $N = 6.023 \times 10^{23}$/mole
R	Ideal gas law constant $R = 1.9872$ cal/(g-mole)(K)

Equations for dimensionless quantities:

$$T^* = T_b/T$$

$$\lambda^* = \frac{\lambda M^{7/6} N^{1/3}}{R^{3/2} T_b^{1/2} \rho^{2/3}}$$

$$\mu^* = \frac{\mu^2 \rho N}{k T_b M}$$

$$\rho^* = \frac{R T_c \rho}{P_c M}$$

$$\omega = \frac{3}{7}\,\frac{T_b/T_c}{1 - T_b/T_c}\log P_c - 1$$

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*Dept. of Chemical Engineering, Tennessee Technological University, Box 5013, Cookeville, TN 38501.

FLOW and STORAGE of LIQUIDS and SOLIDS

An exact solution to the Colebrook equation

Hsi-Jen Chen*

Chemical engineers are familiar with the Fanning (or Darcy) friction factor, f, the Moody chart of f vs. Reynolds number, R_e, and how all of this fits together to calculate pressure drop for a given fluid flow in a given sized pipe. The friction factor is calculated from the Colebrook equation:

$$1/\sqrt{f} = -2 \log[(\epsilon/3.7D) + (2.51/R_e\sqrt{f})] \qquad (1)$$

where ϵ is the absolute pipe roughness, D is the inside diameter of the pipe, and R_e is the Reynolds number for flow. Eq. (1) serves as the basis of the Moody chart. Since the Colebrook equation is implicit in f, it often has to be solved by numerical methods, such as the Newton-Raphson.

For many years, numerous investigators have proposed explicit approximations for Eq. (1), as can be seen in a summary by Serghides [*1*].

Serghides compared nine explicit approximation formulas. Various formulas gave good results but none exactly compatible with the Colebrook equation. Here is a new strategy to solve this equation. The method uses direct substitution repeatedly; the results are quite accurate.

The method

Let us begin with an initial substitution for $2.51/\sqrt{f}$ of Eq. (1). As an estimate, we can use the famous Blasius equation:

$$f/4 = 0.0791/R_e^{1/4} \qquad (2)$$

where 4 converts from Darcy to Fanning factors. Eq. (2) is valid for Reynolds numbers between 2,100 and 10^5 for hydraulically smooth pipe ($\epsilon/D = 0$). This simple equation is useful for making estimates for a range of $0.0178 < f < 0.0467$. Using an average value of 0.0323 for f yields a value for the term $2.51/\sqrt{f}$ of about 14. Combining this value with Eq. (1) gives:

$$1/\sqrt{f} = -2 \log[(\epsilon/3.7D) + 14/R_e] \qquad (3)$$

Substituting Eq. (3) into Eq. (1), we obtain the result of first iteration (where: $A = \epsilon/3.7D$, and $B = 5.02/R_e$):

$$1/\sqrt{f} = -2 \log(A - B \log(A + 14/R_e)) \qquad (4)$$

By repeated direct substitution, we obtain the result of fifth iteration:

$$1/\sqrt{f} = -2 \log(A - B \log(A - B \log(A - B \log(A - B \log(A - B \log(A + 14/R_e)))))) \qquad (5)$$

Although Eq. (5) seems lengthy, it is easy to compute due to its nested form. It is identical to the evaluation of polynomials by Horner's rule—the optimal way to arrange a polynomial for rapid evaluation without performing substantial computations during rearrangement. The evaluation of a polynomial of:

$$P(x) = a_1x^n + a_2x^{n-1} + \cdots + a_{n+1} \qquad (6)$$

requires $n(n + 1)/2$ multiplications and n additions. Horner's rule is achieved by rewriting $P(x)$ as:

$$P(x) = a_{n+1} + x(a_n + x(a_{n-1} + x(\cdots (a_2 + a_1x)))) \qquad (7)$$

which takes only n multiplications and n additions; a_1 to a_{n+1} are constants.

Results

To test the accuracy of Eq. (5), a matrix of 70 points was compared with Eq. (1). This comparison is similar to one by Zigrang and Sylvester [*2*], but also covers the transition region ($2{,}100 < R_e < 4{,}000$). This test matrix consists of seven Reynolds numbers by ten relative-roughness values. The values of R_e are: 2.5×10^3, 4×10^3, 3×10^4, 10^5, 10^6, 10^7 and 10^8. The values of ϵ/D are: 4×10^{-5}, 5×10^{-5}, 2×10^{-4}, 6×10^{-4}, 1.5×10^{-3}, 4×10^{-3}, 8×10^{-3}, 1.5×10^{-2}, 3×10^{-2} and 5×10^{-2}.

The results? In all cases, the accuracy agrees within at least six decimal places with the Colebrook equation. Thus, Eq. (5) is valid for all values of R_e and ϵ/D. In fact, the convergence is so good that any degree of precision can be obtained simply by increasing the number of the iteration.

References

1. Serghides, T. K., "Estimate Friction Factor Accurately," *Chem. Eng.*, Vol. 91, No. 5, p. 63, Mar. 5, 1984.
2. Zigrang, D. J., and Sylvester, N. D., Explicit Approximations to the Solution of Colebrook's Friction Factor Equation, *AIChE J.*, Vol. 28, No. 3, p. 514, 1982.

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*University of New Mexico, Dept. of Chemical and Nuclear Engineering, Albuquerque, NM 87131.

Sizing pipe after steam traps

Michael V. Calogero* and Arthur W. Brooks†

Undersized condensate return-lines create one of the most common problems encountered with process steam traps. Hot condensate passing through a trap orifice loses pressure, which lowers the enthalpy of the condensate. This enthalpy change causes some of the condensate to flash into steam. The volume of the resulting two-phase mixture is usually many times that of the upstream condensate.

The downstream piping must be adequately sized to effectively handle this volume. An undersized condensate return-line results in a high flash-steam velocity, which may cause waterhammer (due to wave formation), hydrodynamic noise, premature erosion, and high backpressure. The latter condition reduces the available working differential pressure and, hence, the condensate removal capability of the steam trap. In fact, with some traps, excessive backpressure causes partial or full failure.

Due to the much greater volume of flash steam compared with unflashed condensate, sizing of the return line is based solely on the flash steam. It is assumed that all flashing occurs across the steam trap and that the resulting vapor-liquid mixture can be evaluated at the end-pressure conditions. To ensure that the condensate line does not have an appreciable pressure-drop, a low flash-steam velocity is assumed (50 ft/s) [*1*].

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*GESTRA, Inc., 385 Prospect Ave., Hackensack, NJ 07601.
†TECHMAR Engineering, Inc., 12 Linden Lane South, Plainsboro, NJ 08536. Mr. Brooks created the nomograph.

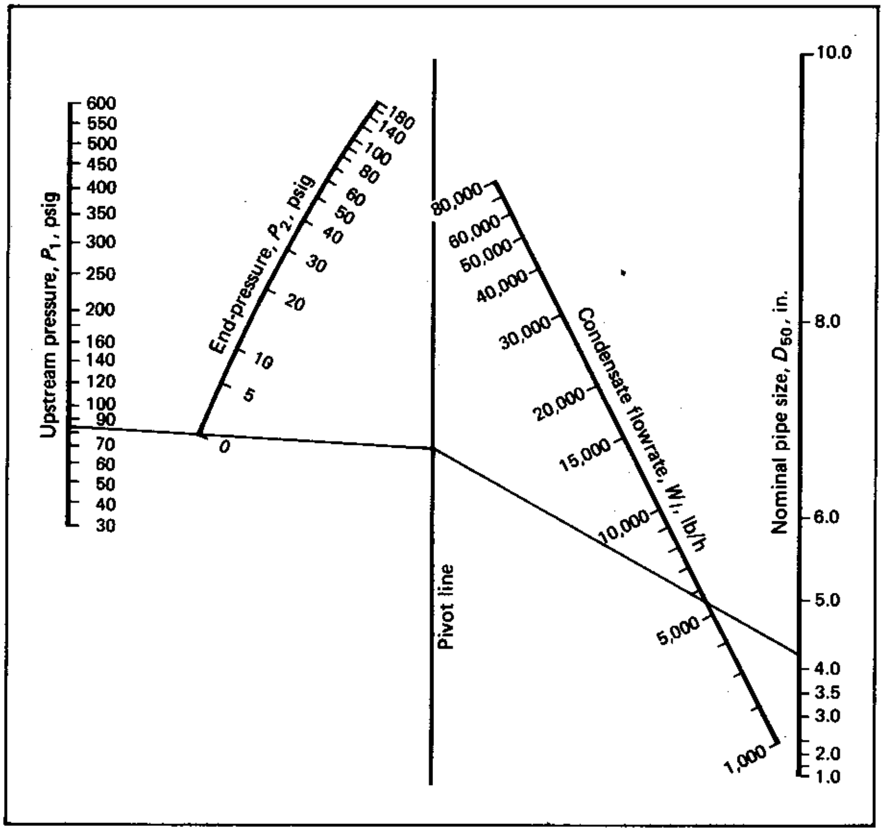


Figure — This nomograph determines recommended pipe size downstream of steam traps

Nomograph

The accompanying nomograph quickly sizes the recommended condensate return-line.

The nomograph employs an enthalpy balance at the upstream and end-pressure conditions to calculate the weight percentage of flash steam that is formed and the flash-steam flowrate.

$$x_{fs} = \frac{(h_{l1} - h_{l2})}{\Delta h_{v_2}} \times 100 \qquad (1)$$

$$W_v = W_l \frac{x_{fs}}{100} \qquad (2)$$

Nomenclature

A_{req}	Required cross-sectional area, ft^2
D_u	Nominal pipe size, based on velocity u, in.
D_{50}	Nominal pipe size, based on 50 ft/s, in.
h_{l1}	Condensate enthalpy at upstream pressure, P_1, Btu/lb
h_{l2}	Condensate enthalpy at end-pressure, P_2, Btu/lb
Δh_{v2}	Latent heat of vaporization at P_2, Btu/lb
Q_v	Flash-steam volumetric flowrate, ft^3/h
u	New flash-steam velocity, ft/s
v_{v2}	Flash-steam specific volume at P_2, ft^3/lb
W_l	Condensate formed at P_1, lb/h
W_v	Flash steam formed at P_2, lb/h
x_{fs}	Flash steam, wt %

The flash-steam volumetric flowrate is then determined:

$$Q_v = W_v v_{v_2} \qquad (3)$$

Based on the assumed velocity, the required cross-sectional area is calculated as:

$$A_{req} = \frac{Q_v}{3{,}600 \times 50} \qquad (4)$$

To simplify the nomograph, the flow area is converted to a nominal pipe diameter. For cases of low pressure drop or high subcooling, it may be necessary to size the condensate line based on the liquid velocity. Generally, a velocity of 3 ft/s is acceptable.

For flash-steam velocities other than 50 ft/s, the nominal pipe size may be approximated by:

$$D_u = \frac{7.07 D_{50}}{\sqrt{u}} \qquad (5)$$

Additional advantages to the nomograph

1. It yields a single result; decision-making is not required, as with other methods.
2. With minimal training, field maintenance personnel can use it.
3. Not only is it applicable to new construction but, more importantly, it can be used to check existing line sizes when trap performance is questionable.
4. It builds on the concept in Ref. [*2*] to provide a complete calculation procedure.

Example

An evaporator is condensing 5,500 lb/h of steam at 150-psig supply pressure. During normal operation, a control valve maintains a pressure of 85 psig upstream of the steam trap. The condensate is returned to a vented tank. What line size is recommended downstream of the trap?

On the chart, connect upstream pressure of 85 psig through vented tank pressure (0 psig) to the pivot line. From pivot line, connect through condensate flowrate of 5,500 lb/h to find 4 in. as the closest size.

References

1. "Condensate Manual 2," Gestra AG, Bremen, West Germany, 3rd ed., 1986, pp. 95–96.
2. Pathak, V. K., How Much Condensate Will Flash?, *Chem. Eng.*, May 14, 1984, p. 89.

A QUICK ESTIMATE FOR CENTRIFUGAL-PUMP EFFICIENCY

Alejandro Anaya Durand,
Instituto Mexicano del Petróleo

To begin the pump-specification process, use this chart and then derive the pump's power rating

In order to calculate the necessary power (i.e., brake horsepower) needed for a centrifugal pump, the design engineer must incorporate the expected pump efficiency in the calculations. This value is provided in the pump performance curve. However, in the early stages of system design, it is convenient to estimate a value for the efficiency. Final values, of course, will depend on the pump that is specified, at the operating conditions that will actually be encountered.

The accompanying chart provides a visual means of obtaining this estimated efficiency. The curves are based on the following equation [1]:

$$E = 80 - 0.2855H + 3.78 \times 10^{-4}QH - 2.38 \times 10^{-7}Q^2H + 5.39 \times 10^{-4}H^2 - 6.39 \times 10^{-7}QH^2 + 4.0 \times 10^{-10}Q^2H^2$$

where

E = pump efficiency, %
Q = flow, gal/min
H = developed head, ft

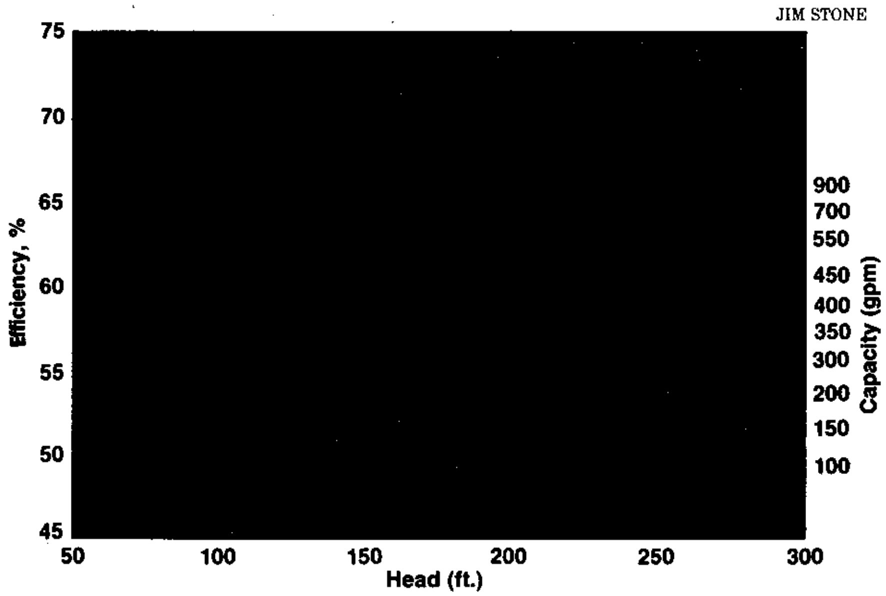
Range of application

The plotted values agree well with reported values from pump-performance curves, especially over the range:

Q = 100 to 1,000 gal/min
H = 50 to 300 ft

For flows in the range 25–99 gal/min, a rough estimate can be obtained using the determined efficiency at 100 gal/min, and then subtracting 0.35 (percentage points) per gal/min, times the difference between 100 gal/min and the low-flow gal/min figure.

JIM STONE



Brake horsepower calculation

To use the derived efficiency value in a brake-horsepower (Bhp) computation, convert it from a percentage figure to a decimal (i.e., 50% = 0.5). Then use the following standard equation:

$$Bhp = \frac{QH \times sp.gr.}{3{,}960E}$$

where *sp.gr.* is the specific gravity of the material being pumped, at the appropriate temperature.

Example: Calculate the pump brake horsepower for a flow of 500 gal/min of water at 60°F (*sp.gr.* = 1), assuming a head of 60 ft.

Solution: From the chart, obtain an efficiency value of 72.5%, convert to 0.725, and calculate:

$$Bhp = \frac{500 \times 60 \times 1}{3{,}960 \times 0.725} = 10.45$$

Reference

1. Branan, Carl, *The Process Engineer's Pocket Handbook*, (Gulf Publishing Co., Houston).

Unusual problems with centrifugal pumps

Solutions to these puzzling pump problems may help you to resolve some of the difficulties that you encounter with pumps.

S. Yedidiah,
Centrifugal Pump Consultant

At first sight, a centrifugal pump seems to be one of the simplest of machines. In practice, however, it is capable of posing an enormous spectrum of different problems [1]. Occasionally, one comes across problems that seem to defy everything we know about centrifugal pumps. We shall report on four such cases.

Case No. 1 — The twin pumps

In a chemical process plant, two identical pumps were installed side by side, to transfer liquid from the same source into the same pressurized container. Each pump had been provided with a separate suction and a separate discharge line. Also, the two pumps were never used simultaneously. While one ran, the other served as a standby.

Everything about these two systems seemed to be identical — except that one pump performed perfectly, whereas the second operated with great noise and vibration. The troublesome pumping system was dismantled several times, but nothing wrong could be found.

The author realized that the successful pump's pipe loop had a 2-in.-dia. discharge, with a reducer connecting directly to a 1½-in. pipeline. However, the troublesome loop had a 6-ft length of of 2-in.-dia. pipe connected to the pump's discharge, and only after this length was the line reduced to 1½ in.

When the 2-in.-dia. section of pipe was replaced by 1½-in. pipe, the problem-causing pump operated satisfactorily.

The reason can be explained by referring to Fig. 1. Up to a certain critical flowrate, Q_c, the net-positive-suction-head (NPSH) requirements of a centrifugal pump increase, approximately, as the square of the flowrate [2]. Above Q_c, however, the NPSH requirements start to increase at a much faster rate — shown schematically in Fig. 1.

In the particular case described above, the frictional losses in the pipelines constituted a very significant part of the total head against which each of these two pumps had to operate. In the discharge line that consisted exclusively of 1½-in.-dia. piping, the resistance to the flow was adequately high. This kept the total head — against which the pump had to operate — well above H_c, the critical pumping head. This, in turn, limited the flowrate to well below Q_c. In the other pipeline, however, the reduced resistance of the 2-in.-dia. pipe section, brought down the total head to well below H_c. This increased the NPSH requirements of the pump well above the available NPSH. Consequently, cavitation developed within the pump. This, in turn, gave rise to the noise and vibration.

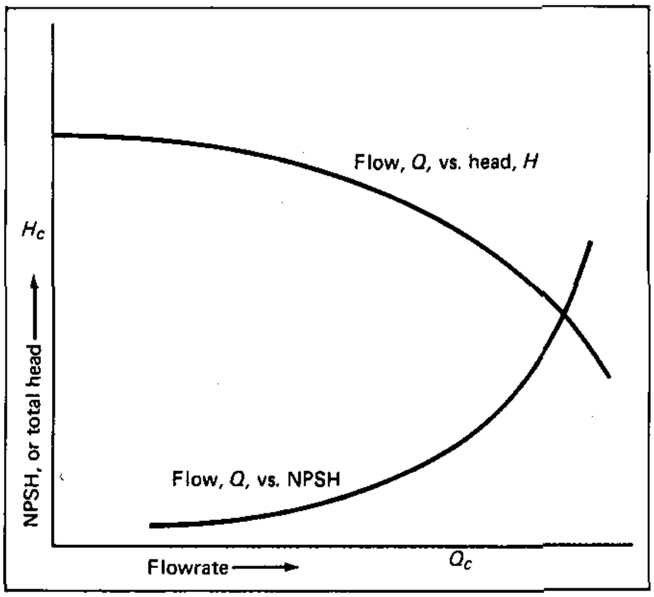


Figure 1 — Effect of flowrate on a pump's NPSH requirements

Case No. 2 — Head in a storage tank

A somewhat related case was encountered with a pump installed in an oil depot. The pump transferred fuel from a storage tank to oil-delivery trucks. Whenever the oil level in the storage tank was low (i.e., when the available NPSH was low), the pump operated satisfactorily. However, when the storage tank became full (i.e., when the available NPSH was high), the pump operated with extreme noise and vibration.

The total head against which a pump operates is defined as the *difference* between the total head existing at a pump's outlet, and the total head available at its inlet. In this particular application, the discharge head tended to be practically constant. This meant that an increase in the available NPSH automatically reduced the total head against which the pump had to operate.

Thus, in accordance with what has been explained in connection with Fig. 1, the author concluded that when the storage-tank was full, the pump operated well below the critical head, H_c. This, however, meant that the pump delivered a flowrate significantly higher than Q_c. However, the flowmeter that was installed in the pipeline, as well as measurements of the time required to fill a a fuel-truck's tank, indicated that the flowrate was well below Q_c. There seemed to be no solution to the puzzle.

However, the author recalled a study he had made earlier on the effects of flowrate on NPSH requirements [3]. According to that study, the most important factor that determines the NPSH requirements of a pump at a given speed is the rate of flow through the impeller.

The flowrate through an impeller is usually slightly greater than that through the pipeline, owing to leakage through the wearing rings. However, if a wearing ring is missing, this short-circuits the impeller discharge to the impeller eye. In such a case, the flow through the impeller may easily be 30 to 40% higher than the flow through the pipeline. In our particular case, this would have brought the total flow through the impeller well above the critical flowrate, Q_c.

The pump was opened, and the front wearing ring was found to be missing. A new wearing ring was installed in the casing, which solved the problem.

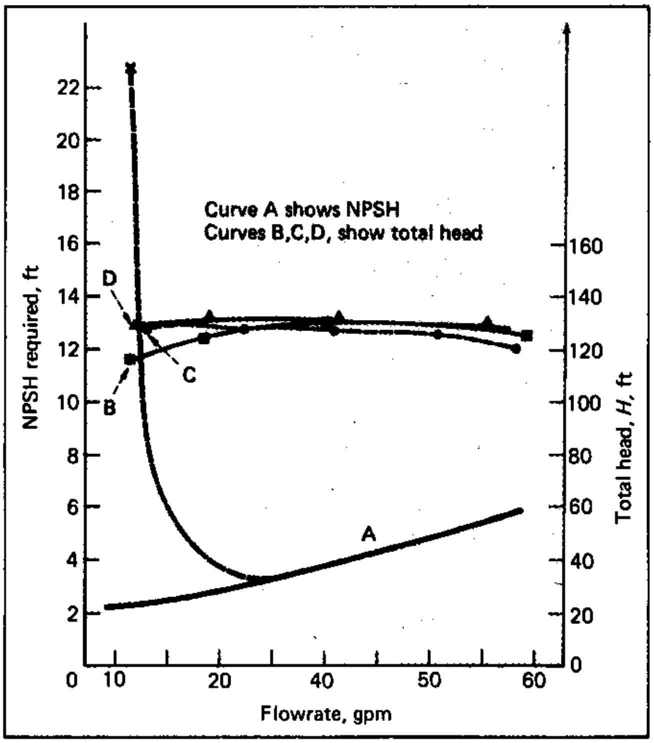


Figure 2 — Effect of length of balancing holes on the suction performance of a centrifugal pump

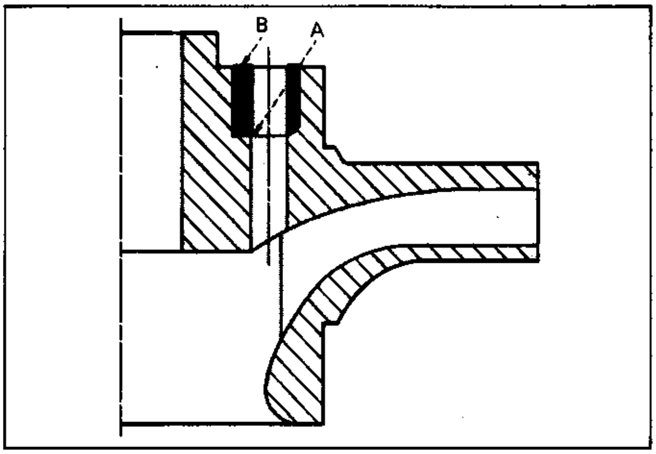


Figure 3 — Modification of impeller hub in tested pump

Case No. 3 — A recessed-hub impeller

A pump that had operated satisfactorily at the NPSH values presented in Fig. 2, Curve A, showed satisfactory performance at 40 gpm, when operating at an available NPSH of 5 ft. The same pump, however, continually failed at much lower flowrates (Curve A, dashed line), although previous tests (Curve A, solid line) showed that it could operate, at these flowrates, at NPSH-values significantly lower than the available 5 ft. Its performance at these times is shown by Curve B in Fig. 2 [4].

In order to eliminate this problem, the impeller was replaced by another of identical design. This time, the pump produced satisfactory results (Fig. 2 Curve C). But the question still remained: Why did the first impeller fail?

A close inspection of the two impellers revealed that the failing impeller had a recessed hub, as shown in Fig. 3 at A, while the second impeller had a solid hub (with no recess). (The same series of tests was repeated with a second, identical pump, with the same results.)

In order to verify whether this difference in the castings was the real cause of the observed differences in performance, the recess of one of the impellers was filled with epoxy and redrilled, as shown in Fig. 3 at B. This immediately restored the full suction capability of the pump, as seen by the test results presented in Fig. 2 by the dashed Curve D.

The tests were performed on pumps ordered by a customer, and there was no time to find out what caused the above effect. It seems, however, that the balancing holes acted here as resonators. The natural frequency of such resonators depends upon their length. Most probably, the shortening of the balancing holes brought their natural frequency in unison with the periodical shedding of vortices, which are known to appear at low partial flowrates. This, in turn, caused the early appearance of cavitation.

This problem occurred at a pump-manufacturer's plant [4], but such a case can easily occur in the field when a worn-out impeller is replaced by a new one. This is especially true when the spare part has not come from the original source.

Case No. 4 — The inconstant pump

This problem, too, was encountered at a pump factory [5], and is also one that can sometimes occur in the field.

A pump designated to operate at 50% of its best efficiency point, when provided with 3 ft of NPSH, was tested at a constant value of NPSH and found to operate perfectly. The pump test was then rerun in the presence of a witness. This time, however, the pump failed completely.

The pump was tested still another time, this time at the constant specified flowrate and variable NPSH. The test produced a two-level curve—Curve A of Fig. 4 [5].

In view of the unsatisfactory and confusing results obtained from the tested pump, both the installation and all test procedures were carefully scrutinized. However, there was no clue as to the source of the observed inconsistencies. As a final check, the entire loop was put under vacuum, and held for 20 min. The whole pumping system was proven to be adequately airtight.

As a last resort, it was decided to find if the problem could have been due to air dissolved in the pumped liquid.

A vacuum pump, located at the top of the airtight suction tank, was allowed to run and, with the aid of regulating valves, air removal from the tank was adjusted to such a rate as to keep a constant vacuum of 15 in. Hg.

After holding this vacuum for about 15 min, to allow a part of the dissolved air to escape, the pump was tested again. This test produced satisfactory results (Fig. 4, Curve B).

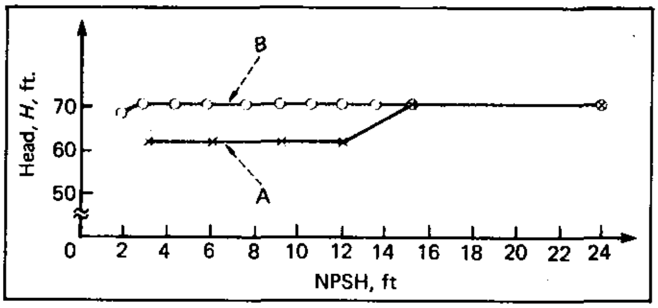


Figure 4 — Time effects of dissolved air on pump's NPSH

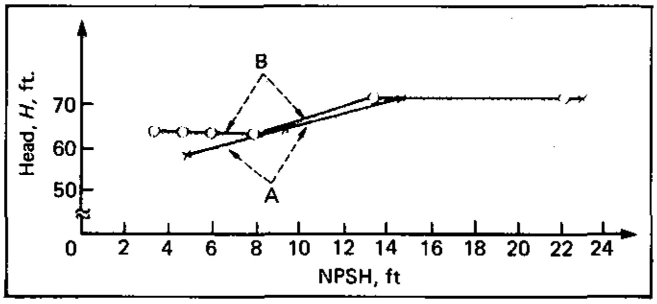


Figure 5 — How dissolved air caused still another failure

Using the same procedure, the test was repeated in the presence of a witness, and the pump failed again! This time, the head vs. NPSH curve had a shape as shown in Fig. 5 by Curve A. A series of additional tests were then carried out, but the results were never consistent. In one particular case, they even produced a curve such as that in Fig. 5, Curve B.

After a significant amount of testing, the answer to the mystery was finally found.

The observed inconsistencies in the test results turned out to be actually due to the air dissolved in the liquid. In addition, however, they were found also to depend upon the number of steps in which the NPSH was lowered to 3 ft, as well as upon the time lapse between two sets of readings.

The first (preliminary) test was carried out at a constant NPSH, and varying flowrate. The tank had been evacuated to 27 in. Hg, and kept under that vacuum until the test engineer was ready to make the readings. This took enough time to allow the vacuum pump to remove any excess of air that had been liberated from the water. Consequently, the pump performed as expected.

With regard to the tests at constant flowrate and varying NPSH, the time effect can be explained as follows:

The initial removal of air was carried out at a vacuum of 15 in. Hg. Consequently, below this pressure, additional air was being liberated from the liquid; this air appeared as a mass of tiny bubbles, dispersed throughout the volume of water. This lowered the specific weight of the pumped mixture.

The amount of air liberated each time the pressure had been lowered depended, of course, on the magnitude of the drop in pressure: The greater the reduction in pressure, the more air was liberated.

When the reduction in pressure was carried out in many small steps, as in the case of Fig. 4, Curve B, only a small amount of air was liberated with each step. This allowed enough time for the vacuum pump to remove the liberated air during the periods between the measurements. Therefore, the centrifugal pump produced satisfactory results.

When the measurements were carried out in large steps, as in Fig. 5 (Curve A), the rate at which the air was liberated from the water exceeded the rate at which it was removed by the vacuum pump. This resulted in a gradual increase in the amount of free air dispersed throughout the pumped liquid, and caused a continuous drop in pressure.

Sometimes, at an intermediate number of steps, a state of equilibrium occurred between the rate at which air was liberated from the liquid, and the rate of air removed by the vacuum pump. In such a case, the pressure readings remained constant, but were lower due to either a reduction in the specific gravity of the liquid caused by the presence of air bubbles, or partial air blockage of the impeller inlet, or both [*1*]. This produced a bilevel curve, like Curve A in Fig. 4.

In the case shown in Fig. 5 Curve B, the available NPSH was lowered, in large steps, down to about 8 ft. This caused the air to be liberated from the pumped water at a higher rate than that which was removed by the vacuum pump. This caused a significant drop in the total head. Below 8-ft NPSH, however, the mechanic started to reduce the available NPSH in significantly smaller steps. This enabled the pump to start to recover a part of the lost head.

***Roy V. Hughson**, Editor*

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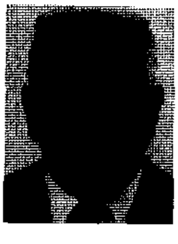


The author

S. Yedidiah is a centrifugal pump consultant, 89 Oakridge Rd., West Orange, NJ 07052, Tel: (201) 731-6293. He has been an active pump specialist since 1938, and has worked for a number of pump manufacturers, including Worthington Pump Inc. He has published many articles, and is the author of "Centrifugal Pump Problems—Causes and Cures," published by PennWell Books. He is a member of the American Soc. of Mechanical Engineers, and is a registered professional engineer.

Check pump performance from motor data

V. Ganapathy*

Plant engineers often need to figure the flow or head of pumps or fans. Such information is needed to check performance or to see if meter readings are accurate; for meters, the flow orifice might be plugged or the meter might be incorrectly calibrated.

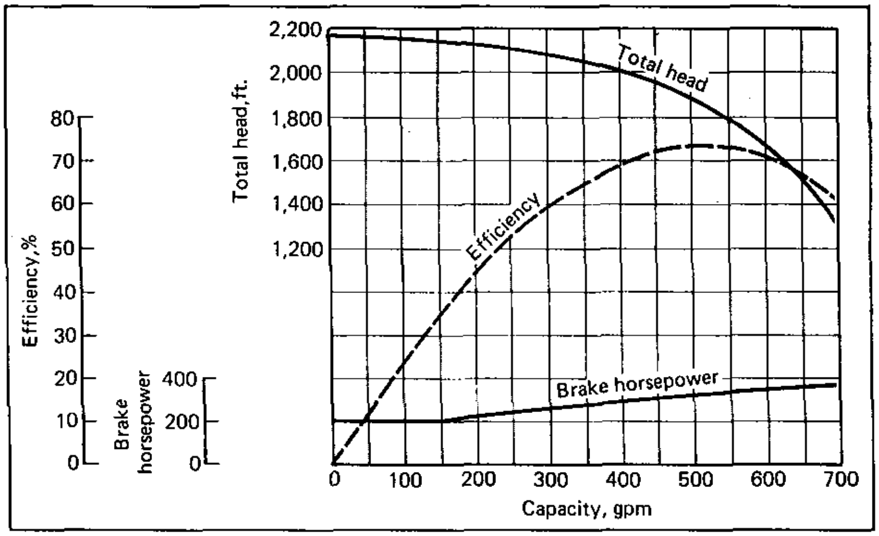


Figure 1 — Typical characteristic curve for a multistage centrifugal pump used to prepare table

Motor data

Electrical-current readings provide a good indication of the flow. The following analysis is developed for pumps, but it may also be extended to fans, since they have similar characteristics.

The power consumed by a pump is:

$$B_{HP} = Q\Delta P/(1{,}715E_p) \qquad (1)$$

The head may be related to the differential pressure by:

$$H = 144\Delta P/\rho \qquad (2)$$

Converting Eq. (1) into kilowatts:

$$P = 0.00043Q\Delta P/E_p \qquad (3)$$

The power output of the motor is given by:

$$P = 0.001732EI\cos\phi E_m \qquad (4)$$

Equating Eqs. (3) and (4):

$$Q\Delta P/E_p = QH\rho/144E_p = 4.03EI\cos\phi E_m \qquad (5)$$

Thus, pump data can be related to motor data.

Comments

The following should be kept in mind:

1. Motor efficiency does not vary much with load. The variation is probably 2–5% for medium-sized motors (50–500 hp) and for practical purposes can be considered constant. So can the power factor and voltage.

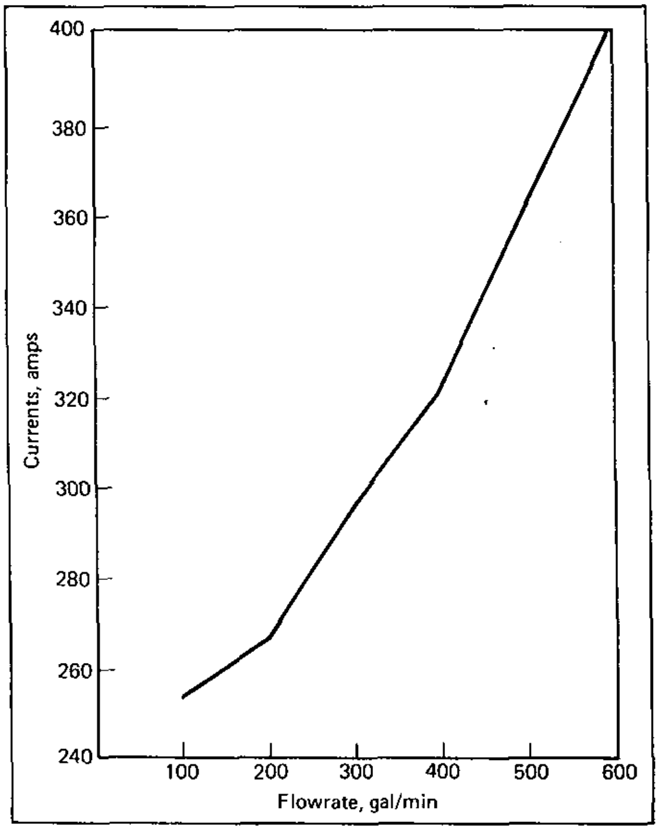


Figure 2 — Using motor data, this plot can be constructed and then used to check pump and meter performance

Nomenclature

B_{HP}	Brake horsepower, hp
E	Voltage
E_p, E_m	Efficiency of pump and motor, fraction
H	Head, ft
I	Current, amps
P	Power consumed, kW
Q	Flowrate, gal/min
	Greek letters
ΔP	Pump differential, psi
ρ	Density, lb/ft^3
$\cos\phi$	Power factor

*ABCO Industries, Inc., 2675 E. Highway 80, Abilene, TX 79604.

Table — Typical operating data for a pump

Q, gpm	H, ft	E_p	QH/E_p	I, amp
100	2,150	0.23	935,000	254
200	2,130	0.43	991,000	269
300	2,100	0.58	1,090,000	296
400	2,000	0.68	1,180,000	321
500	1,880	0.71	1,330,000	361
600	1,690	0.70	1,450,000	394

2. The head, H, and efficiency, E_p, of the pump vary parabolically — see Fig. 1 for a typical pump curve. For a given fluid and flowrate, $QH\rho/144E_p$ is a curvilinear function of Q. Thus, Eq. (5) may be written as:

$$f(Q) = f(I) \tag{6}$$

For a given pump and motor, if E, $\cos\phi$ and E_m are known, Q can be determined as a function of current.

For the pump curve in Fig. 1, we shall determine Eq. (6). First, a table is made of Q, H, and E_p. QH/E_p is calculated (see the table). Assume $E = 460$ V, $E_m = 0.95$, $\cos\phi = 0.9$ and $\rho = 62$ lb/ft^3. Substituting these data in Eq. (5):

$$QH/E_p = 4.03 \times 460 \times 0.9 \times 0.95 \times 144/62 = 3{,}680I \tag{7}$$

By plugging in values from the table, I can be found as a function of Q. This is shown in the table and plotted in Fig. 2.

Using Fig. 2, one can easily relate the flowrate (or head or pump efficiency) to motor current. This is a convenient way of checking the performance of a pumping system or of double-checking meter readings. Thus, having a relationship between flow and current permits monitoring of any significant deviations. For example: What if at some flowrate, the meter shows a current reading that differs from that given by the curve for the same flowrate? Then either the flowmeter has to be checked (if the variation is greater than about 5%) or the pump has to checked for misalignment, overheating, etc. (The same method may, of course, be used on fans.)

The equations take into account friction and heat losses (of both the pump and motor).

Shortcut methods for determining optimum line sizing in SI units

Norbert J. Bell*

This article offers shortcut equations and nomographs that determine exact optimum pipe sizes, and approximate values (usually, within 10%) of the pressure gradient for fluids under turbulent-flow conditions.

The pressure drop

The basic pressure-drop equation (the Darcy formula) for flow through a pipe can be written as:

$$\Delta p = 500 fL\rho v^2/d \qquad (1)$$

The usual practice is to work in terms of the pressure gradient, which is the pressure drop per linear unit of pipe. For instance, substituting $L = 1$ m into Eq. (1) yields:

$$\Delta p_m = 500 f\rho v^2/d \qquad (2)$$

The task of the piping designer is to size the line so that the flow of fluid will create a pressure gradient that is near the economical optimum. The economic optimum conditions take *into account both the capital cost of equipment and operating* costs of pumping, energy, etc.

For any given flow, the optimum pipe diameter, d_{opt}, will be related to optimum pressure gradient by modifying Eq. (2) so that both variables d and Δp_m are at optimum conditions.

Design values for both optimum pressure gradient and the velocity limit for a new petroleum refinery are given in the table. However, these values may be used for other chemical processing applications as well.

In many instances, designers try to stay below these pressure-gradient criteria. This is really not necessary; rather, "coming close to the optimum pressure gradient" is better.

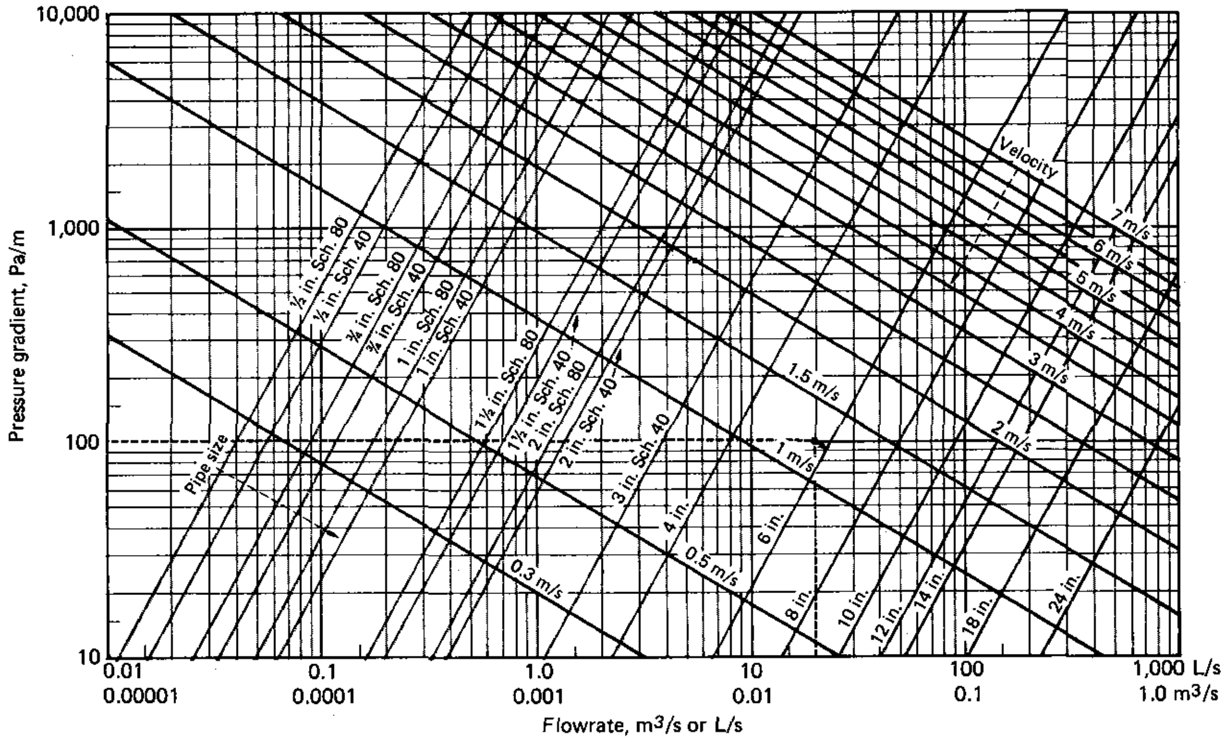
Therefore, one should select the pipe size that — after satisfying the necessary structural requirements — has the standard inside diameter closest to d_{opt}. The resulting pressure gradient will be closest to the optimum pressure gradient. But, in some cases, the actual pressure gradient could somewhat exceed the calculated optimum pressure gradient.

Fluid velocity

For circular pipes, the average velocity is:

$$v = 1.273 \times 10^6\, q/d^2 \qquad (3)$$

Pipe designers normally work with velocities that will not create undue vibrations, and erosion in the piping. Thus velocity limits, such as those provided in the table, are used as a further criterion for optimum pressure gradients in pipe sizing.



Note: Values for pressure gradient are valid only for water at μ = 1(mPa)(s) and ρ = 1,000 kg/m^3.
For fluids at other conditions, use Fig. 2 to determine the correction factor.

Figure 1 — Determine optimum flow conditions from this chart, but for fluids other than water, use correction factor from Figure 2

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*C F Braun & Co., 1000 South Fremont Avenue, Alhambra, CA 91802-3900.

Shortcut pipe-sizing equation

Substituting Eq. (3) into Eq. (2), and solving for d_{opt}, we get:

$$d_{\text{opt}} = 959\,[f\rho q^2/\Delta p_m)_{\text{opt}}]^{0.2} \qquad (4)$$

To use this equation, an estimate of the friction factor is needed. For turbulent flow in clean steel pipes, Simpson† has suggested:

$$f = 0.216/Re^{0.2} \qquad (5)$$

where $Re = \rho dv/\mu$.

Nomenclature

D	Nominal pipe size, in.
d	Pipe inside diameter, mm
f	Friction factor, dimensionless
L	Straight pipe length, m
Δp	Pressure drop, Pa
Δp_m	Pressure gradient, Pa/m
q	Volumetric flowrate, m^3/s
Re	Reynolds number, dimensionless
v	Average fluid velocity, m/s
μ	Dynamic viscosity, (mPa)(s)
ρ	Fluid density, kg/m^3
Subscript	
opt	Economical optimum

As shown in Fig. 1, the optimum line to carry 0.02 m^3/s (20 L/s) of water at a pressure gradient of 100 Pa/m comes out to a nominal size of 6 in. (152 mm). For convenience, a scale for flowrate in L/s has been added to allow input in these flow units. Also, velocity-limit lines have been plotted.

In order to use Fig. 1 to determine pressure gradient for fluids other than water, it will be necessary to use a pressure-gradient multipler (PGM), according to the following equation:

$$\Delta p_m = (\Delta p_m)_{\text{water}}\mu^{0.2}(\rho/1000)^{0.8} \qquad (7)$$

Table — For design purposes, it is not *essential to "stay below" the pressure-gradient criteria*

Liquid	Type of line	Optimum pressure gradient, $(\Delta p_m)_{\text{opt}}$, Pa/m	Velocity limit, m/s
Water	Header	100	4-5
	Lateral	325	4-5
Hydrocarbons	Pump suction, flooded*	95	$(D/6) + 1$, maximum = 2.5
	Pump suction, lift	75	1.3
	Pump discharge, ≤ 1½ in.	800	$(D/6) + 2$
	Pump discharge, 2-3 in.	600	$(D/6) + 2$
	Pump discharge, ≥ 3½ in.	400	$(D/6) + 2$, maximum = 4

*The pump suction is considered flooded when the available NPSH (net positive-suction head) is 7 m or greater.

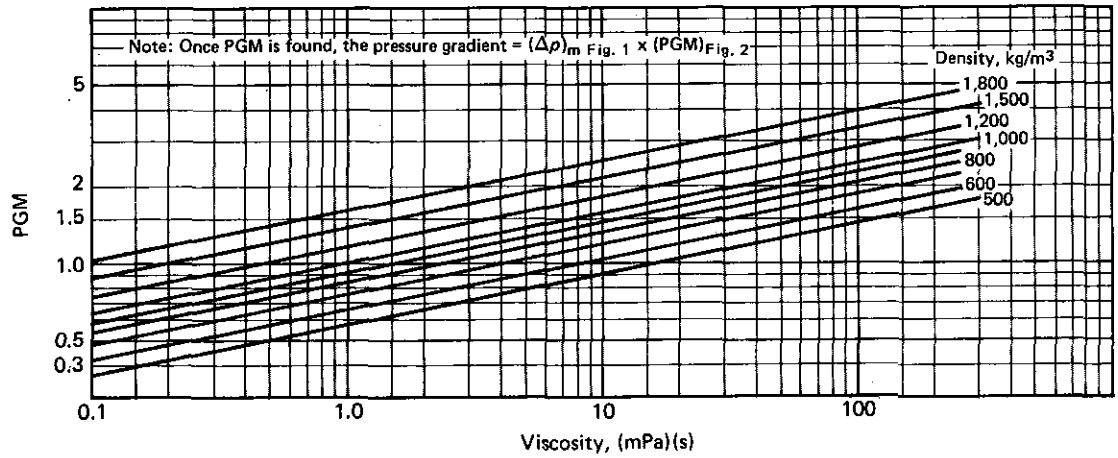

Figure 2 — Use this chart to find the correction factor for pressure-gradient values obtained from Figure 1

Subsituting Eq. (5) into Eq. (4) provides a shortcut pipe-sizing equation:

$$d_{\text{opt}} = 516\,\mu^{0.0417}\rho^{0.167}q^{0.375}/[(\Delta p_m)_{\text{opt}}]^{0.208} \qquad (6)$$

Shortcut nomographs

Nomographs based on Eq. (6) can be constructed that permit quick sizing of pipes for different liquids. For instance, Fig. 1 can be used for water at normal temperatures, when the fluid density is 1,000 kg/m^3, and the viscosity is 1 (mPa)(s), which is also equivalent to 1 cP. To assist in using this equation quickly, another nomograph (Fig. 2) was constructed, in which $\text{PGM} = \mu^{0.2}(\rho/1000)^{0.8}$. Therefore, in order to find a pressure gradient for any fluid whose viscosity is not equal to 1 (mPa)(s) and/or whose density is not 1,000 kg/m^3, we must multiply the values from the two nomographs:

$$\Delta p_m = (\Delta p_m)_{\text{Fig. 1}} \times (\text{PGM})_{\text{Fig. 2}} \qquad (8)$$

For example, for a fluid having a viscosity of 10 (mPa)(s) and with a density of 1,200 kg/m^3, we find from Fig. 2 that PGM = 1.82.

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†Simpson, L. L., *Chem. Eng.*, April 14, 1969, pp. 167-181.

Operating performance of reciprocating or positive-displacement compressors

Michael Polonyi*

Air pressure increases when air flows into a system, and decreases when air flows out. Using this principle, here is a simple method to determine air-compressor performance and compressed-air load for plants using reciprocating or positive-displacement compressors (see figure). Accuracy for this test is approximately 10%.

This technique has proven useful in cases where (1) there is no flowmeter for the compressor; (2) the compressor is first put into service; (3) the unit is running for a longer period than normally expected, or does not stop (in such cases, there is usually no way of telling if this condition is due to higher air consumption by the system, or lower compressor yield); and/or (4) the plant air load is not known.

Plant-air or service-air applications are almost exclusively supplied by using reciprocating or positive-displacement compressors. Since these compressors pump an almost constant amount of air, they must exceed the needed plant-air consumption in order to maintain the required air pressure.

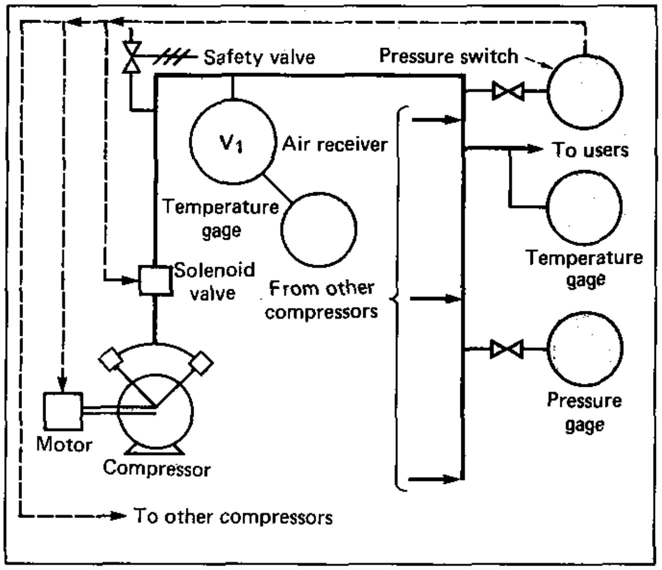

Figure — This compressor-performance test can be run for any given pressure interval or time duration

Accordingly, air pressure varies between two setpoints. At the lower setpoint, the compressor starts pumping, and at the higher pressure, it stops. These setpoints are adjusted via an air-pressure switch, but they cannot be too close — otherwise the compressor would start and stop too often. On the other hand, if the setpoints are too far apart, pressure fluctuations would be too great. Normal values are in the range of 10% to 20%. However, in order to perform the test, it is not necessary to wait to reach these setpoints (the extremes of the system). Any pressure or time interval in which data can be taken accurately will suffice.

If the volume of the system is known, one can measure the time interval it takes for the pressure to reach a specific value. Then apply the following equation:

$$\text{Flow} = \text{Volume} \times (P_1 - P_2)/P_0 \times \text{time}$$

where: Volume = System volume
P_0 = Discharge (atmospheric) pressure
P_1 = Initial pressure
P_2 = Final pressure
time = interval to reach the final pressure

Guidelines to perform the test

1. Determine the air volume of the system. This volume includes the air receiver and the piping. A measuring tape should be sufficient to determine the dimensions of the air receiver. Piping volume need only be estimated.
2. Use any calibrated pressure gage to perform the test. (The more accurate the gage used, the better the test results.) With the compressor running, measure the time interval for either a fixed pressure change (try in the range of 10–20%), or wait until the compressor starts and stops pumping and use that pressure range.
3. Perform two tests. The first should be with the compressor running, which will show the amount of excess air in the system. When the compressor is stopped, perform a second test that will provide the data to determine the actual air being used. *The compressor yield is equal to the sum of the two calculated flows.*
4. If more than one compressor is running, perform the test on the one that starts and stops, and move on, in sequence, to the next one. In calculating, make sure to use the volume for the whole system. The yield for each of the compressors requires the solving of n equations, where n is the total number of compressors actually in service at the test time.
5. Accuracy of results can be increased by including a temperature correction factor, so that the corrected flowrate is equal to the measured flow times the ratio of absolute ambient temperature to absolute compressed-air temperature.

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*170 West 74th St., New York, NY 10023.

Protecting against backflow in process lines

Backflow—a flow-reversal, by which liquids get into places they do not belong—can be serious. Here is how to prevent it.

***Stephen J. Emery**, Hersey Products, Inc.*

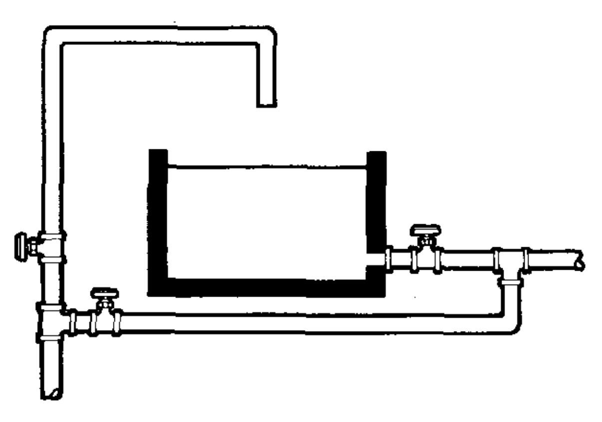
□ Backflow is most simply defined as a reversal of the normal direction of flow of piped liquids. It happens when, for one reason or another, upstream pressure falls below downstream pressure. Backflow can result from a pump failure, heavy upstream drawoff, downstream pressure rise, ruptured lines and many other conditions.

What makes backflow a major cause of concern is that reverse flow of liquids often has highly undesirable consequences. In mixing tanks, for example, backflow of mixed product to feedstock holding-tanks can result in contamination and costly waste. Backflow can also create hazardous conditions, particularly where toxic or explosive fluids are involved.

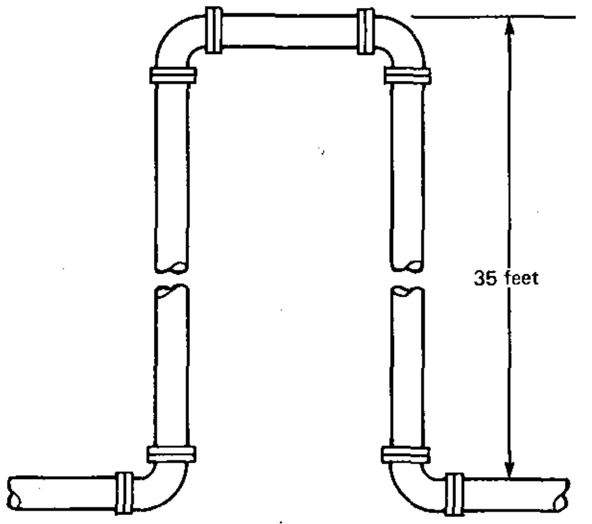
Backflow presents its most serious problems, of course, when human health may be affected. For example, where a single water-supply system furnishes both potable water and process water, great care must be taken to protect potable water from possible contamination. Backflow of water containing impurities or toxins into the potable water supply is, obviously, intolerable. In fact, safe drinking-water legislation and OSHA (U.S. Occupational Safety and Health Administration) requirements have made it mandatory to provide backflow protection on all cross-connections between potable and nonpotable water supplies. Ideally, water or other fluids that should be kept separate would have separate supply systems. In practice, this is often not practical. In some cases, certain facilities, equipment and processes cannot operate satisfactorily without direct cross-connections. The problem is how to maintain control over fluids, so as to prevent backflow or backsiphonage (which is pressure reduction at any point in the water distribution system to atmospheric or lower, that could draw water through the system in a direction opposite to normal).

Open air gap: Best protection

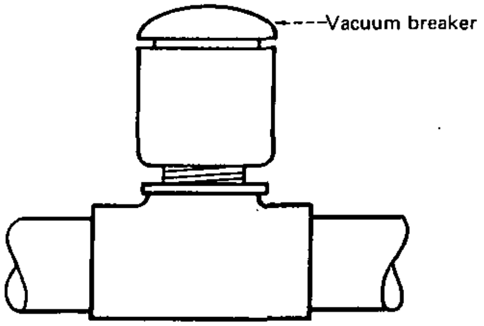
It is generally agreed that the best protection against backflow is the open "air gap." An air gap is created by "interrupting" a piping system. Upstream piping is separated from downstream piping by an air space (the recommended gap is at least twice the diameter of the supply pipe). Typically, fluids pass from the supply line to a holding tank and thence to the downstream piping system. The air gap between supply line and tank makes



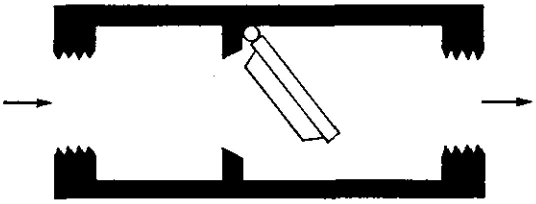
Open air gap is best, but it can be rendered useless by bypass piping **Fig. 1**



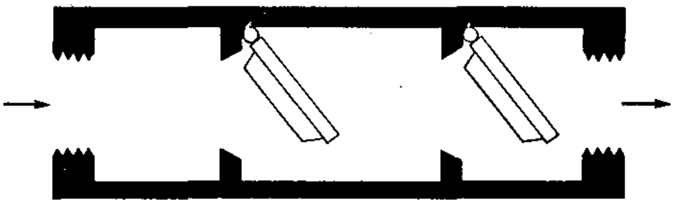
Barometric loop provides protection against backsiphonage but not backpressure **Fig. 2**



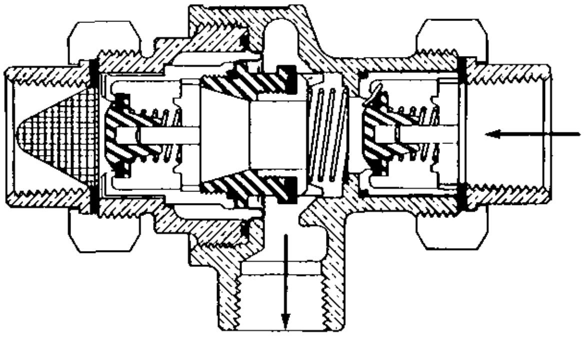
Vacuum breaker provides some protection, but is not recommended for hazardous liquids **Fig. 3**



Single check valve is entirely dependent on the leakproof integrity of the seal **Fig. 4**



Double check provides more protection than single, but still depends on seal integrity **Fig. 5**



Double check with atmospheric vent **Fig. 6**

backflow into the supply system virtually impossible.

The drawback to the air-gap method of backflow control is that line pressure is lost at each air gap and must be reestablished by means of pumps or towers. In a plant with many cross-connections, this would be an extremely expensive proposition. Moreover, because the open air gap is often impractical, codes requiring it may sometimes be ignored.

In many cases, bypass piping is installed while a tank is being repaired or refurbished (Fig. 1). The result is no protection at all. The existence of open tanks may also be undesirable, particularly where fumes or airborne contaminants pose problems.

Barometric loop

Other methods of backflow prevention include the barometric loop (Fig. 2), which operates on the principle that a perfect vacuum can lift water only 34 ft. This method requires the construction of a piping loop that rises 35 ft above the highest fixture in the piping system. While highly effective against backsiphonage, the barometric loop will not provide protection against backpressure. It is also cumbersome, susceptible to freezing, and vulnerable to bypass arrangements.

Vacuum breakers

Vacuum breakers (Fig. 3), of both atmospheric and pressure type, also are used to control backflow. They are at best a partial solution, offering some protection against backsiphonage but none against backpressure.

Single check valve can leak

Another method for controlling both backsiphonage and backpressure is a single check valve (Fig. 4). In theory, a check valve should provide adequate protection. It is, however, entirely dependent on the leakproof integrity of its seals and, as such, is not acceptable for use in situations where backflow into potable-water supplies could include hazardous, noxious or otherwise undesirable materials.

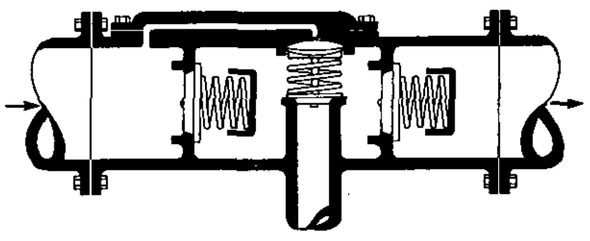
Double check valves: Partial answer

A more effective method of protecting against backflow is the double check valve (Fig. 5). Should upstream pressure drop below a certain level, the two check valves close to protect against backflow. When pressure is restored, the valves open.

As with the single check valve, the effectiveness of double check valves depends entirely on their sealing integrity. Should the seals develop leaks, undetected backflow could occur. For this reason, double check valves should only be used where backflow could be objectionable—but not hazardous—and should never be used where backflow could cause a potential health hazard or other serious condition.

Double check valve with vent

This device, generally known as a continuous-pressure type, is basically a double check valve with an atmospheric vent located between the checks (Fig. 6). In a backpressure condition (downstream pressure exceeds supply pressure) or a backsiphonage condition (supply pressure drops to atmospheric or lower), both checks close and



Reduced-pressure device senses pressure differential and opens relief when needed **Fig. 7**

the atmospheric vent opens. Any leakage at the second check valve flows out through the vent, which also serves as a vacuum break in case of backsiphonage leakage at the first check. Continuous-pressure devices are limited in application to low-hazard cross-connections and pipe sizes of ¾ in. and smaller.

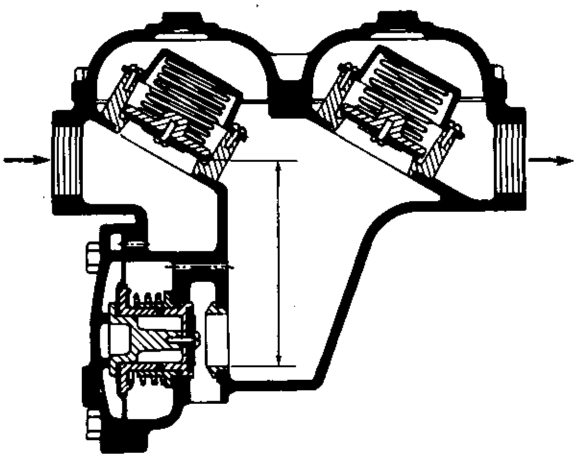
Reduced-pressure devices

Reduced-pressure (RP) devices take over where double check valves leave off. Like the latter, they make it possible to transmit line pressure across a cross-connection, but they are designed to provide a greater degree of protection against backflow where hazardous downstream liquids are involved.

In reduced-pressure backflow preventers (Fig. 7), a zone is provided between the check valves, in which a relief valve continually and automatically maintains pressure at a level lower than supply pressure. Because fluids will not flow from a low-pressure zone to a higher-pressure zone, these devices are able to provide excellent protection against backflow caused by backpressure.

They do not, however, provide the same degree of protection against "backsiphonage." Backsiphonage can occur when supply pressure drops to atmospheric or below. Pressure in the zone drops accordingly and creates a condition in which downstream pressure becomes greater than upstream pressure. In this situation, the only protection against leakage past the check valves is the integrity of the seals. Thus, it is possible to have continuous backsiphonage of hazardous or toxic liquids that may go completely undetected for a long time.

When you want to maintain line pressure, yet need the protection only an air gap can provide, reduced-pressure devices designed with an internal air gap approach this degree of protection (Fig. 8)



Reduced pressure with built-in air gap **Fig. 8**

Like conventional RP devices, these employ a reduced-pressure zone between the checks and, by continuously sensing line pressure, operate a relief valve to keep the pressure in the zone below the supply pressure. However, they offer one very important advantage—the ability to protect against backsiphonage. If supply pressure ever drops to atmospheric or below, the water level in the reduced-pressure zone is automatically lowered to the point where an air gap greater than the diameter of the supply pipe is created within the unit.

Should there be any leakage from the downstream line through the check, it will flow harmlessly out through the relief valve, thereby giving a visual indication that a backflow or backsiphonage condition exists.

Further, should the check valve that protects the supply line leak, the internal air gap will prevent fluids in the reduced-pressure zone from getting past the valve by drawing air into the supply line.

Codes in most states require that backflow preventers be tested and certified on an annual basis. Certified testing personnel provide this service. Generally, because of the critical nature of backflow prevention and the need to maintain devices in proper working order, reduced-pressure backflow preventers should be designed for servicing without removal from the line.

With the chemical process industries in the U.S. now accounting for well over half the water consumed by industry, and with processing plants often containing hundred of potentially hazardous cross-connections, well-planned control programs are becoming increasingly common. Plant operators can determine not only the degree of hazard that exists with cross-connections but the appropriate measures that should be taken.

Roy V. Hughson, *Editor*

The author

Stephen J. Emery is marketing manager of Hersey Products Meter and Controls Group, 20 Elm St., Dedham, MA 02026. He has been employed with Hersey Products since 1978, and has held various positions including those of sales manager and product manager. Prior to joining Hersey, he was employed by USM Corp. as a sales representative.

Predict gas/liquid-flow pressure drop

John J. J. Chen*

Two-phase, gas/liquid flow occurs in many industrial processes. To determine the pressure drop in such situations, the Lockhart-Martinelli method is recommended as the most reliable and simplest of approaches not specific to a particular two-phase system.†

The Lockhart-Martinelli method uses two parameters:

$$\varphi_L{}^2 = (dp/dl)_{TP}/(dp/dl)_{SL} \qquad (1)$$

$$X^2 = (dp/dl)_{SL}/(dp/dl)_{SG} \qquad (2)$$

where φ is the parameter for two-phase-flow frictional pressure drop, X is the two-phase flow modulus, $(dp/dl)_{TP}$ is the two-phase-flow frictional pressure drop, and $(dp/dl)_S$ is the frictional pressure drop assuming that one phase flows alone in the conduit, with subscripts L and G referring to the liquid and gas phases, respectively. (Eq. (1) may also be written using the gas-phase pressure drop as the reference, in which case the subscript L is changed to G.)

When a wide range of data was plotted as φ versus X, Lockhart and Martinelli found that the data fell into four separate curves, corresponding to situations in which the gas/liquid phases were flowing in turbulent/turbulent, turbulent/viscous, viscous/turbulent, and viscous/viscous regimes. These four curves were identified by adding further subscripts — T for turbulent and V for viscous — to the parameter φ_L, thus giving φ_{LTT}, φ_{LTV}, φ_{LVT} and φ_{LVV}. Demarcation between turbulent and viscous regimes was assumed at a superficial Reynolds number of 1,000.

To use the Lockhart-Martinelli technique, $(dp/dl)_{SL}$ and $(dp/dl)_{SG}$ are first evaluated via standard methods. From these, X is obtained using Eq. (2). The appropriate φ_L versus X curve then yields the value of φ_L from which $(dp/dl)_{TP}$ is evaluated by Eq. (1).

The correlation of φ_L versus X is usually presented in graphical form, and the coordinates for plotting these curves are given in Ref. [1]. However, such graphical representation is often inconvenient even for hand calculations.

Other authors have curve-fitted the relationships [2,3]. These expressions are somewhat cumbersome to apply, and exact matching with the curve is really not necessary, given the degrees of error inherent in the method. Simpler empirical correlations have been derived [4,5], but they are not as accurate as the following empirical equations:

$$\varphi_{LTT} = 2.4X^{-0.85} + 1.8X^{-0.10} \qquad (3a)$$

$$\varphi_{GTT} = 2.4X^{0.15} + 1.8X^{0.9} \qquad (3b)$$

$$\varphi_{LTV} = \varphi_{LVT} = 1.84X^{-0.88} + 1.64X^{-0.08} \qquad (4a)$$

$$\varphi_{GTV} = \varphi_{GVT} = 1.84X^{0.12} + 1.64X^{0.92} \qquad (4b)$$

$$\varphi_{LVV} = 1.25X^{-0.95} + 1.36X^{-0.03} \qquad (5a)$$

$$\varphi_{GVV} = 1.25X^{0.05} + 1.36X^{0.97} \qquad (5b)$$

$$R_L = X^{2/3}/(3.5 + X^{2/3}) \qquad (6a)$$

$$R_G = 3.5/(3.5 + X^{2/3}) \qquad (6b)$$

Eqs. (6a) and (6b) for holdup, R, had been derived semi-empirically [6], and are given for the sake of completeness. In the range $0.01 < X < 100$, Eqs. (3), (4) and (5) predict values of φ that deviate from the values given by the Lockhart-Martinelli curve with an averaged absolute deviation of less than 4% and a maximum absolute deviation of 10%. Since the pressure drop is proportional to φ^2, the errors for pressure drop are slightly more than double these values. Such deviations are apt to be acceptable since the Lockhart-Martinelli correlations may sometimes give predictions with errors in excess of ±50% of experimental data.

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*University of Auckland, Private Bag, Auckland, New Zealand.

†For another approach, see *Chem. Eng.*, June 24, pp. 45-50.

Startup of centrifugal pumps in flashing or cryogenic liquid service

Inder S. Rattan[*] and Vijay K. Pathak[†]

Startup of centrifugal pumps in nonflashing service usually is easy, basically requiring priming to remove gases and vapors from the pump casing. The pump is started with the discharge valve closed to prevent motor overload, then the valve is gradually opened fully to allow the pump to operate against system pressure. However, in liquid-hydrocarbon flashing and cryogenic service, pump startup is more difficult due to flashing of the liquid.

There are generally two operations during which flashing may occur in the pump:

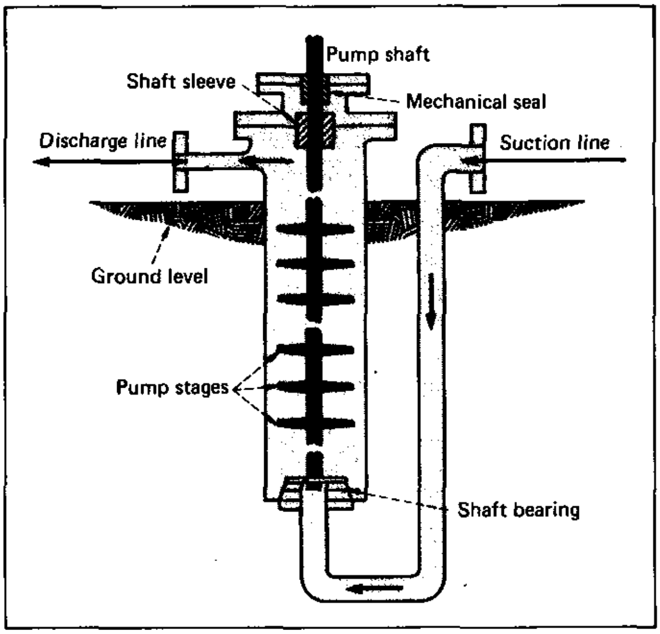

Figure 1 — *Typical multi-stage vertical centrifugal pump*

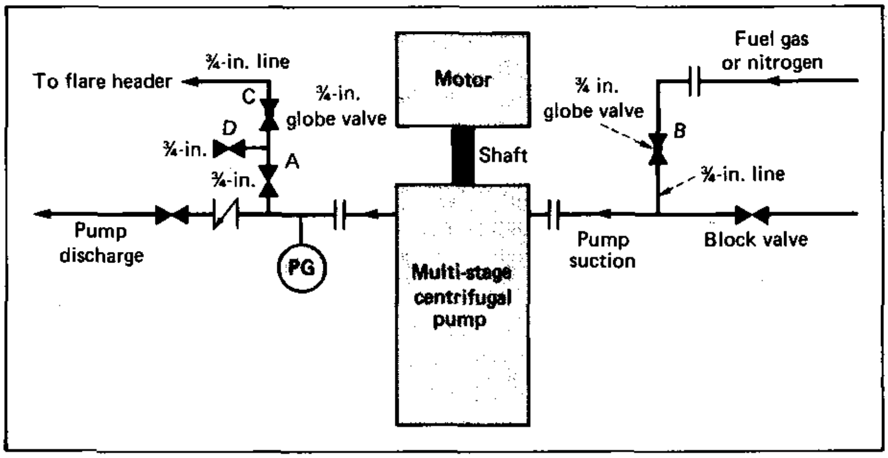

Figure 2 — Recommended arrangement for pump startup

1. During plant commissioning, or after a prolonged plant shutdown when the whole system has been depressured to atmospheric conditions. The pumps are not part of the pressurized system, as the *unit is just being brought up to* operating pressure.
2. During actual plant operation, when it may become necessary to bring a spare pump online. Although in some cold locations the spare is kept full of liquid, this is not the general practice (usually, the spare is kept purged and depressured) because of problems resulting from vaporization of the volatile liquid.

In some cases, a small liquid stream is bled through a hole in the check valve of the spare pump (or through a restriction orifice in a line bypassing the check valve) to renew the liquid in the unit and keep it cool. This would require block valves on the spare unit's suction and discharge lines to be open all the time, which may not please the plant operations people. Moreover, in very hot areas such as the Middle East, a fairly large amount of liquid might have to be bled back to keep the liquid in the spare pump cool, and this would have to be accounted for in the pump design. In addition, there is a possibility that keeping liquid in the spare pump may overpressure the system to above its design pressure.

Effects of flashing

The extent of flashing depends upon the fluid being pumped, but flashing has two basic effects. First, it creates a vapor-lock situation where vapor generation prevents head buildup and causes cavitation, and the pump cannot be started up. Second, in the case of light liquefied products (both hydrocarbon and nonhydrocarbon), flashing within the pump can create very low temperatures and cause auto-refrigeration.

In the case of liquid ethane, for example, temperatures as low as -60°F could be reached, causing the pump material and mechanical specifications to change significantly, since the unit is generally not designed for such low temperatures. Several installations with which we are familiar have experienced failure of such components as mechanical seals, thrust bearings, and wearing rings during pump startup. In addition to being costly in terms of repair, the downtime could result in significant losses.

Starting up the pump

The following method for starting up centrifugal pumps in flashing or cryo-

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[*] Stone & Webster Canada Ltd., 2300 Younge St., Toronto, Ont. M4P 2W6, Canada.

[†] *Monenco Engineers & Constructors, Inc., 801 6th Ave.,* S.W., Calgary, Alta. T2P 3W3, Canada.

genic service is of general applicability. Fig. 1 illustrates the main components of a typical multistage pump arrangement, and Fig. 2 shows the recommended setup for starting up the pump.

A ¾-in. connection is provided on the pump suction downstream of the block valve. It is attached permanently to a gas pressurizing source, such as fuel gas or nitrogen. Obviously, the connection has to be made to a source that can provide gas at pressures above the bubble point of the liquid being pumped.

A ¾-in. connection is provided on the pump discharge, upstream of the block and check valves, and it is connected to the nearest flare header to vent fuel gas and/or hydrocarbon vapor during commissioning. A double block and bleed are provided on the discharge to avoid problems of freezing in cryogenic service.

The recommended procedure for starting up the pump is as follows:

1. Close the pump suction and discharge block valves.
2. Open the ¾-in. block valve labeled A in Fig. 2.
3. Open the fuel-gas/nitrogen globe valve labeled B, which will bring the system pressure up to the level that prevents flashing (this would correspond to the bubble-point pressure of the liquid). The pressure gage shown indicates the system pressure.
4. Open the suction block valve. Liquid in the suction line will not yet be able to enter the pump, due to zero pressure differential.
5. Open slightly the ¾-in. globe valve labeled C, which leads to the flare. As the gas evacuates the suction, liquid will enter the pump without flashing. It is important that Valve C be opened slowly to prevent flashing — opening it too fast or too far may cause gas to leave the system faster than liquid entering it, causing the system pressure to drop. The pressure on the gage should be kept very close to the bubble-point pressure.
6. The pump is full when a liquid/vapor cloud comes out when Valve D is cracked open slightly.
7. Once the pump is full of liquid, close Valves B, A, D and C in that order.
8. Start the pump, and open the discharge valve to the system.

Power consumption of double mechanical seals

Published pump efficiencies are generally based on the use of single mechanical seals during efficiency testing. If double seals are specified, the efficiency of the pump will be lower than that shown by the efficiency curves.

Frederic W. Buse, Ingersoll-Rand Corp.

The performance curves published by manufacturers of chemical process pumps are usually based on data obtained on pumps that have a single, multispring, mechanical seal in the stuffing box (Fig. 1). The curves are usually shown for 3,550 and 1,750 rpm. Many pump manufacturers run their tests at the design speed, 3,550 rpm, and use the affinity laws (summarized later) to calculate performance at 1,750 rpm.

In either case, there is seldom a correction factor or adjustment shown in the manufacturer's literature to account for the additional horsepower consumed by double mechanical seals. When a customer wants a double mechanical seal (Fig. 2), the quotes for efficiency are based on the published curve, not recognizing that the double seal will consume more horsepower than the single one.

As a result, the motor may be overloaded, especially when the pump is operating at 1,750 rpm. The pump efficiency can be one to ten percentage points below that quoted (Fig 3).

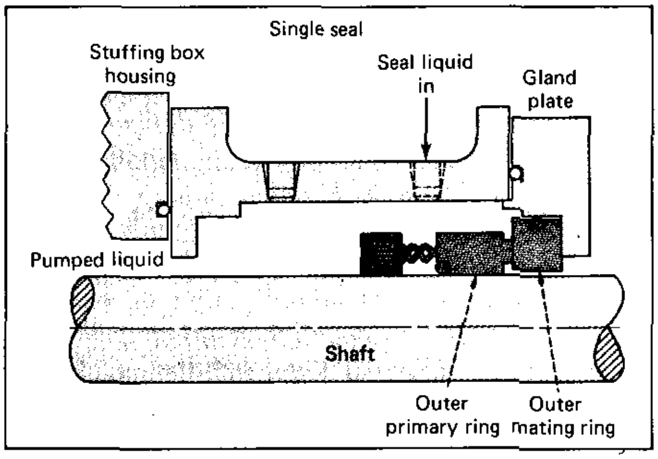


Figure 1 — Single, multispring mechanical seal as used for pumps

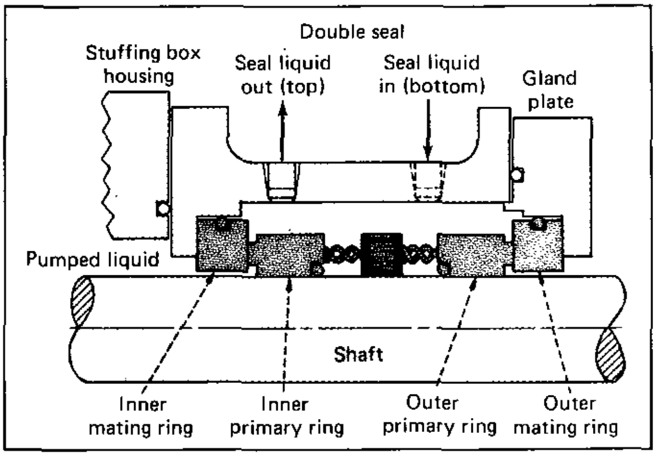


Figure 2 — Double mechanical seal consumes greater horsepower

Testing seal power-consumption

Tests were run on pumps (without impellers) to determine the power consumption of the seals. These were unbalanced single seals, outside seals, and double seals, with carbon on ceramic mating surfaces (the most common materials), at pressures between 25 and 50 psi (the most common pressure range). A standard stuffing box was used, with flush water of 60° to 75°F injected into the seals for lubrication. Motor speed was varied from 1,100 to 3,600 rpm. Base horsepower was determined using a pump with no impeller, and no seal.

Note that the seal's power consumption will change with balanced (rather than unbalanced) seals, with a change in mating materials or mating-surface widths, or with an increase in stuffing-box pressure. Silicon carbide against carbon surfaces will have a higher coefficient of friction than carbon against ceramic, and tungsten carbide against carbon will have a still higher coefficient.

What the tests showed

There was no significant difference in power consumption between an unbalanced single inside seal and an outside seal. However, there was an increase in horsepower when double seals were used on pumps, especially those with low-horsepower drivers at four-pole speeds (Table I).

The additional power may seem insignificant, but when applied to pumps of low horsepower, there can be a substantial change in efficiency. The additional horsepower for the double mechanical seal is added to the specified pump horsepower for the various sizes of pumps to obtain the new efficiencies. Table II shows the brake horsepower (bhp) range where there is a significant change:

To determine the change in pump efficiency for a given seal size, material and construction, *a constant difference in horsepower should be used,* not a constant decrease in points of efficiency (η). Fig. 4 shows pump horsepower at operating conditions vs. loss of efficiency when using double mechanical seals. (Fig. 4 is to be applied to pumps that have been tested at 1,750 or 3,550 rpm.)

Example 1

Find the loss for a double mechanical seal on a 3 × 1½ × 6 pump tested with a single 1⅛-in.-dia. seal, 1,750 rpm, 35 gal/min, 18-ft head, $\eta = 43\%$, liquid sp. gr. = 1.

$$\text{bhp} = QsH/3{,}960$$

where Q = flow, gal/min; s = sp. gr., dimensionless; and H = head, ft.

$$\text{bhp with single seal} = (35)(18)(1)/(3{,}960)(0.43) = 0.37$$

From Fig. 4, the decrease in efficiency for 0.37 bhp is 2.3 points; efficiency with double seal = 43 − 2.3 = 40.7%.

$$\text{bhp with double seal} = (35)(18)(1)/(3{,}960)(0.407) = 0.39$$

Example 2

Find the loss for a double mechanical seal on a 3 × 1½ × 8 pump tested with a single 1⅞-in.-dia. seal, 1,750 rpm, 37 gal/min, 55-ft head, $\eta = 34\%$, sp. gr. = 1.

$$\text{bhp with single seal} = (37)(55)(1)/(3{,}960)(0.34) = 1.5$$

From Fig. 4, decrease in efficiency for 1.5 bhp is 3 points; efficiency with double seal = 34 − 3 = 31%.

$$\text{bhp with double seal} = (37)(55)(1)/(3960)(0.31) = 1.66$$
$$\text{Horsepower increase} = (1.66 - 1.5)/1.5 = 10.7\%$$

If totally enclosed motors were used, the 1.66-horsepower requirement would mean that a 2-horsepower motor would have to be employed, rather than a 1.5-horsepower one.

Effect of using affinity laws

The horsepower consumption of seals is even more significant if the data for a published curve is determined at one pump speed, but the performance for the speed in question has been predicted by use of the affinity laws. These laws state that capacity varies linearly with speed, that head varies with the square of speed, and power with the cube. That is, if you double the speed, you double the capacity, quadruple the head, and approximately octuple the power.

Likewise, when you reduce the speed from 3,550 to 1,750

Table I — How double seals increase power consumption

Speed, rpm	Additional horsepower consumption of unbalanced double seals of size (sleeve dia.), in.: 1⅛	1⅞
1,750	0.020	0.08
2,950	0.025	0.10
3,550	0.030	0.12

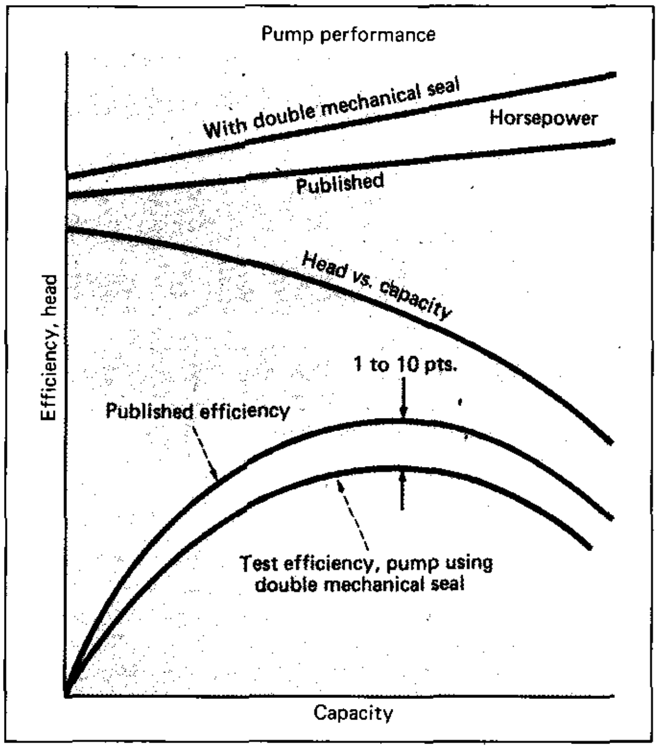


Figure 3 — Pump curves for single and double mechanical seals

Table II — Brake horsepower range where double seal is significant

Seal size (sleeve dia.), in.	Power below which there is substantial efficiency change, bhp: 3,550 rpm	1,750 rpm
1⅛	2	2
1⅞	15	10

Table III — Additional bhp to be added when affinity laws were used

Seal size	Additional bhp for double seal*
1⅛	0.058
1⅞	0.287

*When stepping from 3,550 rpm performance to 1,750 rpm performance on the basis of the affinity laws.

Table IV — Decrease in efficiency when double seals are used

Typical bhp	Typical published η_p, %	Decrease, in points of efficiency for double seal of size shown: 1⅛ in.	1⅞ in.†
0.5	55	6.0*	
2	60		7.5
5	69		4.0
10	65		2.0

*Calculated in the example above.
†Calculated as in the example, but adding 0.287 bhp (from Table II) to the pump's bhp.

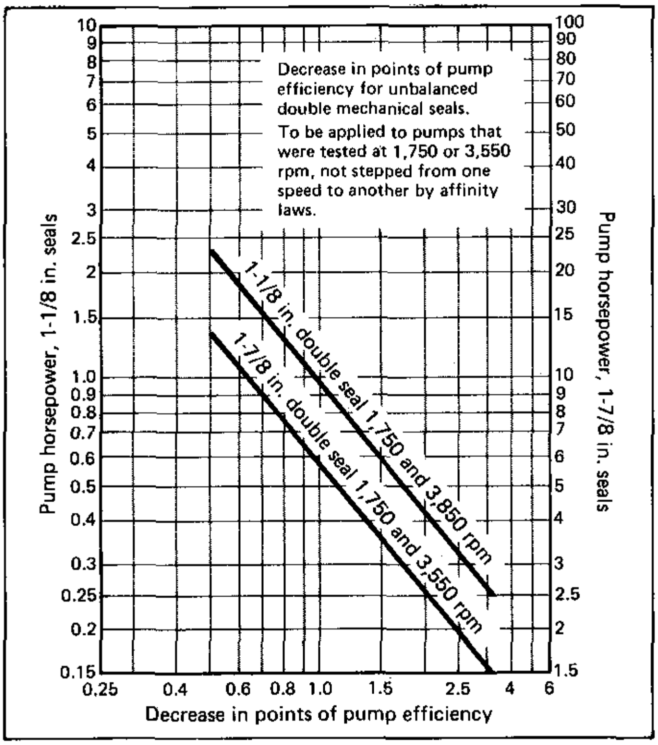


Figure 4 — Loss of efficiency for pumps tested at 1,750 or 3,550 rpm

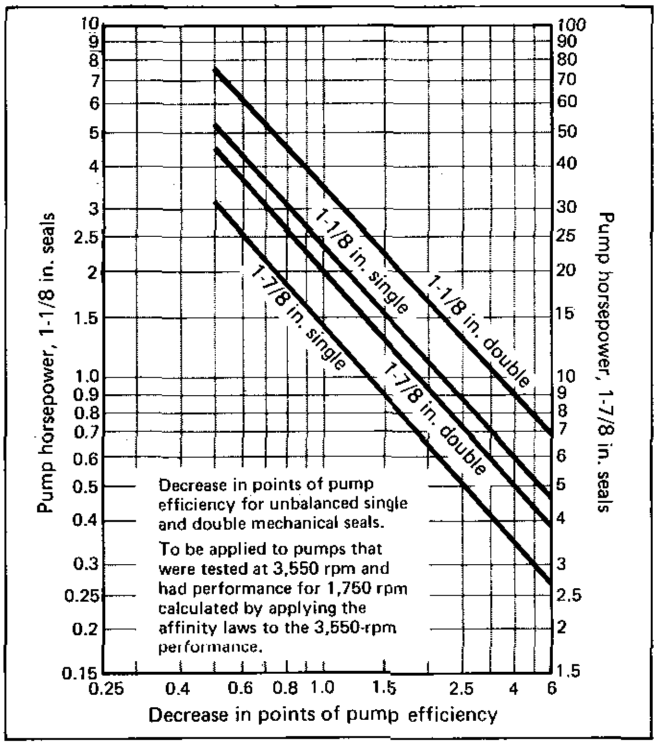


Figure 5 — Efficiency loss in cases where affinity laws were used

rpm, *the capacity will be half, the head will be a quarter* and the power an eighth of that when the pump operates at 3,550 rpm. These losses are from disk friction, discharges and mechanical friction losses. Traditionally, when operating at 1,750 rpm, two points were deducted from the efficiency at 3,550 rpm, to account for the extra losses. (When this rule was established, the losses for different types of mechanical-seal construction were not understood.) To give a more accurate value of the power at the lower speed, instead of taking one-eighth of the power at 3,550 rpm, one should calculate it by using the new capacity, new head, and efficiency at 3,550 rpm, less two points for additional losses.

Tests show that the horsepower consumption of a seal does not follow the affinity laws with change of speed; it is actually much closer to a linear relationship.

Table III shows the additional horsepower that would be added to the calculated pump horsepower that has been adjusted by two or three points for the change in speed.

Example 3

The following sample calculation shows how to determine the decrease in efficiency at 1,750-rpm performance for a double mechanical seal. The published efficiency, η_p, of 55% has already been adjusted 2 or 3 points for the lower speed.

Find the loss for a double mechanical seal on a 1½ × 1 × 6 pump, 1⅛-in.-dia. seal, bhp = 0.5, $\eta_p = 55\%$ efficiency.

From Table IV, for a 1⅛-in. seal, the loss for a double seal, bhp_{seal}, = 0.58 bhp.

$$\text{Efficiency with double seal} = (bhp_p)(\eta_p)/(bhp_p)(bhp_{seal})$$

$$\text{Efficiency with double seal} = (0.5)(55)/(0.5 + 0.58) = 49.3\%$$

$$\text{Decrease in efficiency, } \Delta\eta, = 55 - 49.3 = 5.7 \text{ points.}$$

Table III is a guide to the decrease in points of efficiency due to seal horsepower consumption to be applied to performance that was calculated on the basis of the affinity laws, when stepping from 3,550 to 1,750 rpm.

Fig. 5 shows operating horsepower vs. additional decrease in points of pump efficiency when using a single or double seal, based on performance data that have been converted from 3,550 rpm to 1,750 rpm by use of the affinity laws.

Conclusions

There can be a significant increase in pump brake horsepower when using double mechanical seals on pumps driven by low-horsepower motors at low rpm. Depending on the type of seal and pump horsepower, the decrease in efficiency will vary from ½ to 10 points.

The increase in horsepower required should be applied to the operating pump horsepower to calculate the resulting efficiency. This differential is greater when pump horsepower has been stepped from one speed to another by using the affinity laws (rather than by actually measuring efficiency at the speed in question).

***Roy V. Hughson**, Editor*

The author

Frederic W. Buse is chief engineer of the Standard Pump Div. of Ingersoll-Rand Pumps, P.O. Box 656, #1 Pump Place, Allentown, PA 18105, Tel: (215) 776-6100. He is a graduate of New York State Maritime College, with a B.S. in marine engineering. He is a member of the Hydraulic Institute (which named him "Man of the Year" in 1976) and of the American National Standards Institute (ANSI) committees B215 for centrifugal pumps, and B73.1 and B73.2 for chemical pumps. He has contributed to "Marks' Standard Handbook for Mechanical Engineers" and the "Pump Handbook" (ed. by Igor Karassik), and has just received his 11th U.S. patent on pumps.

Estimate maximum allowable pressures for steel piping

*V. Ganapathy**

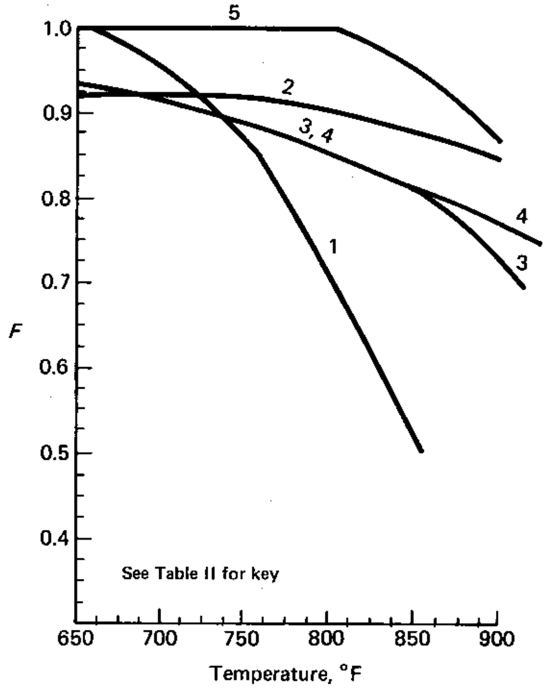
☐ Here is a method to estimate the maximum allowable pressure for ferrous piping according to ASME and ANSI codes without referring to the actual codes' stress tables. In addition, specific pipe dimensions (O.D. and thickness) need not be known, as the schedule number and pipe material are all that are necessary.

Table I gives the maximum allowable working pressure, in psig, based on an allowable stress of 15,000 psi. To correct for the specific material and temperature, multiply the value obtained from Table I by the appropriate *F* factor from the figure. Table II lists the various materials by composition and ASME code specification.

Example

Estimate the maximum allowable working pressure for a 2-in., Schedule 40 carbon-manganese (SA 53B) pipe at 750°F.

Solution: From Table I, a value of 1,782 psi is obtained. From the figure, for SA 53B (Curve 1) at 750°F, *F* is 0.86. Hence, the maximum allowable pressure is 0.86 × 1,782 = 1,532 psig.



See Table II for key

Maximum allowable pressure* **Table I**

Nominal pipe size, in.	Schedule 40	Schedule 80	Schedule 160
¼	4,830	6,833	–
½	3,750	5,235	6,928
1	2,857	3,947	5,769
1½	2,112	3,000	4,329
2	1,782	2,575	4,225
2½	1,948	2,702	3,749
3	1,693	2,394	3,601
4	1,435	2,074	3,370
5	1,258	1,857	3,191
6	1,145	1,796	3,076
8	1,006	1,587	2,970

*Based on allowable stress of 15,000 psi

ASME specifications for various steels **Table II**

Composition	Code designation	Curve on figure
Carbon-manganese	SA 53B	1
Carbon-manganese-silicon	SA 106B	1
Carbon-½Mo (½% molybdenum)	SA 335 P1	2
5Cr-½Mo (5% chromium, ½% molybdenum)	SA 335 P5	3
9Cr-1Mo	SA 335 P9	4
1¼Cr-½Mo	SA 335 P11	5
2¼Cr-1Mo	SA 335 P22	5

Source: ASME boiler and pressure vessel code, Sec. 1, 1980, p. 184, Table PG23.1

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*Struthers Thermoflood Corp., P.O. Box 753, Winfield, KS 67156

HEAT TRANSFER

Use these guidelines for quick preliminary selection of heat-exchanger type

Thane R. Brown*

The table below offers several generalized comparisons to aid in selecting among shell-and-tube, plate, and spiral heat exchangers.

Included among the factors is the "effectiveness index." It is a measure of the cost-effectiveness of an exchanger, and is defined as: I=overall heat-transfer coefficient/heat-exchanger cost. With coefficients expressed in Btu/(h)(ft^2)(°F), and purchase costs in \$/ft^2, EI is given as Btu/(h)(°F)(\$). Such indices are shown in the table, and these are averages for a variety of heat-exchanger sizes (250, 500, 750 and 1,000 ft^2) and services (e.g., water/water, oil/water, and steam/oil).

In the "Usual applications" section of the table, the term "temperature crossing" is used. This can be explained by referring to the figure.

Use the table for preliminary selection. However, as with all generalized guidelines, the correctness of the preliminary selection should be verified during detailed design.

Using the technique

We have to select a heat exchanger for cooling edible oil from 200°F down to 120°F using 85°F cooling water. Corro-

Table — Guidelines for preliminary selection of heat-exchanger type

Shell-and-tube	Plate	Spiral
Typical uses		
Liquid cooling Liquid-liquid heat exchange Steam heating Condensing Low-to-high pressure gas heating and cooling (finned tubes are often used) Reboilers	Liquid cooling Liquid-liquid heat exchange Low-pressure (<50 psig) steam heating	High-temperature gas interchange Column-mounted condensing Reboilers
Effectiveness index*		
Carbon steel 6	Not generally available	4
Type 316 stainless 3	8	2
Hastelloy C 1	3	1
Usual applications		
Services with no-to-slight temperature crossing Low-to-high temperature services Low-to-high pressure services Low-to-medium viscosity fluids (up to 75-100 cP) Low-fouling services (clean fluids)	Services with no-to-high temperature crossing Lower-temperature (<300° F) services Lower-pressure (<150 psig) services Low-to-high viscosity fluids Medium-fouling services Where mechanical cleaning is required Where little space is available Where normal-to-costly construction materials are mandated Where future expansion of heat-exchanger capacity is probable	Services with no-to-moderate temperature crossing Low-to-high temperature services Lower-pressure (<150 psig) services Low-to-high viscosity fluids High-fouling services (including slurries) Where mechanical cleaning is required Where little space is available
Limitations		
Less efficient and more costly with close approaches or crossing temperatures Poor shellside flow distribution causes inefficiency and high fouling rates Mechanical cleaning of shell side is almost impossible Exotic metallurgies costly Requires large amount of space	Not applicable at pressure >150 psig Gasketing limits use to lower temperatures ≤300° F with elastomers ≤450° F with asbestos (sealing is often problem in cyclic services)	Not applicable at pressure >150 psig High costs limit use to special applications

*C. Riesenbeck, internal communication, October, 1985.

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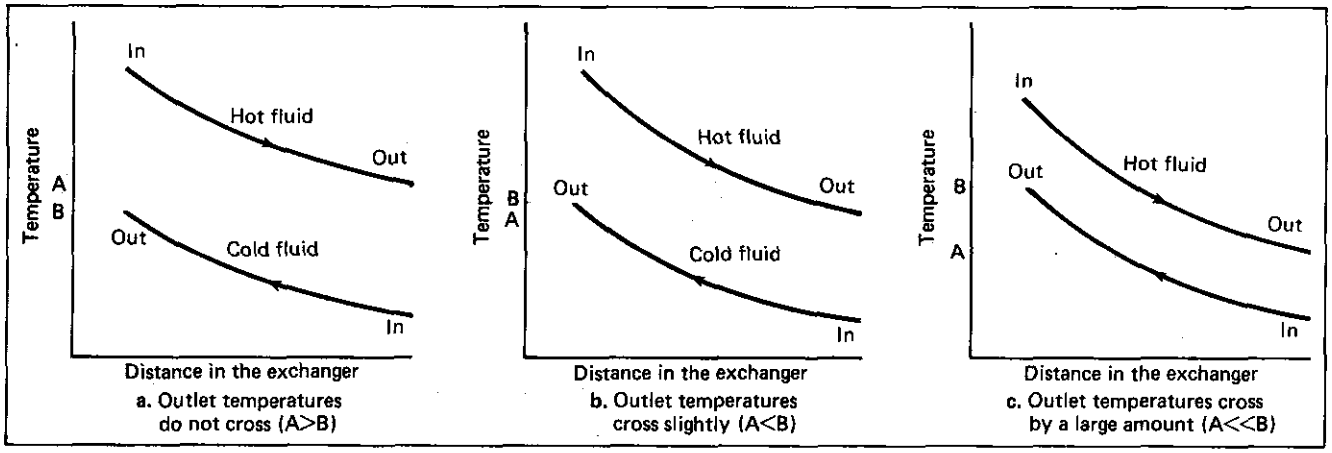
*The Procter & Gamble Co., 6210 Center Hill Rd., Cincinnati, OH 45224.



a. Outlet temperatures do not cross (A>B)

b. Outlet temperatures cross slightly (A<B)

c. Outlet temperatures cross by a large amount (A<<B)

Figure — Determining level of temperature crossing in exchanger

sion studies have shown that the exchanger can be made of carbon steel. The pressure rating will be 125 psig.

The table shows that all three types of exchangers could handle this service. Thus, the exchanger should be selected primarily on the basis of economics. To obtain a comparison of the relative costs, we look up the *EI* values: 6 for a shell-and-tube exchanger in carbon steel, 4 for a spiral unit in carbon steel, and 8 for a plate exchanger in Type 316 stainless since such a unit is not generally available in carbon steel. These numbers indicate that a stainless-steel plate exchanger will be a third more cost-effective than a shell-and-tube unit, and twice as cost-effective as a spiral one.

Evaluate heat exchanger fouling

☐ Generally, Plant Notebook publishes several short articles on various topics of interest to the engineer involved in operating, maintaining or designing a chemical-process-industries plant. In this issue, we deviate from that practice slightly by focusing on a single topic—the fouling of heat-exchange equipment.

Fouling of heat exchangers—the accumulation of deposits on their heat-transfer surfaces over a period of time—results in a degradation of thermal performance, accompanied by an increase in operating costs.

Plant operating engineers must be able to detect fouling and assess its degree and rate of occurrence, then establish a regular maintenance schedule and an as-needed remedial-action plan. These could include sootblowing with air or steam, washing with water, mechanical cleaning, or chemical cleaning with an acid or alkaline solution.

The first of these two articles discusses assessing the degree of fouling in a heat exchanger, while the second presents a graphical method of monitoring fouling over time.

***Cynthia Fabian Mascone**, Editor*

Assessing fouling in heat exchangers

K. N. Murty*

☐ To account for the gradual decrease in performance due to the accumulation of material on heat-transfer surfaces, the engineer incorporates a fouling factor into the design of a heat exchanger and provides excess surface area to allow for some fouling.

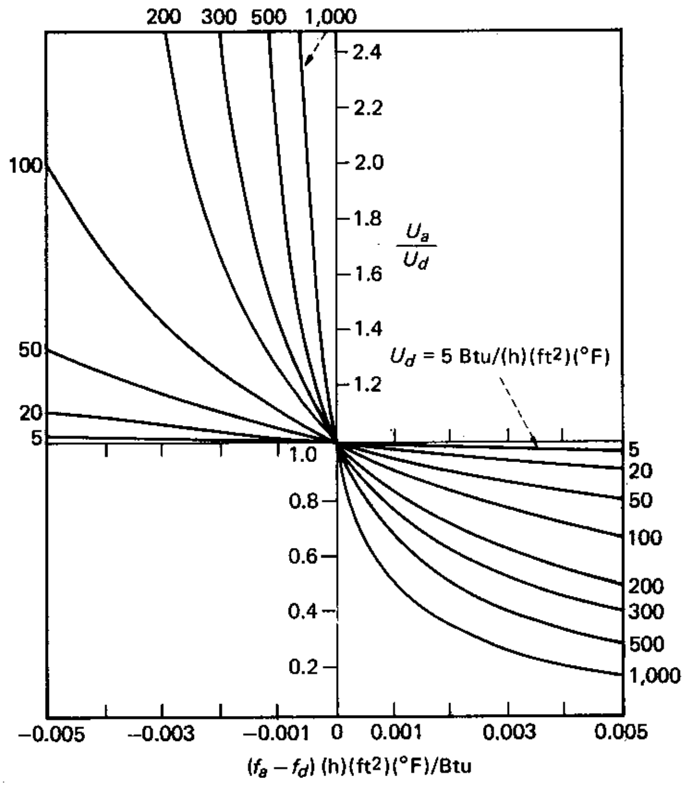
During the initial operating period, or after it has been cleaned, the unit can be expected to perform better than its rated heat-transfer capacity. However, with extended operation, the performance will gradually begin to deteriorate. When the actual fouling resistance (f_a) exceeds that for which the exchanger was designed (the design fouling factor, denoted f_d), the thermal performance (represented by U_a for the actual [measured] overall heat-transfer coefficient) will drop below the rated capability (U_d, the rated [design] overall heat-transfer coefficient).

Coefficient of performance

The exchanger's coefficient of performance, U_a/U_d, can be monitored and is a measure of fouling. This ratio can readily be determined: U_d will be known from design data, and U_a can be calculated by dividing the actual heat duty by the product of the surface area and the log mean temperature difference. When U_a/U_d drops to, say, 80%, cleaning can be done to prevent further deterioration in performance.

U_a/U_d is dependent on U_d and the difference between the actual and design fouling factors, and is given by:

$$\frac{U_a}{U_d} = \frac{1}{1 + U_d(f_a - f_d)} \qquad (1)$$



Coefficient of performance depends on the difference between the actual and design fouling factors

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*Bharat Heavy Electricals, Ltd., Corporate Research and Development Div., Vikasnagar, Hyderabad - 500 593, India.

This can be rearranged and solved for f_a:

$$f_a = f_d + 1/U_a - 1/U_d \qquad (2)$$

The figure illustrates the relationship between U_a/U_d and $(f_a - f_d)$ for different values of U_d.

As can be seen, the same degree of fouling, i.e., $(f_a - f_d)$, will reduce the coefficient of performance by different amounts depending on U_d. For example, if $(f_a - f_d) = 0.001$ (h)(ft^2)(°F)/Btu, the thermal performance of a gas cooler with $U_d = 20$ Btu/(h)(ft^2)(°F) will be reduced by only about 2% and that of a process condenser with $U_d = 200$ Btu/(h)(ft^2)(°F) by about 17%, while that of a steam condenser having a $U_d = 1{,}000$ Btu/(h)(ft^2)(°F) will be diminished by 50%.

Monitor fouling graphically

*V. Ganapathy**

☐ Fouling in a boiler or gas-liquid heat exchanger will cause a decrease in thermal performance, which will be reflected in a lowered value of the overall heat-transfer coefficient. The plant engineer can keep track of the degree of fouling and its rate of occurrence by monitoring U (and thus the fouling factor, ff), as a function of flowrate, over time. When fouling progresses to a certain point, corrective measures (i.e., shutdown for inspection and/or cleaning) can be taken—a reasonable action level would be $ff \geq 0.03$ for products of fuel-oil or coal combustion.

Procedure

The method presented here is quite simple. The gas flowrate and the inlet and outlet temperatures of both fluids are measured. Using the heat-transfer surface area and the heat capacity of the gas, which are known, and the measured temperatures, the overall heat-transfer coefficient is calculated. Then, $1/U$ is plotted against $W^{-0.8}$ if the gas flows inside the tubes or $W^{-0.6}$ if the gas flows outside the tubes. (Note that in the figure the top horizontal-axis scale, for $W^{-0.6}$, was compressed by a factor of ten for convenience. This does not affect the estimation of the fouling factor or the degree of fouling.)

This procedure is initially done when the unit is installed or immediately after it has been cleaned. Several readings are taken over a period of a few days, at different flowrates and/or temperatures—whatever conditions occur due to normal process variations. These points are plotted and a line drawn through them. The line is extrapolated leftward to $W = \infty$, and the intercept B gives an estimate of ff under clean conditions.

Then periodically, once every few months or so, the procedure is repeated, each time taking several readings over varying conditions, within a timespan of about a

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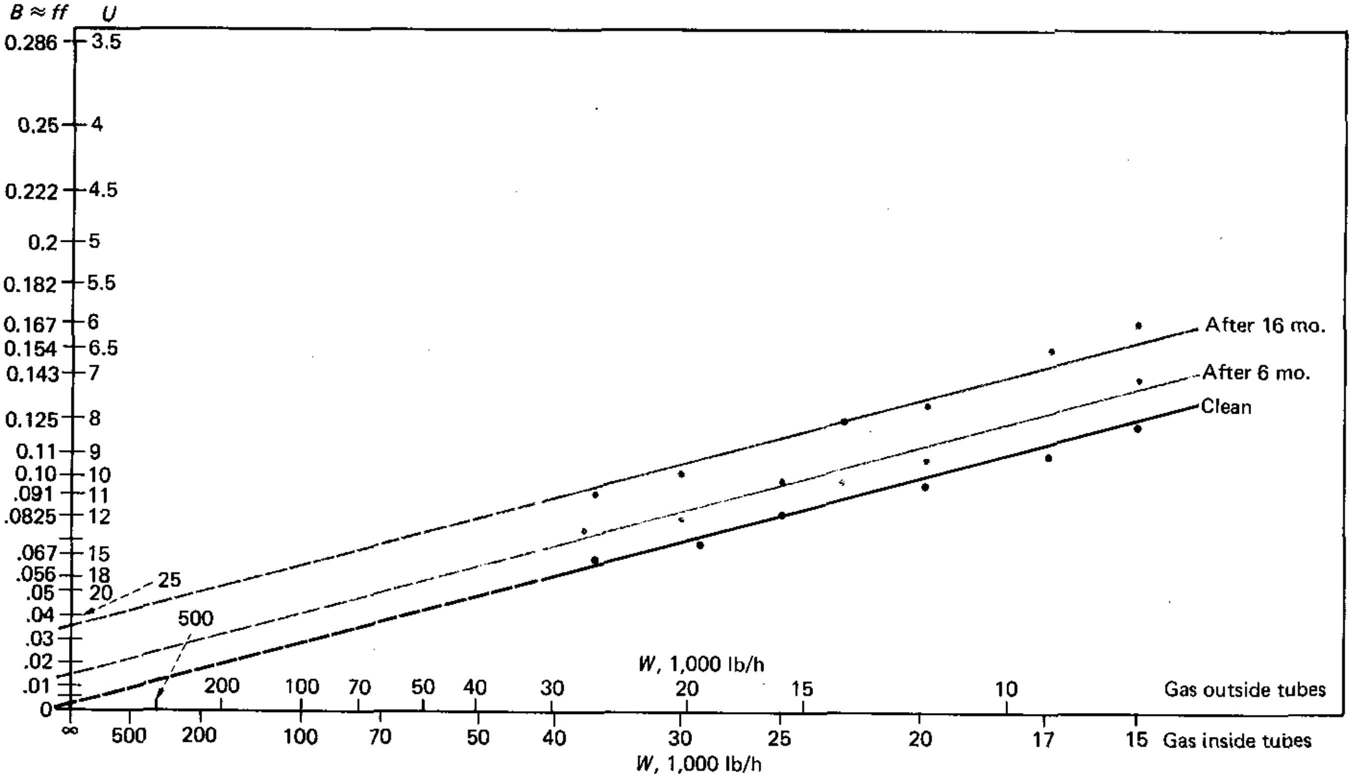
*Abco Industries, P.O. Box 268, Abilene, TX 79604.



Keep track of fouling by monitoring the overall heat-transfer coefficient as a function of flowrate

Operating data for firetube boiler over 16 months

W	T_1	T_2	U
Original condition			
15,000	1,000	460	8
17,000	1,000	461	9
20,000	1,050	477	10
25,000	1,050	494	11.5
29,000	1,050	484	14
37,000	1,000	510	15
After 6 months			
15,000	1,000	483	7
20,000	1,000	484	9.25
23,000	1,000	491	10.25
25,000	1,000	512	10
30,000	1,050	523	12
37,500	1,050	555	13
After 16 months			
15,000	1,000	512	6
17,000	1,000	522	6.5
20,000	1,050	538	7.5
23,000	1,050	555	8
30,000	1,050	565	10
37,000	1,050	598	10.75

week or less, and plotting a line. (The lines should be approximately parallel). The new fouling factor can be approximated by the value of B read from the vertical axis, and the degree of fouling that has occurred is represented visually by the distance between the line for the current conditions and the clean line.

Basis

From the energy balance around a watertube or firetube waste-heat boiler:

$$Q = WC_p(T_1 - T_2) = US\left(\frac{(T_1 - t_s) - (T_2 - t_s)}{\ln\left[\dfrac{T_1 - t_s}{T_2 - t_s}\right]}\right) \qquad (1)$$

the overall heat-transfer coefficient can be expressed as:

$$U = \frac{WC_p}{S} \ln\left[\frac{T_1 - t_s}{T_2 - t_s}\right] \qquad (2)$$

The heat-transfer coefficient of the gas dominates the overall heat-transfer coefficient. For firetube boilers and heat exchangers where the gas flows inside the tubes:

Nomenclature

A, B	Constants
C_p	Specific heat of the hot fluid, Btu/(lb)(°F)
ff	Fouling factor
LMTD	Log mean temperature difference, °F.
Q	Rate of heat transfer, Btu/h
S	Surface area for heat transfer, ft^2
T_1	Hot-fluid inlet temperature, °F
T_2	Hot-fluid outlet temperature, °F
t_1	Cold-fluid inlet temperature, °F
t_2	Cold-fluid outlet temperature, °F
t_s	Saturation temperature of hot fluid, °F
U	Overall heat-tranfer coefficient, Btu/(h)(ft^2)(°F)
W	Flowrate of the fluid controlling the heat transfer, lb/h

$$1/U = AW^{-0.8} + B \qquad (3)$$

and for watertube boilers and heat exchangers where the gas flows over or outside plain or finned tubes:

$$1/U = AW^{-0.6} + B \qquad (4)$$

The constant A, which is the slope of the line of $1/U$ vs. $W^{-0.8}$ or $1/U$ vs. $W^{-0.6}$, incorporates the fluid's thermal and transport properties at its bulk temperature, and B accounts for the contribution of fouling to overall heat transfer (in fact, $B \approx ff$).

Example

The table summarizes observations about a firetube waste-heat boiler made over a period of time. (T_1 and T_2 are the inlet and outlet temperatures of the hot fluid, in °F.)

The points were plotted, using the bottom horizontal-axis scale labeled "Gas inside tubes," as shown on the figure.

The lowest line on the figure represents the initial operating conditions when essentially no fouling was present, and the fouling factor was about 0.002.

After six months, represented by the middle line, fouling had increased somewhat; the new fouling factor was about 0.016, which is still within reasonable limits.

After 16 months, represented by the upper line, fouling had increased significantly, to a new fouling factor of about 0.034, indicating that the boiler tubes should be cleaned.

Weighted mean-temperature-difference for exchangers with phase changes

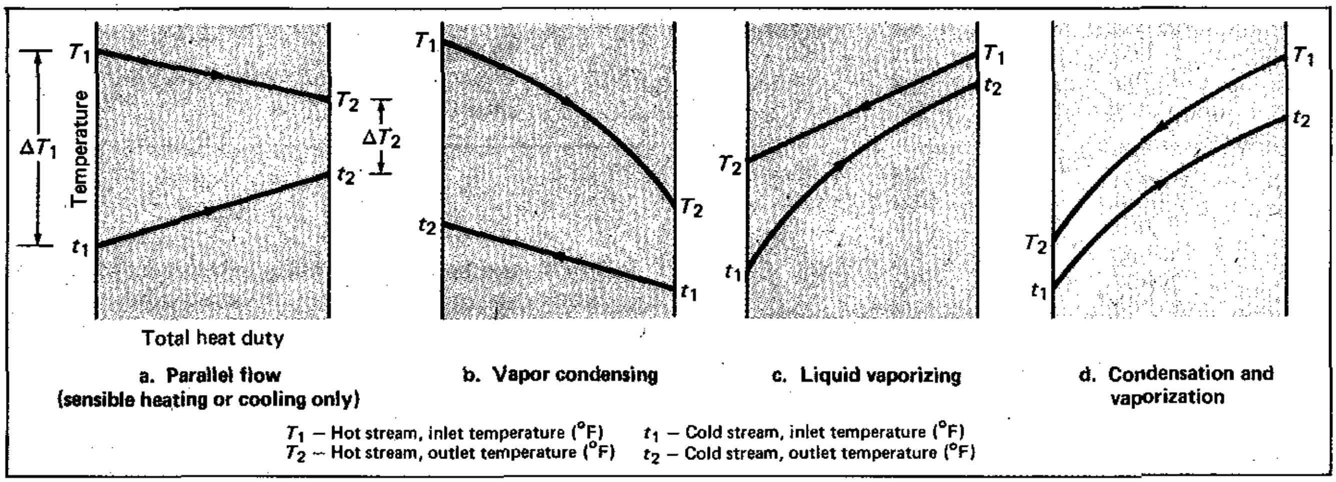
Ben Tammami*

Calculating the mean-temperature-difference (MTD) is a critical factor in designing heat exchangers, where one must determine the temperature difference (ΔT) between the two streams for use in the familiar equation:

$$Q = UA\Delta T$$

where Q = heat duty
U = overall heat-transfer coefficient
A = heat-transfer surface area
ΔT = mean-temperature-difference

When no condensation or vaporization occurs in either the



a. Parallel flow (sensible heating or cooling only)

b. Vapor condensing

c. Liquid vaporizing

d. Condensation and vaporization

T_1 – Hot stream, inlet temperature (°F)
T_2 – Hot stream, outlet temperature (°F)
t_1 – Cold stream, inlet temperature (°F)
t_2 – Cold stream, outlet temperature (°F)

Figure 1 — Divide heat-duty curve into as many zones as necessary to "straighten" the curve

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*Superior Hard-Surfacing Co., Inc., 3209 West 21st Street, P.O. Box 9397, Tulsa, OK 74107.

hot or cold side, an overall logarithmic mean-temperature-difference (LMTD) can be found as:

$$\text{LMTD} = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) \qquad (1)$$

where $\Delta T_1 = T_1 - t_1$ and $\Delta T_2 = T_2 - t_2$, for a parallel-flow exchanger. Calculation of LMTD is based on only sensible heating or cooling (see Fig. 1a) for a representation of such "straight line" temperature differences for a parallel-flow heat exchanger.

Table — Total heat duty is the sum of the heat duty for each zone

Temperature, °F					Corrected
Hot side	**Cold side**	**Δt**	**MTD**	**Correction**	**MTD**
Zone 1					
265	115	150			
↓	↑		154.46	0.9956	153.78
244	85	159			
Q_1 = 1,000,000 – 0 = 1,000,000 Btu/h					
Q_1/ CMTD = 1,000,000 / 153.78 = 6,502.8 Btu/(h)(°F)					
Zone 2					
244	115	129			
↓	↑		126.99	0.9894	125.64
210	85	125			
Q_2 = 1,500,000 – 1,000,000 = 500,000 Btu/h					
Q_2/CMTD = 500,000 / 125.64 = 3,979.62 Btu/(h)(°F)					
Zone 3					
210	115	125			
↓	↑		60.09	0.8505	51.11
120	85	35			
Q_3 = 1,900,000 – 1,500,000 = 400,000 Btu/h					
Q_3/CMTD = 400,000 / 51.11 = 7,826.26 Btu/(h)(°F)					
Q_T = 1,900,000 Btu/h					
Using Eq. (2), WMTD equals:					
WMTD = 1,900,000/(6,502.8 + 3,979.62 + 7,826.26) = 103.78					

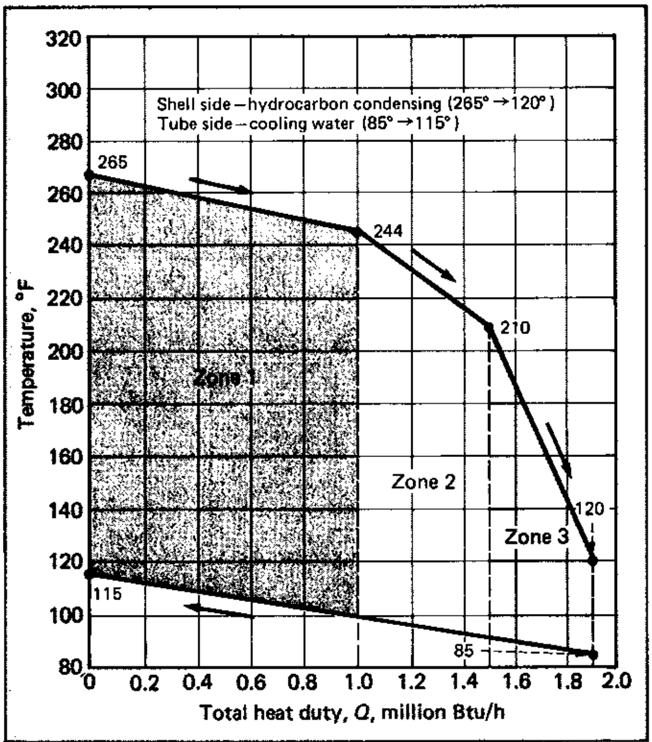

Figure 2 — Use inlet and outlet temperatures in calculating cold-side temperatures for each zone

But, when there is a phase change involved in the process — for example, condensation (Fig. 1b), or vaporization (Fig. 1c), or both (Fig. 1d), LMTD cannot be used directly. In this case, the designer must break the exchanger into two or more zones, and analyze each section separately. The resultant ΔT is called the weighted mean-temperature-difference (WMTD), and is expressed as:

$$\text{WMTD} = Q_T/[(Q_1/\text{LMTD}_1) + (Q_2/\text{LMTD}_2) + \ldots + (Q_n/\text{LMTD}_n)] \qquad (2)$$

where Q_T = Total heat duty of the unit, and subscripts 1, 2, and n are the first, second, and nth zones, respectively. The number of zones to be used is up to the engineering judgment of the designer. In general, there should be as many zones as necessary to provide straight-line sections of the heat-duty curve.

Practical design considerations

- If the heat exchanger is designed for true countercurrent flow (which usually is the most desirable design arrangement since less surface area is required for a given rate of heat flow), no correction factor need be applied to the calculated LMTD.† Also, a large temperature cross (i.e., where the outlet temperature of the cold side exceeds the exit temperature of the hot side) can be tolerated for such units.
- Countercurrent flow can always be assumed, if thermally possible, regardless of the range of the temperature cross as long as the temperature approaches (e.g., hot outlet and cold inlet temperatures) are greater than 5°F. Also, only TEMA "E" and "F"-type shells should be used for countercurrent flow designs, provided that the number of shell and tube passes are the same.
- Whenever a temperature cross exists, either (1) design the heat exchanger as a countercurrent unit, if thermally practical, or (2) increase the number of shells until the correction factor is satisfactory. (As a general rule, a minimum correction factor of 0.80 is considered satisfactory.) In the latter case, the correction factor should be based on intermediate temperatures (the temperatures between the shells) rather than temperatures from the plots for each zone.

An example

Hydrocarbon vapor is being condensed, using water in the tube side of a 1-2 exchanger, with no temperature cross (Fig. 2). The condensing curve is broken into three zones. In each zone, the temperature differences for the hot side and cold side are calculated, as can be seen in the table. Temperatures and heat duties can be read directly from Fig. 2. Notice, however, in the table that for the cold side, the inlet and outlet temperatures (85° and 115°F, respectively) are used for each zone. In multiple-tube-passes situations, this approximation provides a sufficiently good approach-temperature reading.

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†For complex heat-exchanger arrangements, the mathematical expression for the MTD becomes quite complex. The usual procedure is to modify the LMTD by a correction factor, which can be found in charts by R. A. Bowman, A. C. Mueller and W. M. Nagle (*Trans. of ASME*, May 1940, pp. 283-293), and by the Tubular Exchanger Manufacturers Assn. (TEMA). Also, see *Chem. Eng.*, Feb. 23, 1981, pp. 77-81, for another method of determining LMTD correction factors.

A two-stage deaerator can eliminate surplus low-pressure steam.

CONSERVING STEAM

Jan Barton*

Large volumes of low-pressure steam are probably wasted in your plant. Low-pressure steam, of course, heats process media and buildings, and deaerates feedwater and heats buildings. While high-pressure steam-consumption depends somewhat on relative humidity and plant-air temperature (especially for turbines at constant performance), use of low-pressure steam can fluctuate more, by 20% or greater, due to seasonal variations.

Thus, substantial surpluses of low-pressure steam may be generated in a system. This steam is simply condensed or vented to the atmosphere.

In those systems where low-pressure consumption varies only slightly, the capacity of the plant's conventional deaerator setup can be used to accommodate steam consumption: The liquid level in the deaerator is adjusted according to instantaneous consumption, by a method given by Doyle [1]. Of course, this method cannot offset a longterm reduction of low-pressure steam consumption. However, this longterm surplus can be eliminated by rethinking the deaeration concept and adopting the layout shown in the figure, suggested by the author [2].

The figure illustrates a simple system having medium-pressure (MP) and low-pressure (LP) steam. MP steam is consumed in a turbine having two LP-steam exhausts, E_1 and E_2. E_1 is held at constant pressure. E_2 may fluctuate and is always lower than that of E_1. Various steam consumers (boilers, heat exchangers, etc.) are connected to E_2. So is the first stage of feedwater dearation, S_1, a stripper open to the atmosphere via a throttling valve.

Feedwater, introduced at the top of S_1, is a conditioned mixture of LP-steam condensate from the E_2 fraction and demineralized fresh water. LP steam, also from E_2, is fed into the bottom of S_1 and condenses. A small amount of steam, along with the first fraction of gases in the water, is vented. The aqueous phase from S_1 is passed to S_2, as is the E_1 LP-fraction.

This two-stage deaeration enables the temperature of the steam-boiler feedwater, which comes from the second-stage deaerator, to be held constant. The feedwater boiling point depends on the pressure of the steam fed to the deaerator.

Because the pressure of this steam is kept constant, the feedwater temperature also stays constant, at the boiling point of water at the pressure of the LP steam, E_1.

More to the point, controlling the pressure in the LP subsystem, E_2, makes it possible to regulate the amount of this steam such that a change in the consumer needs will not lead to surpluses of LP steam. If the LP-steam users consume less steam, the pressure of E_2 is decreased too.

A constant power output of the turbine can be achieved at a lowered LP steam demand, while the MP-steam parameters (temperature and pressure) remain constant. The MP-steam flowrate is decreased.

A constant power output could also be achieved by raising the temperature and pressure of the MP steam at a constant pressure of the LP steam. This, however, would be uneconomical, as it would increase the costs of steam generation.

References

1. Doyle, John B., Storing Steam by Changing Water Level in a Deaerator, *Chem. Eng.*, July 7, 1986, Vol. 93, No. 13, p. 77.
2. Bartoň, et al., Two-stage Degassing of Feedwater for Steam Boilers, *Chem. Prum.*, Vol. 35, No. 9, 1986, p. 459.

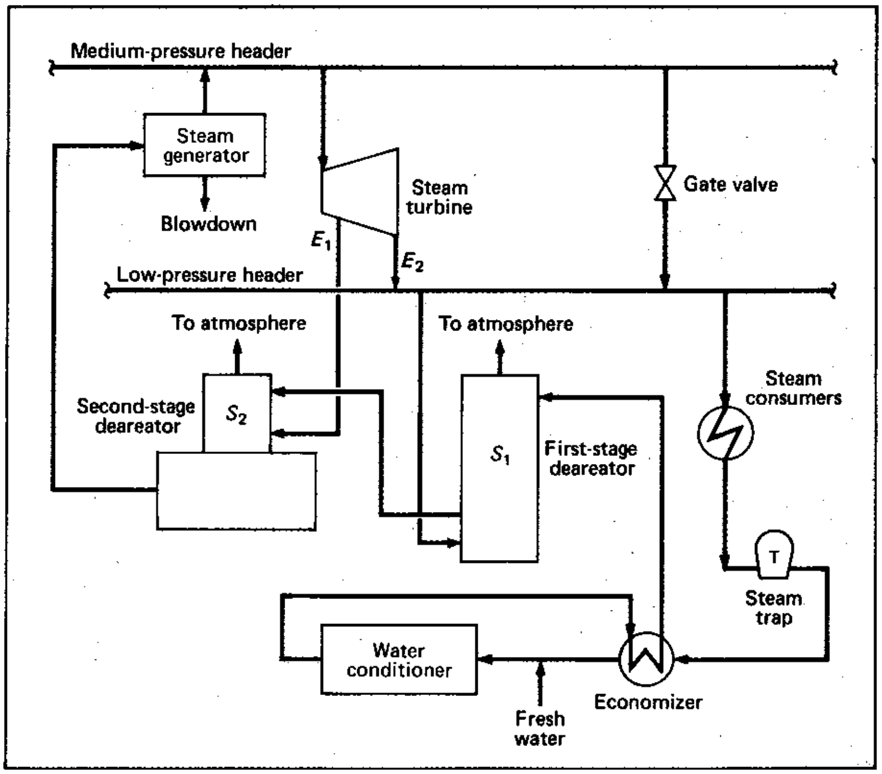

Figure — This two-stage scheme for deaerating feedwater eliminates the wasting of low-pressure steam

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*Chemopetrol, Research Institute of Inorganic Chemistry, Systems Engineering Laboratory, 400 60 Ústí nad Labem, Czechoslovakia.

Operating boilers intermittently

In many cases, particularly in batch operations, steam generating equipment can be shut down for part of each day (or on weekends), saving money on fuel. But this can cause problems if you are not aware enough to forestall them.

Charles A. Faust,
Hercules Inc., Water Management Chemicals

During periods of economic downturn, many plants find it necessary to limit production. For steam generation equipment, this translates into idle periods and intermittent operation. For instance, shutting down steam production at night, when it may not be needed, can save considerable sums of money. Under such a condition, several problems must be faced.

First, as equipment cools, it can pull in air that will dissolve in any available water or moisture. (Fig. 1 shows that the solubility of oxygen increases as temperatures drop.) This aerated water is highly corrosive to boiler metals. During periods of intermittent operation, steam and condensate lines are in particular jeopardy.

Most operators of this equipment are well prepared to minimize oxygen corrosion while the units are running. They also know how to prevent corrosion in boilers that are to be laid up for long periods of time. The real challenge is to prevent or reduce the corrosive effects of oxygen when the equipment is shut down on a weekly, or even daily, basis.

Another common problem is that particulate and soluble iron returns to the boiler by way of the condensate and feedwater. When water circulation is discontinued, these solids are much more likely to settle out on internal surfaces. Frequent outages increase the opportunity for these deposits to grow to the point where problems can result, since cleaning of the surfaces rarely takes place under these circumstances.

During shutdowns, water can also condense on the fireside of the boiler. Each shutdown results in a recurrence of this condensation phenomenon. The moisture can combine with acidic residue on the tubes and cause severe corrosion. If allowed to continue, tube failures can occur. The problem is most pronounced in boilers burning sulfur-laden fuels, even when such fuels are burned only occasionally.

During idle periods, fresh or partially exhausted ion-exchange equipment will be shut down, and flows will be discontinued. If the exchangers are not properly rinsed before being put back into operation, they can release hardness ions (or other unwanted ions) into the feedwater.

While dollars can be saved by shutting down the steam generating equipment when steam is not needed, there is a cost to the boiler in terms of loose rolled joints and cracked or loosened refractory. The effects of frequent heating and cooling cycles can be minimized by keeping the boiler warm with an auxiliary heat source.

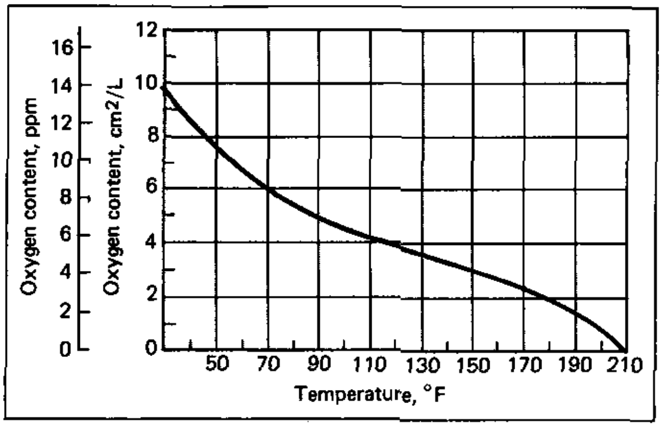

Figure 1 — Solubility of oxygen in water goes up as temperature falls

The problems cited above are currently being experienced by many industrial steam-generating plants. While there is no simple solution to all of them, a number of steps can be taken to lessen the adverse effects.

Boiler corrosion

Boilers that always maintain some pressure do not need supplemental treatment. Those that cool sufficiently to allow air to enter the steam space can be given additional protection by feeding extra oxygen scavenger, such as sulfite or hydrazine. A sufficient residual in the boiler will react with any oxygen that is contacted. An extra dose of oxygen scavenger to the boiler in the hours (up to a day) before shutdown will be beneficial. If the boiler is to be shut down each night, a continuously elevated level of sulfite or hydrazine is recommended.

The water in the storage section of the deaerator should also receive a higher than normal level of oxygen scavenger for two reasons. First, this water will not be deaerated again before going to the boiler, so any oxygen that is dissolved will be available for boiler and condensate-line corrosion. Further, the deaerator storage section is not usually designed for corrosive conditions, and so would be susceptible to its own oxygen attack.

The spaces above the water level in the boiler and in the deaerator may experience flash rusting, but this should not be too serious over very short periods of time. If the boiler is to remain idle for more than a day or two, it should be completely filled with deareated water containing a substantial residual of scavenger. Steam or nitrogen blanketing of the steam space and the deaerator will help on longer shutdowns. These recommendations are consistent with those that would be made for long-term storage of boilers. (**Important:** Appropriate safety precautions should be observed when entering vessels that have been blanketed with nitrogen.)

Corrosion

$$\underset{\text{magnetite}}{4\,\mathrm{Fe_3O_4}} + \mathrm{O_2} \longrightarrow \underset{\text{hematite}}{6\,\mathrm{Fe_2O_3}}$$

Passivation

$$\underset{\text{hematite}}{6\,\mathrm{Fe_2O_3}} + \mathrm{N_2H_4} \longrightarrow \underset{\text{magnetite}}{4\,\mathrm{Fe_3O_4}} + 2\,\mathrm{H_2O} + \mathrm{N_2}$$

Figure 2 — Reaction of hematite and hydrazine to form magnetite

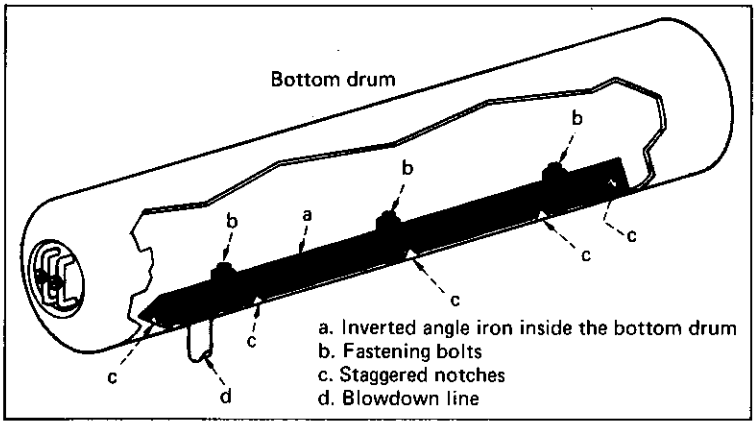

Figure 3 — Angle iron installed in mud drum will aid in removing particulates

Steam/condensate lines

These lines are especially vulnerable during even brief shutdowns. Neutralizing amines, which are normally used for protection during operation, do little to prevent oxygen corrosion. Corrosion products can return to the boiler and contribute to deposits on internal surfaces. Condensate maintained at a pH of 8.5 to 9.0 with neutralizing amines will experience a moderation of oxygen corrosion, owing to the pH alone. However, pitting rates could still be unacceptably high.

Filming amines provide protection from oxygen attack, but in a non-operating system the film is not permanent. These amines will provide some corrosion protection for short idle times.

Hydrazine can be fed to the condensate system during operation to help promote the formation of a magnetite film in the the return lines. This surface is more corrosion-resistant than the bare metal, and will provide some protection during short outages. Fig. 2 shows the simplified reaction of hematite and hydrazine to form magnetite.

The combined use of a filming amine and hydrazine may provide the best corrosion protection available for an idle steam/condensate system. Not only will the metal be protected by a resistant magnetite layer, but the amine film will keep oxygen-laden water and atmosphere away from the surface.

Membrane filter tests can help to determine the degree of protection a given treatment is providing. Based on the results, it may be desirable to consider dumping condensate for some period of time upon startup. This could result, for instance, from excessive iron levels being present.

The primary incentive for returning condensate is to recover its heat value. The first condensate that returns after a shutdown will have very little recoverable heat value and will contain higher levels of unwanted corrosion products. However, the level of corrosion products will probably be higher than normal even when hot condensate starts returning to the boiler house.

Ion-exchange equipment

Softeners, polishers and demineralizers, even those that are freshly regenerated, will produce high-hardness effluents when started up after an idle period, due to ion reversal. To ensure that these troublesome ions are not inadvertently sent to the boiler, the ion-exchange units should be rinsed prior to being put back into service. In some situations, demineralizers will benefit if water is recirculated through the unit continuously.

Boiler iron deposits

As stated above, intermittent operation can cause varying concentrations of iron in the condensate system. Heavily iron-fouled condensate should be dumped if its use in the feedwater would be detrimental to good boiler operation. The criteria should be the ASME Guidelines and past plant history. Some form of condensate polishing may be worth considering if the condition is severe or continuous.

Since more iron will be entering the boiler during a start-up, an increased amount of sludge conditioner should be added for the first day. Further, if iron is an ongoing problem, an iron-specific component should be added to the sludge-conditioning program. Generally, sludge-conditioning materials can safely be added at substantially higher than normal operating dosages, with no negative effects. Care must be taken to ensure that the conditioner being fed does not exhibit problems on overfeed. In addition, carryover potential would have to be watched during these periods.

Mechanically, an inverted angle-iron of suitable design in the mud drum will aid in removing unwanted particulates from the boiler. Such a device is shown in Fig. 3.

Increasing the frequency of the manual blowdowns during the first day of operation following a shutdown will help reduce suspended solids. Obviously, boiler design must be considered prior to changing any of the mechanical devices or operational procedures. For instance, heavy manual blowdown on boilers with small headers instead of large mud drums could lead to starved tubes and rapid failures. In such a case, other methods would have to be evaluated.

Problems often develop when iron in the boiler is deposited on furnace wall tubes adjacent to the flame. This can be a function of conditions on the fireside of the boiler as well as on the waterside. In order to prevent overheating of furnace wall tubes, every effort must be made to keep the flame from impinging on boiler tubes. Such flame impingement can cause problems even with excellent water quality and treatment. Factors that can cause a faulty, or spread, flame pattern are:

- Incorrect positioning of the burner.
- Too little excess air/too much fuel.
- Low atomizing steam pressure.
- Worn-out burner tip.
- Low oil temperature.
- Wet atomizing steam.

Table — Checklist: Items to consider for intermittent boiler operation

Reminder checklist*

Prior to shutdown

☐ Is condensate pH 8.5–9.0?
☐ Has filming amine been used?
☐ Is there a volatile oxygen scavenger in condensate?
☐ Are there high levels of oxygen scavenger in boiler?
☐ Are there high levels of oxygen scavenger in deaerator?
☐ Have tubes been sootblown in proper sequence?
☐ Has an effective fuel-oil additive been used?
☐ Has magnesium oxide been blown into furnace?

During shutdown

☐ Is there a steam blanket in steam drum and deaerator?
☐ Is there a nitrogen blanket in steam drum and deaerator?
☐ Is there warm dry air throughout furnace?

Prior to and during startup

☐ Has feedwater iron been monitored? (Dump condensate if it is of unsatisfactory quality.)
☐ Have softeners and demineralizers been rinsed to achieve suitable quality effluent?
☐ Has extra sludge dispersant been used?
☐ Have more-frequent bottom-drum blowdowns been carried out?

*Not all items will apply to all situations. Modify to suit.

The most commonly used fuels, oil and coal, leave a residue on the boiler and economizer tubes. The composition of such deposits will vary, depending on the fuel and its source, but none have beneficial characteristics. When the boiler is taken off line, and the tubes cool, moisture can be absorbed by these deposits, leading to corrosion; this can be particularly severe if sulfur is present. The burning of fuels low in sulfur is one way to help, but this by itself will not cure the problem. Since fireside corrosion causes thinning of the tubes, it may take several years to be noticed. By then, the problem is often widespread and severe.

To reduce deposition on boiler tubes, the soot blowers should be used just prior to shutdown. The tubes farthest from the stack should be blown first, followed sequentially by the others, with the tubes closest to the stack being blown last.

Boilers that are idle for long periods are normally washed down and dried, but this is not practical when the unit is taken out of service frequently for short periods of time. Other methods must be evaluated for these conditions.

Keeping the tubes warm and dry will prevent serious corrosion damage; passing warm dry air through the firebox is one way to accomplish this. Another method is to have a steam line that runs inside the length of the mud drum and the returns, to act as a heat exchanger for keeping the water and tubes hot. Steam must come from an operating boiler.

Blowdown water from an adjacent boiler can be piped into the bottom of the idle boiler, with the overflow from a point at the top of the steam drum going to a suitable drain. This will protect the waterside from the intrusion of oxygen and will help to keep the tubes hot and dry on the fireside. If this method is employed, protection of the superheater must be taken into consideration.

The continuous use of effective fuel additives, such as the newer generation of organometallics, not only will help to minimize the quantity of deposit but also will increase the deposit's pH. As an alternative approach, powdered products containing magnesium oxide can be blown into the furnace during each shift for a day or two prior to shutdown. This will cover the existing deposit and tube with a coating of significantly higher pH. However, magnesium oxide blown into the furnace will tend to follow the fast-moving gas path and provide very little protection to the firebox and floor tubes.

Continuous addition of effective organometallics is more desirable because this method of feed will affect the entire deposit and not just the surface area. Plants that burn mostly gas, with heavy fuel oil being burned only part of the time, also are prone to the problems mentioned above.

Conclusion

During times of intermittent operation, every effort should be made to prevent oxygen- and acid-caused corrosion of the equipment throughout the steam generating system. Some methods are accomplished quite simply, while others will require more effort. The system will benefit from every improvement that can be made. The comments and suggestions outlined above should serve as no more than a starting point for developing specific procedures. It has been said that most boiler damage takes place when the equipment is not operating. This potential can be controlled with well-designed procedures for long- and short-term storage.

The table contains a checklist of items to consider.

***Roy V. Hughson**, Editor*

The author

Charles A. Faust is technical support manager for Water Management Chemicals, Hercules Inc., Wilmington, DE 19894, where he has also worked as a technical representative and in field-sales management. He studied engineering at the University of Lowell (Mass.), received a B.A. from Boston University, and did graduate work at the State University of New York, Albany. His 17 years in water treatment include troubleshooting boiler and cooling-water-treatment problems, emphasizing both chemicals and equipment in many types of industries. He has also given seminars, lectures and training sessions on these topics.

Surface condensers

1. Find the most compact surface condenser

K. N. Murty*

Manufacturers of plain- or finned-tube surface condensers generally present heat-transfer data as overall heat-transfer coefficients and tubeside heat-transfer coefficients. These coefficients are obtained from tests on a single tube. Heat-transfer rates are generally based on the inner-tube surface area.

These data must be carefully interpreted to obtain overall heat-transfer coefficients for condenser tube bundles, e.g., horizontal condensers. Based on h_{cs} for a single tube, the condensing heat-transfer coefficient for a bundle, h_{cb}, can be found from Eq. 3.4.6-22 of Ref. [*1*]:

$$h_{cb} = h_{cs}F_i^{-1/6} \tag{1}$$

From Ref. [*1*]:

$$F_i = 0.78D_b/P_t \tag{2}$$

Thus:

$$F_i^{-1/6} = 1/1.077(D_b/2P_t)^{-1/6} \tag{3}$$

Since Eq. (3) uses vertical tube-pitch, it accounts for triangular and square pitches.

In Eq. (3), D_b is related to D_s by:

$$D_b = D_s - 2C_{bs} \tag{4}$$

D_s, however, is obtained from equations that relate N_t, P_t, n and constants that depend on whether the pitch is square or triangular. The variables are related using Eqs. 10-1 and 10-2 of Ref. [*2*].

It is well known that the number of tubes affects the overall heat-transfer coefficient. For example, consider 3/4-in.-O.D. tubes arranged in a 1-in. triangular pitch, with 4 passes on the tubeside. The number of tubes, N_t, is related by Eq. (10-1) [*2*]:

$$N_t = \frac{\left[(D_s - 1.08)^2\frac{\pi}{4} - 0.9\right] - 1.96(D_s - 1.08)}{1.223} \tag{5}$$

or:

$$D_s = 2.3278 + \sqrt{2.703 + 1.5572N_t} \tag{6}$$

Assuming $C_{bs} = 1/2$ in., and using Eqs. (4) and (6), the tube-bundle diameter is given by:

$$D_b = 1.3278 + \sqrt{2.703 + 1.5572N_t} \tag{7}$$

Since $N_t \gg 1$, Eq. (7) can be approximated by:

$$D_b = 1.248\sqrt{N_t} \tag{8}$$

Using Eqs. (1), (3) and (8), the condenser heat-transfer coefficient for a bundle is (since $P_t = 1$):

$$h_{cb} = 1.00443h_{cs}N_t^{-1/12} \approx h_{cs}N_t^{-1/12} \tag{9}$$

In Eq. (9), the value of h_{cs} is calculated from manufacturer's data for a single tube:

$$\frac{1}{h_{cs}} = \frac{1}{U_s}\left[\frac{A_o}{A_i}\right] - \frac{1}{h_i}\left[\frac{A_o}{A_i}\right] \tag{10}$$

Or

$$h_{cs} = \frac{A_i}{A_o}\left[\frac{U_sh_i}{h_i - U_s}\right] \tag{11}$$

From Eqs. (9) and (11), and neglecting the tube-wall thermal resistance, the expression for the overall heat-transfer coefficient for the bundle is:

$$\frac{1}{U_b} = \frac{A_o}{A_i}\left[\frac{h_i - U_s}{U_sh_i}\right]N_t^{1/12} + f_o + \left[\frac{1}{h_i} + f_i\right]\frac{A_o}{A_i} \tag{12}$$

In Eq. (12), N_t can be replaced by nN_p.

Eq. (12) relates U_b to N_t and helps to quickly evaluate U_b, for a given total number of tubes, N_t, or vice-versa, for horizontal surface condensers.

Example I

Find U_b for a horizontal condenser, given the following:

$h_i = 8{,}000$ kcal/(m^2)(h)(°C)
$U_s = 400$ kcal/(m^2)(h)(°C)
$f_o = 0.0001$ m^2–h–°C/kcal
$f_i = 0.0002$ m^2–h–°C/kcal

Nomenclature

A_i	Tube inner-surface area, m^2
A_o	Tube outer-surface area, m^2
A_{ol}	Outside surface area of tube per unit length, m^2/m
C_{bs}	Clearance between bundle and shell, in.
D_b	Dia. of tube bundle, in.
D_s	Inside dia. of shell, in.
F_i	Kern's factor for bundle due to inundation
f_i	Fouling factor for tubeside fluid, m^2–h–°C/kcal
f_o	Fouling factor for shellside fluid, m^2–h–°C/kcal
h_{cb}	Condensation heat-transfer coefficient for tube bundle, kcal/(m^2)(h)(°C)
h_{cs}	Condensation heat-transfer coefficient for single tube, kcal/(m^2)(h)(°C)
h_i	Tubeside heat-transfer coefficient, kcal/(m^2)(h)(°C)
L	Tube length per tubeside pass, m
N_p	Number of tubes per tubeside pass
N_t	Total number of tubes
n	Number of tubeside passes
p_t	Vertical tube pitch, in.
Q	Heat duty, kcal/h
ΔT	LMTD (logarithmic mean temperature difference), °C
U_b	Overall heat-transfer coefficient for tube bundle, kcal/(m^2)(h)(°C)
U_s	Overall heat-transfer coefficient for single tube, kcal/(m^2)(h)(°C)

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*Bharat Heavy Electricals Ltd., Vikas Nagar, Hyderabad 500-593, India.

$$A_o/A_i = 3.5$$
$$N_t = 1{,}000$$

Using Eq. (12), $U_b = 62.42$ kcal/(m^2)(h)(°C). In general, similar calculations can be made for given values of h_i, U_s, f_o, f_i, A_o/A_i and N_t, for various types of tube pitches, (triangular, square, etc.), tube sizes and tube-pitch values. For a given heat duty, Q, of a horizontal condenser:

$$Q = U_b A_{ol}(nN_p)L\Delta T \qquad (13)$$

Since $U_b = f(nN_p)$ from Eq. (12) for given fluids and working temperatures, it follows from Eq. (13):

$$Q = A_{ol}L\Delta T\Phi(nN_p) \qquad (14)$$

where:

$$\Phi(nN_p) = nN_p f(nN_p)$$

Thus, for a given Q and ΔT, a suitable combination of A_{ol}, L and nN_p can be arrived at, using Eq. (14). This results in a compact condenser, as illustrated in Example II.

Example II

A Freon-11 condenser has integral low-fin tubes. Data are:

$Q = 3.0 \times 106$ kcal/h
$A_o/A_i = 3.5$
Saturation temperature of Freon-11 = 40°C
Cooling-water inlet temperature = 32°C
Cooling-water outlet temperature = 37°C
h_i (cooling-water side) = 8,000 kcal/(m^2)(h)(°C)
U_s (based on A_i) = 4,000 kcal/(m^2)(h)(°C)
$f_i = 0.0004$ m^2–h–°C/kcal
$f_o = 0.0002$ m^2–h–°C/kcal

Find the most compact condenser among these four alternatives:

Alternative 1: $N_t = 2{,}400$, $A_{ol} = 0.15$ m^2/m. Eq. (12) gives $U_b = 347.9$ kcal/(m^2)(h)(°C). Since $\Delta T = 5.098$°C, using Eq. (14), $L = 4.699$ m. From Eq. (6), $D_s = 63.483$ in. = 1,612.5 mm.

Alternative 2: $N_t = 2{,}400$, $A_{ol} = 0.20$ m^2/m.

For this case, $L = 3.524$ m, $D_s = 1.477.2$ mm.

Alternative 3: $N_t = 2{,}000$, $A_{ol} = 0.15$ m^2/m.

For this case, $L = 5.613$ m, $D_s = 1.477.2$ mm.

Alternative 4: $N_t = 2{,}000$, $A_{ol} = 0.20$ m^2/m.

For this case, $L = 4.21$ m, $D_s = 1.477.2$ mm, making this condenser the most compact.

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2. Does your surface condenser have spare capacity?

Adnan Y. Irhayem*

Often, there exists spare capacity in a surface condenser, one that condenses exhaust steam from a centrifugal compressor or from a steam turbine under vacuum. This spare capacity may often be over 10% of the normal operating capacity and can be attributed to one or more of the following:

- The condenser may have been designed with a spare duty.
- The unit is not at full capacity on the process side.
- Performance improvements because of positive changes in the cooling water (i.e., its quality, velocity, etc.).

It is easy to check the existing duty from the enthalpy of steam and condensate, knowing the flowrate of both, and compare values with design figures. The calculated results also coincide with the heat removed by the cooling medium, normally water.

Plant experience

In one case, a spare duty of more than 6.0 million Btu/h (1.5 million kcal/h) was found to exist in a condenser. This duty could be utilized to condense over 2.5 tons/h of nearby waste low-pressure steam to a useful condensate. As a safe margin, we decided to condense only 1.5 tons/h. No additional driving force was necessary, since the steam pressure was always higher than that of the surface condenser. The condenser normally operates at a pressure below atmospheric, so that the change involved only adding suitable-size piping, valves, plus a flowmeter.

Nomenclature

d	Internal pipe dia., in.
L	Length of pipe, mile
P_s	Standard pressure, psia
P_1	High pressure, psia
P_2	Low pressure, psia
Q_s	Steam flowrate, lb/h
S	Density of steam (or gas), lb/ft^3
T	Fluid temperature, R
T_s	Standard temperature, R

To add another steam line to the surface condenser, one can either make a tie-in to the existing steam inlet or create a new entry point. For a new point, a suitable impingement plate is needed. Even if lengthy piping is needed, the arrangement is still economical, offering a short payback.

Finding pipe size

To estimate pipe size, the Weymouth equation can be used for low-pressure steam or gas flow:

$$Q_s = 443.45\left(\frac{T_s}{P_s}\right)d^{2.267}{\frac{(P_1^2 - P_2^2)}{LST}}^{0.5}$$

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*Engineering Dept., Daura Refinery, Baghdad, Iraq.

Oil removal from condensate

It is normally economical to return condensed steam to boilers. However, if the condensate becomes contaminated with oils, problems can arise. Here is how to detect and remove such impurities.

Samuel S. Mills
Betz Laboratories

Maximizing the return of condensate to a boiler's feedwater system becomes more important as plants strive for lower operating costs and capital expenditures. Because of its significant heat value and high quality, condensate is an extremely valuable commodity. Its use as boiler feedwater reduces internal chemical-treatment costs and makeup water requirements.

Condensate, however, may not always be usable as boiler feedwater, due to the presence of certain process contaminants (see Table I). When steam is used for heating, to power equipment, and for varied process applications, condensate comes into contact with oil and other impurities. Contamination from fuel-oil heaters or leaking seals in steam-driven equipment, such as pumps, turbines or compressors, is common. Also, the raw water used as plant makeup, may contain significant levels of solvents, fuel oils, gasolines and greases. In addition to these general contamination sources, each industry has its unique ones (Table II). If any of these contaminants become entrained in the returned condensate stream, numerous and sometimes unpredictable problems in the boiler and related systems can occur.

Foaming

With long-chain hydrocarbons, such as oil, where hydrophobic and hydrophilic portions of the molecule exist, surface-tension effects will be significant, and may result in foaming and possible acceleration of carryover. The alkalinity of the boiler water may saponify any fatty acids present, producing a crude soap that may foam. Soaps, sulfated oils and alcohol, sulfonated aliphatics and aromatics, quaternary ammonium compounds, nonionic organic ethers and esters, and various fine particles, act as emulsifiers that can increase foaming.

Deposits

Oil can cause feedwater hardness to form hard adherent scales rather than a soft sludge that could be removed during blowdown. In a phosphate treatment program, the oil can delay precipitation of hardness until it reaches the hotter areas of the boiler, where it then forms a tight adherent scale. The oil may also serve as a nucleus or binder for scaling in hot areas of the boiler. In addition, any sludge or suspended particles present in the boiler can be coated with oil and become sticky. The particles agglomerate and then stick to the tube walls. The oil in these deposits may then volatilize or char at high temperatures and leave a hard scale on the surface. Occasionally, loose oil-bearing scales will form in water-wall tubes. If these scales break loose and block tubes, failure due to starvation will result.

pH depression

Many organics, particularly sugars, completely carbonize at medium boiler pressures. This carbonization produces organic acids as the sugars are reductively decomposed. In high-pressure boiler systems using coordinated phosphate/pH programs, these acids can drive the pH down to 3 or 4.

Condensate contamination problems

Oil in the condensate system can cause serious problems. Alone or in combination with corrosion products, which it can cause to slough off of pipe walls, oil can plug steam traps,

Table I — Contaminants commonly found in condensate

Sugars
Greases
Naphtha
Kerosene
Hexanes
Fatty acids
Gasoline

Fuel oils, Numbers 1 through 6
Animal and vegetable fats and oils
Machining and cutting oils
Lubricating oils (natural and synthetics)
Black liquor (lignosulfonate)
Reduced crude oil
Furfural

Table II — Condensate contamination sources in various industries

Industry	Contaminant	Source
Hydrocarbon processing	Oils, crude oil and derivatives	Process units [1] Ejectors Crude distillation units Vacuum towers Catalytic reformers Fluid-catalytic- or hydrocrackers
Pulp and paper	Black liquor	Multiple-effect evaporators Indirect-contact digester
Automotive	Machine and cutting oils	Heating coils in cleaning and plating tanks
Tire and rubber	Oils, rubber particles, release agents	Tire molds
Food processing	Animal fats and oils Fatty acids Sugars	Heating coils in soaking pits, cooking vessels, evaporators

valves, and heat exchangers, as well as increase boiler deposition. Besides the additional maintenance required for replacing and unsticking traps and valves, considerable inefficiency and energy loss can occur. If condensate with oil is passed through condensate polishers, the oil will coat the resin and cause poor ion removal.

High oil concentrations can also cause plugging of the bed. This leads to excessive pressure drops, resulting in frequent bed cleaning, downtime, and high maintenance costs.

Detection and monitoring

Visual detection is the simplest and quickest way to detect and monitor contamination. This method may be enhanced by an inspection tank equipped with glass portholes. A light should be mounted outside one porthole so that a person can look through the opposite one and see the light reflected on the water. Oily contamination, unless it is truly soluble, will produce a rainbow effect on the water surface.

The tank should be portable so that it can be moved to various receivers and condensate-return taps around the plant. This will enable isolation of contaminated-condensate streams for dumping or treatment.

Another quick visual detection method is conducted by placing a few slivers of camphor into the sample to be tested. If the sample contains oil, the slivers will remain stationary (they spin in an oil-free sample). The camphor test will detect oil in any water sample (even water that appears oilfree to the naked eye) and is 100% accurate.

Analysis

There are several laboratory tests available for identifying the type of contaminant and its concentration. If onsite laboratory facilities are not available, these tests can be conducted by an independent laboratory or a water-treatment consultant. Tests include gravimetric analysis, infrared analysis, spot testing, chromatography and measurement of total organic carbon (TOC).

Equipment is available that can employ laboratory analytical techniques in instruments suited for online monitoring and detection of oily contaminants. These instruments use the principles of solvent extraction, infrared, spot testing, TOC measurement, and photometric measurement of visible and ultraviolet adsorbence. They are not subject to interferences from suspended solids, air bubbles, color, oil droplet size, or dissolved solids in the water.

Prevention of contamination

Oily contamination can be kept from entering the boiler by:

Fixing leaks — Although this may be the most obvious prevention method, it is not always the simplest or most convenient. In a large refinery, for instance, where there are hundreds of heat exchangers, reboilers, and steam-heated coils containing thousands of tubes, locating and fixing a leak can be time consuming and very difficult. Similarly, replacement of a reciprocating air compressor may be too expensive, even though boiler problems have occurred.

Dumping contaminated condensate — This is the most effective stopgap measure to apply until the contamination source is located or repaired. It may also be the only answer if contamination levels are extremely severe or no removal equipment is available. Dumping contaminated condensate, however, may not be practical, since it may constitute a large percentage of the the plant's boiler feedwater.

Removal techniques

Even with special precautions, condensate may become contaminated with oily contaminants. Since the condensate is too valuable to dump, oil removal or cleanup is required.

Filtration/coagulation — This is one of the most effective oil-removal techniques. Pressure filters have had long use in removing hydrocarbons from oil refinery condensate.

Choosing the filter medium to be used in a pressure filter is a very important consideration. The medium should be selected primarily for its ability to retain the contaminant to be separated, and to have an acceptable life in the filter environment. Filter media are manufactured from cotton, synthetic polymers, glass, cellulose, metal, carbon, refractories, and other porous or perforated solids, and sand and other particulate solids. The filter's effectiveness may be enhanced by the use of filter aids.

Ultrafiltration — This process has been successful with mixtures difficult to separate, such as oily machining wastes and oily wastewater. A pressure-driven filtration membrane separates multicomponent solutes from solvents, according to molecular size, shape and chemical bonding. Substances below a preselected molecular size are driven through the membrane by hydraulic pressure, while larger molecules, such as oil droplets, are held back. Effluent oil concentration depends on influent concentration, but properly operated ultrafiltration units can produce oilfree water (less than 0.1 ppm for all practical purposes).

Polymeric adsorption resins — A series of polymeric adsorbents has recently been developed that can adsorb hydrocarbons from aqueous systems. They derive their adsorptive properties from a combination of macroreticular porosity, pore size distribution, high surface area, and the aliphatic nature of their structure. Adsorbent resins can produce water containing less than 1 ppm oil.

A recently developed variation on these polymeric resins is the carbonaceous adsorbents. They are composed of hard, black, nondusting beads whose chemical composition is between that of activated carbon and polymeric adsorbents.

These adsorbents are designed for removing and recovering organics from other liquids and may be used at temperatures greater than 100°C. They have a capacity many times that of the best activated carbon for removing low concentrations of nonpolar or halogenated organics and can be regenerated by solvents, steam stripping, or both.

Coalescence (liquid-liquid separators) — Liquid-liquid separators are able to treat large volumes of oil-contaminated waters — up to approximately 2,000 gal/min. Effective separation (to 0.1% of original concentration) of two liquid phases having a specific gravity difference of 0.1 or less is easily accomplished. Liquid-liquid separators are used in some process industries to separate unstable emulsions found in steam distillation, solvent extractions, and organic reactions. They are not applicable for miscible liquids, chemically stabilized emulsions, or emulsions containing contamination, particularly of a gelatinous nature.

The concept of liquid separation is based on a porous membrane's being impervious to the flow of any liquid under certain conditions, provided the size of the capillaries through the membrane are smaller than the molecules of the liquid, and the critical pressure is not exceeded. The miscible liquid fills the cavities of the porous membrane and readily passes through, while the immiscible liquid is retained at the interface formed at the surface openings of the capillaries.

Prior to separation, coalescing is carried out as a preconditioning step. The enlarged drops then settle more rapidly.

Steam stripping — This technique has long been used on sour water and process condensate in refineries. Steam stripping is also becoming more attractive in other industries as the demand for water conservation increases. In a number of process plants, high-quality process condensates are recovered, stripped, and reused. Effluent oil concentrations of less than 50 ppm are achievable in stripping columns but actual concentration depends on influent concentration, tower design and contaminant type.

Hot-processing softening — An alternative method of oil removal, although not necessarily the most desirable, is to return the condensate to a hot-process softener. In the softener, the oil is removed by sorption on the normal precipitates formed during the softening process. This method effectively removes influent oil at concentrations of approximately 30 to 40 ppm. Effluent oil concentrations of less than 5 ppm can be obtained.

Flotation processes — Air flotation can be an effective method of treating oily condensate. This process speeds up emulsion separation by mixing dissolved or dispersed air bubbles with the oil droplets. The air decreases the oil density, which increases the rise rate of the droplets.

Three general methods are used in forming the minute air bubbles. One involves aerating the stream through submerged diffuser plates. The second, called induced air flotation, uses an agitator to physically beat the air into the water. In the third method, a sidestream of dissolved air-flotation effluent is aerated under 2 to 3 atm of pressure and then fed to the incoming contaminated stream. The sudden decrease in pressure on the sidestream causes breakout of minute bubbles, which attach to contaminants in the stream.

Treatment and cleanup procedures

Even after prevention and removal techniques are applied, oil or hydrocarbon contaminants can still enter the boiler system. Here are methods that can be applied during and after contamination of the boiler:

Foaming and carryover — Oils can cause or accentuate the foaming and carryover potential of a high-solids boiler water. No internal treatment can be totally relied upon to overcome oil-induced carryover. There are, however, highly-effective antifoaming agents that can significantly reduce carryover tendencies. In general, polymerized esters, alcohols, and amides are the most effective agents yet developed.

Deposit control — Internal treatment programs affect a boiler's tolerance to oil intrusion. For example, chelant-based treatment programs are much more tolerant than phosphate-precipitating ones because precipitation results in sludge that attaches to the oil. In a precipitating program (phosphate), oil in the feedwater delays the precipitation of hardness as sludge until it reaches the hot areas of the boiler, where it can form a hard scale as the oil volatilizes or chars. Oil contaminants are notorious deposit binders, due to polymerization and degradation reactions. The chelant program, on the other hand, maintains hardness in solution and, in combination with polymers, does not allow sludge to lay down on boiler heat-transfer surfaces.

Another method of controlling oil-contamination-related deposition, which may be used as an effective adjunct to the internal treatment program, is to run higher-than-normal boiler hydroxide levels. This is usually applicable on low-pressure systems where increased conductivity and alkalinity due to caustic addition may be tolerated without causing carryover. The increased alkalinity reduces the surface tension of the oil-coated bubbles and causes them to burst. In addition, this higher alkalinity increases the oil's solubility.

pH depression — Hydrocarbons can rapidly depress boiler pH to very acidic levels. When this occurs, it is extremely important to get the pH back to normal as rapidly as possible. Corrective action during low-pH excursions involves feeding caustic to elevate the pH, increasing the blowdown to remove solids generated, and raising the dosages of polymeric dispersants to aid in iron removal and prevent iron redeposition. Low-pH excursions are equivalent to performing an online acid cleaning, which results in removing and transporting large quantities of iron from the boiler heating surfaces, and thus in severe metal loss.

Cleanup

When severe oil contamination requires a boiler tube to be taken offline, the oil must be removed from the boiler's heat-transfer surfaces. This can be accomplished by alkaline boilout. If this is insufficient, an oxidizer must be applied.

Oxidizers have been very successful in instances where an alkaline boilout has not been so. A common situation where this arises is the heavy fuel-oil contamination of the oil heating system as a result of steam coil corrosion. The coking or decarbonizing of the organic contaminant increases the likelihood that an oxidizer will be necessary.

If oil contamination in the boiler is too severe, or the previously described cleanup techniques are unsuccessful, a reputable chemical-cleaning service should be consulted.

Roy V. Hughson, Editor

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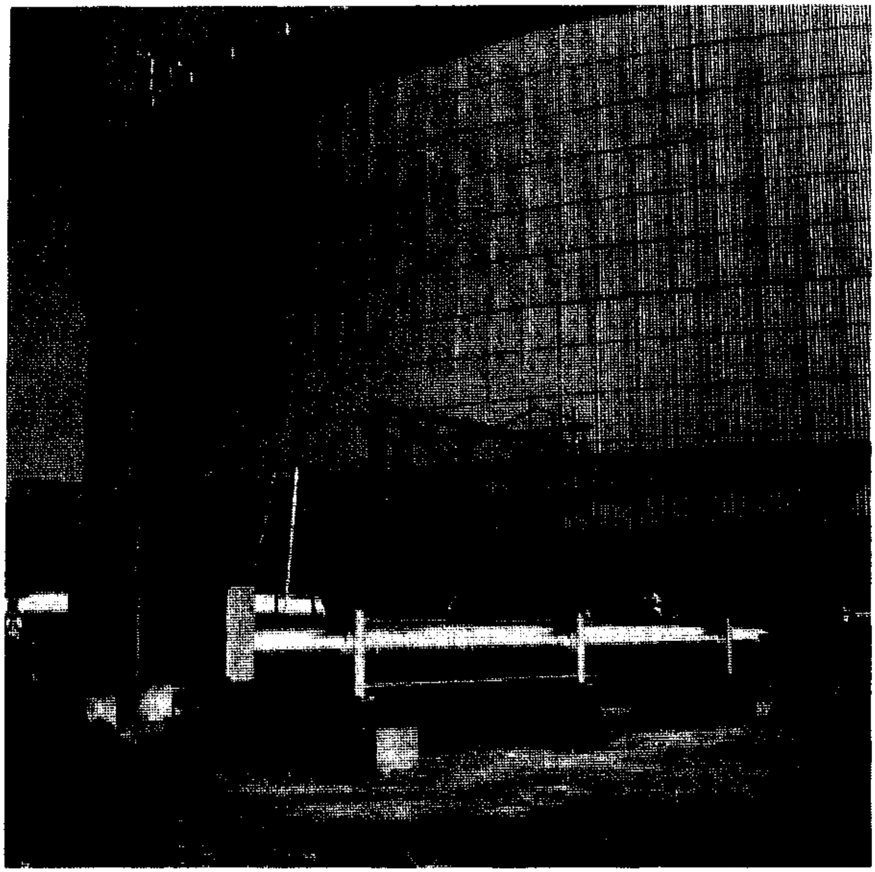
The author

Samuel S. Mills is a staff engineer in the Technical *Marketing Services, Boiler Group*, at *BETZ Laboratories*, Somerton Rd., Trevose, PA 19047, tel: (215)355-3300. His responsibilities include providing technical and marketing backup and support and training of the field sales forces. He holds a B.S. in chemical engineering from the University of Dayton.


Hynes Electric Heating Co.

This tank is steam heated to flow temperature. An electric inline heater (foreground) in the feedline brings fuel oil up to the higher temperature needed for combustion.

IMPROVE EFFICIENCY OF ELECTRIC HEATING

Here are several ideas for optimizing the use of electric process heating equipment.

George L. Koester, III
Consultant to Hynes Electric Heating Co., a subsidiary of Indeeco

Many plants in the chemical process industries (CPI) employ electric heating for warming large storage tanks, vapor- and liquid-phase heat transfer, trace heating, high-temperature process-gas heating, railroad tank-car heating, and other applications. Facilities can save power and reduce energy costs by making the most efficient use of electric heating.

Significant results may be obtained, for example, by: heating insulated tanks during off-peak hours; rescheduling batch operations; storing material in two tanks at different temperatures; installing feedline heaters; and using staged heaters. This article discusses these and other energy-saving approaches to electric process heating.

The first step in optimizing electric heating is to analyze the plant's total electricity usage for process heating. Start by answering these questions for each stage of the process:

- What are the requirements for the stage (actual, not assumed)?
- What is the hourly usage?
- Are there steps that run intermittently, and what is their timing, relative to each other?
- Which heating methods could be supplemented or replaced by electric heating? (Include factors such as the energy consumed by pumps to supply fuel and to return condensate.)

The answers to these questions will indicate which process areas could benefit most from more-efficient use of electric heating.

Managing power usage

Managing power consumption effectively is important, especially for those plants subject to "demand billing." Demand billing is based on a formula, rather than on actual usage alone; this formula includes the plant's highest demand level in any given period, in addition to actual usage, surcharges, and so on. (The utility must have that much power available if it is needed.) On the bright side, electrical process heating energy can be "free" if it fills the gap between actual usage and the limit.

For instance, upgrading the insulation on storage tanks makes off-hours heating an effective procedure. Enough warmth is retained through the daytime, when electric rates are high, to keep the material at process temperature; power for heating is drawn at night when rates are low.

Similarly, one easy schedule readjustment would be to have deliveries made at the end of the day for materials that had to be heated. The warming could be done overnight with cheaper power. And in a batch process, where energy goes for mixing, conveying, pumping, heating, and cooling, one could schedule the batching to synchronize some energy-consuming operations with the off-periods of other energy-consuming equipment.

In one example, a major distribution facility on the U.S. East Coast stores a variety of heavy petroleum fractions. Only tanks soon to be discharged are heated, with one inline heater serving multiple tanks sequentially. Valves isolate each tank and ensure that only one tank discharges (and thus requires heating) at a time.

CHECKLIST FOR ENERGY SAVINGS

Could the plant install:
- ☐ **More insulation,** and heat materials during off-peak hours?
- ☐ **Staged heaters** to maintain appropriate temperatures?
- ☐ **Auxiliary contacts** on motors, to reduce electricity usage?
- ☐ **Two storage tanks**, one small and hot, one large and warm?
- ☐ **Inline heaters** for specific point or line heating?
- ☐ **Suction heaters** so that no warmth is lost?
- ☐ **Oversized feeders,** and storage hoppers?

Or, could procedures be changed to accommodate:
- ☐ **Evening deliveries** and overnight warming?
- ☐ **Orchestrated power** use, for moderate, even, usage?
- ☐ **Synchronized batching** to take advantage of lower rates?

Tailoring demand can reduce capital investments related to electric supply. For instance, good energy management can minimize transmission feeder-line sizes and plant transformer sizes. If all plant equipment were running, a 5,000-A feeder might be needed, but with effective energy management and process scheduling, a lower-cost 4,000-A feeder may suffice.

Equipment enhances efficiency

In addition to scheduling and procedural changes to improve energy efficiency, equipment additions or modifications might be made. Instead of heating a single large storage tank, consider a two-tank approach: A large tank holds a long-term supply and is warmed only enough to ensure that the contents flow to a smaller tank, which holds a one-day supply at process temperature. There is much less heat loss with one small hot tank and one large warm tank than with one large hot tank alone.

If operations run 24 hours a day, a two-tank setup may not be practical. But there is an alternative: Install an inline heater in a feedline from the large low-temperature storage tank and warm the material as it is fed. For example, a Southern U.S. petroleum refinery burns heavy hydrocarbon material for fuel, which must be at 425°F for atomization in burners. The material is distributed from a 5,000-bbl heated tank to several burners through a mile-long 12-in. pipe. The tank and line are insulated, but heat loss from their large surface area is substantial.

Rather than maintain a high-temperature throughout the line, the refinery stores the material at flow temperature, 325°F. Separate inline heaters at each burner raise the fuel to atomization temperature. This "trim" heating often improves burner efficiency as well. Savings from reduced heat loss more than offset the cost of adding and operating the trim heaters.

A variation of the inline heater is the baffled "suction heater." Inserted in a large tank and surrounded by baffles, this heater warms a limited region of the material around the outside of the baffle. The temperature is high enough for the warmer material to flow into the heater, where it reaches the process temperature. Colder, more-viscous material moves down, is warmed, and then flows into the heater. Virtually no heat is lost, because the surrounding tank contents are warmed, making this type of heating very efficient.

Staged heaters represent a different approach, which is effective when two or more heat levels are required. A staged heater has several different settings, rather than just on and off, and can be operated incrementally so that only a modest amount of energy is consumed when the material being heated is close to its desired temperature.

An example is the use of inline staged heaters on a line supplying asphalt to a shingle-manufacturing machine that produces shingles of either conventional asphalt/felt, or asphalt/glass-fiber. The asphalt temperature at the machine must be higher than the temperature at which the asphalt can be pumped through the system, and each kind of shingle requires a different temperature. Consequently, inline heaters installed immediately before the machine heat the asphalt to application temperature. Heat losses from the line are reduced because the asphalt is at a lower temperature. Staging in the inline heater matches heating to the different temperature requirements; thus, warmth is applied only where and when it is needed.

Energy consuming equipment that operates continuously could be resized and run intermittently. For example, a conveyor resized for more capacity than necessary for continuous operation could be run intermittently to feed gravity storage hoppers during times when energy use is below peak. Similarly, a heater can have a low-energy stage to limit peak power consumption during times of high demand.

A related idea is to add an auxiliary contact to equipment such as the motor starter or an intermittent-operation high-horsepower motor. When the motor operates, the contact ensures that the heaters are off, and another contact ensures they do not all run at once.

The author

George L. Koester, III, is a consultant to Hynes Electric Heating Co. (1157 Globe Ave., Mountainside, NJ 07092; tel. 201-789-2900). He joined the company in 1972 and has held positions as salesman, sales manager, and vice president of sales; from 1983–1987, he served as president of the firm. Before joining Hynes, he was technical training supervisor for American Poclain Corp. Several of his articles on process heating have appeared in technical magazines. He holds a B.S. in industrial engineering and an M.B.A., both from Cornell University.

HOW TO PREDICT BATCH-REACTOR HEATING AND COOLING

John McEwan, PE
Paducah, Ky.

Calculating temperatures for jacketed batch reactors

Starting up a jacketed batch reactor requires control of the heat-up and cool-down rates. To do this, the jacket heat-transfer-fluid temperatures have to be determined and set. This can be done by trial-and-error experiments, but it is often quicker and more straightforward to simply make a trial heat-up and then plug the results into time-dependent heat-transfer equations. Here's how to do this for steam or hot-water jacketed reactors.

The time required for isothermally heating or cooling the reactor can be calculated by the equation

$$(1) \quad \ln\left(\frac{T_1 - t_1}{T_1 - t_2}\right) = \frac{UA\phi^{(1)}}{MCp} \qquad [1]$$

Rearranging to solve for the heat-up time gives

$$(2) \quad \phi = \ln\left(\frac{T_1 - t_1}{T_1 - t_2}\right)\frac{MCp}{UA}$$

Eq. (2) can also be used to calculate the heat-up time for nonisothermal heating (such as hot-water jacketing), provided that the difference between the outlet and inlet jacket temperatures is not greater than 10% of the difference between the batch and average jacket water temperature [2].

The heat transfer area, reaction mass and heat capacity of the vessel contents are generally known. The overall heat transfer coefficient, however, is a function of five resistances and can be difficult to estimate.

These resistances include the inside-the-jacket film heat-transfer coefficient (HTC) and fouling factor, the inside-the-reactor HTC and fouling factor, and the reactor-wall resistance.

Assuming U, M, Cp and A are constant, Eq. (2) can be rewritten as follows:

$$(3) \quad \phi = \ln\left(\frac{T_1 - t_1}{T_1 - t_2}\right) \times \frac{1}{K}$$

where K = UA/MCp
Rearranging Eq. (3) to solve for the jacket temperature as a function of time gives

$$(4) \quad T_1 = \frac{t_1 - t_2 e^{K\phi}}{1 - e^{K\phi}}$$

By taking a series of readings during a trial heat-up, K can be calculated. The heat-up and cool-down times for varying jacket temperatures can then be predicted.

Nomenclature

A = heat transfer area
Cp = heat capacity of reactor contents
K = constant = UA/MCp
M = mass of the reactor contents
T = average jacket temperature
t_1 = initial temperature of reactor contents
t_2 = final temperature of reactor contents
U = overall heat transfer coefficient
Φ = time

Example: The overall cycle time for a batch reaction is to be 8 h. The cycle time will include 2 h for heat-up and 3 h for cool-down. The batch will be heated from 20°C to the reaction temperature of 60°C, then cooled to 40°C. What are the jacket temperatures required for heat-up and cool-down?

Using a hot water jacket temperature of 75°C, it was found that it took 20 min to heat the batch from 20°C to 30°C.

Then, following Eq. (3),

$$\ln\left(\frac{75^\circ - 20^\circ}{75^\circ - 30^\circ}\right) = K \times 20\text{ min},$$

$$K = \frac{\ln\left(\frac{55^\circ}{45^\circ}\right)}{20\text{ min}}$$

$$K = \frac{0.01}{\text{min}}$$

From Eq. (4),

$$T_1 = \frac{20 - 60e^{(0.01/\text{min} \times 120\text{ min})}}{1 - e^{(0.01/\text{min} \times 120\text{ min})}}$$

T_1 = 78°C for heat-up

$$T_1 = \frac{60 - 30e^{(0.01/\text{min} \times 180\text{ min})}}{1 - e^{(0.01/\text{min} \times 180\text{ min})}}$$

T_1 = 24°C for cool-down

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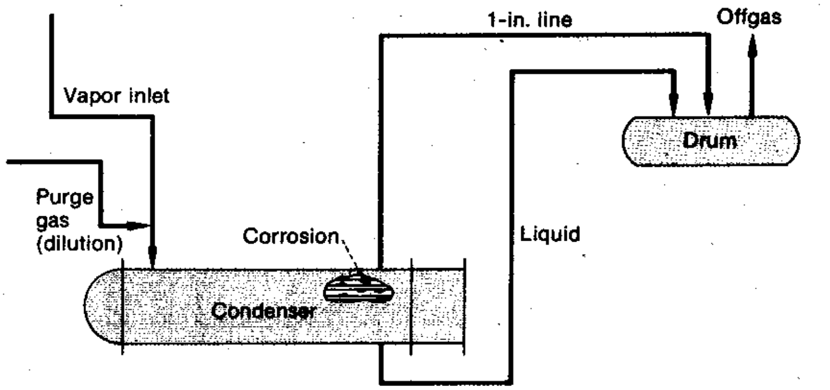
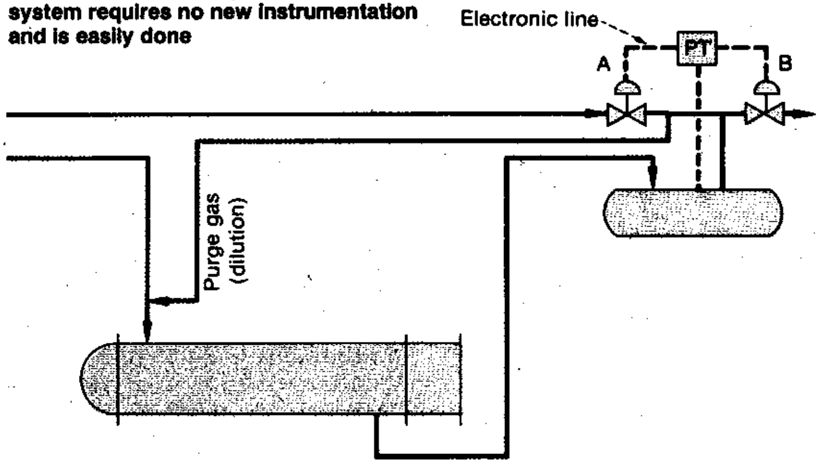


FIGURE 1. Adding a purge gas to an overhead condenser helps to eliminate corrosion of tubes

FIGURE 2. Converting to a new control system requires no new instrumentation and is easily done



PURGING PREVENTS CONDENSER CORROSION

In condensing sour gases, certain areas of tube bundles suffer corrosion. Such attack is prevented by diluting these harmful gases.

Adnan Y. N. Irhayem*

Overhead low-pressure condensers, used with strippers and fractionators, sometimes corrode quickly, especially with sulfur-containing liquids, unless the tubes are fabricated from high-grade alloys. Such attack occurs because of a relatively high percentage of light hydrocarbons present in the condenser. Since light hydrocarbons do not condense in typical low-pressure condensers, as the gas stream moves through the unit, the condensate becomes progressively acidic. This is because a small concentration of acid in the incoming gas becomes a large liquid concentration, owing to the amount of light components remainig in the gas. Thus, corrosion occurs near the liquid outlet, where the acid concentration is highest (see Fig. 1).

In our plant, an overhead water-cooled condenser has a shell of carbon steel and tubes of admiralty brass, and handles a sour hydrocarbon stream, in the shell. The tube bundle suffered general corrosion and wear on its outside surface and required replacement every two years, for the past 16 years. This despite the use of a recommended corrosion inhibitor.

The tube bundle showed attack at its end, as shown in Fig. 1. Rather than replace the bundle with a more-resistant alloy, the following measures were applied (see colored lines in FIg. 1):

- The corrosive gases were diulted, using a small amount of a hydrocarbon gas (not more than 5% is needed relative to the offgas stream).
- The trapped noncondensibles were released directly to the overhead accumulator through a small (1 in. dia.) added line. The gases included the light hydrocarbons and any noncondensible acid gases.

As a result, tubelife was more than doubled, and the concentration of corrosion inhibitor was lowered.

New control-system

A split-range control-system is normally used for an overhead low-pressure condenser (see Fig. 2). In this setup, Valve A controls makeup gas to the accumulator, and Valve B releases excess pressure. Valve A opens or closes according to a preset system pressure, delivering makeup gas whenever needed.

In the modified setup (see colored lines in Fig. 2), Valve A feeds the makeup gas directly into the condenser; the makeup also functions as the purge gas, to dilute and sweep out residual trapped corrosives (as well as noncondensible light components). The split-range system is converted to a dual-purpose one that regulates the pressure and controls corrosion, without any noticeable effect on the controlled pressure.

Overall, the new control system (and modified purge-gas piping) are inexpensive. The cost of the new setup is less than that of doubling the life of the tubes.

*Engineering Dept., Daura Refinery, Baghdad, Iraq.

How effective are finned tubes in heat exchangers?

K. N. Murty*

In heat exchangers, fins are used on tubes (externally or internally) to enhance the available surface area for heat transfer. For a "plain tube" heat exchanger, the overall heat-transfer coefficient is given by:

$$\frac{1}{U_p} = \frac{1}{h_o} + R_t + R_{fo} + \left(\frac{1}{h_i} + R_{fi}\right)\frac{A_p}{A_i} \qquad (1)$$

Note that A_p, rather than the traditional A_o (shellside surface-area), is used to simplify equations.

Now consider a finned-tube unit with the same I.D. and O.D. tubes. Service conditions are kept identical—i.e., fluid velocities, temperatures, etc., are constant. The increases in h_o and R_t over plain-tube exchangers are assumed negligible, especially for phase-change applications, such as condensation or evaporation on the fins. Thus:

$$\frac{1}{U_f} = \frac{1}{h_o} + R_t + R_{fo} + \left(\frac{1}{h_i} + R_{fi}\right)\frac{A_f}{A_i} \qquad (2)$$

From Eqs. (1) and (2): $$\frac{U_f}{U_p} = \frac{c_1 + c_2(A_p/A_i)}{c_1 + c_2(A_f/A_i)} \qquad (3)$$

where the constants are defined:

$$c_1 = \left(\frac{1}{h_o} + R_t + R_{fo}\right) \qquad (4)$$

$$c_2 = \left(\frac{1}{h_i} + R_{fi}\right) \qquad (5)$$

Since $A_f > A_p$, it follows from Eq. (3) that $U_f < U_p$—the overall heat-transfer coefficient for finned-tube units is always less than that for plain-tube units. However, the real advantage to using finned tubes is the substantial increase in the surface area. This increase in area more than compensates for the reduction in the overall heat-transfer coefficient, resulting in a greater amount of heat that is transferred in finned-tube units.

The ratio of heat transferred in a finned-tube unit to that in a plain-tube exchanger working within the same temperature limits is:

$$Q_f/Q_p = U_fA_f/U_pA_p \qquad (6)$$

Nomenclature

A_f	Finned-tube outside surface-area
A_i	Finned- or plain-tube inside surface-area
A_p	Plain-tube outside surface-area
h_i	Tubeside heat-transfer coefficient
h_o	Shellside heat-transfer coefficient
Q_f	Heat transferred for finned tubes
Q_p	Heat transferred for plain tubes
R_{fi}	Tubeside fouling resistance
R_{fo}	Shellside fouling resistance
R_t	Tubewall thermal resistance
U_f	Overall heat-transfer coefficient for finned tubes
U_p	Overall heat-transfer coefficient for plain tubes

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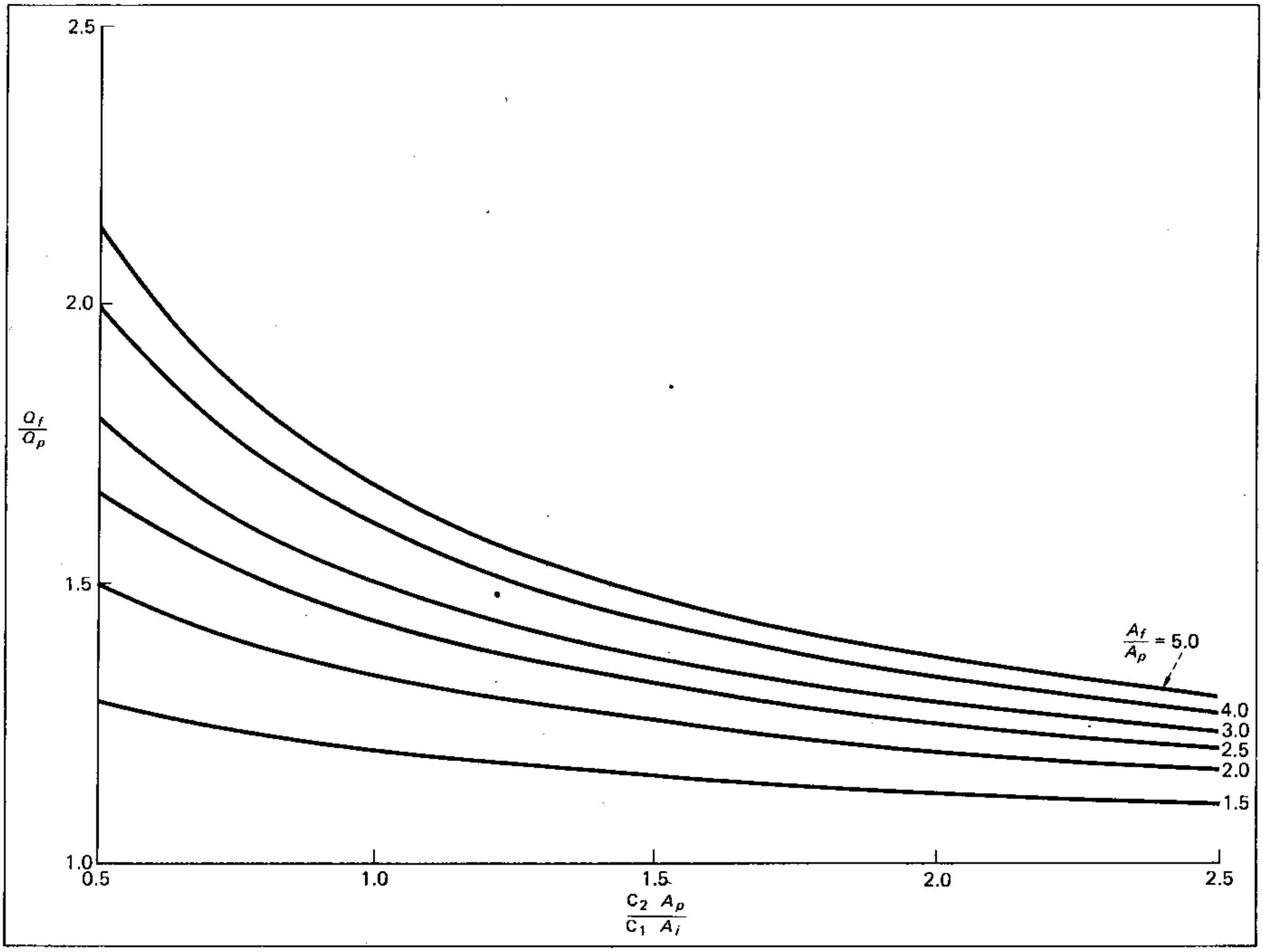
*Bharat Heavy Electricals Ltd., Vikas Nagar, Hyderabad 500 593, India



Figure—Finned-tube heat exchangers can transfer more heat than do conventional "plain-tube" units

From Eqs. (3) and (6):

$$Q_f/Q_p = \left[\frac{1 + \dfrac{c_2A_p}{c_1A_i}}{1 + \dfrac{c_2A_f}{c_1A_i}}\right]\left[\frac{A_f}{A_p}\right] = \frac{1 + \dfrac{c_2A_p}{c_1A_i}}{\dfrac{A_p}{A_f} + \dfrac{c_2A_p}{c_1A_i}} \qquad (7)$$

Eq. (7) shows that Q_f is always greater than Q_p, since $A_p < A_f$.

Rearranging Eq. (7):

$$Q_f/Q_p = \frac{\left(1 + \dfrac{c_2A_p}{c_1A_i}\right)\left(\dfrac{A_f}{A_p}\right)}{1 + \left(\dfrac{c_2A_p}{c_1A_i}\right)\left(\dfrac{A_f}{A_p}\right)} \qquad (8)$$

Eq. (8) is plotted in the figure for c_2A_p/c_1A_i, varying from 0.5 to 2.5, and A_f/A_p, ranging from 1.0 to 5.0. For $c_2A_p/c_1A_i = 0.5$, twice the amount of heat can be transferred with finned tubes than with plain tubes ($Q_f = 2Q_p$) for $A_f/A_p = 4.0$.

Example

Find the increase in the heat transferred for a finned-tube exchanger over a plain-tube exchanger, if $c_2A_p/c_1A_i = 1.5$ and $A_f/A_p = 2.0$. From the figure, $Q_f/Q_p = 1.25$, i.e. an increase of 25%.

MASS TRANSFER
and
SEPARATION OPERATIONS

Relating transfer units and theoretical stages

Sorab R. Vatcha*

Vapor-liquid mass-transfer operations, such as absorption, stripping and distillation, are carried out in packed and plate columns. The key difference is that counterflowing vapor and liquid are contacted continuously with packings, and discretely with plates. The equilibrium and operating lines of packed and plate columns are identical under the same operating conditions—feed and product flowrates and compositions, temperature and pressure. Models for the design and analysis of packed columns are based on their close analogy to plate devices.

Transfer units vs. stages

However, two different models of packed columns are used. One is in terms of transfer units; the other, in terms of equivalent theoretical plates. Both must yield the same final results.

We shall connect and reconcile these models. The number and height of transfer units and equivalent theoretical plates will be quantitatively related via equations and a graph.

This information is useful for analyzing and designing packed columns, for comparing them with plate columns, and for selecting the right type of column for a specific application. One can directly design a packed column from design or experimental data of a plate column, or vice versa, for the same service without repeating the entire calculation. This includes evaluating a proposed new packing to replace an existing one, or the conversion of a plate column to a packed column.

A worked-out example will follow.

Packed-column model

The height of packing required under a given set of operating conditions may be expressed in terms of gas-phase transfer units, Eq. (1); liquid-phase transfer units, Eq. (2); or equivalent theoretical plates, Eq. (3):

$$h = HTU_{OG} NTU_{OG} \tag{1}$$

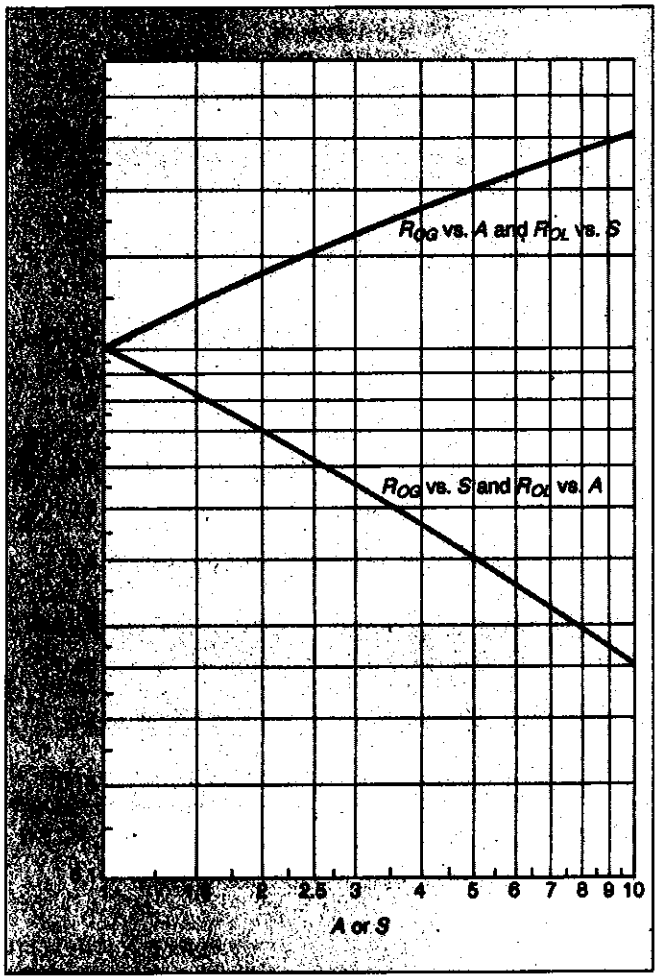
$$h = HTU_{OL} NTU_{OL} \tag{2}$$

$$h = HETP \times N \tag{3}$$

Thus, the number and the height of transfer units and equivalent theoretical plates (or stages) can be related by two ratios, defined as:

$$R_{OG} = HETP/HTU_{OG} = NTU_{OG}/N \tag{4}$$

$$R_{OL} = HETP/HTU_{OL} = NTU_{OL}/N \tag{5}$$



Ratios R_{OG} and R_{OL}, defined in Eqs. (4) and (5), are functions of the absorption factor and stripping factor

In general, these two ratios depend upon the operating conditions and the phase equilibria in a complex manner. However, if the absorption factor or the stripping factor, defined in Eq. (6), is nearly constant throughout the column, then the two ratios become simple functions of A or S only (this assumption is generally valid for dilute, nonreactive vapor-liquid systems):

$$S = 1/A = mV/L \tag{6}$$

$$R_{OG} = AR_{OL} = (A \ln A)/(A - 1) = (\ln S)/(S - 1) \quad \text{when } A \neq S \neq 1 \tag{7}$$

$$R_{OL} = SR_{OG} = (\ln A)/(A - 1) = (S \ln S)/(S - 1) \quad \text{when } A \neq S \neq 1 \tag{8}$$

$$R_{OG} = R_{OL} = 1 \quad \text{when } A = S = 1 \tag{9}$$

Graphs of R_{OG} and R_{OL} vs. A and S are shown in the

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*P.E., Consultant, 87 Jaques St., Somerville, MA 02145-1804.

figure, based on Eqs. (7-9). Note that R_{OG} decreases and R_{OL} increases as A decreases or as S increases. Therefore, the number and height of transfer units and equivalent theoretical plates satisfy Eq. (10), which serves as a useful qualitative check of design calculations and data:

$$\text{When } S \gtreqless 1, \text{ and } A \lesseqgtr 1\text{: } \mathrm{HTU}_{OL} \gtreqless \mathrm{HETP} \gtreqless \mathrm{HTU}_{OG}$$

$$\text{and} \qquad NTU_{OG} \gtreqless \mathrm{N} \gtreqless \mathrm{NTU}_{OL} \qquad (10)$$

Nomenclature

A	Absorption factor
b	y-intercept of the equilibrium line
h	Height of packing
$HETP$	Height equivalent to a theoretical plate
HTU	Height of a transfer unit
L	Molar flowrate of liquid
m	Slope of the equilibrium line
N	Number of theoretical plates
NTU	Number of transfer units
R	Ratio of the number of transfer units to the number of equivalent theoretical plates
S	Stripping factor
V	Molar flowrate of vapor
x	Mole fraction in the liquid
y	Mole fraction in the vapor
Subscripts and superscript	
A	Component A
i	Inlet
o	Outlet
OG	Overall, gas phase
OL	Overall, liquid phase
$*$	Equilibrium value

The inlet and outlet stream-compositions, the number of transfer units, and the absorption or stripping factor are related by:

$$NTU_{OG} = \frac{\ln\left[(1-S)((y_{Ai}-y^*_{Ao})/(y_{Ao}-y^*_{Ao}))+S\right]}{(1-S)}$$

$$\text{when } S \neq 1 \qquad (11)$$

$$NTU_{OL} = \frac{\ln\left[(1-A)((x_{Ai}-x^*_{Ao})/(x_{Ao}-x^*_{Ao}))+A\right]}{(1-A)}$$

$$\text{when } A \neq 1 \qquad (12)$$

$$NTU_{OG} = NTU_{OL} = \frac{(y_{Ai}-y_{Ao})}{(y_{Ao}-y^*_{Ao})} = \frac{(x_{Ai}-x_{Ao})}{(x_{Ao}-x^*_{Ao})}$$

$$\text{when } A = S = 1 \qquad (13)$$

The values of y^*_{Ao} and x^*_{Ao} are the outlet stream-compositions that would be in equilibrium with the inlet liquid and vapor streams, respectively:

$$y^*_{Ao} = mx_{Ai} + b \qquad (14)$$

$$x^*_{Ao} = (y_{Ai} - b)/m \qquad (15)$$

The inlet and outlet stream-compositions and flowrates satisfy the overall mass balance of the transferring component, A:

$$V(y_{Ai} - y_{Ao}) = L(x_{Ao} - x_{Ai}) \qquad (16)$$

Discussion

The stripping factor is the ratio of the slopes of the equilibrium and operating lines, Eqs. (6, 14-16). Therefore, it is constant when these two lines are straight, and it is =1 if they are parallel. The stripping factors of packed columns are constant when both streams are: dilute; in plug flow; are not reacting chemically; have negligible heat of absorption. This situation is often obtained in absorption and stripping columns.

Even if the stripping factor varies appreciably along the height of the column, a constant effective value can be used [*1*]; or different values can be selected for each section of the column, and the total height calculated as the sum of the heights of each section. So the assumption of a constant stripping factor that underlies Eqs. (7-16) and the figure is realistic and not restrictive, especially for purposes of preliminary design.

The derivation of Eqs. (7-13), and charts based on Eqs. (11-13), are given in Perry's [*1*] and in textbooks [*2*, *3*], so they are not repeated here.

The number of equivalent theoretical plates is intermediate between NTU_{OL} and NTU_{OG}. Similarly, the height equivalent to a theoretical plate, $HETP$, is intermediate between HTU_{OL} and HTU_{OG}. Transfer units and theoretical plates are equal in number and height if and only if S is =1 (Eq. 10).

The application of the concepts and methods developed here is now illustrated:

Example

A waste-air stream containing 2 mol % of impurity A is purified to 0.02% by scrubbing with pure water in a countercurrent packed-column. The column provides 7 overall gas-phase transfer units in a height of 1.75 m. The equilibrium mole fractions of A in the vapor and liquid are related according to:

$$y_A = 1.75x_A. \qquad (17)$$

Find HTU_{OG}, HTU_{OL}, NTU_{OL} and $HETP$ for this system. How many theoretical stages would be required to achieve the same separation in a plate column? What is the composition of the exiting liquid stream?

Solution:

First summarize the data in the proper nomenclature: $m = 1.75$, $b = 0$, $y_{Ai} = 0.02$, $y_{Ao} = 0.0002$, $x_{Ai} = 0$, $NTU_{OG} = 7$, $h = 1.75$ m.

Since the terminal compositions of the gas phase and NTU_{OG} are given, the stripping factor can be found by solving Eq. (11) for S, or by using the charts based on this equation in the references [*1-3*]:

$$y^*_{Ao} = 1.75(0) + 0 = 0 \qquad \text{by} \quad (14)$$

$$\ln\left[(1-S)(0.02/0.0002)+S\right] = 7(1-S) \quad \text{by} \quad (11)$$

$$S = 0.4186.$$

Since $S \neq 1$, the ratios R_{OG} and R_{OL} are obtained from Eqs. (7) and (8) or from the figure.

$$R_{OG} = (\ln 0.4186)/(0.4186 - 1) = 1.498 \quad \text{by} \quad (7)$$

$$R_{OL} = (0.4186)(1.498) = 0.627 \quad \text{by} \quad (8)$$

Now the number and height of transfer units and equivalent theoretical plates are obtained directly:

$$HTU_{OG} = 1.75/7 = 0.25 \text{ m} \quad \text{by} \quad (1)$$

$$HETP = (0.25)(1.498) = 0.3745 = 0.37 \text{ m} \quad \text{by} \quad (4)$$

$$HTU_{OL} = 0.3745/0.627 = 0.5973 = 0.60 \text{ m} \quad \text{by} \quad (5)$$

$$NTU_{OL} = 1.75/0.5973 = 2.9299 = 2.93 \quad \text{by} \quad (2)$$

$$N = 1.75/0.3745 = 4.6732 = 4.67 \quad \text{by} \quad (3)$$

A plate column of 4.67 theoretical stages would be required to yield the same separation as this packed column.

The composition of the exiting liquid stream can be found by the mass balance (Eq. 16) and the definition of the stripping factor (Eq. 6). It can also be found by solving Eq. (12) for x_{Ao}, but the former is easier in this case:

$$V/L = S/m = 0.4186/1.75 = 0.2392 \quad \text{by} \quad (6)$$

$$x_{Ao} = x_{Ai} + (V/L)(y_{Ai} - y_{Ao}) \quad \text{by} \quad (16)$$

$$= 0 + (0.2392)(0.02 - 0.0002) = 0.004736$$

$$= 0.47 \text{ mol\% } A$$

Since the concentrations of the transferring component A in both phases are quite low, the assumption of a constant stripping factor would be valid in this case. As a check, note that the results satisfy the criteria of Eq. (10). These calculations were simple enough to be done on a calculator, yet accurate enough for preliminary design purposes.

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This calculation simplifies the analysis of flooding capacity for mass-transfer columns

A NEW CORRELATION FOR SIEVE TRAYS

Fig. 1.

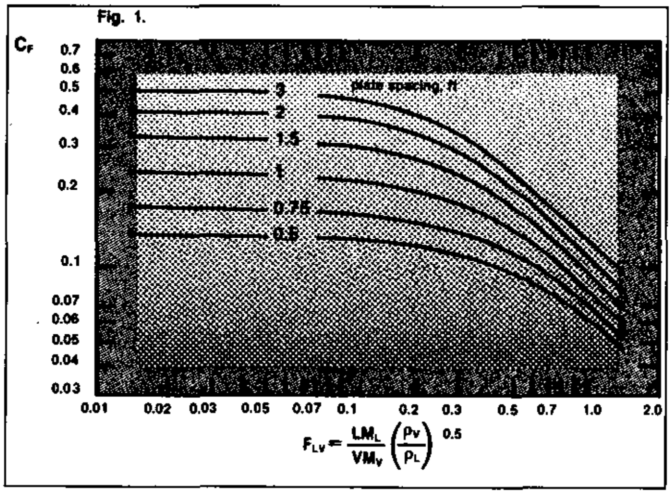
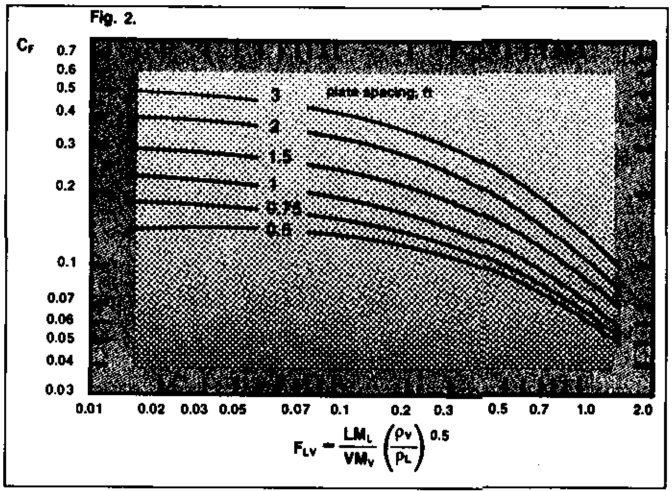


Fig. 2.



LAWRENCE DANIELS

Thomas J. Ward,
Clarkson University, Potsdam, N.Y.

The calculation of column diameter for distillation and absorption columns [*2,4,6*] is usually based on the flooding velocity, which, in turn, requires values of the flooding capacity factor, C_F. Fair's flooding-capacity plot for sieve trays [*1*] correlates the flooding capacity factor with a flow parameter F_{LV} for each tray-spacing value, t, as shown in Figure 2. The flow parameter involves the liquid mass flow, LM_L, and vapor mass flow, VM_V (both in lb/s), as well as the densities of the two streams.

This flooding-capacity calculation is also used as a design factor for bubble-cap and valve trays. There is a need to have these curves correlated in an analytical form suitable for computer-based design calculations. Now, there is a single equation that provides an adequate fit of the six curves of Figure 2. The development of the correlation is a novel application of process control methodology to mass transfer.

Nomenclature

C_F = flooding capacity factor, ft/s
F_{LV} = flow parameter
t = tray spacing, ft

The prior art

An early correlation [*5*] was of the form

$$C_F = a \log_{10}(F_{LV}^{-1}) + b \qquad (1)$$

where a and b were given linear functions of the tray spacing, t. However, this was fitted to the curves only in the region where $F_{LV} > 0.1$. The C_F values in the rest of the diagram were assumed constant at the value of C_F where $F_{LV} = 0.1$. This correlation is also given by Lockett [*4*]. A reasonably accurate single-equation fit of the Fair plot was also presented by Lygeros and Magoulas [*5*].

More recently, quadratic and cubic spline fits of the curves were presented [*3*]. The first required a look-up table of eighteen constants. The second required the selection of one of two sets of equations and a 48-element look-up table.

A new approach

Each of the curves in Figure 2 is similar in shape to a familiar form in process control — the log-log Bode amplitude-ratio curve for a first-order system. The amplitude rate curve is fitted by the equation

$$A = \frac{K}{\left[1+\left(\frac{w}{w_b}\right)^2\right]^{0.5}} \qquad (2)$$

where A is the variable on the logarithmic ordinate, w is the variable on the logarithmic abscissa, and both K and w_b are parameters. The amplitude ratio curve is characterized by a horizontal straight-line portion at low values of w, and a straight-line portion with a slope of -1 at large values of w. K is the A value at low values of w, and w_b is the value of w at the intersection of the two straight-line asymptotes. This provides a graphical method to estimate the two parameters.

This same graphical procedure is applied to each of the curves in the capacity factor plot of Figure 2 to give a pair of parameters that characterize each curve. Each of the parameters is then plotted as a function of the tray spacing, t, and fitted by:

$$K = 0.26t - 0.029t^2 \qquad (3)$$

$$1/w_b = 2.451t^{0.3749} \qquad (4)$$

The capacity-factor correlation

These values of K and w_b are substituted into the Bode form above to give the sieve-tray capacity factor correlation.

$$C_F = \frac{0.26t - 0.029t^2}{\left[1 + 6F_{LV}^2\, t^{0.7498}\right]^{0.5}} \qquad (5)$$

where the tray spacing, t, is in feet and C_f is in ft/s. This single equation with four constants is an adequate engineering correlation of the sieve-tray curves for all six tray spacings over the entire range of Figure 2.

This new correlation gives the curves shown in Figure 1, which can be compared with the original plot of Fair. The correlation provides the best fit at the ends of the curves. If a better fit is desired in the middle range, the gains (K values) can be adjusted to match the curve in any range. The equation is readily programmed, and does not require the selection of an equation set or region, nor a stored look-up data table.■

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Using CAD systems to prepare McCabe-Thiele diagrams

J. Neil and S. Jane Stone*

The increasing use of microcomputer-based computer-aided drafting (CAD) systems is making these tools available to the whole design team, not just to the drafting professionals. Thus, applications other than pure drawing sometimes arise.

McCabe-Thiele diagrams

One such case was to prepare a McCabe-Thiele diagram for a wide concentration range. The diagram was needed in the design of an HCl absorber, in which the inlet-gas partial pressure of 22.75 mm Hg had to be reduced to 0.00406 mm Hg to meet environmental requirements. This gave rise to a gas-phase concentration range of four orders of magnitude.

To create the needed McCabe-Thiele diagram would have required either several sets of graphs with different scales or use of logarithmic scales, both of which would have had inaccuracies and transcription errors, as well as being inconvenient to draw. In contrast, by using CAD, a single diagram covered the whole range and stepping off stages was done easily and accurately. The reason behind this is the internal accuracy of CAD. Although the resolution on the screen is no better than on graph paper, the curves and steps are stored to many decimal places. Thus, by simply changing the magnification on the screen the areas of low concentration can be enlarged or reduced at will.

The figure shows the operating diagram for the HCl absorber. Since the operating line is drawn at a stage efficiency of 0.68, the representation depicts real trays. More than 7 trays are needed, but 8 would overshoot the design point. This accounts for the way the stages are drawn.

Below the fifth tray, the steps on the diagram become indecipherable. However, an enlargement of the area shows how the construction continues into the lower concentration range. A further enlargement is needed to see the points for

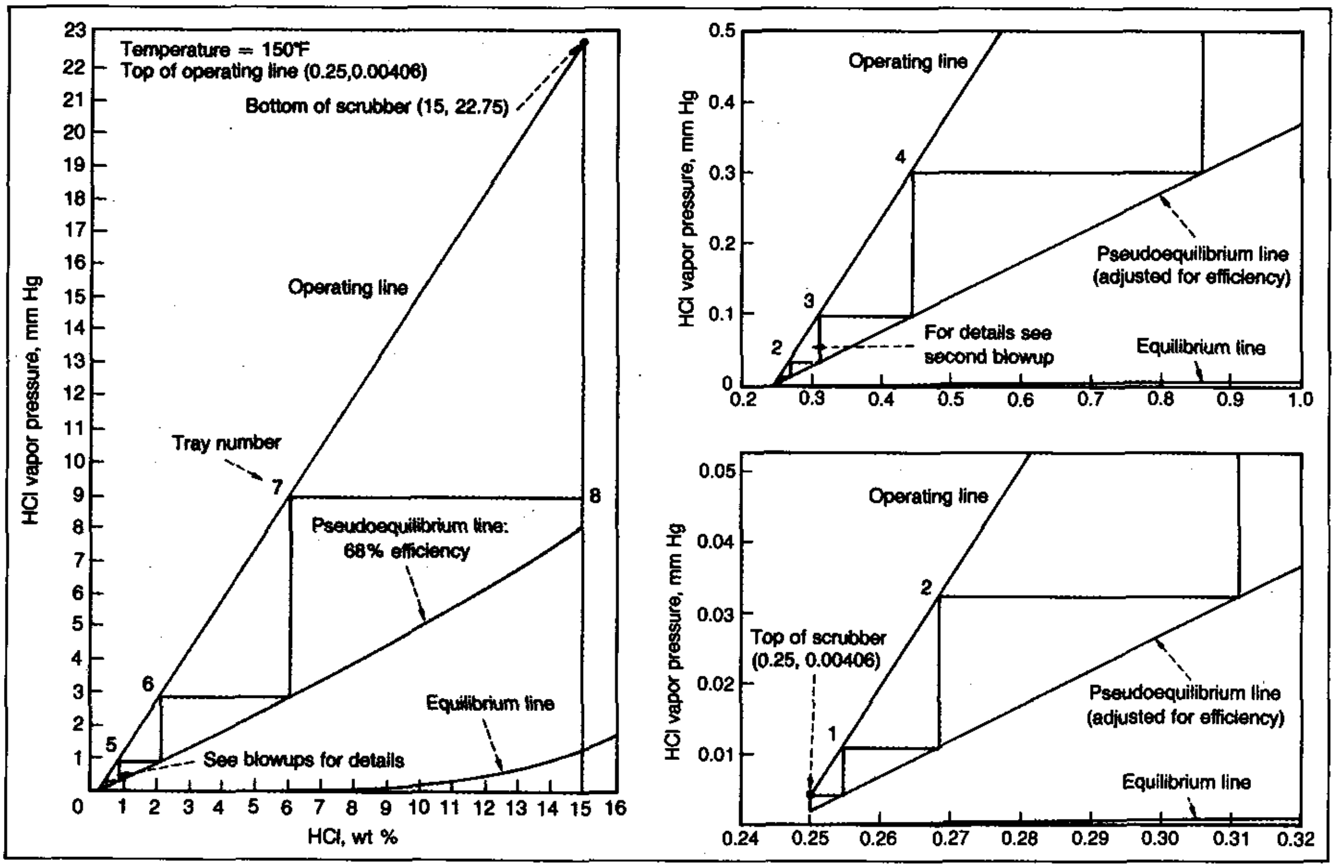

Figure — By using CAD, two exploded views of this McCabe-Thiele diagram can be easily created, to provide enough detail to see the first four stages

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*Ledge Engineering Inc., 179 Lansdowne Ave., Kingsville, Ont., N9Y 3J2, Canada the top two trays. These blownup views are easily created by pressing a few keys and need never be plotted.

Such features, which are probably offered with most CAD systems, can augment the accuracy and convenience of the graph. In this case, the equilibrium curve was automatically drawn by entering a series of points and having the system fit them to a curve.

Also, the pseudoequilibrium line, which allows for tray efficiency, was constructed by a LISP-based routine, written to automatically correct the equilibrium line for the tray efficiency of 0.68. [LISP is a language used in artificial-intelligence applications.]

Although many of the graphs used in chemical engineering can be created by mathematical computer models, use of CAD provides the accuracy of the computer, while making the results clear through easily understood diagrams.

Evaluating turndown of valve trays

George F. Klein*

☐ Valve trays, widely used vapor-liquid contacting devices, provide excellent turndown. However, little published information is available to calculate the lower operating limit, also known as the weep point, represented by V_{Hmin}.

This article presents a correlation for estimating V_{Hmin} (based on data from Glitsch, Inc., Bulletin 4900, 3rd ed., p. 28) and equations to establish the maximum valve hole area (A_{Hmax}) and the maximum number of valve caps (N_{max})—important parameters in design calculations.

Data for estimating weep points for flat-orifice (Type V-1) and venturi-orifice (Type V-4) valve caps are plotted in the figure and are represented by the equation:

$$V_{Hmin} = CS^{0.615}\sqrt{\frac{\rho_L}{\rho_V}} \qquad (1)$$

where C is 0.35 for V-1 units and 0.63 for V-4 units, and the tray submergence, S, is:

$$S = h_W + 0.4\left[\frac{F_L Q_L}{L_W}\right]^{2/3} \qquad (2)$$

The valve area can be represented in terms of volumetric flow as:

$$A_H = Q_V/V_H \qquad (3)$$

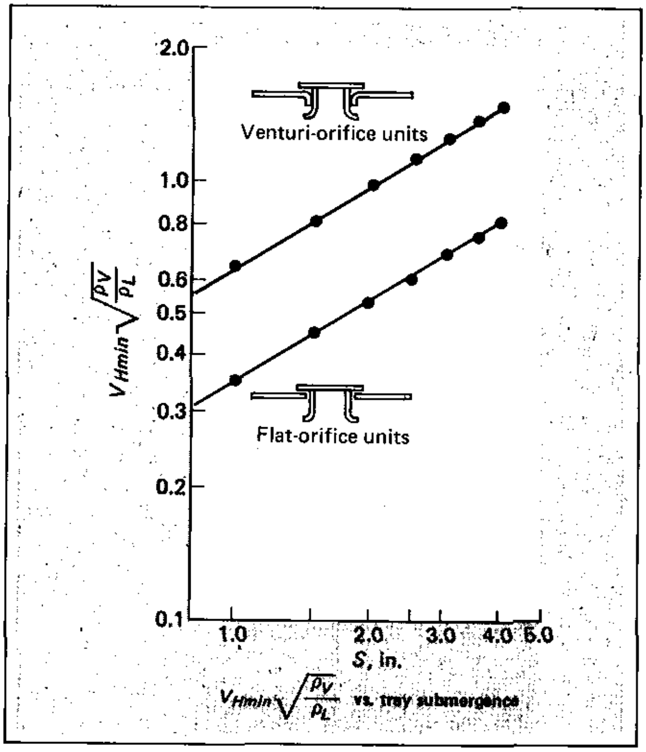
The maximum valve area required may be determined by combining Eq. (1) and (3):

$$A_{Hmax} = \frac{F_V Q_V}{CS^{0.615}\sqrt{\frac{\rho_L}{\rho_V}}} \qquad (4)$$

This is the maximum allowable hole area to avoid weepage upon turndown. The actual area used may be less, and would result in greater flexibility, provided the pressure-drop requirement is satisfied.

Eq. (4) can be modified to find the maximum number of valves on the tray. The total hole area can be expressed as:

$$A_H = NA_C \qquad (5)$$



$V_{Hmin}\sqrt{\frac{\rho_V}{\rho_L}}$ vs. tray submergence

where A_C = 0.01279 ft^2. Substitution into Eq. (4) yields:

$$N_{max} = \frac{78.2\ F_V Q_V}{CS^{0.615}\sqrt{\frac{\rho_L}{\rho_V}}} \qquad (6)$$

The following limitations apply to Eq. (6):

- The orifice diameter is the standard 1¹⁷⁄₃₂ in.
- The maximum valve density is 14 per ft^2 of tray active area.

Example

Given the following information, design a valve tray with flat orifice units for 40% vapor-liquid turndown ($F_V = F_L = 0.4$): Q_L = 114 gal/min; Q_V = 4.75 ft^3/s; ρ_L = 30 lb/ft^3; ρ_V = 2 lb/ft^3; h_W = 2 in.; L_W = 41.3 in.; and the tray active area is 5.23 ft^2.

First, calculate the weep-point velocity. From Eq. (2), $S = 2 + 0.4[(0.4)(114)/41.3]^{2/3} = 2.43$ in. Substituting this into Eq. (1) yields $V_{Hmin} = 0.35(2.43)^{0.615}\sqrt{30/2}$ = 2.34 ft/s.

Next, determine the valve hole area, the number of valve caps, and the cap density. Using Eq. (4), A_{Hmax} = (0.4)(4.75)/2.34 = 0.81 ft^2. From Eq. (6), N_{max} = [(78.2)(0.4)(4.75)]/2.34 = 63.5, or say 63. The cap density is 63/5.23 = 12, which is less than 14, the maximum.

Nomenclature

A_C	Area of valve unit (perforation in tray) = 0.01279 ft^2
A_H	Total valve hole area, ft^2
C	Constant; 0.35 for flat orifice, 0.63 for venturi orifice
F_L	Liquid turndown, fractional value
F_V	Vapor turndown, fractional value
h_W	Weir height, in.
L_W	Weir length, in.
N	Number of valve units on a tray
Q_L	Liquid flow from tray, gal/min
Q_V	Vapor flow to tray, ft^3/s
S	Tray submergence, in. of liquid
V_H	Vapor velocity through valve hole, ft/s
ρ_L	Liquid density, lb/ft^3
ρ_V	Vapor density, lb/ft^3

Subscripts

max	Maximum
min	Minimum (at weep point)

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*Crawford & Russell, Inc., 17 Amelia Pl., Stamford, CT 06904.

Quickly determine multicomponent minimum reflux ratio

K. Venkateswara Rao and A. Raviprasad*

A shortcut method is given to estimate the minimum reflux ratio for multicomponent mixtures. Two examples will be presented, comparing the results of this method to those of three standard ones. The standard procedures will be summarized, to provide a handy reference.

Calculation methods

Of the standard shortcut methods, the Underwood [1], Gilliland [2], and Colburn approximation [3] (all shown later on) require fewer calculation steps than the Brown-Martin [4] and Colburn rigorous [5] approaches, and were thus selected for comparison.

This new method is based on the equation given by Robinson and Gilliland [6] for the minimum reflux ratio for binary mixtures, for the limiting condition $x_D = x_B = 1.0$:

$$R_m = 1/(x_F(\alpha - 1)) \tag{1}$$

Eq. (1) is rewritten for multicomponent mixtures by defining the effective light-key composition in the feed in a manner analogous to the effective product composition of Hengstebeck [7]. The α in Eq. (1) is replaced by $(\alpha_{LK})_{av}$. The minimum reflux ratio is given by:

$$R_m = 1/((x_{FLK})_{eff}[(\alpha_{LK})_{av} - 1]) \tag{2}$$

where:

$$(x_{FLK})_{eff} = x_{FLK}/(x_{FLK} + x_{FHK}) \tag{3}$$

Examples

Data for the two examples are given in Table I. The R_m values, as determined by Eq. (2), appear in Table II. This table shows that the values of R_m calculated by Eq. (2) are closest to those of the Underwood method, which is more often the approximation preferred in preliminary multicomponent distillation design. Although it yields slightly lower values, Eq. (2) offers the advantage of being simple and taking less computational time. Further, it does not involve trial and error.

While Eq. (2) has been found satisfactory to employ for many cases of multicomponent distillation, use this equation with caution, as it may produce high values of the minimum reflux ratio when the composition of the light key in the distillate is not predominant.

Standard procedures

1. Underwood:

$$\sum_{i=1}^{n} \frac{\alpha_i x_{iF}}{\alpha_i - \Theta} = 1 - q \tag{4}$$

$$\sum_{i=1}^{n} \frac{\alpha_i x_{iD}}{\alpha_i - \Theta} = R_m + 1 \tag{5}$$

Assumptions: Constant molal flow, and constant or an averaged value of relative volatility are used. Calculate:

- Θ from Eq. (4), knowing α_i, x_{iF} and q.
- R_m for a given x_{iD}; substitute the value of Θ in Eq. (5).

2. Gilliland:

$$R_m + 1 = A_1(A_2 + A_3 + A_4) \tag{6}$$

Table I — Two examples are used to compare this method to those of Underwood, Colburn (approximate) and Gilliland

[illegible]

Component	α_{av}	Composition, mole % Feed	Distillate	Bottoms
Benzene	6.78	2.2	22.8	~0
Toluene, light key (LK)	2.45	7.4	72.2	0.5
Ethylbenzene, heavy key (HK)	1.00	43.4	5.0	47.5
Styrene	0.71	47.0	~0	52.0

Feed, 100 moles/h; distillate, 9.63 moles/h; bottoms, 90.37 moles/h; condenser type, total; pressure of operation, 21.3 kPa; all streams are saturated liquids.

[illegible] (from Ref. [5])

Component	α_{av}	Composition, mole % Feed	Distillate	Bottoms
Methane	7.356	5	13.0	~0
Ethane, light key (LK)	2.091	35	83.0	5.0
Propylene, heavy key (HK)	1.000	15	2.5	22.8
Propane	0.901	20	1.5	31.6
i-butane	0.507	10	~0	16.2
n-butane	0.408	15	~0	24.4

Feed, 100 moles/h; distillate, 38.5 moles/h; bottoms, 61.5 moles/h; condenser type, total; pressure of operation, 2,757 kPa; all streams are saturated liquids.

Table II — This formula compares most favorably with results from Underwood's method

Method	Minimum reflux ratio Example 1	Example 2
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]

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*Dept. of Chemical Engineering, Andhra University, Visakhapatnam 530003, India.

Nomenclature

A_1, A_2, A_3, A_4	Constants in Gilliland method
B	Bottoms rate
D	Distillate rate
n	Number of components
q	Liquid mole fraction in feed (0 for dewpoint, 1.0 for bubblepoint)
R_m	Minimum reflux ratio
r_F	Ratio of light key to heavy key composition in the feed
x	Liquid mole fraction
Subscripts	
av	Average
BHK	Heavy key in bottoms
DHK	Heavy key in distillate
DLK	Light key in distillate
eff	Effective
F	Feed
FHK	Heavy key in feed
FLK	Light key in feed
i	Component
iB	*i*th component in bottom product
iD	*i*th component in distillate
iF	*i*th component in feed
HHK	Heavier than heavy key
HK	Heavy key
LK	Light key
nHK	Upper pinch composition of heavy key
nLK	Upper pinch composition of light key
Greek letters	
α_i	Relative volatility of component i based on heavy key
Θ	Variable in Underwood method

where:

$$A_1 = \left(\frac{x_{DLK}x_{FHK}}{x_{BHK}x_{FLK}} - \frac{x_{DHK}}{x_{BHK}}\right) \qquad (7)$$

$$A_2 = \left(1 + (\alpha_{LK})_{av}\frac{x_{FLK}}{x_{FHK}}\frac{x_{BHK}}{(\alpha_{LK})_{av} - 1}\right) \qquad (8)$$

$$A_3 = \sum \frac{\alpha_i x_{iB}}{((\alpha_{LK})_{av} - \alpha_i)} \qquad (9)$$

$$A_4 = \frac{D}{B}\sum \frac{\alpha_i x_{iD}}{\alpha_i - 1} \qquad (10)$$

Assumption: Average value of relative volatility is used. Calculate:

- A_1 from Eq. (7) for given values of feed, distillate, and bottom compositions of light and heavy keys.
- A_2 from Eq. (8).
- A_3 from Eq. (9) by summing over all components heavier than heavy key.
- A_4 from Eq. (10) by summing over all components lighter than light key.
- R_m by Eq. (6).

3. Colburn approximation:

$$R_m = \frac{1}{((\alpha_{LK})_{av} - 1)}\left(\frac{x_{DLK}}{x_{nLK}} - (\alpha_{LK})_{av}\frac{x_{DHK}}{x_{nHK}}\right) \qquad (11)$$

where: $$x_{nLK} = r_F/((1 + r_F)(1 + \Sigma\alpha_{HHK}x_{FHK})) \qquad (12)$$

$$x_{nHK} = x_{nLK}/r_F \qquad (13)$$

$$r_F = x_{FLK}/x_{FHK} \qquad (14)$$

Assumption: Average value of relative volatility is used. Calculate:

- r_F from Eq. (14).
- x_{nLK} via Eq. (12), using ($\Sigma\alpha_{HHK}x_{FHK}$) summed over all the nonkey components heavier than heavy key.
- x_{nHK} using Eq. (13).
- R_m by Eq. (11) for given values of the light and heavy keys, x_{DLK} and x_{DHK}, respectively.

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Extend the life of slurry agitators

S. Raghavachari*

When slurries are being handled, process upsets (even relatively minor ones) can cause the material to thicken or harden, resulting in equipment damage. Repair can be expensive and inconvenient, and often results in costly plant downtime and quality-control problems.

Fig. 1 shows an agitated preneutralizer for mixing phosphoric acid and ammonia in a fertilizer plant. The slurry is very thick, with a specific gravity of 1.5 to 1.6. The agitator paddle is mounted on a 14-ft-long stainless-steel shaft suspended on the output shaft of a roof-mounted gear reducer.

During process upsets, the slurry hardens, causing the paddle shaft to bend, and sometimes resulting in damage to the gear reducer as well.

This problem can be overcome by modifying the system as shown in Fig. 2. The agitator shaft is extended into an additional bearing at the bottom of the vessel to add rigidity. Also, a fluid coupling is substituted for the flexible one between the motor and the gear reducer; in the event of excessive load, the fluid coupling slips, preventing the shaft from bending and avoiding damage to the gear reducer.

In our facility, this system is performing satisfactorily, and agitator failures and the resulting production outages have been avoided. Operations are now running smoothly, production is higher, and product quality has improved.

*Madras Fertilizers Ltd., Manali, Madras - 600 068, India.

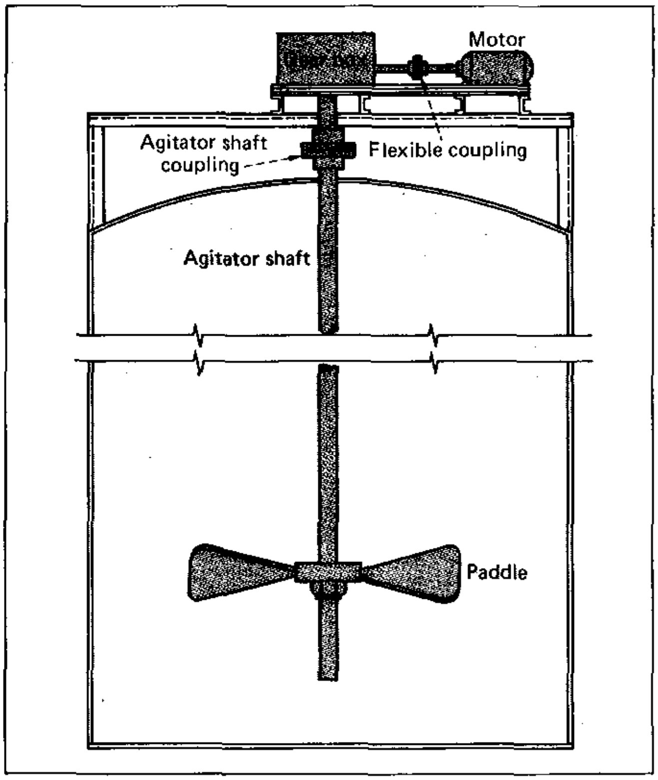


Figure 1 — Slurry agitator in original configuration was often damaged during process upsets

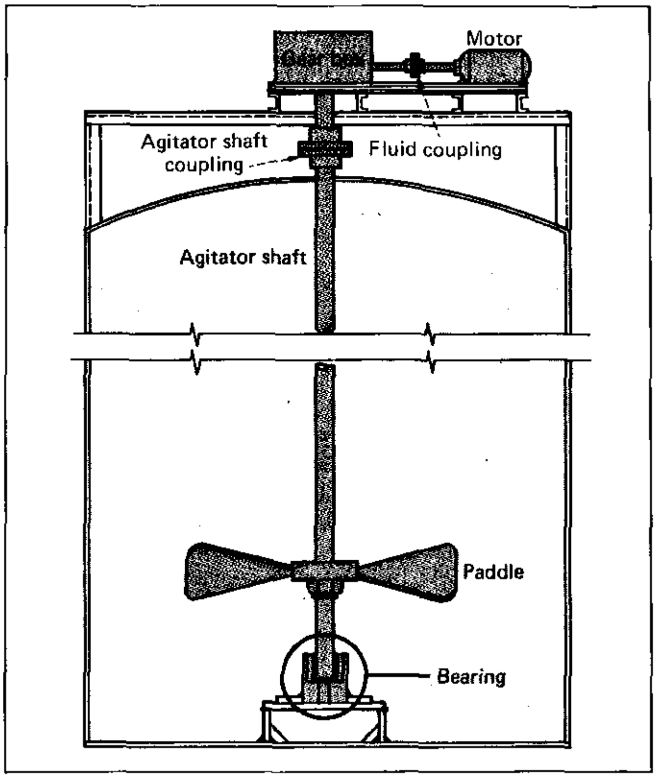


Figure 2 — Extending agitator shaft to bearing, and using a fluid coupling between motor and gear reducer, prevents damage

Critical-velocity plots for moisture separators

P. S. V. Kurmarao*

☐ In a gas compressor train, moisture must be removed from the gas before the latter goes on to the next stage. This is done either by scrubbing in a knitted-wire-mesh separator, or with a vane-type separator that takes advantage of impingement, centrifugal force and surface tension.

If the gas velocity increases above the separator's critical maximum velocity, the water will be reentrained and carried away by the gas. Thus, critical velocity is one of the most important factors in the sizing of a moisture separator, since the thickness of the wire mesh and the width and depth of the vanes are normally standardized.

The densities of water and the gas are the variables that have the greatest influence on the critical velocity. Critical velocity, U_{max} (ft/s), can be represented as:

$$U_{max} = K[(\rho_L - \rho_V)/\rho_V]^{1/2}$$

where K is the separator constant, normally taken as 0.35 ft/s when the liquid is water, and ρ_L and ρ_V are the densities of water and the gas, respectively.

Fig. 1 and 2 plot critical velocity as a function of pressure for four gases—air at three different temperatures, and carbon dioxide, natural gas, and ammonia synthesis gas all at 68°F. These charts eliminate the need to evaluate the properties of water and the gas at the various temperatures and pressures of interest, and can be used to estimate the size of the separator: The cross-sectional area, A (ft^2), is simply equal to the volumetric flowrate of the gas, Q (ft^3/h), divided by the gas's critical velocity, U_{max}.

Example

Determine the cross-sectional area of standard 6-in.-thick wire mesh required to separate water droplets from saturated air. The air flowrate is 70,000 ft^3/h, the flowrate of the entrained water is 100 lb/h, and the air enters the unit at 7 atm and 100°F.

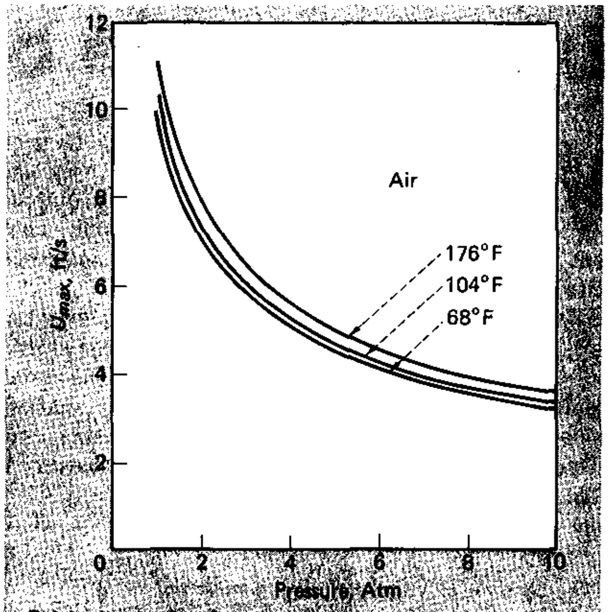
Solution: The maximum allowable velocity for air at 7 atm and 100°F, from Fig. 1, is $U_{max} = 3.9$ ft/s. Thus, the cross-sectional area needed is:

$$A = (70{,}000/3{,}600)(1/3.9) = 4.986 \text{ ft}^2$$

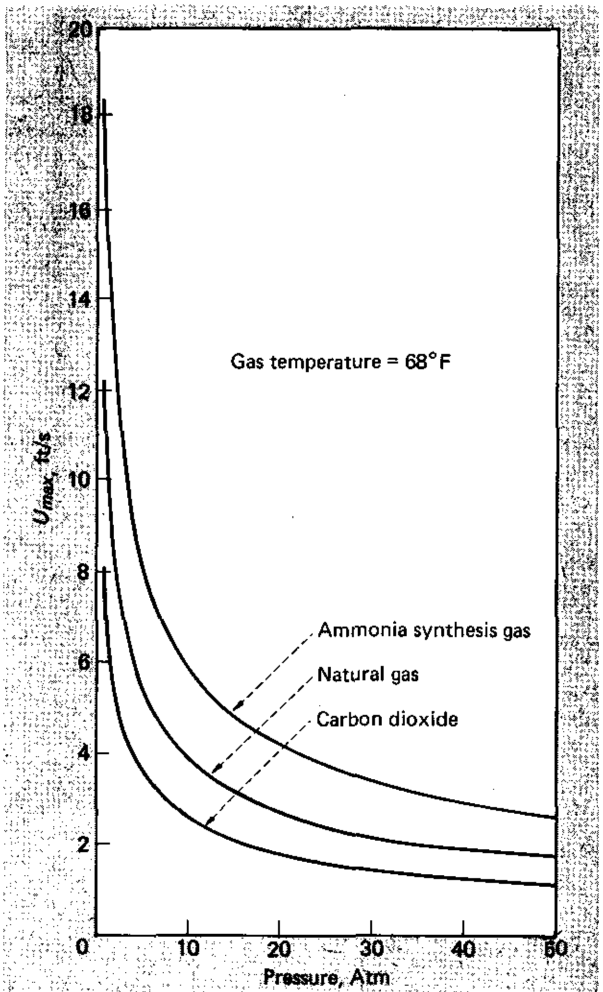
Since the separator achieves its maximum efficiency when the velocity is in the range of 20% to 100% of U_{max}, 20% more area should be allowed as a safety margin, to take into account liquid loading and surges.

Thus, the area of the mesh to be used should be:

$$A = 4.986/0.8 = 6.25 \text{ ft}^2$$



Critical velocity for air, as a function of temperature and pressure Fig. 1



Critical velocity of three gases at 68° F, as a function of pressure Fig. 2

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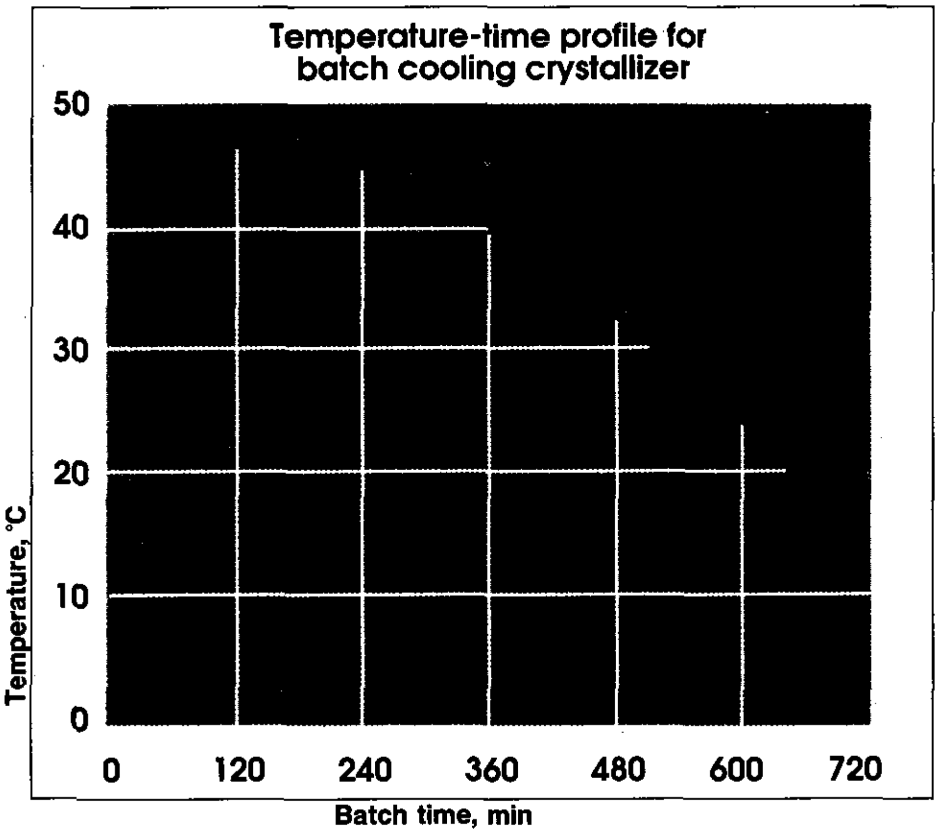
*EDC (Heat Exchangers), Bharat Heavy Electricals, Ltd., Hyderabad - 500 032, India

HOW TO SIMPLIFY BATCH CRYSTALLIZER DESIGN

K. A. Ramanarayanan,
Hoffman-La Roche Inc.*



Temperature-time profile for batch cooling crystallizer

The classic approach to crystallizer process design involves simultaneous solution of the (crystal) population balance equation with the mass balance equation. Larson and Garside [*1*] used this approach for the design of a batch cooling crystallizer. In most industrial situations, however, process design involves determination of an explicit *crystallizer vs. time profile, for a* given vessel. A simplified method can be used that will aid in obtaining relatively larger particles with a specific morphology, flow properties and purity. A similar procedure can be developed for evaporative batch crystallizers.

During the past several years, in situ microphotographic observation of contact nuclei and the subsequent growth of these nuclei has revealed a number of interesting phenomena. These can be summarized as:

- The nuclei are produced in a variety of sizes ranging up to about 40 μmeters. This phenomenon is called the "birth size dispersion." Similarly, different crystals grow at different rates, a characteristic called "growth rate dispersion"
- The individual nuclei grow at a constant rate under given conditions. This *growth rate is determined by the intrinsic* crystal properties and is not related to size during the growth rate
- There is no correlation between the birth size and the growth rate of an individual crystal

These observations have been summarized in a statistical-mathematical equation called the Constant Crystal Growth (CCG) Model [*2*]. The CCG Model provides a generalized engineering approach to account for distribu*tions in* birth size, *growth* rate and residence time. For a batch crystallizer, however, there is no residence time distribution, and the model can be simplified considerably. Further, for a seeded crystallizer containing a monodispersed seed size and no growth rate distribution (all the crystals grow at the rate of G), the CCG Model reduces to

$$L = L_s + Gt \qquad (1)$$

Consider a batch crystallizer at its satu-

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*Dr. Ramanarayanan wrote this article while at Hoffman-La Roche, in Nutley, N.J. He currently works at Dow Chemical U.S.A., Pharmaceuticals Process Research, Bldg. 845, Midland, MI 48667

Sample problem parameters

maximum allowable growth rate (G) = 3.0 μm/min
product crystal size (L_p) = 2,000 μm
available seed size (L_s) = 100 μm
production rate (P) = 500 kg
feed concentration (C_o) = 0.25 kg A/kg B

feed temperature (T_o) = 50°C
density of feed solution (ρ_o) = 1.05 kg/L
density of final mother liquor (ρ_f) = 1.05 kg/L
density of solute crystals (ρ_s) = 1.8 kg/L
desired final batch temperature (T_f) = 20°C

ration temperature, seeded with monodisperse seeds of size L_s. On cooling the crystallizer, the seeds grow at the expense of solute depletion in the liquid phase. At any time, t, the seeds have grown from size L_s to size L at growth rate G.

If W is the quantity of solvent and the concentration of the solute is expressed as C on a solute-free basis, the solute depletion can be expressed as $W(C_o - C)$. An increase in the mass of seed crystals accrued during growth is equal to the mass of crystals m at time t, less the mass of seeds m_s. This can be expressed as:

$$N \rho_s k_v (L^3 - L_s^3)$$

where k_v is the volumetric shape factor for the crystals and N is the total number of crystals in the system. Thus, writing a solute balance and equating the two terms gives:

$$W(C_o - C) = N \rho_s k_v (L^3 - L_s^3) \qquad (2)$$

By introducing the term for the seed mass, m_s, Equation 2 can be modified into an operating line equation for the batch crystallizer:

$$C = C_o - (m_s/W)[(L_s + Gt)^3/L_s^3 - 1] \qquad (3)$$

Equation 3 is the operating line relating the solute concentration in the crytallizer liquid phase with the batch time, t. The solute concentration C in the crystallizer is the sum of the saturation concentration, C^*, and the unrelieved supersaturation in the liquid phase. For the sake of simplicity, the supersaturation term is neglected, so that $C = C^*$. Hence, the solute concentraiton in the liquid phase in the crystallizer can be related to the crystallizer temperature via the solubility relationship,

$$C^* = C^*(T) \qquad (4)$$

The actual solubility data can be fitted to, say, a polynomial of temperature and used in place of Equation 4.

The operating line (Equation 3) can be solved simultaneously with the equilibrium line (Equation 4) to obtain the temperature-time profile for the batch crystallizer.

It is assumed in this analysis that there is no secondary nucleation during the batch crystallization process. Secondary nucleation will tend to increase the number of crystals and thereby reduce the final average product size. Hence, the temperature profile obtained from the above analysis should only be used as a first approximation for process design.

Sample problem

Design a batchwise, seeded cooling crystallizer for a solute, A, crystallizing from solvent B. The concentrations are expressed on a solute-free basis. The crystallization is accomplished by cooling a feed solution at an initial concentration of 0.25 kg solute A per kg solvent B from from 50°C to 20°C. The operating conditions and the physico-chemical properties of the system are provided in the Table. Determine:

a) crystallizer volume
b) quantity of seeds to be added and temperature of seeding
c) batch time
d) theoretical yield
e) temperature-time profile

Assume that there is no secondary nucleation in the crystallizer.

Nomenclature

C	concentration of solute, kg solute/kg solvent
ρ	density of seeds, kg/L
G	growth rate of seeds, μmeter/min
k_v	volumetric shape factor
L	crystal size, μmeter
m_s	mass of seeds, kg
N	total number of crystals
P	production per batch, kg
t	batch time after seeding, min
T	crystallizer temperature, °C
V	crystallizer volume, L
W	mass of solvent B, kg

Subscripts

b	batch time
o	initial
p	product
s	seeds
f	final

Solution

A certain amount of process data is normally available, or must be derived from laboratory studies (e.g., kinetic growth data). These values are given in the Table. The solubility relationship is given by

$$C^* = 0.01 + 0.005T$$

where T is in °C and C^* is expressed as kg solute A per kg solvent B.

Crystallizer volume

The first step is to determine the crystallizer volume. The final crystallizer temperature is 20°C. From the solubility relationship, the solute concentration at 20°C can be calculated as

$C_f = 0.01 + 0.005 \times 20 = 0.11$ kg solute/kg solvent

From a solvent balance,

$$W(C_o - C_f) = 500$$

or the solvent required for crystalliztaion, W,

$= 500/(0.25 - 0.11)$
$= 3{,}571$ kg of solvent.

The total solution mass equals the solvent mass plus the solute mass:

$= W + W \times C_f$
$= 3{,}571 + 3{,}571 \times 0.11$
$= 3{,}964$ kg of solution.

Using the density figure from the Table, this weight can be converted to an initial volume of 3,775 L of solution. The production rate, 500 kg, gives the quantity of crystals; their volume is 500/1.8 kg/L, which equals 278 L of solids. The total volume is the sum of the solution and solid volumes, or 4,053 L.

Seeds and seeding

In the absence of secondary nucleation, the total number of seeds is the same as the total number of product crystals, N. Then

$$m_s = N \rho_s k_v L_s^3$$

$$P = N \rho_s k_v L_p^3$$

Dividing one by the other,

$m_s = P(L_s/L_p)^3$
$= 500(100/2{,}000)^3$
$= 0.0625$ kg or 62.5 g.

The seeds have to be added to a saturated if not supersaturated solution. The saturation temperature corresponding to a feed concentration of 0.25 kg/kg can be calculated to be 48°C from the solubility relationship. It is therefore necessary to cool the feed from 50°C to 48°C, seed the crystallizer with 62.5 g of

100-μmeter seeds, and follow the temperature profile as determined below.

The batch cycle time can be calculated from the CCG Model:

$L_p = L_s + Gt_b$
$2{,}000 = 100 + 3\,t_b$
or t_b equals 633 min.

Theoretical yield

The solute in feed to crystallizer equals $W \times C_o$, which equals $3{,}571 \times 0.25$ or 893 kg. The crystal production is 500 kg. Then, yield equals 500/893, or 56%.

Temperature profile

The temperature-time profile can be calculated from the simultaneous solution of Equations 3 and 4. Hence,

$$0.01 + 0.005T = C_o - (m_s/W)[(L_s + Gt)^3/L_s^3 - 1]$$

This is a cubic equation relating temperature with time. Substituting the relevant parameters,

$$T = 48 - 0.0035[(1 + 3t/100)^3 - 1]$$

Note that at $t = 0$, the solution is at its saturation, corresponding to a concentration of 0.25 kg/kg. This cooling curve is plotted in the accompanying graph. Note the the cooling rate is very slow initially, followed by a faster cooling rate toward the end of the cycle.

From a practical perspective, the initially slow cooling rate provides enough supersaturation to balance growth. Typically, this would help in producing a small number of larger crystals compared to the case when the cooling rate is initially high. In the latter case, the solution will tend to cross the metastable limit for the system, resulting in spontaneous nucleation.

The above analysis suggests that natural exponential cooling is typically undesirable as it tends to cool faster initially due to the larger driving force, followed by slower cooling toward the end of the batch. This generally would result in spontaneous nucleation and a large number of smaller particles. ■

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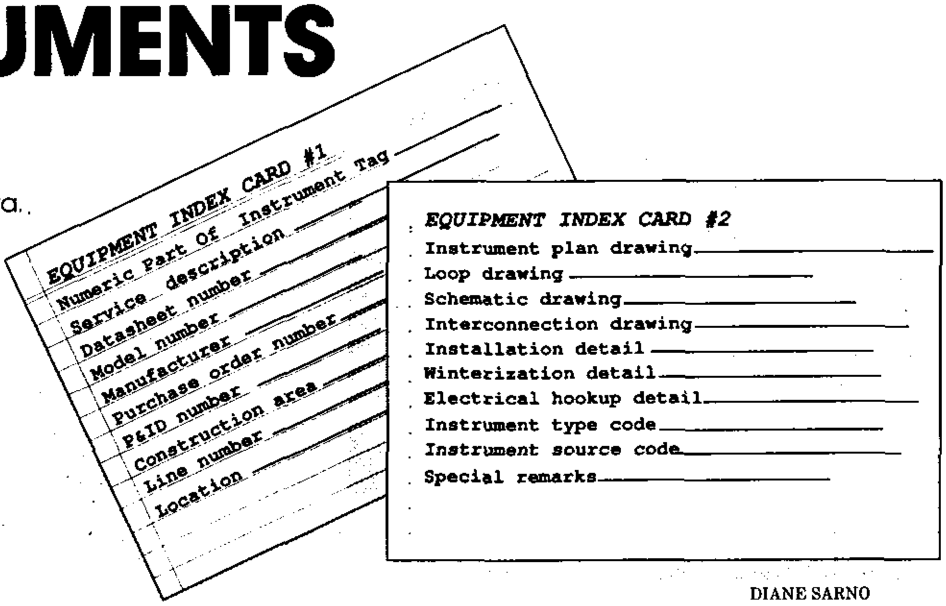
PROCESS MEASUREMENT and CONTROL

HOW TO KEEP TRACK OF INSTRUMENTS

Walter Driedger, P.Eng.,
Fluor Daniel Canada, Inc., Calgary, Alta.



EQUIPMENT INDEX CARD #1
Numeric Part Of Instrument Tag
Service description
Datasheet number
Model number
Manufacturer
Purchase order number
P&ID number
Construction area
Line number
Location

EQUIPMENT INDEX CARD #2
Instrument plan drawing
Loop drawing
Schematic drawing
Interconnection drawing
Installation detail
Winterization detail
Electrical hookup detail
Instrument type code
Instrument source code
Special remarks

DIANE SARNO

When the design or revamp of a chemical processing plant is completed, there may be thousands of sensors, transmitters, valve positioners and similar instruments put in place. Keeping track of the design or maintenance status of these units can be burdensome. The usual recordkeeping measure, an "instrument index" list, is commonly looked on as a low-level clerical function, and is sometimes treated as a chore that can be put off until after commissioning.

But this index can be a far more useful tool. To realize its full potential, it must be initiated at the very beginning of the design process and must be handled by a senior engineer. Moreover, computerizing this index as a database makes it accurate and efficient, and a real time-saver. I have found that by following the procedures outlined here, the time spent generating the index can be shaved by 20%, and the value of the index is significantly higher.

Start at the beginning

It has been my practice to begin the first review of the piping and instrumentation diagrams (P &IDs) with a blank set of index update forms (either a paper index card, or the individual records in a computerized database). At this point I assign the tag numbers and fill service descriptions into the form. A number of other decisions are made while filling in the form:

- Is the item part of an equipment package assembled by someone else?
- Will the item have a data sheet?
- Will it have a loop drawing?
- Will it have an isometric drawing?
- Will it appear on a location plan?
- Will it have a wiring diagram, an installation detail, a winterization detail?

The answers to each of these questions are usually obvious. In those cases where it is not, a design decision is made. In all cases where the answer is "no", "N/A" is filled into the form. This step is extremely important because it shows that the question has been considered and not ignored. If it is conscientiously done from the beginning, work lists can be immediately produced from the index.

A well-organized database, kept current, helps maintain project status

In general, a blank anywhere in the index represents work to be done. For example: As soon as the first P & ID review and tagging is completed, a listing of all items with a blank in the loop-drawing field is a complete list of loop-drawing requirements. At the end of the project the index will have no blanks whatsoever. During the course of the project, the various blank fields are filled in. This must be done as the work is completed, not in advance. For example, as each piping isometric (ISO) is checked by instrumentation, the ISO number is filled in. This is evidence that the work has been completed and checked. The same approach is used for installation details and other fields that document the instrument.

At any stage of the job it is possible to come up with "percentage complete" figures:

$$\text{Percent complete} = \frac{\text{number of fields not blank and not N/A}}{\text{number of fields not N/A}} \times 100$$

For example, applying this formula to the datasheet field gives the percent complete for datasheets. It is, of course, a simple matter to produce a list of those items which do not yet have datasheets.

Quality control

The sorting capabilities of a modern database program offer many opportunities for quality checks. This happens in *three ways: Firstly, unusual data items* tend to sort to the beginning or the end of a sequence. This makes oddball items stick out. Secondly, a sorted list of any field will make similar items come together. For example: A sort by purchase-order number will have orifice plates in one row and control valves

Database programs offer many opportunities for quality checks

in another. Errors are readily apparent. Even sorting the "Remarks" field will turn up discrepancies that may have passed unnoticed despite months of use.

The third function of sorted lists is for checking. Toward the end of the job, various quality control checks must be made on every project. My way of rechecking P & IDs is to list the entire index sorted by P & ID number. All items are then marked one by one on both the drawing and the printout. A similar check is done datasheets, location drawings, loop drawings, etc. The type of error uncovered by this method are deletions or additions that haven't been followed through. It never ceases to surprise me how many such errors are found on every project. Only commissioning will tell how many slip through if this type of check is not performed.

During the course of the job it is very useful to produce specialized lists for various purposes. These requirements will vary from job to job. In fact, it is impossible to predict in advance what these various requirements will be. An index that is at all times current will make it possible to satisfy these requirements on short notice. Examples I have encountered include:

- A list of all items to be supplied by a certain compressor vendor. The purchase order number was the key to this list
- A list of all items used for utilities within the plant. This was required for tax purposes. The P & ID number was the key to this list
- A list of all items to be installed by each area field contractor. The instrument plan number was the key to this
- A list of all devices requiring flexible insulation, based on equipment specifications
- A list of all items required to commission the emergency generator. Again the P & ID was the key. In addition, a list of those emergency generator items for which schematics had not yet been issued

Finally, commissioning lists can be readily produced for marking up during checkout. On one project, the field inspectors required lists of all instruments expected on each piece of packaged equipment. The few missing items were noted and found well before the actual startup.

Database management

All these functions can be readily accomplished on a microcomputer using a commercially available database-management program. These programs are extremely powerful. They offer the flexibility of tailoring an index to the needs of any project. Since it is rarely possible to anticipate all the special fields that may be required, the ability to restructure the data base in the middle of a project is very useful.

For example: Financial considerations made it necessary to divide a project into several phases. The commercial database we were using allowed us to add a "phase" field to the index. Any items listed for Phase I but not completed on time were promoted to Phase II. In this way, worklists could be supplied to the field in order of priority.

No two projects are ever alike, and it is impossible to compare what is with what might have been. However it has been my impression that a computerized index of the type outlined above costs about 80% of a simple manual index. To achieve this cost reduction, two major considerations must be kept in mind:

The index must be simple. It is impossible, within the scope of a single project, to develop an integrated system tying directly into P & ID drafting computers, scheduling systems or any other complex system. Until such integrated systems are available for purchase or unless your company has vast development resources, keep it simple.

One person must be responsible. He or she must have sufficient authority to control access to the data base. It is very easy for one sloppy worker to cause major damage. The responsible engineer must also be competent to judge the requirements for various special needs and to be able to provide them promptly. Much of the value of the special lists depends on timeliness.■

Don't overspecify control valves

Tight shutoff costs money. You can save on valve costs by understanding just what degree of shutoff is really needed.

***Mark Adams**, Fisher Controls International, Inc.*

☐ Eliminating overspecification of capital equipment can reduce costs significantly. In a recent comparison of two nearly identical petrochemical plants, one of them specified American National Standards Institute (ANSI) Class IV shutoff for control valves, and the other Class II. The savings from less-stringent shutoff requirements amounted to as much as 20% of the cost of the tighter-shutoff valves.

What did the one company gain from its extra expenditure of thousands of dollars? Perhaps some manager gained peace of mind, or some engineers were relieved of the responsbility of deciding if a lower shutoff classification would suffice in each case.

Chances are, that the plant gained nothing in performance, and that most of the valves are not even being maintained to tight shutoff levels. In short, thousands of dollars may have been wasted.

Understanding shutoff

Shutoff is one of the most misunderstood basic valve concepts. Often valve suppliers discover that customers really have not considered which shutoff class they actually need.

Tight shutoff in a valve can cost a substantial amount, both initially and in maintenance. Considerable savings can often be realized by understanding the various ANSI leakage classes and by carefully determining the class truly needed in each application.

Shutoff classifications

Shutoff (leakage) classes for valves, standardized in 1976 by ANSI B16.104, are summarized in the table. (See the original standard for test media, pressures and required procedures.) These classifications should be familiar to anyone concerned with valves, whether for throttling or on-off applications, and whether manual or power-operated. The specifications are not complex, but a few important facts should be noted.

Class I is not a quantitative leakage classification. The specification merely calls for the "manufacturer's standard" shutoff capability for the valve in question and does not even require a factory leak test. It is used only in special circumstances at the agreement of customer and vendor.

Classes II, III and IV call for progressively tighter shutoff, expressed as a percentage of valve capacity. Classes V and VI are a bit more complicated (see the table).

Since test conditions will seldom match the valve's service conditions, the actual leakage in service may be significantly different from that specified in the standard. This may be due to temperature, fluid type, in-service wear, differential pressure, or seat load. Each of these is discussed below.

ANSI standards specify the test-fluid temperature to be 50 to 125°F (10 to 50°C). Leakage is affected by the different thermal expansion rates of the various valve parts and by the changing plasticity of the sealing members.

Test conditions may also differ from service conditions with respect to the fluid involved. The ANSI standard requires tests with either air or water for Classes II,

Control valve seat leakage classes

ANSI/FCI 70-2-1976 (R1982)*

Leakage class	Maximum allowable leakage
I	(No specification)
II	0.5% of rated capacity
III	0.1% of rated capacity
IV	0.01% of rated capacity
V	0.0005 mL/min of water, per inch of port diameter, per psi differential
VI	See table below

Class VI seat leakage allowable

Nominal port diameter, in.	mm	Leak rate mL/min	Bubbles/min†
1	25	0.15	1
1½	38	0.30	2
2	51	0.45	3
2½	64	0.60	4
3	76	0.90	6
4	102	1.70	11
6	152	4.00	27
8	203	6.75	45

*Formerly ANSI B16-104.

†See Standard for measurement specifications.

III and IV, water for Class V and air or nitrogen for Class VI. Process fluids that have higher viscosity or greater solids content can be expected to leak less than do the test fluids.

Differences may also be due to the condition of the valve after it has been in service for a time. For example, a Class V valve can be quickly turned into a Class II valve (or worse) by erosion due to abrasive solids, cavitation, or the "wire drawing" effects of flow across valve seats when the valve is shut off or nearly so.

Perhaps the most drastic difference between test and service conditions is the differential pressure (psid). Except for Class V, all leak-class tests are conducted at either 50 psid or the service differential specified on the order, whichever is lower.

For example, consider a Class IV valve ordered for service at a shutoff differential of 1,000 psid. There is no easy way to predict leakage at differentials above 50 psid, even with air at room temperature. One reason for this unpredictability is that "choked flow" will develop in the narrow leakage path at a lower differential pressure than in the open valve. This effect will tend to prevent the leakage from rising in proportion to full flow as the differential pressure is increased. In addition, leakage is very sensitive to the force load of the plug on the valve seat.

If the valve configuration is such that the fluid pressure assists closure, the leakage rate could conceivably decrease with increasing pressure.

On the other hand, if unbalance tends to open the valve instead, leakage may be greater at higher differentials unless specific actuator sizing considerations are applied.

The unpredictable effects of seat load could be eliminated if all valves, regardless of ANSI class, were tested at the actual seat loads they will experience at the full service differential. However, the ANSI standard calls instead for testing with the full actuator force—a force that is the sum of seat load, unbalance force on the plug, and friction forces. In some cases this test condition results in significantly higher seat load at test than in service.

Because of all these differences between test and service conditions, shutoff specifications should be used mainly for comparing the capabilities of alternative valves rather than in an absolute sense. The buyer knows that a Class IV valve will leak much less than a Class II valve of a similar configuration—although one can never be certain just how much less, or exactly how much leakage can be expected in either case.

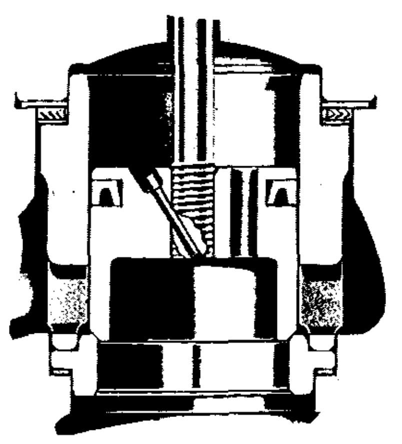
In most situations, past experience is a good indication of what is acceptable.

If the actual amount of leakage is critical to a given application, the manufacturer can usually simulate the service conditions and give a more accurate and meaningful test. It is necessary, however, to justify the increased cost of this test work. For most applications, the ANSI test criteria provide an adequate technical base for judgment.

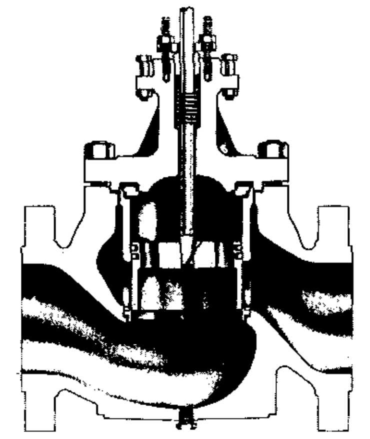
The cost of tight shutoff

The fact that actual leakage in service cannot be accurately predicted from the shutoff specification may reinforce a purchaser's inclination to overspecify. However, the cost of progressively tighter shutoff specifications for sliding-stem control valves of the same general family, in the same application, ranging in size from 1 to 6 in., is worth noting. Actuator costs increase too, since greater seat loads are required for the tighter shutoff classes.

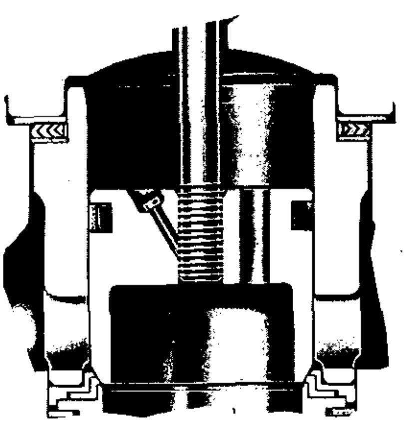
A third factor is increased maintenance costs. There is considerable wear on the soft parts generally used to achieve tight shutoff. Therefore, to maintain the shutoff specification, it is necessary to rebuild the valve more often—leading to increased cost of maintenance personnel, repair parts,



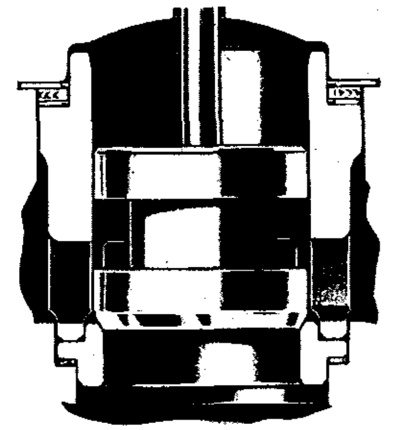
a. Basic balanced plug with metal seating (Class II shutoff)



b. Balanced valve with optional double piston ring (Class IV)



c. Balanced valve with TFE seat ring; moderate temperatures only (Class V)



d. Basic unbalanced plug valve (Class IV to Class VI with optional constructions) parts inventory, and downtime. The cost of these, especially downtime, can be significant.

Sliding-stem valves fall into two general categories: unbalanced and balanced. These terms refer to the design of the plug, and to the amount of force required to operate it. Unbalanced valves are simple, but require high closing forces due to pressure effects. Balanced valves are slightly more complex, but are less affected by pressure, thereby allowing easier operation by smaller, lower-cost actuators.

For each valve size, the progression toward tighter shutoff begins with a balanced valve with metal seats (Part *a* of the figure), rated at ANSI Class II. To bring that valve up to Class III, the standard carbon-filled TFE piston ring that seals the plug against the cage can be replaced with a graphite ring. Furthermore, the plug must be lapped against the seat ring using a grinding compound, and the seat load must be doubled. Each step involves a higher price.

Achieving Class IV in balanced valves requires going to a more complex piston-ring arrangement: a seal ring with a separate backup ring (Part *b* of the figure). The metal seat-ring remains the same, and seat lapping of the plug and ring is provided on a standard basis. Seat load may have to be increased as well. Again, obtaining Class IV in a balanced valve involves additional cost.

Moving up to Class V entails using a seat ring containing a TFE (tetrafluoroethylene) or elastomeric disk (Part *c* of the figure). Less seat load is required, but the valve's temperature limits are significantly reduced. The "soft seat" is also more subject to damage from erosive particles, and is more expensive than a simple metal one.

Due to leakage past the plug-to-cage seal, Class VI shutoff cannot be achieved with a balanced valve. Various versions of unbalanced valves provide Class IV, V and VI shutoff (Part *d* of the figure). No special lapping is required, except perhaps to correct nicks or misalignment that occur during manufacture or maintenance.

The unbalanced valve may be moved up to the more expensive Class V by lapping the seat and significantly increasing the seat load.

Class VI in the unbalanced sliding-stem control valve, in turn, is achieved by using the same soft seat required for Class V in the balanced plug valve. The seat load decreases drastically to the same value required for the soft seat in the balanced valve. However, the unbalanced forces may require a significantly larger actuator.

Leakage varies with seat load. Valve manufacturers publish experimentally-determined seat loads required for the various shutoff classes. Seat load requirements are an important part of the actuator sizing calculations specified in product literature.

The seat loads are intentionally large enough so that the valves can be returned to their original shutoff capabilities by routine maintenance such as lapping seats and replacing seals. High seat load also increases the length of time tight shutoff can be maintained against high pressure drops, by reducing small leaks that can erode into big ones.

Implications for valve selection

Several conclusions can be drawn by comparing the costs of tight-shutoff valves with those of valves in ANSI Class II. For 1-in. through 2-in. valves, tighter shutoff does not cost much—typically 2% to 9% more than Class II for Class III, IV, or V shutoff ratings. In fact, in these small valve sizes, the unbalanced valve with Class IV may cost no more than the balanced valve with Class II.

But for 4-in. valves and larger, tighter shutoff carries a stiff price, mainly due to the larger actuators required for higher seat loads and unbalance force. In 4-in. and 6-in. sizes, the increase to ANSI Class IV or V in a balanced design can cost from 2 to 18% more than Class II. Obtaining Class IV or V shutoff in large unbalanced designs can cost considerably more.

Requiring tight shutoff in large valves is particularly expensive in applications where spring and diaphragm actuators are preferred or where positioners normally would not be required. This is due to the necessity of changing to piston actuators in order to get the necessary thrust.

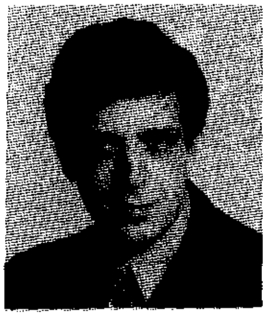
The cost of tight shutoff is particularly unnecessary with a valve whose only purpose is throttling control in a continuous process. Such a valve does not need to shut off tightly at all, and the Class II rating is more than sufficient. In this application, a valve typically throttles flow over a range of ten to one or less, for weeks or even months at a time. When the process has to be shut down, the line can be shut in—if necessary, by means of block valves that traditionally accompany a throttling control valve. Yet even for applications of this sort, many users specify Class IV. Perhaps not trusting the piston rings of balanced valves, they often further add to the cost by requiring unbalanced valves. For a 6-in. valve, the cost increase for this unnecessary tightness can be as much as 30%.

The only application in which a throttling control valve must shut off tightly is when it is required to do double duty as a remote-controlled block valve. There are a number of instances where this is a legitimate requirement. For example, there are turbine bypass valves that need to conserve energy when closed, but open quickly and throttle when the turbine is tripped off the line.

In a great majority of applications, including sliding-stem control valves, the implications of cost analysis are plain: rather than arbitrarily specifying a tight-shutoff class, one should generally pick the lowest class required for the application. This will help ensure that the most cost-effective valve is chosen.

Roy V. Hughson, *Editor*

The author



Mark Adams is Senior Product Manager, Fisher Controls International, Inc., 205 South Centre St., Marshalltown, IA 50158; tel: (515) 754-3011. He has worked at Fisher in various capacities since 1973, when he received his B.S. in physics from Iowa State University. He has published many technical articles and papers, and is a member of the Instrument Soc. of America.

Analyzing batch process cycles

Roger V. Morrison[*]

☐ Batch processes often must be monitored to determine capacity restrictions, to analyze time-cycle problems, or to optimize the cycles. Here is a simple graphical technique for analyzing batch cycles easily and rapidly.

First, the cycle is broken down into its component activities. Table I represents a reaction consisting of a sequence of seven steps. The allotted time is the design cycle time, budgeted cycle time or a judgmental optimum time for each activity; it is the standard against which actual performance will be compared.

Second, the batch activities are translated into a series of milestones, or batch events, as shown in Table II. These are specific instants in time that denote the beginning or end of an activity, and they must be easily recognizable by plant operators. The schedule is derived by accumulating the activity times listed in Table I.

Next, operating log sheets are developed based on the actual performance of the cycle, as shown in Table III.

Finally, the data from the operational log are plotted on a performance chart, as illustrated in the figure. The horizontal axis is a 24-h linear scale to show one operating day. The schedule is plotted along the vertical axis—using the same linear time scale as the horizontal axis—and the batch events are identified to the left of it.

Batches whose cycles conform to the schedule will be plotted along a 45° line descending from left to right (several are shown on the figure). Non-conforming batch cycles will deviate from the 45° slope, enabling problems to be identified quickly.

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[*] Plastic Coatings Corp., 2 Belle Glades Lane, Belle Mead, NJ 08502

Batch reaction consists of a series of activities **Table I**

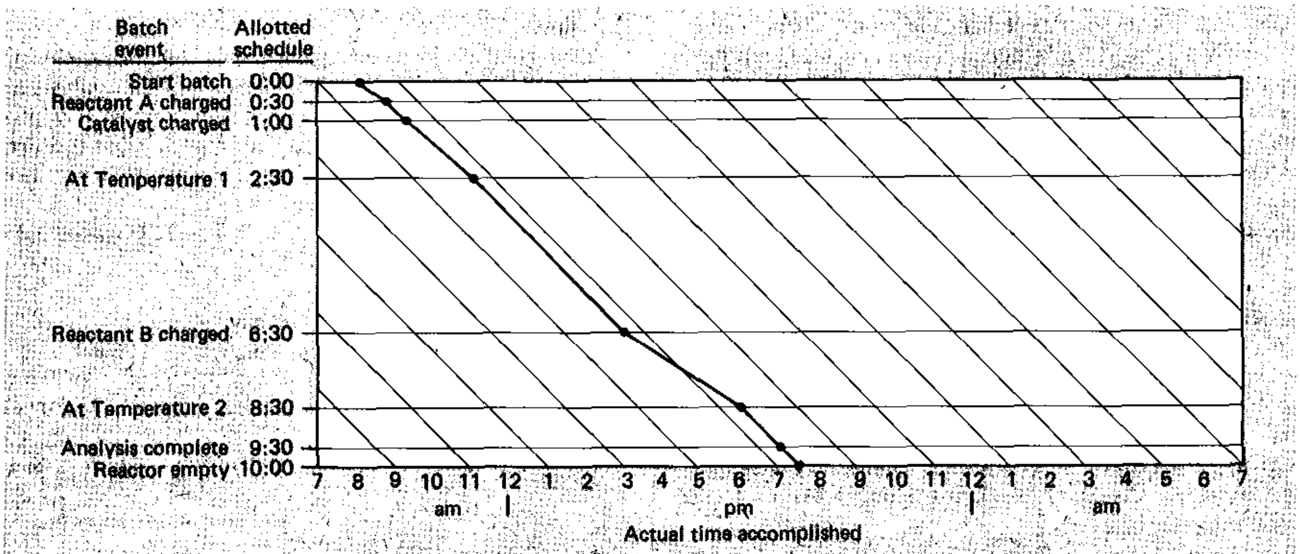
Batch activity	Allotted time for activity
Charge Reactant A	0:30
Charge catalyst	0:30
Heat to Temperature 1	1:30
Add Reactant B	4:00
Heat to Temperature 2	2:00
Sample and analyze	1:00
Empty reactor	0:30
Total allotted time	10:00

Events denote completed batch activities **Table II**

Batch event	Schedule
Start batch	0:00
Reactant A charged	0:30
Catalyst charged	1:00
Temperature 1 reached	2:30
Reactant B charged	6:30
Temperature 2 reached	8:30
Analysis complete	9:30
Reactor empty	10:00

The operational log is a record of the batch cycle **Table III**

Batch events	Actual time accomplished
Start batch	8:00 am
Reactant A charged	8:45 am
Catalyst charged	9:15 am
Temperature 1 reached	11:00 am
Reactant B charged	3:00 pm
Temperature 2 reached	6:00 pm
Analysis complete	7:00 pm
Reactor empty	7:30 pm



Performance chart is used to compare actual batch cycles with schedules

Control disturbances? Check grounding

Unexplained control disturbances may be caused by grounding problems. Here is how to track down these tricky faults.

James McAlister, *Arkansas Power & Light Co.*

☐ Imagine that your plant is running smoothly, and all control loops are on automatic. Suddenly, several loops trip to manual. After resulting process upsets have settled down, the loops go back on automatic, only to have the problem repeat later in the day. All attempts to identify the cause are unsuccessful. What is happening?

Such unexplained disturbances often point to grounding problems in controls, computers, or vital electronic systems. Proper grounding is complex and often misunderstood, but is critical to proper system operation.

Since grounding problems often manifest themselves in a diversity of ways, precise identification is sometimes long in coming. A thorough, detailed examination of improper grounding is beyond the scope of this article, but reviewing the history of a few real problems can provide valuable insight to help engineers get on the right track quickly in solving unexplained disturbances.

Problem No. 1

At one of our plants, a boiler's control loops would mysteriously trip from automatic to manual, and valves would sometime change position for no apparent reason. These symptoms, along with a lack of hard facts, suggested transient electrical noise as an explanation.

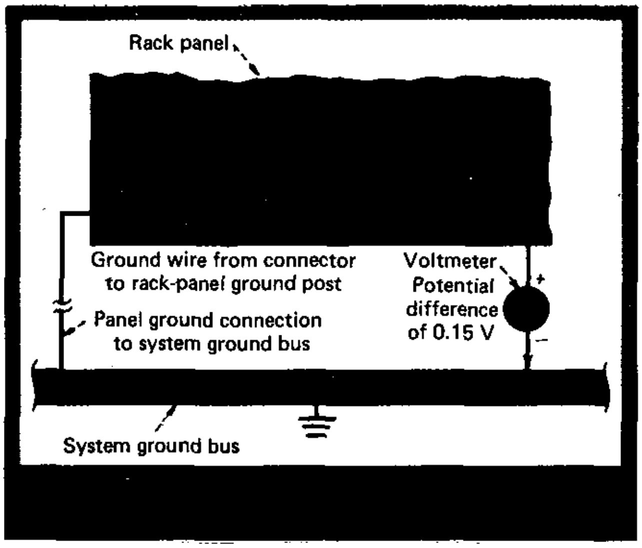
In the control racks, the individual circuit-card connectors were bolted to the rack panels and each card's ground wire was affixed to a small adjacent grounding post, which also was bolted to the rack panel (Fig. 1). Voltage checks between the system ground bus and the connector's ground pins showed rather wide variations in potential difference. In some cases, voltages as high as 0.15 V were recorded when, theoretically, all measurements should have been zero. These voltage levels were, of course, unacceptable in a 6-volt logic system and helped explain some of the irregular operations.

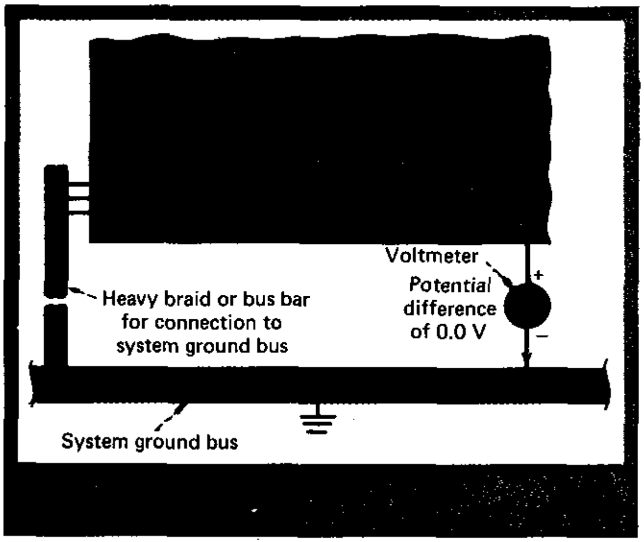
The grounding scheme was modified (Fig. 2) to alleviate the problems. Separate ground wires were run directly to each individual connector and then attached to the system ground bus by means of a heavy ground strap. Ground potentials were reduced to only a few millivolts and erratic operation was eliminated altogether.

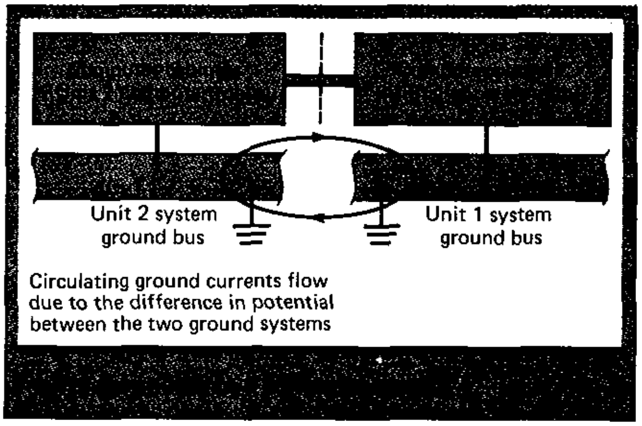
Problem No. 2

In another instance, one plant's computer frequently tripped, especially during lightning storms. Again, grounding problems were suspected and tests outlined by the manufacturer pointed to multiple grounds rather than the single ground prescribed. Resolution evolved in several phases.

First, the computer cabinets were physically lifted







from the concrete floor, high enough to allow insertion of sheets of electrical insulation between them and the floor. The machine's main ground was removed, but the computer-to-ground resistance did not fall within specification. An inspection of the computer's internals revealed a small cooling fan that was rotating very slowly in the reverse direction even though all power had been removed from the machine. Circulating ground currents seemed to be the only explanation, and the stray ground allowing the currents was finally located. It was in an operator's console, which was located in another generating unit in the same station (Fig. 3).

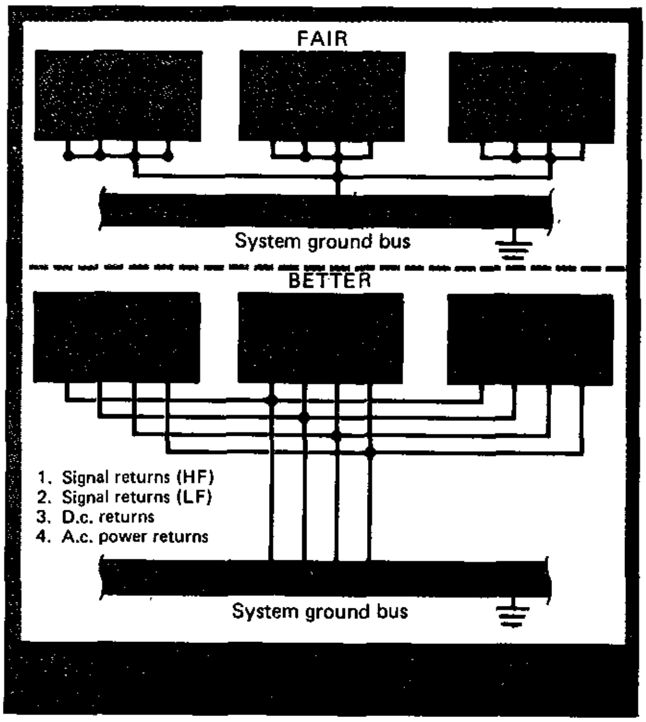
The console was connected to the ground bus of one unit and the computer itself to the ground in the other unit. If ground currents can produce such significant current flow in a power circuit, consider the effects that might be seen in a sensitive electronic circuit. The ground for the console was connected back to the computer, but tests indicated yet one more extraneous ground. It was eventually located in a high-accuracy flowmeter that served as an input to the computer. There was a pilot lamp in the meter, and one side was grounded. Removal of the lamp brought the computer within specifications and the mysterious trips ceased.

Problem No. 3

This final problem was not actually one of grounding, but the principles involved are quite similar. In this case, a computer suffered the loss of a power supply, and no spare was available to replace it. There was, however, another supply with the same voltage rating within the same machine. The two supplies fed circuits that were electrically isolated. Due to the emergency, they were interconnected and powered from the working supply.

Everything worked well as long as the computer cabinet door was left open. With the door shut, the machine would run for only a few seconds.

Apparently, the proximity of the cabinet door provided the path for interaction between the two circuits, resulting in recurring computer error. The exact mechanism involved is not known, but it is similar in some respects to multiple-grounded circuits that exhibit interactions because of the grounds. Such grounds can accidentally be applied to sensitive circuitry through the third wire in a three-wire power cord to test instruments or through the shield of a shielded cable, to name just two. The point to note is that extreme care should be exercised when even the possibility of coupling two isolated circuits exists. The effects are not always predictable.



Other points

Some systems contain a mixture of circuits: for example, high frequency, low frequency, direct current, and a.c. power. The different types of circuit grounds should be separated, as illustrated in Fig. 4, to avoid possible undesired coupling effects. In addition, all lines carrying signals above 50 kilohertz frequency should be shielded, with the shields grounded at both ends. Grounds should be as short as possible. Twisted, shielded wire should be used for all power lines, and the shield should be grounded at the transmitting end only. Power circuits should be separated from low-level circuits as far as is practical. Twisted-pair cable should be used for all audio-frequency circuits.

In summary, these case histories, though not exhaustive, do illustrate several of the types of problems that you may expect to encounter in trying to run down unexplained control-system disturbances.

***Roy V. Hughson,** Editor*

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The author

James McAlister is Manager, Special Projects and Applied Research, for Arkansas Power & Light Co., P.O. Box 551—FCB 8, Little Rock, AR 72203. Tel: (501) 371-8204. He is responsible for the design and implementation of a major quality-improvement effort at all of AP&L's generating stations. He holds B.S. and M.S. degrees in electrical engineering from the University of Arkansas, and a certificate in Business Management from American Management Assns. He is a registered professional engineer.

Ensuring the security of field control cabinets

*Eric R. Lichtenheld**

☐ Unauthorized access to control cabinets in production areas can sometimes be a problem: Unlocked ones invite tampering, while the security of a locked cabinet is inversely proportional to the number of keys that have been distributed.

Consider, for example, an automated process with a manual control station located in a field cabinet. The following parties must have access to it:

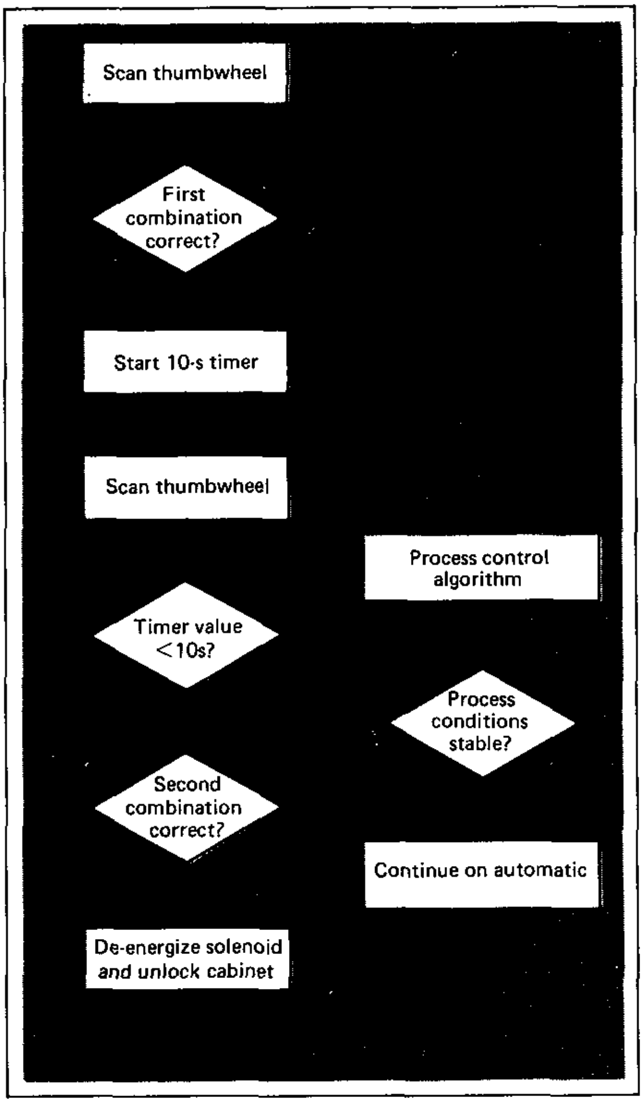
- Production—Upon failure of microprocessor control, the process must be operated manually.
- Maintenance—Periodic calibration and repair of the field hardware will be required.
- Engineering—Software modifications and controller tuning will sometimes be needed.

To control access to the cabinet, a solenoid lock and thumbwheel can be installed on the door of the field cabinet and wired to input/output points on the controller. The access code is specified by the system programmer and should be a pair of combinations that will be easy for the maintenance workers to remember. Upon entry of the correct code or a process instability, the solenoid will be de-energized to permit entry. This arrangement makes it unnecessary for production personnel to know the access code, and the unlock algorithm detailed in the figure uses a 10-s timer to frustrate clever operators.

As with any locking system, the following fail-safe features are necessary:

- The solenoid lock must secure the cabinet when energized, so that the removal of power permits entry.
- An external access port must be accessible to the system programmer so that the solenoid lock can be "forced off" or the access code changed.

In addition, counters and clocks can be added to the software to indicate the time of the last entry to the cabinet.



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*Armour Pharmaceutical Co., P.O. Box 511, Kankakee, IL 60901.

Avoid erratic control operation by suppressing relay arcs

James McAlister*

Almost everyone who has worked with electronic control systems has experienced a familiar problem—erratic operation due to noise induced by arcing relays. Because electronic control systems operate at low power levels, they are susceptible to energy induced by outside, higher-level electrical sources. One of the most common sources of this type of energy is the arc created by interruption of the current flow in an inductive circuit. The intensity of the arc depends upon such factors as the circuit current, the type of load, and the speed with which the current is interrupted. The arcs may be short, but they radiate intensive, undesirable electromagnetic energy that can potentially be "picked up" by sensitive circuitry. System designs that do not provide for proper rejection of this energy are prime candidates for problems.

In any kind of plant, there are many things, such as equipment location and cable routing, that are almost impossible to change after their installation. Obviously, the place to solve the problems caused by radiated energy is in the initial system design and installation, by using proper grounding and shielding techniques. However, even if that is done, problems still may exist.

Arc suppression

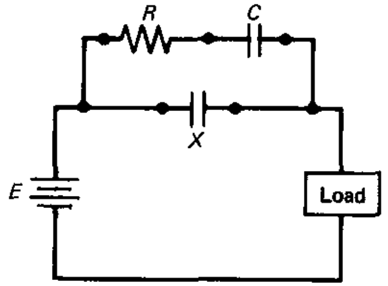
Suppression of the arc at its source also is often possible and may provide a satisfactory solution. In many cases, this is quite easy with the proper combination of a resistor and capacitor connected in series across the arcing relay contacts, as shown in Fig. 1.

The following equation can be used to calculate the optimal values of the required resistance and capacitance:

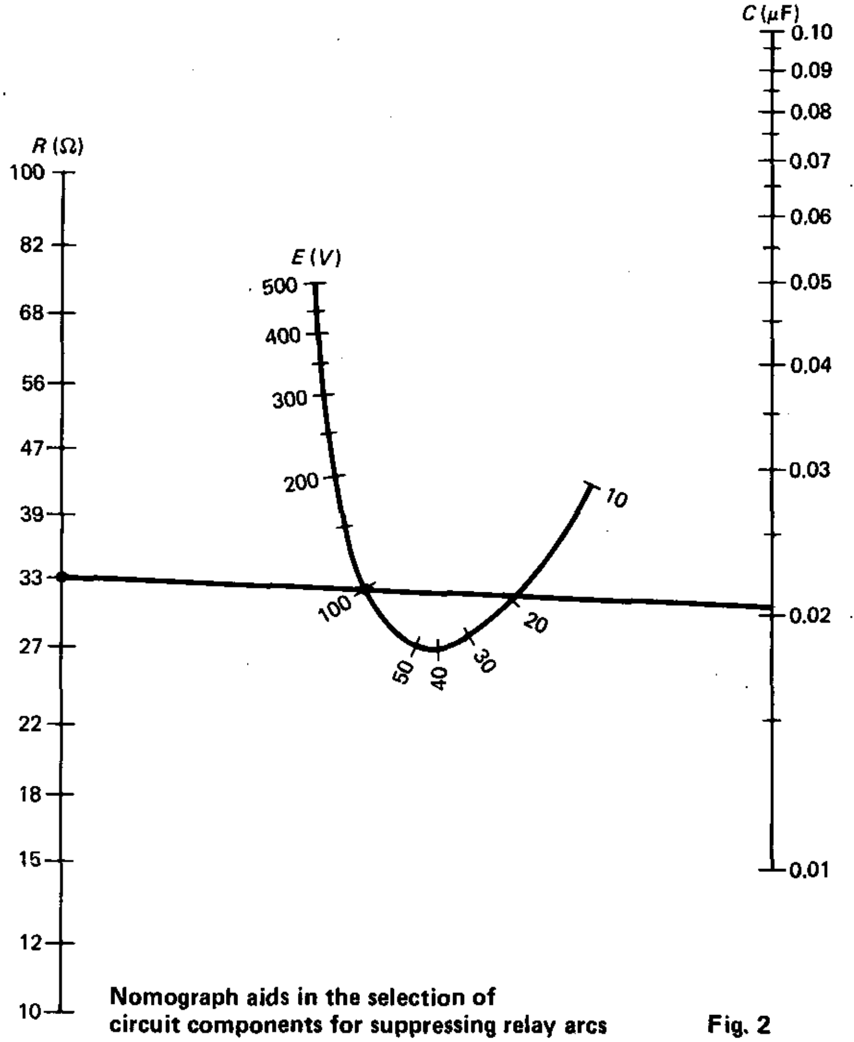
$$R = \frac{E}{10\,(3.16\sqrt{C})^{1+50/E}}$$

where R = resistance, ohms, C = capacitance, microfarads, and E = open-circuit voltage, volts.

There are an infinite number of R-C combinations possible for a given open-circuit voltage. The final selection will depend on such factors as component power ratings, circuit



Resistor/capacitor network suppresses relay arcs **Fig. 1**



Nomograph aids in the selection of circuit components for suppressing relay arcs **Fig. 2**

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*Arkansas Power & Light Co., P.O. Box 551, Little Rock, AR 72203.

voltage, and availability of parts. Fig. 2 can be used to estimate a variety of combinations very quickly.

Example

Relay contacts (denoted by X in Fig. 1) are in a circuit with an open-circuit voltage (E) of 100 V d.c. Find the R-C combination needed to suppress the arc, assuming that a 33-Ω resistor is readily available.

Solution: A line drawn through 100 on the E-scale and 33 on the R-scale intersects the C-scale near 0.02. Thus, a 0.02-μF capacitor would be satisfactory for the 100-V/33-Ω combination. Notice that this same R-C combination would be proper also for a 20-V circuit.

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A CONTROL ALGORITHM FOR BATCH PROCESSES

Simplify scheduling and provide automatic rate adjustment

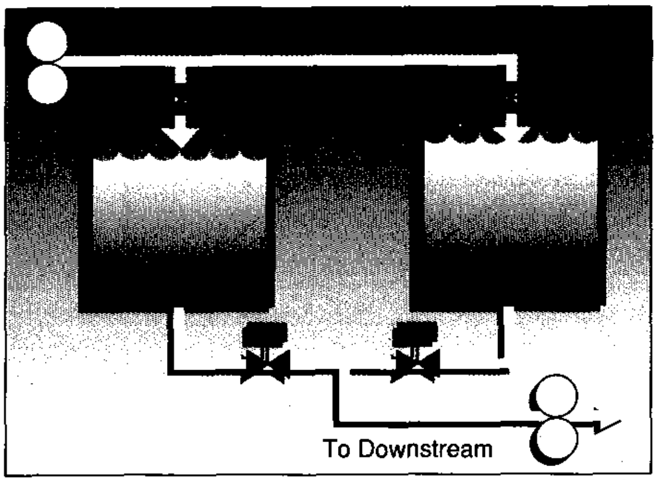

FIGURE 1. The problem: scheduling the fill steps for these tanks

Brian Philbin, Honeywell Industrial Automation and Control Div.*

In many batch applications, scheduling and coordination of the beginning and end periods of steps within the production process are required. With modern control algorithms, multistep processes can be designed where production rates can change and a batch schedule will automatically adjust the time to complete a phase or step. Here's how it could be done in one instance.

Figure 1 shows an example of a simple batch process with two vessels that cycle back and forth to feed a downstream process. Initially, tank T_1 will be filled. After it is full and the batch is completed, the bottom solenoid valves open up and feed a downstream process at varying rates depending on the overall production rate.

At some time before T_1 empties, tank T_2 must be filled and the batch completed. The fill steps for the components can vary either in the time they begin or the time that they take to complete the fill. Rather than trying to calculate when to begin the fill steps for each component, the rate at which the component is put into the vessel can be calculated so that the fill step finishes at a desired time.

The control strategy estimates the time at which the emptying vessel will become empty, and subtracts a specified time from that value. This becomes the estimate for the time to complete the fill of the other vessel. The

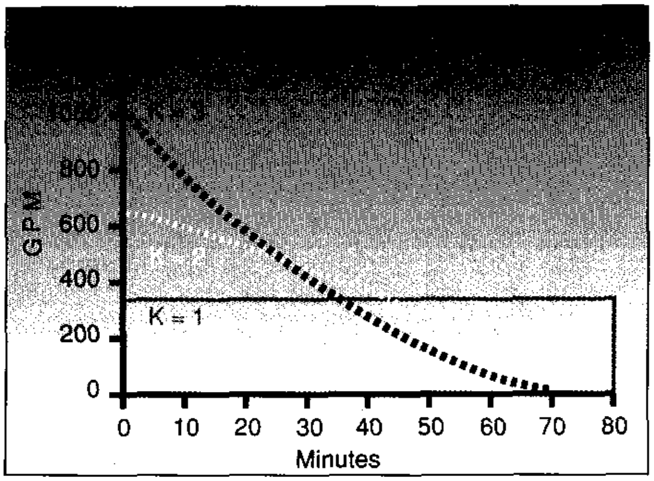

FIGURE 2. The choice of K factors affects response

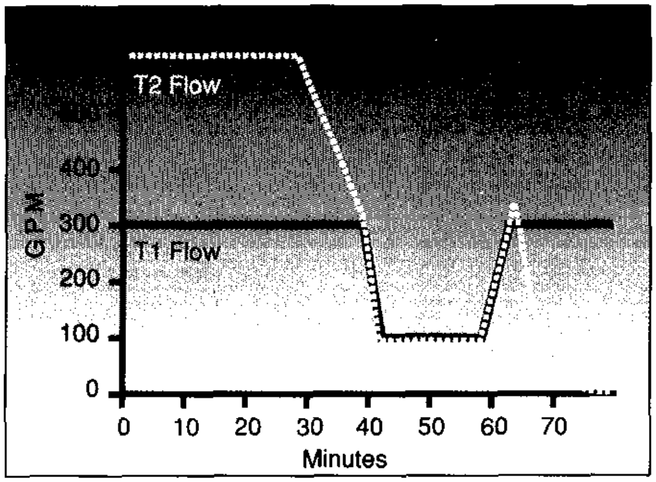

FIGURE 3. Flowrate for Tank T_2 moves in step with Tank T_1

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*16404 North Black Canyon Highway, Phoenix, AZ 85023

Control the process by estimating the time to empty a tank and subtracting a specified amount from that

flowrate of the component (or components, if more than one is involved) are then calculated. A tuning factor is used to affect the shape of the response.

Dynamic response

The dynamic response of the feedrate is calculated as follows:

$$T_e = (L_e - L_{et})/R_e \tag{1}$$

$$T_f = T_e - T_d \tag{2}$$

$$R_f = K \times (L_{ft} - L_f)/T_f \tag{3}$$

$$L_e = L_{e0} - \int k_e R_e \, dt \tag{4}$$

$$L_f = L_{f0} - \int k_f R_f \, dt \tag{5}$$

The scale factors represent the differences in units between level and rate, and the relative calibrations of the rate meters and level meters.

Substituting Equations 1, 2, 4 and 5 into Equation 3 results in the following integral expression:

$$R_f = \frac{R_e K(L_{ft} - L_{f0} - k_f \int R_f \, dt)}{L_{e0} - k_e(\int R_e \, dt) - L_{et} - R_e T_d} \tag{6}$$

Assuming that the production rate is constant, the solution is:

$$R_f = C_0[L_{e0} - L_{et}\, k_e - R_e T_d - k_e R_e T]^{(Kk_f/k_e-1)} \tag{7}$$

where

$$C_0 = KR_e(L_{ft} - L_{f0})/(L_{e0} - L_{et} - R_e T_d)^{(Kk_f/k_e)} \tag{8}$$

Example

Assume a batch process as shown in Figure 1 with the following conditions:

$L_{ft} = 5{,}000$ gal
$L_{et} = 500$ gal
$L_{e0} = 5{,}000$ gal
$L_{f0} = 500$ gal
$R_e = 300$ gal/min
$k_e = 1$
$k_f = 1$

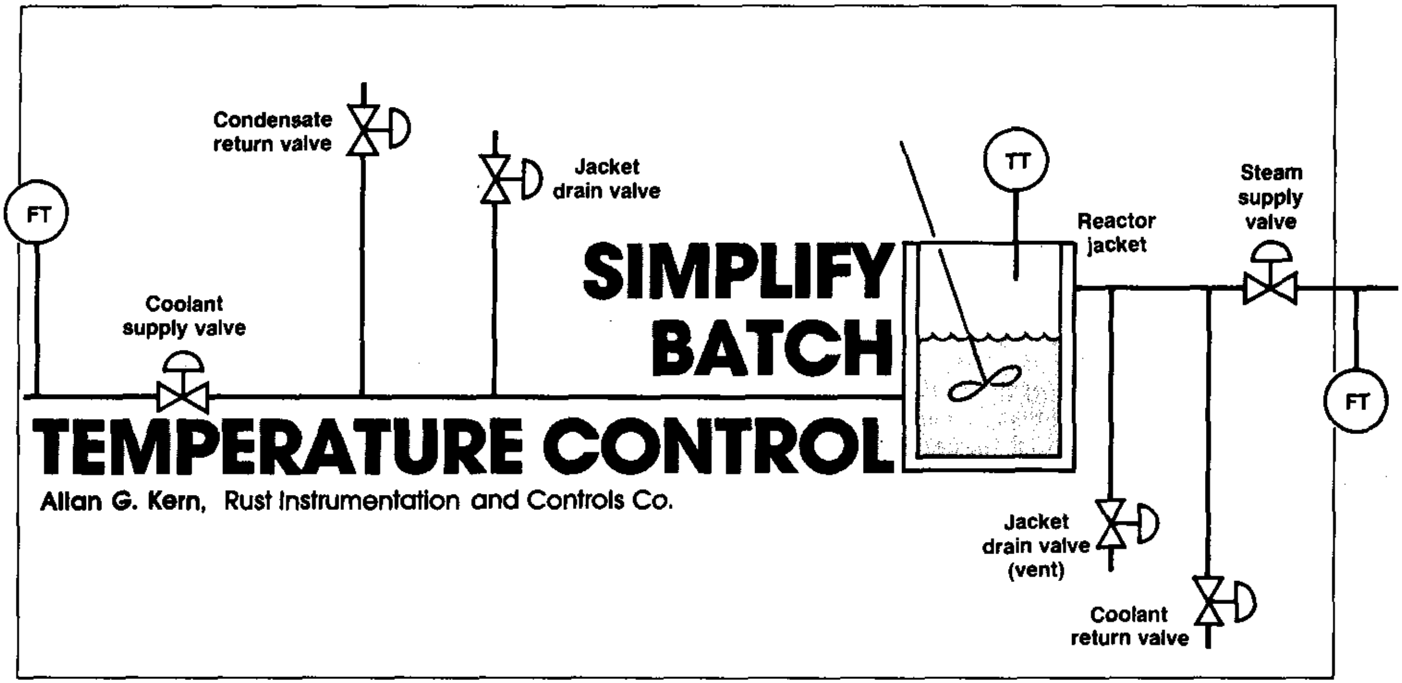
Figure 2 shows the response curves for different values of K. As one can see, if K is greater than 2, then the flow response slope at the time that the flow approaches 0 is 0. If K is less than 2, then the slope is infinite.

It is generally desirable to have the flow approach a low rate toward the end of the fill step, so that the target volume or weight can be accurately achieved. In many cases, the flow will be reduced to a dribble value. The tradeoff is that a larger pump, feeder or conveyor is required in order to provide the higher flowrates during the initial filling of the vessel.

In Figure 3, the response is shown when $K = 3$, the maximum flowrate is 600 gal/min, and the emptying rate varies from 300 to 100 gal/min. As can be seen by the response, the controller adjusts the feedrate (into tank T_2) to track the changes in the emptying rate (out of tank T_1) and still achieve the target fill time. ■

Nomenclature

T_e	=time to empty, min. T_f, time to fill; T_d, time difference between complete fill and complete empty
L_e	=current volume of the vessel which is emptying, gal. L_f, level of vessel that is filling; L_{et}, target empty level; L_{ft}, target fill level; L_{e0}, L_{f0}, emptying and filling tank levels at time 0
R_e	=current emptying rate, gal/min.
R_f	=feedrate
K	=tuning constant (dimensionless)
k_e, k_f	=scaling factors



SIMPLIFY BATCH TEMPERATURE CONTROL

Allan G. Kern, Rust Instrumentation and Controls Co.

Control the temperature of a batch reactor with one intelligent controller, instead of a slew of gages, valves and motors.

Most batch reactors today employ, at best, only partially automatic control. The operator must keep track of numerous valves, motors, and flow and temperature gages — just to control temperature, and just for one reactor. Failure to continuously coordinate these devices leads to varying temperature profiles from batch to batch, and, thus, inconsistent or unacceptable product quality.

By employing a single, off-the-shelf, microprocessor-based controller that incorporates a second algorithm (representing the control criteria for the entire process), control is much simpler, and more versatile, accurate and reliable, than conventional, auto/manual, single-loop PID (proportional-integral-derivative) controllers and the accompanying process hardware.

The concept

The temperature cycle for an exothermic batch reactor typically includes heating to initiate the reaction, cooling to control the rate of reaction, and adding more heat to drive the reaction to completion. Often, intermediate reactants must be added and more heating/cooling cycles are required. Further complicating the control problem, the reactor jacket and/or coils must be drained of coolant before steam is applied, or else the hammering that results when the steam collapses against the coolant can stress and crack the reactor. And, if the reactor has internal coils, separate top and bottom jackets, multiple coolant pumps, or multiple coolants, the picture becomes even more complicated.

This setup can be controlled easily with a single intelligent controller that includes a second algorithm — the process' control criteria — as shown in the figure on the next page. The reactor temperature acts as the input to a conventional PID control algorithm. The output, a value of 0–100% (of heating demand), acts as the input to the second algorithm, rather than being sent directly to an individual valve. Since the 0–100% value is a measure of the heating demand at any time, the second algorithm can coordinate all of the valves based on it.

This can be thought of as an enhanced split-range scheme. Aside from the fast and accurate temperature cycles that result, the beauty of this concept is that the operator has only one controller to deal with. In addition, whenever the controller mode is switched to manual, the output ramped up and the steam accordingly comes on, the second algorithm shuts down the coolant, drains the jacket, and checks any other interlocks.

The second algorithm

Remember that the objective is to determine the proper controller output (based on a 0–100% measure of heating demand) for the coolant valves, the drain valves, the steam valve, and the condensate valve. Because the thought process outlined here is very straightforward, additional equipment can readily be included in the control scheme; although the algorithm will get more complicated, approaching it one valve at a time will generally lead to success.

The next three sections of this article discuss the control criteria for each type of valve. Taken as a whole, these criteria represent the second algorithm.

Implementing these criteria requires no special functions, and can be done on nearly any intelligent controller having sufficient input/output and memory capacity. For example, just about any programmable logic controller that supports analog I/O can handle it, and just about every manufacturer of distributed systems provides a single-loop controller with built-in computing power that will handle this application nicely.

Coolant valves

The coolant supply valve is generally a fail-open throttling valve; when the PID algorithm output is 100% (which typically corresponds to 20 mA and 15 psig of air), this valve is fully closed. The coolant return valve is generally a fail-open on/off valve; when the PID output for this valve (a discrete output) is on, the valve is closed.

The control criterion for the coolant return valve is very simple and depends only on the position of the supply valve: Whenever the PID output to the supply valve is less than 100%, the return valve

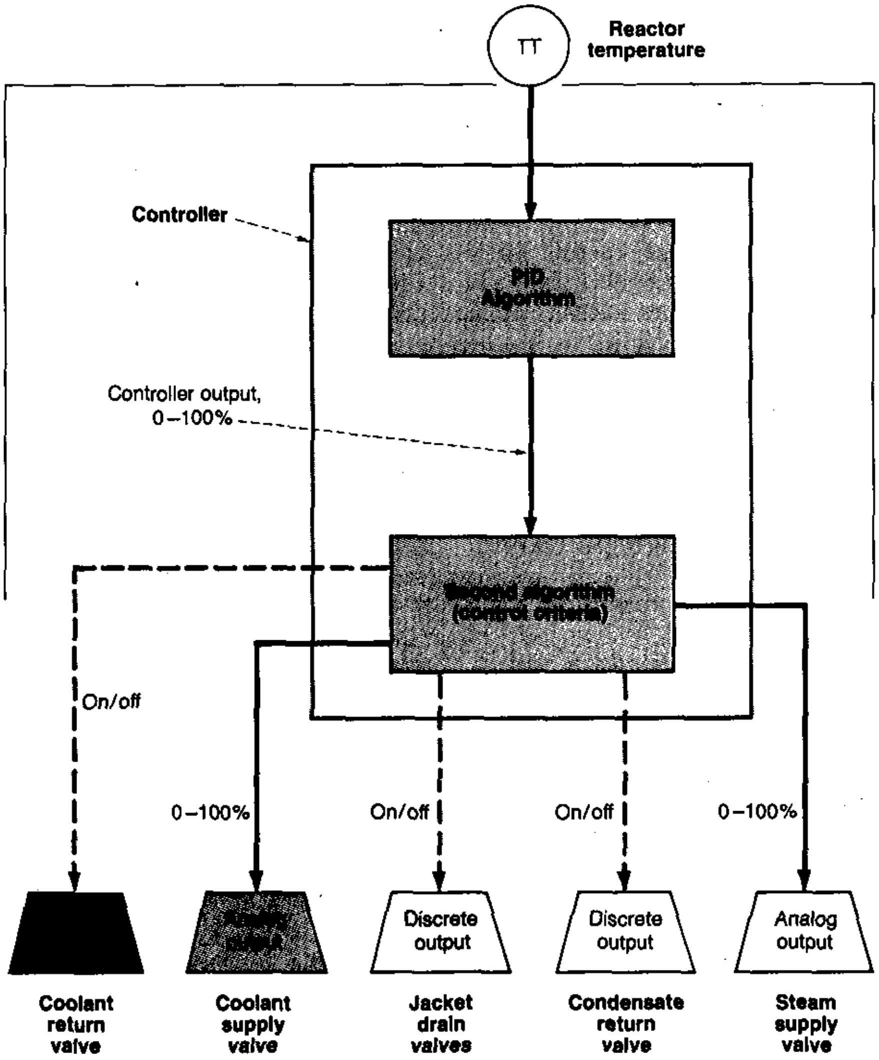


Figure — Control concept simplifies temperature control

must be open to accommodate the flow.

The control criteria for the coolant supply valve are determined by the output from the PID algorithm in a split-range fashion. When the output is 0%, meaning that maximum cooling is required, the output to the coolant valve should also be 0%, for fully open. When the output is 50%, meaning that neither cooling nor heating is required, the output to the valve should be 100%, for fully closed. (Thus, the coolant valve output is twice the PID algorithm output, with an upper limit of 100%.)

This leaves the upper range of PID output (51–100%) for the steam valve, as will be discussed later.

Jacket drain valves

The jacket must always be drained before steam is applied, which involves opening both drain valves while all other valves remain closed. The jacket can be drained for a set amount of time as indicated by a timer, or a flow switch could be installed in the drain pipe to indicate when draining is complete.

The drain valves should be opened whenever the PID output is greater than 50% (i.e., if steam is required), if the timer hasn't expired. The timer starts as soon as the drain valves are opened, and gets reset when the PID output goes below 50%, or when the coolant flow is greater than zero (or whenever something else happens that is a reliable indication that coolant has entered the jacket).

Rarely, if ever, is it necessary to open one of the drain valves and not the other, so a single output is all that is usually required for both of these valves.

Steam valves

Control of the steam valve is also quite simple. It is normally handled in the conventional split-range manner, using the upper half of the PID output range. That is, the output to the steam valve is twice the controller output minus 100%, with a practical lower limit of 0%. Of course, if the jacket drain timer has not expired, the steam output must be forced to 0%. The steam valve logic will also normally incorporate several other interlocks.

The condensate return valve, like the coolant return valve, may be controlled a number of different ways, based on the output to the supply valve. For example: When the steam supply output is greater than 0%, the return valve should open.

Tuning

Unfortunately, this method of control does not solve any of the inherent problems of nonlinear, exothermic temperature control. But it does not prevent the use of any of the traditional remedies, such as adaptive gain, bidirectional gain, or anti-reset windup, since these are all modifications to the PID algorithm.

At first glance, it may appear that draining the jacket of coolant when the controller is asking for steam might result in controller overshoot. This turns out not to be a problem for two reasons: First, the time it takes to drain the jacket, compared with the speed of most reactor temperature loops, is usually insignificant. Second, depending on the jacket volume, draining the jacket can indeed have a noticeable heating effect as the water (which acts as a heat sink) is removed.

In many processes, there is no point in throttling the steam: Either the steam is used only to initiate the reaction, which requires bringing the reactor to a certain temperature as fast as possible, or the heating rate becomes limited by the condensation rate as soon as the steam valve is even minimally open. In these situations, the steam supply valve is on/off actuated based on a controller output of 100%, the coolant is ranged over the full 100% span, and the jacket is drained at 100% controller output.

The more a process reflects one of these two conditions, the higher the coolant/steam split-range breakpoint should be. For example, temperature performance may be best when the coolant range is 0%–75% and the steam range is 75%–100%. To facilitate tuning of this breakpoint, it should be defined as a common variable when programming the algorithm.

The setpoint profile

Microprocessor-based control strategies like this one generally include a selectable temperature-setpoint profile that is based on product grade. In other words, the operator can push a button or select from the keyboard the desired temperature profile. A setpoint-generator algorithm then continuously calculates the setpoint as a function of time and feeds it to the controller. The controller, then, controls to this setpoint.

Setpoint profiles almost always consist of straight-line segments (either horizontal or sloped), and they are programmed as discontinuous functions.

Of more concern here is determining when the batch is done. Ideally, when the setpoint profile is complete, so is the batch. However, even if the setpoint generator functions perfectly, there usually are several other variables in the overall length of the cycle and in the degree of control. Varying steam pressure and coolant temperature, for example, directly affects heatup time, overshoot, and cooldown time. Just because the setpoint profile is complete, do not assume the batch is done.

The solution to this problem is handled by the time/temperature integral calculation — i.e., determining when the batch is done, based on the reaction temperature and the duration at each temperature. When the integral reaches a certain value, the setpoint is automatically changed to the final cooldown setting. A reaction-rate equation is often available from the original process design, and the coefficients can be fine-tuned based on product-quality data of ongoing operations.

Interlocks

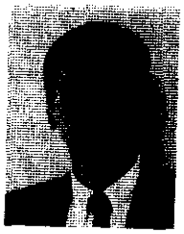
For failsafe reasons, the coolant valves are generally fail-open with no interlocks, the steam valves are fail-closed with several interlocks, and the jacket drain valves are fail-closed with some interlocks. The definition and implementation of interlocks are, however, topics too broad to be covered here.

In the case at hand, the interlocks could be implemented right within the second algorithm, or the the outputs from the algorigthm could actually be internal, to be further acted upon by a separate interlock program.

Cynthia Fabian Mascone, Editor

The author

Allan G. Kern is a sales engineer for Rust Instrumentation and Controls Co. (8070 South 1300 West, West Jordan, UT 84084; tel. 801-566-7878). Previously, he was a systems engineer with Fisher Controls, a process engineer with Hercules Specialty Chemical Co., and a software engineer for Eaton-Kenway. He holds a B.S. in chemical engineering from the U. of Wyoming, and is a member of the Instrument Soc. of America.



MAKING SMALL-SCALE ADDITIONS

Here is one scheme to control adding small amounts of liquids, and another for small volumes of gases. Both are alternatives to more conventional, costly metering techniques.

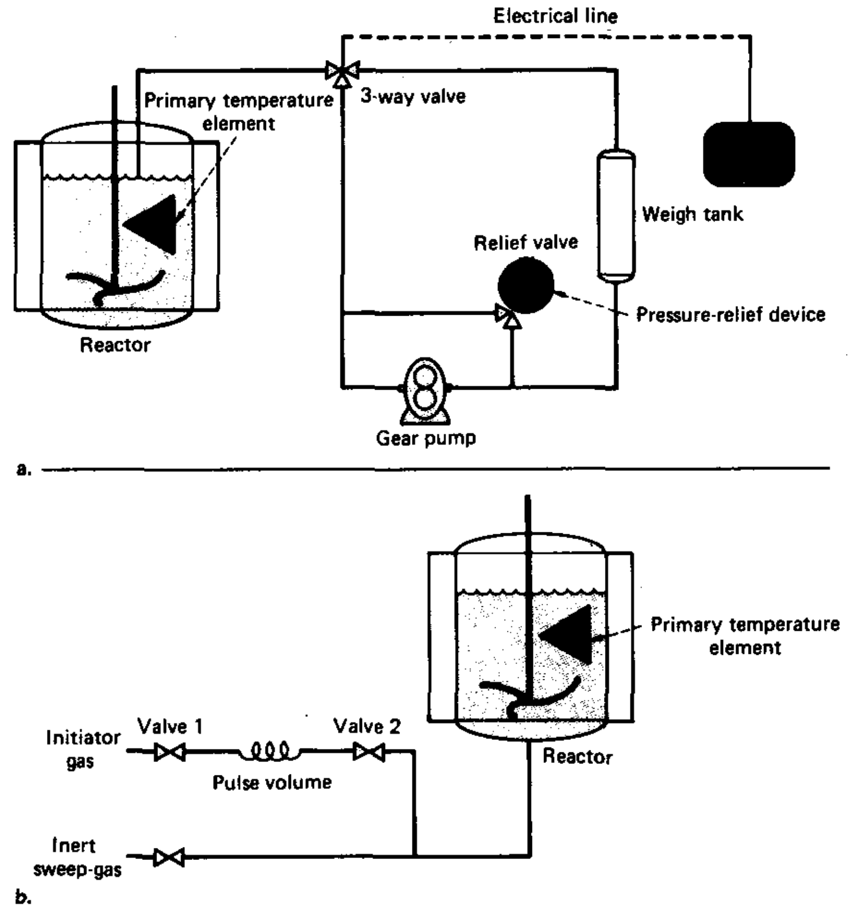


FIGURE a. A three-way valve accurately and inexpensively controls the addition of small amounts of liquid to the reactor
FIGURE b. Controlling the number of pulses of a gas is a means to precisely measure flowrate

1. METER SMALL AMOUNTS OF PROBLEM LIQUIDS

James R. Birch*

This is a cheaper, and just as reliable, alternative to using devices such as diaphragm pumps. In pilot plants, often, liquid streams are continuously added to reactors at low flowrates. Normally, a piston or diaphragm pump would be used.

However, if the stream is one that is corrosive, or a slurry, or one that contains a dispersed gas, pumps that rely on check valves for their operation, such as those mentioned, usually cannot be used.

Fig. a shows the solution. The tank contains the problem liquid and is mounted on a weight-measuring device. The recirculation pump is preferably a gear type, since the output of such a unit is fairly insensitive to the discharge pressure. The pump is sized to provide a turnover ratio that will ensure good tank-mixing.

The three-way valve intermittently diverts flow from the recirculation loop to the reactor. The timer controls the duration and frequency of the valve diversion, which sets the overall addition rate.

Example

A simple example illustrates the setup: An engineer needs to add 3 gal of an organic liquid to a 10-gal reactor over 90 min (0.0333 gpm). The organic is saturated with hydrochloric acid and contains a significant amount of a partially dissolved Lewis-acid catalyst [chemistry buffs will remember that, in 1923, G. N. Lewis introduced the useful concept that any acceptor of an electron pair is an acid, and an electron donor is a base].

The contents of the addition tank (5 gal) is recirculated by a small gear pump at 1.5 gpm to maintain the suspension. To achieve the proper addition rate, the three-way valve forwards solution to the reactor about 2% of the time (0.0333 x 100/1.5 = 2.222). Therefore, the timer directs the valve forward for about 2 s and recirculates the solution for about 98 s. Based on observed weight loss in the weigh tank, minor adjustments are made to the timer by either a process controller or an operator.

While formulated for problem liquids, this metering technique can be used for any type of liquid. Greater system flexibility is possible via the use of a variable-speed drive. The cost of such a setup is often less than that of one employing small metering pumps for adding liquids.

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*Agricultural Products & Processes Research., Michigan Div., Dow Chemical U.S.A., 969 Building, Midland, MI. 48667.

2. MEASURE SMALL GAS-FLOWS

Now, we shall present an inexpensive means to meter small flows of gas. Accurately controlling and measuring such flowrates is often difficult. Control valves with small trims coupled to differential-pressure transmitters having small orifices (or long capillaries) are prone to plugging and calibration troubles. On the other hand, most mass flowmeters are expensive for small-scale uses.

A simple, but often overlooked, control alternative is the gas-phase analog of the piston pump. In this scheme, the gas is added as a series of small gas-pulses of known volume and pressure. The pulse frequency sets the addition rate; applying an equation-of-state to the volume of each pulse determines the mass flowrate. A pulse counter totals the gas added. For large changes in flowrate, the pulse volume or pressure can be changed quickly and easily.

Example

An engineer wants to add 0.5 cm^3/min of a gaseous initiator to a 50-gal pilot-plant reactor. The reaction is highly exothermic, so the gas flow is controlled by the reactor temperature (see Fig. b). With the temperature below the setpoint, a controller opens Valve 1 to fill the injection volume at room temperature and 30 psig. After a few seconds, Valve 1 closes and Valve 2 opens, injecting the volume into the flowing inert-gas stream at near-atmospheric pressure. After a few seconds, Valve 2 closes, and waits for the next cycle.

Almost any on/off-mode timer or controller is suitable for regulating Valve 1. In this case, a process-control computer looks at the reactor temperature and, based on the controller error, adjusts the pulse frequency to maintain the setpoint. The loop volume is 0.12 cc, and about four pulses per minute are added to the reactor.

An alternative for applications where pressure differences cannot be used is to employ a combination of valves or a multiport valve to direct the sweep gas into the injection coil, similar to operating a chromatography injection port.

The method presented here is also used in commercial reactors where poor flow-control can be serious.

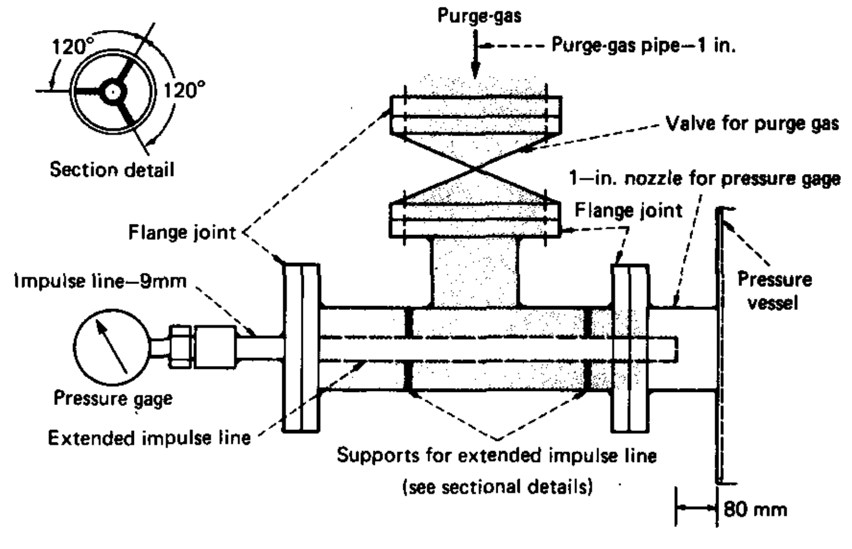


FIGURE. Modified tap is simple to build and guarantees that pressure readings are accurate

TRY THIS SIMPLE PRESSURE TAP TRICK

Here's an accurate way to measure pressure when using taps with gas purges.

K. Chauhan*

If you are using a purge-gas-type pressure tap, you may be taking inaccurate readings or choking the device. Such problems can be eliminated by simply extending the pressure tubing nearer to the vessel wall.

Choking-type gases (those that plug or block orifices due to dust deposition, solidification, or precipitation of process gases or vapors) are normally followed by a gas purge — either nitrogen or a reactant gas, such as shown in the figure. Generally, the purge gas is throttled to a minimum flowrate when a pressure reading is taken to avoid an erroneously high pressure. However, if the purge gas is reduced too much, the pressure tap may well choke.

The dashed lines show a solution to the problem: Extend the pressure tubing to about 80 mm from the wall. The purge gas need not be throttled, since the process gas is close to the tubing and the purge-gas backpressure will not affect the pressure sensing. Still, the pressure-sensing point is continuously flushed by the purge gas. Supports are needed to hold the tubing in the center of the pipe.

This setup has been working well for one of our reactors. The temperature is about 400°C, and ammonia, which is one of the process components, serves as the purge gas as well.

*Gujarat State Fertilizers Co.

Maintain maximum liquid level in pressurized vessels

*N. J. Ricord**

☐ In pressurized vessels where feed liquor is pumped into the top and out of the bottom, venting gases to maximize the liquid level in the tanks may present a problem. Differential pressure cells and nuclear level detectors are too sophisticated for this application.

A simple solution is to use a dual-conductivity-probe level control to activate an automatic valve for venting the gases from the vessel. And even a severe scaling problem does not affect this device, as it would the differential pressure cell and the nuclear level detectors.

Manual venting of digester vessels to maximize the residence time of a slurry solution was automated using a dual-conductivity-probe relay system.

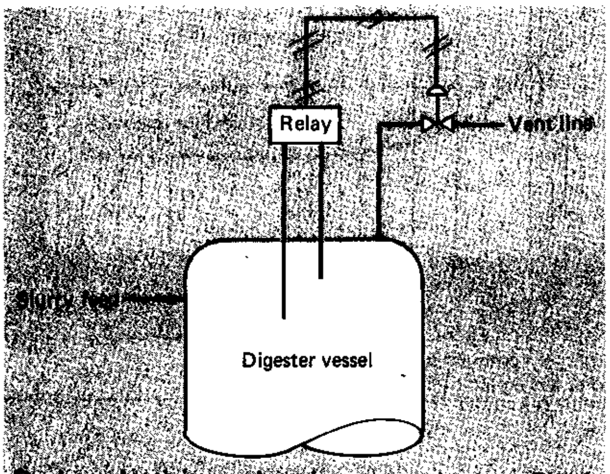
A slurry solution is fed into the top and out of the bottom of each digester vessel in a train. As the slurry flows through the vessels, the resultant pressure drop causes some flashing of the liquid and in-gassing of the digester vessels, which displaces some of the liquid in the vessel and reduces the slurry's residence time.

The manual venting system consisted of having an operator open the vent valve and listen for a noise change. Usually liquor flowed from the vent before the operator knew the vessel was completely vented. This resulted in plugged vent lines.

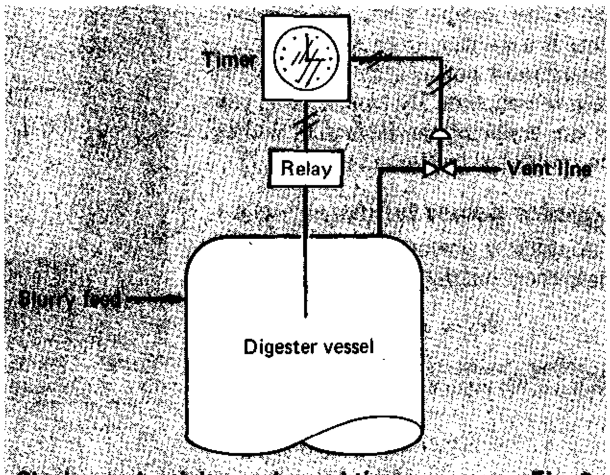
Fig. 1 illustrates the automated venting system. The dual-probe unit has a long probe and a short one. When the long probe is not covered by slurry, the relay opens an automatic control valve, which vents the tank. When slurry contacts the short probe, the valve closes and the venting is completed. The length differential between the probes determines the venting interval.

A variation of the same idea uses a single probe and a timer (Fig. 2). The timer is activated when the slurry level falls, discontinuing contact between liquid and probe. After a preset time interval, the system opens the vent control valve. When the slurry again contacts the probe, the valve closes and the timer is reset.

*Ormet Corp., P.O. Box 15, Burnside, LA 77038.



Dual-conductivity-probe relay system Fig. 1



Single conductivity probe and timer Fig. 2

Rudolph E. Lisa*

DETERMINE VISCOUS-REACTION ENDPOINT EASILY

A simple method based on power draw solves a sticky problem: Telling when the reaction is over.

This approach uses power as a function of molecular weight to locate the endpoint. Many viscous polymerization reactions do not lend themselves to either online endpoint determinations or rapid laboratory analyses.

Often, such reactions occur at high temperatures, and material may freeze up in a normal process viscometer during runs or between batches.

Also, sampling and analysis may result in certain problems: production delays due to laboratory analysis, inaccurate determinations if the reaction continues, thermal hazards due to taking high-temperature samples, air leakage at the sample port, etc. However, there is a solution to these problems — the agitator power-draw is directly related to endpoint. By plotting this power, the endpoint can be found, without the problems mentioned above.

Agitator power fundamentals

The agitator power draw is related to the critical mixing parameters by:

$$N_p = K\,(N_{Re})^m \qquad (1)$$

where the dimensionless power and Reynolds numbers are defined by:

$$N_p = Pg_c/\rho N^3 D^5 \qquad (2)$$

$$N_{Re} = D^2 N\rho/\mu \qquad (3)$$

The relationship between the power and Reynolds numbers is represented

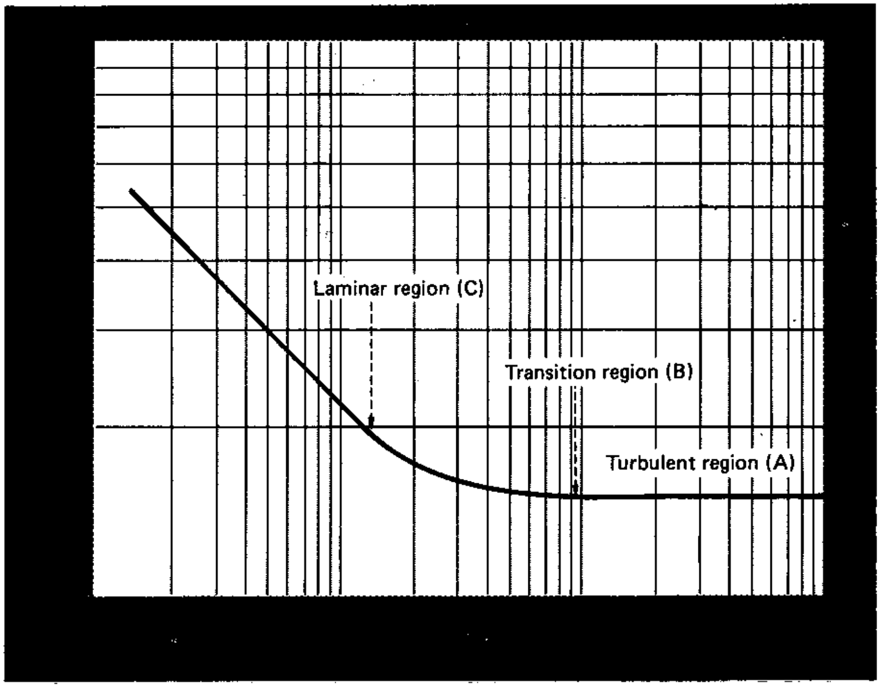


Figure — The higher values of the molecular weight of viscous materials correspond to the laminar mixing region

*BASF Corp., Chemicals Div., Chemical Engineering Dept., 1419 Biddle, Wyandotte, MI 48192-3736.

by a log-log plot [1]. A typical agitator mixing curve is shown in the figure. In the flat region (A), mixing is turbulent, as in the start of polymerization. The viscosity is low and the Reynolds number large. The motor consumes a relatively low percentage of its rated horsepower.

As viscosity begins to build, the Reynolds number decreases and the mixing condition in the tank moves to the left on the figure, toward the viscous regime. As long as the Reynolds number remains in the flat region of the curve, the power number, and hence the power draw, both remain constant if the material's density does not change appreciably.

As viscosity builds further, a slight power increase will be noticed as the reaction moves into the transition region (B) where the mixing curve begins to bend upward. At a certain point, viscosity begins to build rapidly, and the reaction enters the viscous or laminar region (C). A rapid increase in power draw will be noted on the wattmeter if the motor is not greatly oversized. (A power draw of 35–50% of rated capacity at endpoint is usually sufficient.)

This power will be uniquely determined by viscosity, and the reaction can be stopped at a given degree of polymerization (viscosity), once the desired power-draw/viscosity correlation is known.

The mixing curve can be obtained for any reactor/agitator system. Equipment vendors usually have a curve for a particular system, provided the agitator blades, baffling and other internals have not been altered. An alternative is to build a geometrically similar model in the laboratory and determine the mixing curve with a mixing kit [2,3].

Wattmeter vs. ammeter

Engineers assume that an ammeter will provide an accurate measure of power draw. This is true if the motor is near full load. Since current and voltage are sinusoidal, the power draw is the product of the instantaneous current and voltage times the cosine of the phase angle between them:

$$P = V_i I_i \cos\phi \qquad (4)$$

Current and voltage may be at maximum values, but be out of phase, drawing relatively less power. As load increases, the phase angle ϕ approaches 0 and $\cos\phi$ moves toward its maximum of 1, drawing maximum power.

Nomenclature

D	Agitator dia., ft
g_c	Gravitational constant, lb-ft/(lb$_f$)(s^2)
I_i	Instantaneous current, amps
K	Constant
m	Exponent, variable
N	Agitator speed, rev/s
N_p	Power number, dimensionless
N_{Re}	Reynolds number, dimensionless
P	Power, ft-lb$_f$/s
V_i	Instantaneous voltage, volts
Greek letters	
μ	Viscosity, cP
ρ	Density, lb/ft^3
ϕ	Phase angle, deg.

Table — By measuring the amount of power drawn by the mixer, the molecular weight of the material can be found

Molecular weight	Power draw, hp
170	7.3
3,000	7.3
11,000	8.1
27,000	10.2
53,000	14.8
85,000	15.9
150,000	16.8
250,000	19.0

Notes: For three sets of agitator blades, for a 350-gal reactor.

Thus, sensitivity can often be lost with an ammeter that measures only current, depending on how close the motor is to full rated capacity at endpoint and how precise the endpoint must be.

A wattmeter, however, automatically takes the phase angle into account and computes power directly. This provides an accurate measurement. Wattmeters are relatively inexpensive and can be installed by any electrician. Several measurement ranges (numbers of coils) can be built into one unit so that the same wattmeter can be used on pilot and plant-scale equipment.

Example

Given: A 350-gal batch polymer reactor has a desired molecular weight endpoint of 150,000. Mixing parameters: $D = 1.625$ ft., $N = 190$ rpm (or 3.17 rev/s), $\rho = 68.6$ lb/ft^3, and starting viscosity is 100 cP or 0.0672 lb/(ft)(s). The reactor has three independent sets of mixing blades.

Find: The endpoint of the reaction.

Solution: A wattmeter is used to correlate power draw to molecular weight (see the table) and the reading is then used to reliably determine the batch endpoint. The reactor's mixing curve (for one set of blades) is obtained from the vendor and is given in the figure. The initial Reynolds number:

$$N_{Re} = 1.625^2 \times 3.17 \times 68.6/0.0672 = 8{,}545$$

This is in the flat (turbulent) region of the curve. At the endpoint, the power draw is used to determine apparent viscosity.

Power (from the table) = 16.8 hp × 550 ft-lb/(s)(hp) = 9,240 ft-lb/s for three sets of blades, or 3,080 per set of blades:

$$N_p = 3{,}080 \times 32.2/68.6 \times 3.17^3 \times 1.625^5 = 4.01$$

From the figure, $N_{Re} = 26$. The apparent viscosity is then:

$$\mu = D^2 N\rho/N_{Re} = 1.625^2 \times 3.17 \times 68.5/26 = 22.1 \text{ lb/(ft)(s), or } 32{,}900 \text{ cP}$$

An existing 10,000-gal reactor can be evaluated for scaleup. By the above method, the turbulent Reynolds number for the proposed vessel can be found. Using the pilot-scale apparent viscosity, the laminar Reynolds number at endpoint can be set, and with the use of the reactor's mixing curve, the new power number and agitator power draw can be calculated.

Such a determination shows that, at the endpoint, the motor will not overload, and that the mixing is well into the laminar regime and the power draw is measurable. The first large-scale batch is successfully stopped at the proper molecular weight as determined by a wattmeter.

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Onstream calibration of pressurized gas-flow controller

Dean P. Montgomery[*]

In pilot-plant operation, flow controllers are often used to maintain constant flow of pressurized gas. Unfortunately, these devices are subject to drift — due to orifice clogging, temperature changes and other causes — and thus often need to be recalibrated. The setup described here is used periodically (or on demand) to determine the actual flow of controlled, pressurized gas, and then to reset the controller to achieve the intended flow. Conventional hardware and instruments are used, and are coupled to a computer that may also perform nonrelated supervisory and background functions.

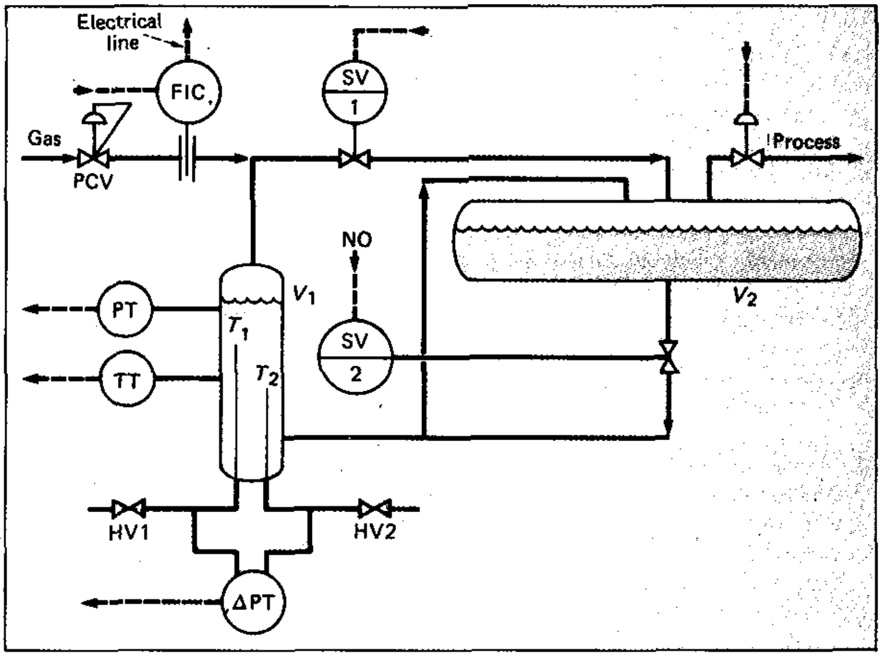

Figure 1 — Basic setup for calibrating gas flow-controller

The system

Fig. 1 shows how the apparatus works. The left-to-right flow of pressurized gas is controlled by the controller, computer and pressure-reducing regulator. The flow normally goes through SV1 and V_2.

Periodically or on demand, the computer simultaneously closes SV1 and SV2. Liquid then flows from V_1 to V_2, displacing gas from V_2 to the process. However, differential pressure of the liquid is not detected by the ΔPT until the falling liquid-level reaches the open end of tube T_1, inside of V_1. Thereafter, the differential pressure increases steadily until the liquid level reaches the open end of tube T_2, at which point the differential again becomes steady.

(The volume, V_o, between the ends of T_1 and T_2 was previously determined by draining the liquid to dryness through HV1 and HV2, and correcting for the internal volume of T_1.)

Constancy testing

Meanwhile, the computer periodically scans and stores ΔP values, and tests these for constancy. Elasped time from one constant ΔP value to the other is the time of flow for the gas volume, V_o. The temperature and pressure of V_1 are read and stored by the computer. After the constant maximum-ΔP value has been attained, the computer reopens SV1 and SV2, allowing liquid to return to V_1 and reestablishing the original gas-flow path. Vessel V_2 is larger than V_1, so that all liquid can be held in V_2 if there is a system failure.

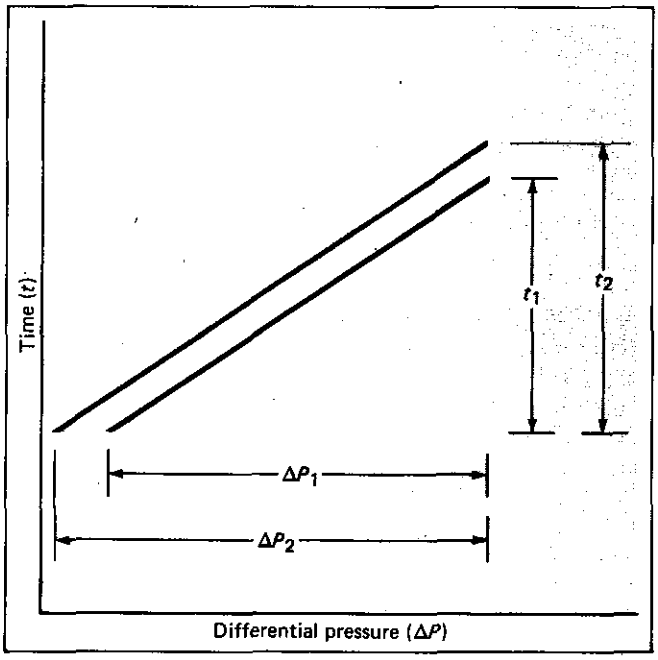

Figure 2 — The slope of displacement line is constant

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[*] Bartlesville, Okla. Deceased.

Nomenclature

FIC	Flow-indicating controller
HV	Hand-operated valve
NO	Normally open
PCV	Pressure-reducing regulator (self-contained)
PT	Pressure transmitter
SV	Solenoid valve
T	Tube
t	Time
TT	Temperature transmitter
V_o	Volume
V	Vessel
ΔP	Differential pressure
ΔPT	Differential-pressure transmitter

Determining true flow

Fig. 2 shows the displacement time for the known volume, V_o, of liquid and gas. The time, t_1, from the first non-constant ΔP value to the last non-constant one, ΔP_1, is accurately known from recorded data points. Therefore, the slope of the line of time vs. differential pressure is accurately known, as well. This slope multiplied by the difference in constant-value pressure-differentials, ΔP_2, is the total time, t_2, for the flow of V_o:

$$\text{Time} = (t_1/\Delta P_1) \times \Delta P_2 = \text{slope} \times \Delta P_2$$

Using the time, temperature, pressure, volume and nonideality correction (if needed), a background computer program calculates the true flow of gas, and changes the flow controller's setpoint.

It is useful to perform the flow check hourly if the deviation is less than 5%; otherwise, twice hourly.

There are several favorable features of this setup: Temperature changes that are great enough to affect the flow properties of gases do not significantly affect the volume, V_o. Density changes of the displacement liquid are irrelevant, since the differential-pressure transmitter serves as a timing device. The apparatus can easily be adjusted for variations in flow ranges by changing the lengths of T_1 and T_2, by altering the pressure in V_1, and by changing the computer's scan rate.

Desuperheater control-system cools gas

Ratan L. Agrawal*

A desuperheater control-system can be used to regulate the temperature in a centrifugal compressor. Such a system can be used to meet the maximum discharge-temperature of the compressor.

The system was designed for a process that recycles methyl chloride gas, but is applicable to many compounds. For the methyl chloride setup, the gas goes through drying, compression and other recovery operations (see Fig. 1).

There was a problem after the compressor started up. When a newly regenerated gas dryer was put on cycle after the system was running, the temperature at the dryer's outlet increased rapidly, which caused the compressor's second-stage suction and discharge temperatures to correspondingly increase. This resulted in alarm and tripout conditions in the compressor. There were unexpected shutdowns, production outages and poor overall compressor efficiency.

Design approach

To solve the problem (Fig. 2), variable amounts of liquid methyl chloride are injected as a fine spray, upstream of the compressor, to control the gas temperature. Liquid methyl chloride from a supply source enters the desuperheater at constant pressure, flows down the jacket and enters the main line through exposed, tangentially drilled ports.

The amount of liquid injected depends on the number and size of ports exposed. Control is accomplished by withdrawal of a stem through the control cylinder in response to a pneumatic control-signal. This signal reacts to a downstream temperature-sensing element.

The desuperheater is a standard configuration. Consult

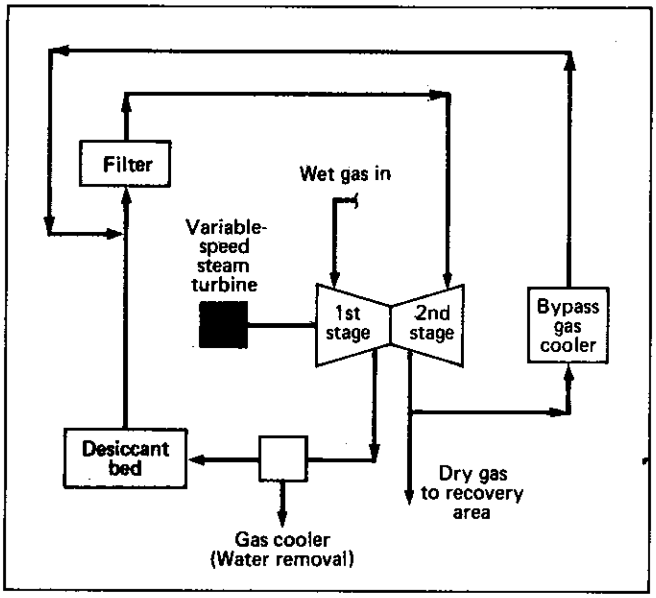

Figure 1 — A method was needed to keep temperatures in the second stage of the compressor from getting too high

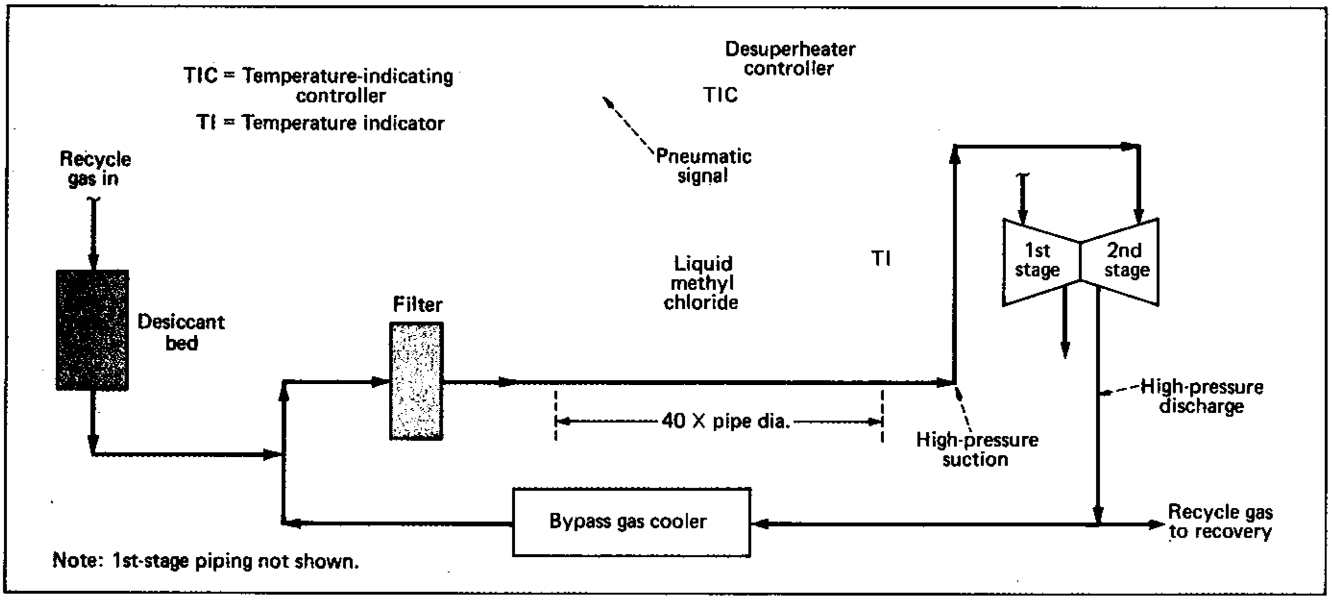

Figure 2 — Desuperheater injects variable amounts of liquid methyl chloride to control the gas temperature

*Polysar Ltd., 201 North Front St., Sarnia, Ont. N7T 7V1 Canada manufacturer's literature regarding the desuperheater's size and hole setup. The direction of the holes and their taper impart a high rotational velocity to the liquid. As the stem is moved, the holes become progressively larger.

The differential pressure (above 50 psi) between the liquid and the recycle gas makes it possible to inject the liquid as a finely atomized spray. To ensure complete vaporization of the liquid, the desuperheater controller must be installed at the equivalent of 40 pipe diameters ahead of the temperature-sensing element.

Performance data

The equipment's performance was evaluated during a hot summer month. After an initial debugging, the equipment performed within design limits. A performance evaluation appears in the table.

Table — Typical data for desuperheater control-evaluation reveal satisfactory performance, as seen from temperature ranges

	Compressor temperature			
Reading	**Suction**	**Discharge**	**Control valve opening, %***	**Remarks**
0	84.1	Normal	0	Normal condition
1	84.3	Normal	0	
2	84.7	Normal	0	Dryer swing
3	86.5	+1	13.5	Valve partially open
4	86.1	+2	49.9	Valve almost half open
5	93.5	+7	103.6	Valve fully open
6	94.4	+7	104.4	Valve fully open
7	95.0	+9	105.0	Maximum discharge temperature reached
8	94.5	+7	104.4	|
9	93.1	+5	102.9	Compressor discharge
10	90.8	+1	100.6	Steady temperature fall
11	88.0	Normal	100.0	↓
12	86.0	Normal	94.6	Normal condition

The readings were taken at 10-min. intervals. Ambient conditions: hot and humid, 91°F
Suction setpoint: 85°F

*Capacities were based on flowrate; actual capacity could exceed design values.

Prevent plug valves from sticking and jamming

Adnan Irhayem*

Plug valves frequently jam or become sticky, making it difficult for the inner plug to rotate, despite the use of a sealant. Possible reasons for this include:

- The sealant may not be suitable for hot service.
- The sealant or lubricant may not cover the entire moving surface.
- The lubricant may not be applied regularly.
- Wear of the moving plug enlarges the gap between the plug and the body, allowing fluid to seep along the plug.
- Frequent maintenance also enlarges the gap between the plug and the body.
- In foul atmospheres, deposits of dirt and corrosion products can cause the plug to jam.

In our facility, there were three 8-in. plug valves (Fig. 1) that operated pneumatically via the action of a piston on a toothed wheel. The valves became difficult to rotate, even after new pistons were installed, and had to undergo frequent maintenance (machining and lapping), which reduced the diameter of the tapered plug and enlarged the gap between the plug and the body. The plug would jam and stick, and eventually the problem became so frequent and severe that we decided to replace the plug valves with valves of a different type.

However, after studying the various forces that affect the movement of the plug, we determined that, due to seepage of gases along the plug, the space below the plug was constantly filling with gases. These gases were under pressure equal to the line pressure, and exerted an upward force on the base of the plug, causing the plug to wedge.

The problem was solved very simply by bleeding the trapped gases to the atmosphere, as shown in Fig. 2. The plug was then able to rotate freely, the valves did not need to be replaced immediately, and maintenance was reduced.

*Daura Refinery, Baghdad, Iraq.

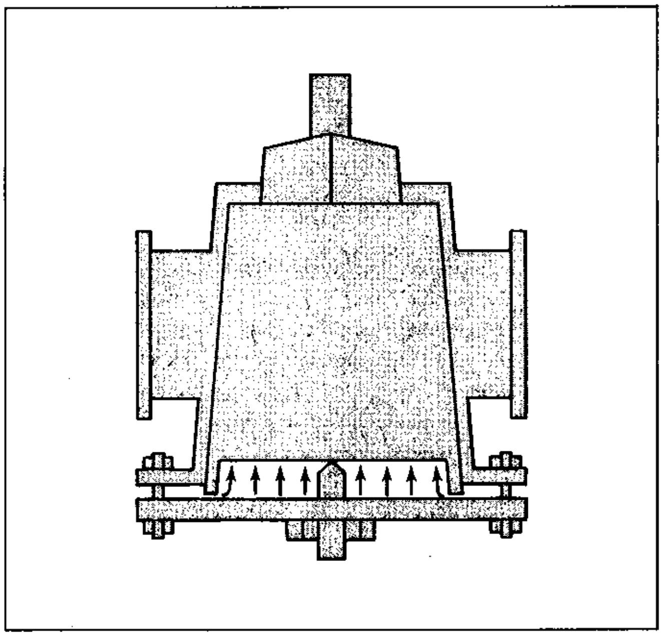

Figure 1 — Trapped gases exert upward pressure and cause plug to jam

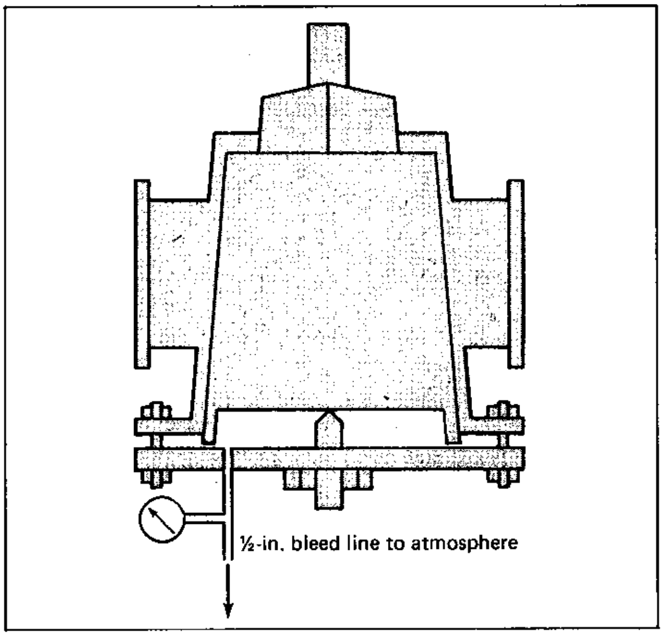

Figure 2 — Bleeding trapped gases to atmosphere dislodges plug, making rotation easier

OTHER ENGINEERING TECHNOLOGY

REDUCING EQUIPMENT-CLEANING WASTES

Carl H. Fromm and **Srinivas Budaraju**, Jacobs Engineering Group, Inc.,
Hazardous and Toxic Materials Div.

These strategies will minimize wastes and save a plant money on maintenance and waste disposal.

TABLE I. TYPICAL ORIGINS OF DEPOSIT FORMATION IN PROCESS EQUIPMENT

Origin	Description
Crystallization	Major problem in evaporators and crystallizers; *very frequent in food processing*
Sedimentation	Major problem in petroleum-refinery crude unit desalters and oil storage tanks; also present in cooling-tower basins
Chemical reaction or polymerization	Buildup on internal reactor surfaces, and in crude-oil storage tanks, is often encountered
High-temperature coking	Carbonaceous material is deposited on walls of furnace tubes
Corrosion	Common problem in heat exchangers
Bacterial growth (biofouling)	Major problem on cooling-water side of heat exchangers in electric power production
Clingage	Residual coat of process liquid left after drainage; major problem in reactors and mixers in paint-manufacturing industry, and generally in all high-viscosity liquid-transfer operations

Because of the high costs of waste disposal and treatment, regulatory pressures, and concerns about legal liabilities, industries are scrutinizing their hazardous-waste-generation practices. This applies not only to process wastes, but also to wastes from the cleaning of process equipment.

A primary objective of these waste-minimization efforts has been to reduce the quantity and/or the toxicity of the waste. For equipment-cleaning residues, this can be done, as outlined here, by determining: Why is the deposit present and how was it formed? How can cleaning frequency be reduced? Which cleaning method and medium will generate the least amount of the least-toxic waste when cleaning is unavoidable?

Equipment cleaning is a maintenance function that is typically performed to:

- Restore or maintain the operating efficiency of equipment.
- Avoid or limit contamination of the product.
- Minimize corrosion and extend equipment life.
- Allow for inspection and repair of equipment.
- Improve the appearance of exterior surfaces.

Cleaning is needed because deposits form on the surfaces exposed to the process (Table I). Understanding how and why these deposits are formed, one can concentrate on reducing cleaning frequency, and reducing the quantity and toxicity of cleaning waste.

REDUCING CLEANING FREQUENCY

Generally, cleaning can be avoided, or its frequency reduced, by: inhibiting the rate of fouling or deposit formation, maximizing the dedication of equipment to a single formulation or function, scheduling production properly, and *avoiding unnecessary cleaning*. Of course, the process implications of these actions, whether favorable or unfavorable, must also be considered.

INHIBIT FOULING

It is particularly important to inhibit fouling in heat-transfer applications. This can be accomplished by:

Using smooth surfaces. Deposits, or their precursors, adhere less to smooth surfaces than to rough ones. In one case, switching to electropolished stainless-steel tubes in a black-liquor evaporator at a paper mill reduced the cleaning frequency from once a week to once a year. Smooth, nonstick surfaces such as Teflon are also good; all-Teflon heat exchangers are commercially available, as are units of Teflon-coated steel.

Lowering film temperatures and/or increasing turbulence. The rate of heat-exchanger fouling depends on fluid velocity and, quite often, on film temperature. Film temperature controls the speed of chemical reactions that yield deposit-forming compounds, whereas fluid velocity controls the shear rate at the fluid/deposit interface. Hence, lowering the temperature of the heating medium and increasing the fluid velocity can reduce the fouling rate. An economic analysis of the increased pumping costs and the decreased cost of cleaning (and other possible savings) is usually warranted.

A review of thermal and hydrodynamic aspects of heat-exchanger fouling appears in Refs. [*1,2*]. Minimization of fouling through operational and design factors is addressed in

Refs. [3,4]. Oversurfacing of heat exchangers in fouling service results not only in increased equipment costs but also in more cleaning waste, because operating velocities will be lower than the design velocity, and the reduced turbulence results in increased deposits (see Ref. [5]).

Choosing the proper exchanger. The type of exchanger influences cleaning frequency. For example, spiral-plate units are often used in fermentation plants, owing to the ease of solid resuspension and the devices' nonplugging characteristics. A rod-baffle design provides more-effective shellside turbulence at a lower pressure drop than a segmented-baffle unit, so the former is likely to have less fouling.

Controlling fouling precursors. Maintaining cooling-water quality in open circulating systems is important to minimizing water-side scaling. Hardness, pH, corrosivity and biofouling can be controlled through careful monitoring. Biocides must propagate the entire cooling-fluid path in order to be deposited and function at all locations in the exchanger; and acid-feed equipment that maintains pH in the 6–7 range must be reliable, or rapid scaling or corrosion occur. In general, removing fouling precursors by filtration, softening, desalting and other such treatments can inhibit particulate fouling, scaling or metal-catalyzed polymerization.

The need for slowing the rate of deposit formation is not limited to heat exchangers, but also is important for other types of equipment. For example, in storage tanks containing crude oil that is exposed to atmospheric oxygen for long periods of time, gums and resins can form on the walls; floating-roof tanks and inert-gas blanketing may reduce buildup. Agitation to keep solids in suspension reduces or eliminates the need to clean petroleum storage tanks.

DEDICATE EQUIPMENT

Dedicating equipment to one function or formulation also reduces cleaning frequency. Dedication can be maximized by converting from a batch to a continuous process, or by using the unit intermittently for just one product.

The choice of continuous or batch mode is governed by production volume, and tradeoffs between capital and operating costs. The batch process is better where production volumes are small and/or product diversity large.

Dedicating a piece of equipment to a single formulation in a batch process means that the equipment remains dormant between production runs. Cleaning after each run can be avoided if the materials left in the equipment do not deteriorate with time or corrode the internals. The cost penalties of underutilizing equipment must be outweighed by cleaning and waste-disposal costs incurred when the equipment is used for more than one formulation.

SCHEDULE PRODUCTION PROPERLY

This is another way to decrease cleaning frequency. Equipment-utilization strategies and the resulting production schedules should meet the desired production goals, with due consideration of such constraints as available equipment, cost of turnaround, labor availability, and storage, and at the same time minimizing cleaning frequency.

Common strategies include increasing the size of single-product campaigns (often resulting in an increased product inventory), and scheduling consecutive production runs to maximize compatibility between batches so as to avoid cleaning or reduce its frequency. (A review of developments in production scheduling is given in Ref. [6].)

AVOID UNNECESSARY CLEANING

This should be a goal of waste-minimization programs. At times, equipment is cleaned routinely, with little or no consideration of the reasons for it. In one case, a ball mill was used periodically to wet-grind a mixture of inorganic powders. The ball mill had corrosion-proof internals and was totally dedicated to one formulation. Yet, the ball mill was cleaned after each use; the reason: other, nondedicated, ball mills at the facility were cleaned after every use.

REDUCING QUANTITY AND TOXICITY

When cleaning is necessary, it should be done with minimum generation of waste. The main factors to consider are the choice of cleaning medium and cleaning technique, as well as of waste-disposal option (not discussed here.)

There are two types of cleaning: chemical and mechanical. Chemical cleaning is an effective technique that employs substances (see Table II) that chemically attack the deposits and render them either solvent- or water-soluble. The basic reactions are oxidation, reduction, chelation, or conversion of insoluble oxides into soluble salts. Cleaning formulations also include surfactants that lower the surface tension of the solution to allow for faster penetration and breakup of deposits.

Mechanical (or physical) cleaning breaks the adhesion of the deposit to a surface, by: (a) using mechanical devices, such as scrapers, squeegees, rags, drag lines, "pigs," or lances; (b) surface scouring, e.g., using high-velocity water jets (hydroblasting); or (c) application of surface vibration, e.g., by ultrasound. Mechanical and chemical cleaning are often combined — for example, when high-velocity jets spray a caustic solution.

CLEANING TECHNIQUES

There are six basic techniques for cleaning process equipment [7]:

Fill-and-empty. A vessel is isolated from other equipment and filled with a cleaning solution. The solution can be heated and agitated, and is drained after four to eight hours. Rinsewater, or a dilute alkali or acid solution, is then used to remove residual cleaning chemicals. The drained chemicals and rinse solutions are either reused, treated, recycled or landfilled. The fill-and-empty method uses large quantities of chemicals and requires substantial downtime. It is typically applicable to small vessels, tanks or heat exchangers.

Circulation. The vessel is filled with cleaning solution and allowed to stand for a short time, after which the solution is either agitated to increase internal circulation, or circulated through an auxiliary system (called a clean-in-place system, as discussed later). Fresh makeup solution can be pumped in if used solution is withdrawn. In boilers, nitrogen can provide agitation for more-effective scale removal.

Flow over. This technique involves spraying the cleaning solution onto the surface being cleaned. It is applicable to large tanks where cleaning by filling or recirculation would require excessive quantities of solution. Extra safety precautions are usually necessary.

Gas propulsion. Here, cleaning agents (which must not be overly corrosive at high temperatures) are propelled through the system by steam. This method is useful for pipelines.

Process "simulation." This technique is applied to equipment that is easily fouled and for which spare parallel units are provided. The fouled equipment is isolated, drained of process fluid, and filled with the cleaning solution; the process operation is then simulated, thereby cleaning the equipment. An example of this route is the removal of iron oxide and copper deposits from high-pressure steam generators, using ammoniated ethylenediaminetetraacetic acid (EDTA) solution.

Onstream cleaning. This is the most preferred method because it relies on process fluid to do the cleaning during normal operation, making it good for such equipment as reactor jackets and heat exchangers. In-service cleaning of large, circulating, cooling-water systems is often done by intermittently varying pH between acid and neutral.

Among the many mechanical devices that may be employed in onstream cleaning are: ram valves for cleaning plugged nozzles, and moveable heat-exchanger-tube inserts propelled by reversing process fluid. Fluidized beds of inert solids (e.g., sand) have also been found useful in heat-transfer applications with extreme fouling, such as heat recovery from geothermal brines; here, the solid particles constantly abrade the deposit away, maintaining high transfer rates.

SELECTING METHOD AND MEDIUM

The following guidelines (in addition, of course, to cost) apply to the selection of cleaning method and medium:

- Minimize the amount of cleaning solution used.
- Choose the medium that results in the least-toxic waste.
- Determine in advance how the cleaning waste will be disposed of.

Chemical cleaning can result in large quantities of hazardous wastes that must be treated prior to disposal. Since onsite facilities are not available in every case, mechanical cleaning and onstream cleaning are preferred. According to several cleaning contractors, the trend is toward hydroblasting and onstream cleaning, and this can be directly attributed to the rising costs of waste disposal and treatment.

When chemical cleaning is unavoidable, the least-toxic medium should be chosen. However, if the toxicity of the deposit is the controlling factor, the cleaning agent with the highest potential for recovery and reuse should be used, to minimize the amount of contaminated solution to be disposed of.

CLEAN-IN-PLACE SYSTEMS

In offline-cleaning applications, a clean-in-place (CIP) system is an attractive alternative to arrangements calling for multiple containers of solvent and waste, and/or manual labor. A CIP system includes tanks, heat exchangers, filters, pumps, piping and instrumentation permanently installed as auxiliary equipment. It circulates a controlled amount of cleaning solution through isolated process equipment, often using internal spray manifolds or liquid jet nozzles. CIP systems are popular in the food, pharmaceutical and paint industries (see Ref. [8]).

CIP systems generally reduce the amount of cleaning medium required. They are especially effective when coupled with high-velocity, automated, jet manifolds and staged countercurrent rinsing. An 80-90% reduction in aqueous waste was achieved by paint manufacturers after installing high-pressure spray nozzles for tank rinsing [9].

REUSE CLEANING SOLUTION

The reuse of cleaning solution is desirable, especially if the solution can be made a part of the product. For exam-

TABLE II. SOME CHEMICAL CLEANING COMPOUNDS AND THEIR USAGE

Cleaning compound	Chemical action	Usage	Remarks
Hydrochloric acid	Dissolves most water scales and corrosion products	On boilers, heat exchangers, pipelines, etc.	Corrosive to steel; temperature must be below 175°F
Nitric acid	Dissolves most water scales and corrosion products	On stainless steel and aluminum	Cannot be used on copper and ferrous alloys
Hydrofluoric acid	Dissolves silicate deposits	As an additive to HCl (as ammonium bifluoride)	
Sulfamic acid	Dissolves calcium salts	As an additive to HCl	
Citric acid	Dissolves iron oxides	Mainly to clean boilers; frequently with added ammonia and oxidizers	Not good for removing water scale
Caustic soda, soda ash	Dissolves oil and grease	To remove oil and grease before acid cleaning and to neutralize the acid after cleaning	
Ammonia	Forms soluble complexes with copper ions	To remove copper from large boilers	
Ethylenediamine-tetraacetic acid (EDTA)	Dissolves water scales at alkaline pHs	For cleaning water systems without shutdown	Expensive

THE COSTS OF EQUIPMENT CLEANING

The cost of cleaning is made up of direct and indirect expenses. Among the direct costs are: equipment disassembly/reassembly (in some cases); cleaning chemicals and supplies; waste treatment and disposal; cleaning labor and supervision; cleaning-equipment depreciation; and utilities. Among the indirect costs are: planning and scheduling; lost production; lost raw-material inventory; inspection and testing; and process-equipment deterioration.

Equipment cleaning is often done by outside contractors with specialized equipment. Frequently, those contractors assume responsibility for hauling away the waste and disposing of it properly. (Of course, this will be reflected in the cost for the service.)

Cleaning costs vary widely, depending on the medium, method and application. Typical relative costs for contract heat-exchanger-cleaning service:

Method	Relative cost
Hydroblasting	1
Rodding	4 to 5
Chemical cleaning:	
Onsite waste disposal	0.3 to 3
Offsite waste disposal	2 to 4

In most cases, cleaning costs are included with other maintenance expenditures. As a result, the actual costs associated with cleaning are not visible to plant management, possibly reducing its support for waste minimization. Often, however, when management learns of the true expenditures, actions to lower cleaning costs are quickly initiated.

ple, reactor cleanup waste was reduced by a firm that applied a two-step rinse to a batch kettle for phenolic-resin synthesis. The first rinse used a small amount of water and generated a concentrated stream that could be recycled to the process; the second, full-volume rinse generated wastewater containing far less toxic material than a previously used single-rinse method [*10*].

Other examples include reuse of rinsewater from latex-tank cleaning as part of latex formulation in the paint industry [*11*], and use of warm oil to flush deposits out of crude-oil storage tanks in an oil refinery, followed by solids separation in the slop oil system [*12*].

Another approach to reusing cleaning solution is a multistage countercurrent cleaning process. This type of system will produce a more-concentrated effluent than a single-stage unit, and requires less solution. In many cases, a more-concentrated waste stream is also better suited for onsite recycling.

Onsite recycling has lower costs and liabilities than offsite commercial recycling. For aqueous cleaning media, removal of solids by sedimentation or filtration is often sufficient. For most solvents, onsite recycling typically becomes economically justifiable at the level of only eight gallons of solvent waste generated per day [*13*].

REDUCE CLINGAGE

A related aspect of cleaning-waste reduction is the minimization of clingage, which is material left inside equipment after draining. In operations involving viscous fluids, such as paint manufacturing and resin compounding, clingage is an important concern, since it is expensive to dispose of, and it represents a loss of product or raw material.

To reduce clingage, consider the following measures, most of which are practiced throughout industry:

- Providing adequate drainage time.
- Choosing low-adherence surfaces.
- Using mechanical wall wipers.
- Using manual wipers or squeegees.
- Selecting square-cylindrical or spherical geometry to minimize the wetted surface area.
- Spinning the agitator after batch dumping, to reduce clingage on the blades.

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The authors

Carl H. Fromm is a chief engineer in the Hazardous and Toxic Material Div. of Jacobs Engineering Group, Inc. (251 S. Lake Ave., Pasadena, CA 91101-3063; tel. 818-449-2171), where he leads waste-minimiztion programs, and oversees the technical aspects of projects for the division. He holds a B.S. in chemical engineering from Wayne State U., is a registered chemical engineer in California, and is a member of AIChE, Sigma Xi, and the American Assn. for the Advancement of Science.

Srinivas "Bud" Budaraju is a chemical engineer, also in the Hazardous and Toxic Materials Div. of Jacobs Engineering Group in Pasadena. He holds an M.S. in chemical engineering from the University of Southern California and a B. Tech. from the Indian Institute of Technology, is a registered chemical engineer in California, and belongs to AIChE and the Institute of Electrical and Electronics Engineers.

The Pareto chart. . . Tool for problem solving

The Pareto principle says that most of a plant's problems stem from a few basic causes. The Pareto chart, based on this principle, is a tool for selecting the critical causes of problems — the ones that should be resolved first.

Davis M. Woodruff and Felix M. Phillips
Management Methods

One of the usual responsibilities of an engineer is problem solving, e.g., process troubleshooting. Using an established technique of problem analysis, you can become a better problem solver by knowing which problem to attack first. The Pareto chart is one of the most powerful tools in problem solving, yet is often overlooked in U.S. industry.

A Pareto chart (or diagram) arranges items in order of importance. Therefore, it helps us select the problems that need to be solved, in their relative order of importance. They may be viewed in terms of percentages, dollars, units, or any measure that is of value to you.

Background

The discovery of the Pareto principle has been credited to an Italian economist, Vilfredo Pareto. In the late 1800s he learned, through his studies at Lucerne University, that some things were not equally distributed. Pareto discovered that 80% of the wealth was owned by 20% of the people, and he also learned that 80%, or more, of crimes were committed by 20%, or less, of the people. Hence, the Pareto principle is often referred to as the 80/20 rule. Other research shows that many things will follow the Pareto principle:

- One planet, Jupiter, contains most of the orbiting mass of the solar system.
- A few of the biological species account for the majority of the animal population.
- Most of a company's inventory dollars are tied up in a small percentage of the total parts or items.
- The majority of defects in production result from a small percentage of the possible causes.

In one job-shop operation, we found that about 80% of the company's profits came from 20% of the jobs or contracts. By using this information, it was possible to design management systems and material flows around these few jobs that led to increased productivity and profit.

Use of Pareto charts in your job

How can this principle help you become a better problem solver? Well, it has been shown that, on many occasions, the 80/20 rule is also true in industry. Fig. 1 shows a Pareto chart of quality defects in one operation, and Fig. 2 shows a Pareto chart of downtime causes in a processing plant. One can see from these charts that about 80% of each total is covered by 20% of the possible causes. Thus, as many have said, the "vital few" can be separated from the "trivial many." The strength of the technique lies not in an exact 80/20 ratio but in the concept that only a small portion of the whole makes the largest contribution. By applying the Pareto principle in your operation, you can sort the major problems from the many minor ones. This will allow you to devote your time to the areas that can make the biggest impact.

The preceding examples depict the use of Pareto charts in quality and manufacturing. However, these charts are applicable to almost any job or operation. We have seen them used in manufacturing, quality control, inventory control, purchasing and personnel. Think about your personal finances for a minute. More than likely, at least 80% of your income is utilized for 20%, or less, of your budgeted items.

Constructing a Pareto chart

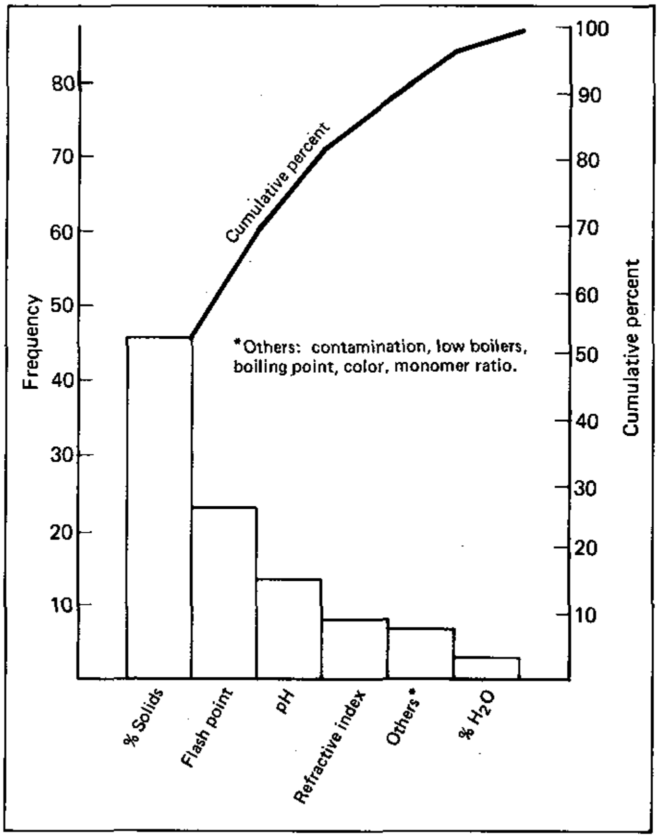
We will use an example to develop a step-by-step method for constructing a Pareto chart.

Example: You have just received the monthly quality report in a chart format, as shown in Table I.

By studying the chart, it is possible to determine the major problems, but remember, this is a highly simplified example. Usually, a monthly quality summary is more detailed and lists many more than ten reasons for product rejection.

We will now construct a Pareto chart, using six easy steps:

1. Label the chart with all the pertinent information of your process. For example, title, date, time period, department, plant operation, etc. In our example, the chart is labeled by using the data from the quality report.
2. Arrange your data by category in a chart or table (see table). This will make the calculation step easier. This step may be omitted, but be careful when you begin actually constructing the chart if you choose to go directly to calculating the percentages, etc.
3. Using the table from Step 2, calculate the percent of the total represented by each category, and the cumulative percentage as you proceed down the chart.
4. Determine the scales for the two vertical axes of the



*Others: contamination, low boilers, boiling point, color, monomer ratio.

Figure 1 — Pareto chart showing quality defects by percentage

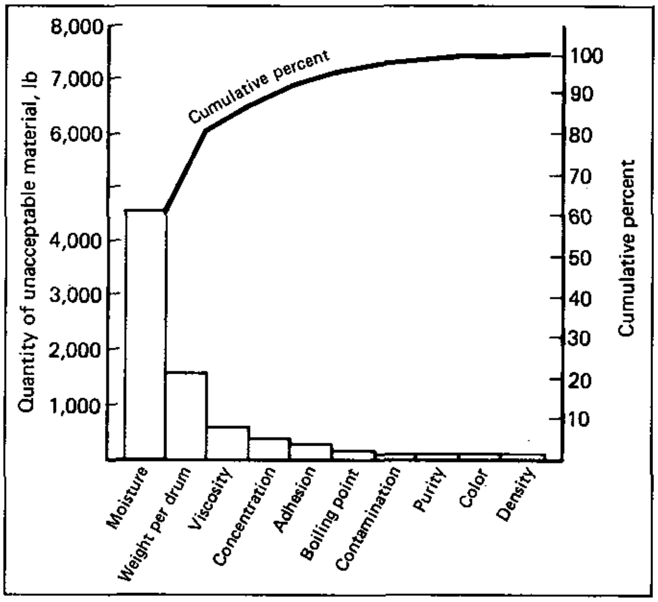
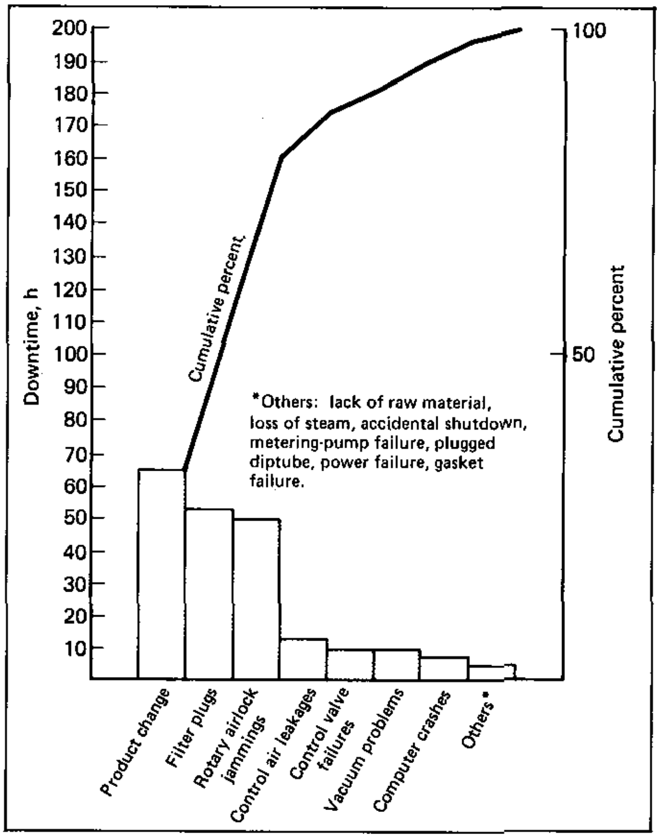


Figure 3 — Pareto chart of quality losses, by cause, for April 1985

chart. The left side is usually the percent in a particular category — frequency of occurrence, dollars, pounds, etc. It may be any units that are of benefit to you. In our example, we will use unacceptable pounds as the left vertical axis. The right vertical axis will be used for the cumulative percentages.



*Others: lack of raw material, loss of steam, accidental shutdown, metering-pump failure, plugged diptube, power failure, gasket failure.

Figure 2 — Pareto chart of quarterly downtime summary

5. Label the horizontal axis of your chart according to the categories in the table you generated in Step 2. These categories should begin with the largest on the left side of the chart and continue in descending order of importance. Draw in the bars of equal width, with the heights corresponding to the left vertical axis.

6. Plot the cumulative line across the top of the chart, using the vertical axis on the far right. You will notice a rapid ascent in the line, which will flatten as you approach 100%. This is telling you that you are reaching the "trivial many."

Interpretation

By arranging the categories in descending order of importance, we have made the interpretation easy. One can readily determine which areas to work in first. We have separated the "vital few" from the "trivial many." Be careful to avoid misinterpretation. In this example, the major reasons for quality losses are easily seen. However, these may not be the major sources of lost dollars due to poor quality. Another Pareto analysis could be done to establish those causes. Once the charts are completed, you can define and implement improvement efforts.

Summary

The Pareto chart is an established tool for problem analysis and problem solving. The 80/20 rule holds in chemical processing as well as in hard-goods operations. Using this simple tool can improve your problem-solving skills as an engineer or manager. After all, if we can become better

Table I — Example of monthly quality report in chart format

ABC Company
Monthly Quality Report
Polymer VR-999
April, 1985

Defect	Lb of defective material
1. Concentration	300
2. Color	75
3. Weight per drum	1,515
4. Density	75
5. Adhesion	225
6. Moisture	4,535
7. Contamination	80
8. Viscosity	530
9. Purity	75
10. Boiling point	150

Table II — Data from monthly quality report, arranged for plotting

ABC Company
Monthly Quality Report
Polymer VR-999
April, 1985

Defect	Quantity, lb	%	Cumulative %
Moisture	4,535	60	
Weight per drum	1,515	20	80
Viscosity	530	7	87
Concentration	300	4	91
Adhesion	225	3	94
Boiling point	150	2	96
Contamination	80	1	97
Purity	75	1	98
Color	75	1	99
Density	75	1	100
Total	7,560		

problem solvers, we become better engineers, and more valuable to our employers.

Use of Pareto charts on a continuing basis also provides documentation of improvements. The charts can become a record of the problem-solving efforts, as well as the basis of determining dollar value of savings, quality improvement, etc., that you have contributed to your company.

***Roy V. Hughson**, Editor*

The authors

Davis M. Woodruff is the owner of Management Methods, a management and consulting firm, P.O. Box 1484, Decatur, AL 35602. Tel. (205)355-3896. Previously he worked, for ten years, in various technical and management positions for 3M Company. He has developed and presented many training programs and seminars for industry. He holds a B.S. in chemical engineering from Auburn University, is a member of AIChE, ACS and the American Consultants League, and is a registered professional engineer in Alabama.

Felix M. Phillips is associated with Management Methods, serving as a consultant to industry, with emphasis on statistical process control. He has had over 12 years of experience in both technical and management assignments. He, too, holds a B.S. in chemical engineering from Auburn University and is a registered professional engineer.

Improve production efficiency via EVOLUTIONARY OPERATION

Evolutionary operation is a technique for fine-tuning a process by systematically varying process parameters. Here is how you can obtain valuable information on how to improve a product, while making it salable.

Warren P. Scarrah, Montana State University

Evolutionary operation (EVOP) is a simple technique that allows operators of industrial processes to continually improve efficiency during routine production. The basic philosophy behind EVOP is that "it is inefficient to run an industrial process in such a way that only a product is produced, and that a process should be operated so as to produce not only a product but also information on how to improve the product" [*1*].

It must be emphasized that EVOP is a routine method for permanent process operation, not an experimental procedure, and that it is used on the plant, rather than pilot, scale. It was developed to avoid undesirable characteristics of full-scale process experiments (e.g., short-term investigations that require special personnel and facilities, and usually make off-specification product). And, EVOP requires no special staff and can be done by the usual plant personnel after brief training.

The first steps in implementing EVOP are to identify the pertinent process variables, and then to design a cycle of process runs. One run usually consists of the normal values (or levels) of the variables, and the other runs involve levels of the variables that differ slightly from the normal values. These differences are deliberately kept small to avoid the production of off-specification product.

Fig. 1 illustrates two typical combinations of runs, with reference points in the center. The advantage of using the normal operating conditions as the central reference point is that one can easily estimate the profit or cost of each run relative to routine operations. Note that it is impractical to simultaneously investigate more than three variables in an industrial process using operating personnel.

The runs in each cycle usually satisfy a factorial design — this requires a run for every combination of the proces variables at their high and low levels (i.e., above and below the reference level). The factorial runs correspond to the corners of the sketches in Fig. 1 — four runs for two-variable investigations, eight runs for three-variable ones.

The cycle of runs is routinely repeated to provide information about the process characteristics in the immediate vicinity of normal operation; the repeated running through of a cycle is called a phase. After enough information has been collected, it is then possible to decide whether the process efficiency can be improved by selecting different levels of process variables for normal operation.

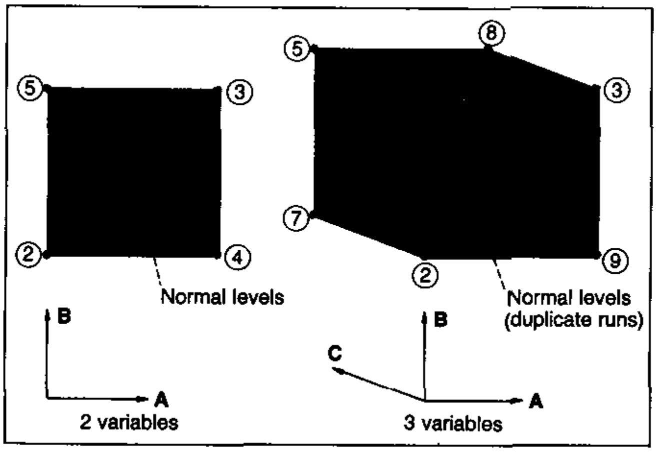


Figure 1 — Each cycle consists of a reference run (normal conditions), plus runs for which the variables' levels are slightly different

Subsequent phases could be used to eventually reach optimum conditions. Even then, it would be advantageous to continue EVOP, so that process variables could be adjusted to remedy the effects of such changes as deteriorating equipment and use of alternative raw materials.

An appealing feature of EVOP is its instinctive approach — after exploring the effects of process variables near the current operating conditions, adjustments can be made in the direction that improves the process efficiency.

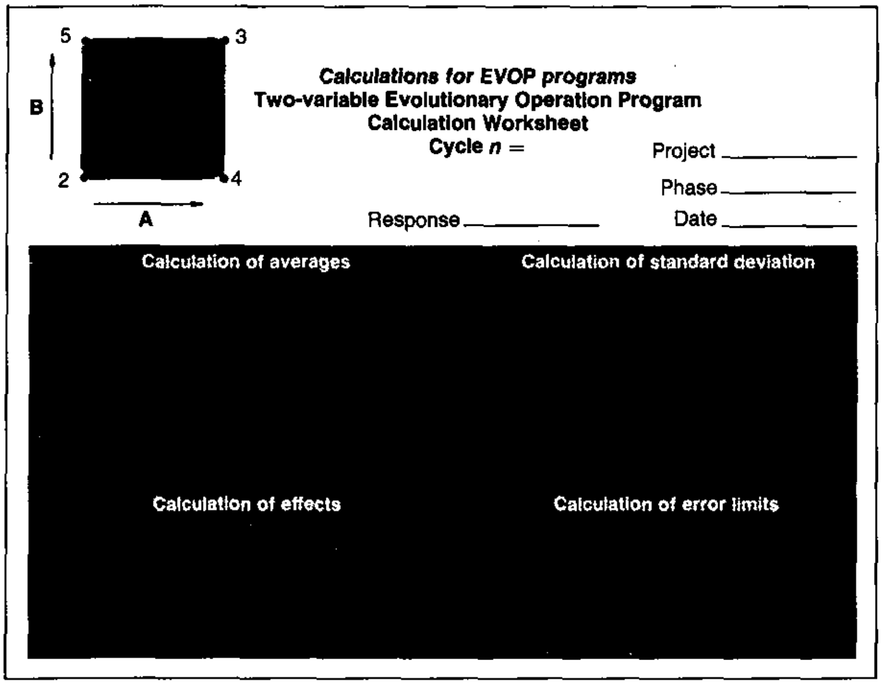
The EVOP philosophy is so natural that it can be (and undoubtedly has been) practiced without relying on statistics. However, the addition of a few simple statistical techniques can help avoid the economic penalties accompanying the production of off-specification product. By running many cycles in each phase, the small, virtually undetectable effects on the product in the individual runs can be evaluated, using statistics. Another advantage of statistical analysis is that interactions between variables are readily identified.

Another benefit of an EVOP program is an improved understanding of the process derived from scrutinizing it in preparation for introducing EVOP — potentially important variables must be identified and desirable goals established. In addition, the EVOP results should be studied at monthly intervals by a small group of technical people having a wide range of expertise outside of production, and not involved with the daily operation of the process. The broad perspective provided by these advisors supplements the obvious changes indicated by EVOP results; review and discussion can lead to suggested changes not initially identified as having potential benefits. This also allows production personnel to acquaint the members of the advisory committee with process differences peculiar to full-scale operation and to identify any annoying deficiencies built into the plant.

How to start an EVOP program

The first step is to determine whether the process is indeed suitable for optimization using EVOP. Characteristics that indicate a potentially favorable application are [2]:

- The process must be repetitive. Either the production run should last a reasonably long time, or it should be repeated frequently enough to justify the optimization efforts.



Calculations for EVOP programs
Two-variable Evolutionary Operation Program
Calculation Worksheet
Cycle n =

Project ____

Phase ____

Response ____ Date ____

Calculation of averages

Calculation of standard deviation

Calculation of effects

Calculation of error limits

Figure 2 — Worksheet [3] summarizes EVOP calculations

- The potential benefits from improving the process must be significant. This is not a serious constraint for any reasonably-sized operation, because seldom is a process being operated at optimum conditions.
- Due to the strategy of repeating cycles of runs with slightly different variable levels, the levels of the process variables must be easily changed between runs.
- To reduce the time required to obtain measurable results, the process must stabilize fairly rapidly during each EVOP run, and it must be possible to quickly measure the process response being improved.

Of course, as with any guidelines, a successful EVOP program can sometimes be initiated even though a process does not satisfy all of these criteria.

As the primary link between operating and technical groups, the process engineer is the natural person to initiate an EVOP program. His or her troubleshooting achievements in solving previous process problems lend the credibility demanded by operating personnel before they will consider any process modifications.

Because EVOP is a departure from normal procedures, the need for adequate preparation cannot be overemphasized. Preparation activities will convince the engineer of the genuine advantages of the technique, and develop the confidence required for the new role as local EVOP "expert."

There are three areas that must addressed during preparation: statistical concepts, calculations and strategy.

You need not spend much time attempting to master rigorous statistical theory. Because EVOP was developed to be used by nontechnical process operators, it can be applied without any knowledge of statistics. However, be prepared to address the operators' tendency to distrust decisions based on statistics. Concepts that you should understand quantitatively include: the difference between a population and a sample; the mean, variance, and standard deviation of a normal distribution; the estimation of the standard deviation from the range; standard errors; sequential significance tests; and variable effects and interactions for factorial designs having two and three variables. Illustrations of statistical concepts (e.g., a normal distribution) will be valuable tools.

You must also be thoroughly familiar with the EVOP calculations. Fortunately, worksheets have been developed that clearly show the purpose of the calculations [3]; a sample worksheet is shown in Fig. 2.

The pertinent information obtained from the worksheet is a summary of the effects on the desired response of each variable, the interaction effects between variables, and the standard errors for each of the effects. The importance of each variable is determined by comparing the magnitude of any effect in which it is involved to the corresponding standard error for that effect — a ratio of effect to error greater than unity indicates that the variable probably has a significant effect.

Your preparation should also include following a realistic simulation of a process through a complete phase of eight to ten cycles. "Realistic simulation" implies that measurements of the response to be optimized (e.g., yield, profit, etc.) include some error, or noise, typical of that caused by such factors as raw-material variations, equipment fluctuations and instrument deviations. Because of the conservative nature of EVOP, the response changes and the noise may occasionally have comparable values; however, as the number of cycles increases, the simulation will show that even though the best run may vary from cycle to cycle, the effects of the noise will eventually average out so that the true responses and variable effects can be determined.

At the completion of a phase, the worksheet calculations will clearly indicate the effects of the variables being investigated. However, the strategy for selecting subsequent phases involves engineering judgment. In fact, Ref. [*1*] points out that "... the actions to be taken in various circumstances are not precisely specified. Human judgment is a very important part of an EVOP investigation, as it should be in any investigation. Automatic rules which would in effect take the responsibility for the running of [the] process out of the hands of the process superintendent are bad both psychologically and practically."

Two means of developing a sound EVOP strategy are (1) to "learn as you go" during the initial application of EVOP to an industrial process, and (2) to practice the technique by optimizing a process simulation. Because of the conservative nature of EVOP, there is little risk in allowing a strategy to evolve while the efficiency of a process is being improved. The only obvious drawback to this commonly practiced approach is that, until the benefits of an EVOP program have been demonstrated, the conviction to ensure that the program will be completed is sometimes absent.

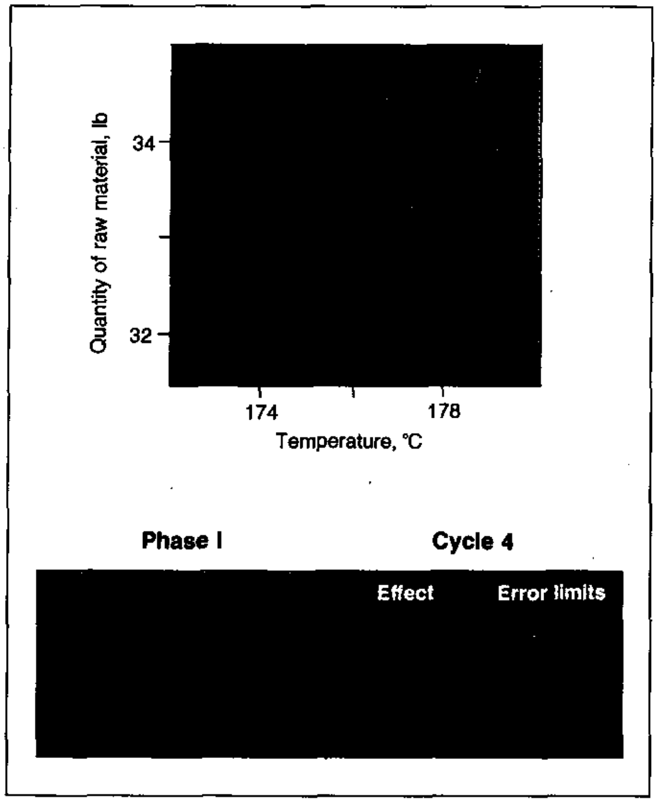


Figure 3 — Information board promotes employee involvement

Using a realistic simulation provides a chance to develop a viable EVOP strategy while simultaneously becoming convinced of its value. Enthusiasm can be generated by using a simulation that performs all the calculations for a complete EVOP phase — time is then available to evaluate the data and explore alternative sequences of phases until the process has been optimized. It is especially instructive to start from different initial values of the process variables; usually all the investigations will terminate at about the same optimum operating conditions. This emphasizes that EVOP tends to be self-correcting — if a poor decision is made in selecting the runs to be included in a phase, this will become obvious and can be corrected when choosing the next phase.

(To provide this type of exercise, the author has developed a simulation based on an industrial pharmaceutical process [*4*]; a complimentary program listing can be obtained by writing to him. — *Ed.*)

A bonus for the process engineer who successfully introduces an EVOP program is the satisfaction resulting from actually improving the process. It is a decidedly different experience from that of solving problems by troubleshooting, deemed to be successful when the process is returned to the previous status quo.

Communicating the results

Poor communication has been identified as a principal reason for EVOP not being more widely practiced [*2*]. An extreme example cited was the refusal of one plant manager to inform the personnel involved in an EVOP study of the results on the grounds of company security.

One approach to improving communication is to use an "information board" [*1*]. The board should be readily accessible to everyone involved in the operation of the process. To simplify interpretation of the data, it is important that information be presented graphically — sketches similar to Fig. 1 should have the mean responses for each run posted at the appropriate locations, as shown in Fig. 3.

Such a pictorial representation makes it easier to decide whether a variable has a positive, negative or neutral effect on the response. The adjacent tabulations of effects and error limits simplify the identification of important variables or variable interactions — i.e., an effect value greater than or similar to the corresponding error limit value is considered significant. The change in mean effect indicates whether the EVOP program has had a positive or negative effect on the response, relative to normal operating conditions.

Daily EVOP decisions should be based on contributions actively solicited from all sources having a vested interest in the operation of the process — production (supervisory and hourly personnel), process engineering, and quality control.

***Cynthia Fabian Mascone**, Editor*

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The author

Warren P. Scarrah is a professor of chemical engineering at Montana State University (Bozeman, MT 59717; tel. 406-994-5926). He holds B.S. and Ph.D. degrees from that school, and an M.S. from Purdue University, all in chemical engineering. He has had twelve years of experience in the food processing, chemical, nuclear and petroleum industries, and has spent thirteen years in academia, at MSU and the University of North Dakota. His research interests include supercritical-fluid extraction, renewable fuels from vegetable oils, and the use of computers to introduce realism into the classroom. Scarrah is a member of AIChE and the American Soc. for Engineering Education, and a registered professional engineer in Montana and North Dakota.

Directly determine reaction order

Sorab R. Vatcha*

A method is presented to directly determine the order of the kinetics of a reaction from the kinetic data, without trial and error. This method replaces the conventional one of integral analysis that generally requires trial and error, and data analysis by graphical or linear-regression methods.

The principle underlying the method is that the ratio of the times at which two specific conversions occur depends solely on the order of the reaction. The method is applicable to single irreversible reactions of any order—positive or negative, integral or fractional—as well as to reversible first-order reactions.

From the kinetic data of concentration vs. time, one computes the ratio of the times corresponding to conversions of X_A = ¾ and ½, or ⅔ and ⅓. Look up the nearest value listed in Table I, then read off the reaction order. Use the equations derived below for other conversion values and/or reaction orders not listed in Table I (this rarely need be done).

Derivation

The power-law kinetic model for a single irreversible reaction is:

$$\text{Rate} = -d[A]/dt = k[A]^n \tag{1}$$

The order and the rate constant are unknown. Integrating Eq. (1) with the initial condition:

$$[A] = [A]_o \text{ at } t = 0 \tag{2}$$

and solving for the residence time:

$$t = -(\ln(1 - X_A))/k \quad \text{for } n = 1 \tag{3}$$

$$t = \frac{(1 - X_A)^{1-n} - 1}{(n - 1)k[A]_o^{n-1}} \quad \text{for } n \neq 1 \tag{4}$$

where X_A is the fractional conversion of A in an irreversible reaction (Eq. 5); or the fractional approach to equilibrium of a reversible reaction (Eq. 6); or the progress of an irreversible or reversible reaction in terms of a physicochemical property that is a linear function of concentration, [*1*], e.g., electrical conductance (Eq. 7):

$$X_A = \frac{[A]_o - [A]}{[A]_o} \tag{5}$$

$$X_A = \frac{[A]_o - [A]}{[A]_o - [A]_\infty} \tag{6}$$

$$X_A = \frac{P_o - P}{P_o - P_\infty} \tag{7}$$

The ratio of the times at which two distinct conversions occur is given by:

$$\frac{t_{X_{A1}}}{t_{X_{A2}}} = \frac{\ln(1 - X_{A1})}{\ln(1 - X_{A2})} \quad \text{for } n = 1 \tag{8}$$

$$\frac{t_{X_{A1}}}{t_{X_{A2}}} = \frac{(1 - X_{A1})^{1-n} - 1}{(1 - X_{A2})^{1-n} - 1} \quad \text{for } n \neq 1 \tag{9}$$

(Continued on p. 196.)

Table I — Reaction order can be found from kinetic data of concentration as a function of time

Reaction order, n	$t_{3/4}/t_{1/2}$	$t_{2/3}/t_{1/3}$
−1	1.25	1.60
0	1.50	2
0.5	1.71	2.30
1	2	2.71
2	3	4
3	5	6.40

Eqs. (10, 11) and (12, 13) should be used for computing other values not listed above.

Nomenclature

$[A]$	Concentration of A, the limiting reactant
k	Rate constant in the power-law kinetic model
n	Reaction order in the power-law kinetic model
P	Physicochemical property that varies linearly with concentration
t	Residence time or contact time
X	Fractional conversion or approach to end-point

Subscripts

o	Initial value
∞	Final value

*Consultant, 87 Jaques St., Somerville, MA 02145-1804.

Table II — Data for decomposition of diazobenzene chloride

Time, min	Conc. DBC, mol/m^3	Conversion, X_{DBC}
0	69.7	0
6	46.7	0.330
9	38.7	0.445
12	30.8	0.558
14	26.7	0.617
18	20.4	0.707
20	18.01	0.742
22	15.98	0.771
24	14.18	0.797
26	11.92	0.829
30	9.59	0.862
∞	0	1

Source: Ref. [2].

Table III — Data for disproportionation of *p*-toluenesulfonic acid

Time, min	Conc. PTA, mol/m^3	Conversion, X_{PTA}
0	100	0
15	86.3	0.137
30	75.2	0.248
45	64.0	0.360
60	56.8	0.432
120	38.7	0.613
180	29.7	0.703
300	19.6	0.804

Source: Ref. [1].

Two particular cases of Eqs. (8) and (9) from which Table I was generated are:

$$t_{3/4}/t_{1/2} = \ln(1/4)/\ln(1/2) = 2 \quad \text{for } n = 1 \tag{10}$$

$$t_{3/4}/t_{1/2} = (4^{n-1} - 1)/(2^{n-1} - 1) \quad \text{for } n \neq 1 \tag{11}$$

$$t_{2/3}/t_{1/3} = \ln(1/3)/\ln(2/3) = 2.71 \quad \text{for } n = 1 \tag{12}$$

$$t_{2/3}/t_{1/3} = (3^{n-1} - 1)/(1.5^{n-1} - 1) \quad \text{for } n \neq 1 \tag{13}$$

Note that these time ratios depend on n alone. Only one time ratio is required to determine n, but a second time ratio may be used for confirmation.

The above derivation is for data taken in a constant-volume batch reactor, with t denoting the residence time. The derivation for flow reactors is closely analogous, with space-time as the corresponding variable.

Examples

The method is illustrated by two examples based on actual kinetic data reported in the literature:

Example 1: Data for the decomposition of diazobenzene chloride (DBC) in aqueous solution at 50°C are presented in Table II. Determine the kinetic order with respect to DBC.

Noting that this reaction is irreversible, first compute the conversions via Eq. (5). We obtain the times corresponding to ¾, ½, and ⅔, ⅓ conversions by linear interpolation in Column 3, and thence find the time ratios:

$$t_{3/4} = 20.6 \text{ min}, \; t_{1/2} = 10.5 \text{ min}, \; t_{3/4}/t_{1/2} = 1.96 \approx 2.$$

$$t_{2/3} = 16.2 \text{ min}, \; t_{1/3} = 6 \text{ min}, \; t_{2/3}/t_{1/3} = 2.7.$$

Both these time ratios indicate that this reaction is first-order in DBC, according to Table I.

Example 2: Data for the disproportionation of *p*-toluenesulfonic acid (PTA) in acidic solution at 70°C are presented in Table III. Determine the kinetic order with respect to PTA.

The following time ratios are obtained as before:

$$t_{3/4} = 236 \text{ min}, \; t_{1/2} = 82.5 \text{ min}, \; t_{3/4}/t_{1/2} = 2.9 \approx 3.$$

$$t_{2/3} = 156 \text{ min}, \; t_{1/3} = 41.4 \text{ min}, \; t_{2/3}/t_{1/3} = 3.8 \approx 4.$$

Both these time ratios indicate that this reaction is second-order in PTA.

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Determine saturation and stability indexes graphically

*Yi-Shon Chen**

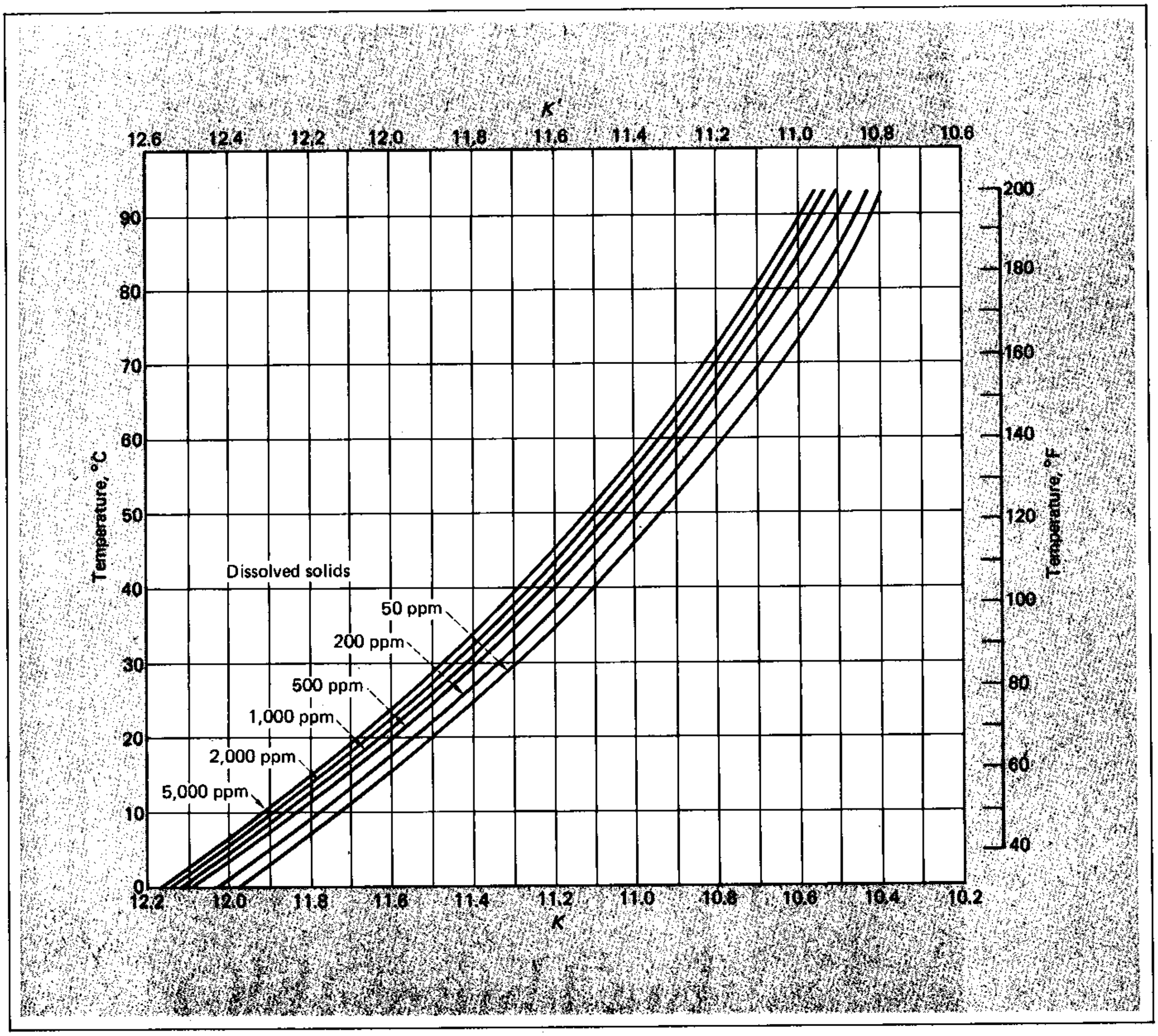
☐ Water treatment engineers use the Langelier saturation index (LI) and Ryznar stability index (RI) [*1,2*] to evaluate the scaling and corroding tendencies of water. Here is a graphical method for determining these parameters as a function of total dissolved-solids concentration and temperature.

The equations used to calculate these indexes are:

$$\text{LI} = \text{pH} - \text{pH}_s \qquad (1)$$

$$\text{RI} = 2\text{pH}_s - \text{pH} \qquad (2)$$

where pH = actual pH of the water and pH_s = pH at



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*Stone & Webster Engineering Corp., P.O. Box 2325, Boston, MA 02017.

which the water is in equilibrium with calcium carbonate. pH_s can be calculated by either equation:

$$pH_s = K - \log [Ca] - \log [Alk] \quad (3)$$

$$pH_ss = K' - \log [Ca^*] - \log [Alk] \quad (4)$$

where K and K' are constants dependent on temperature and dissolved-solids concentration, and they are obtained from the graph; [Ca] is the calcium concentration in ppm as Ca; [Alk] is the total alkalinity in ppm as $CaCO_3$; and [Ca*] is the calcium concentration in ppm as $CaCO_3$.

Normally, the LI is maintained between 0 and +0.5 and the RI is maintained between 6 and 7 for both corrosion and scale ($CaCO_3$) control. When the LI is positive and the RI is less than 6.5, the water has a tendency to produce $CaCO_3$ scale on piping and heat-exchanger surfaces. When the LI is negative and the RI is greater than 7, the water is corrosive—tends to disintegrate the surface of metal components via electrochemical reaction (e.g., hydrogen evolution and oxygen depolarization).

A water contains 100 ppm calcium (or 250 ppm calcium as $CaCO_3$) and 100 ppm total alkalinity. The measured pH is 7.0. The total dissolved solids is 500 ppm and the temperature is 40°C. From the graph, $K = 11.2$ and $K' = 11.6$, so from either Eq. (3) or (4), $pH_s = 7.2$. Thus, LI$=-0.2$ and RI$=-7.4$, slightly corrosive.

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Protect turbomachinery from magnetism

Ashok K. Mittal*

Turbines and compressors can occasionally act as electric generators due to high shaft currents, resulting in equipment damage and production losses. Engineers unaware of the real problem might simply replace wornout parts, only to find that after a period of time the problem recurs. This article explains how shaft currents are generated, how to detect them, and how to control them.

Shaft currents are caused by residual magnetic fields in stationary and rotating parts of the equipment. These magnetic fields, which may be either strong or weak, can be created during electric arc welding, magnetic particle inspection, or when magnetic tools (e.g., magnets and magnetic chucks) are used during manufacturing or maintenance. When equipment with sufficient magnetic field rotates, it acts as an electric generator.

The current and voltage depend upon the strength of the residual magnetism, the size of the magnetized mass, and resistance of lubricating oil film in bearings or seals. Machines having heavy parts and running at high tip speeds and bearing velocities are more susceptible to magnetic shaft currents than slower equipment. In addition, self-excitation results as currents flow throughout the equipment.

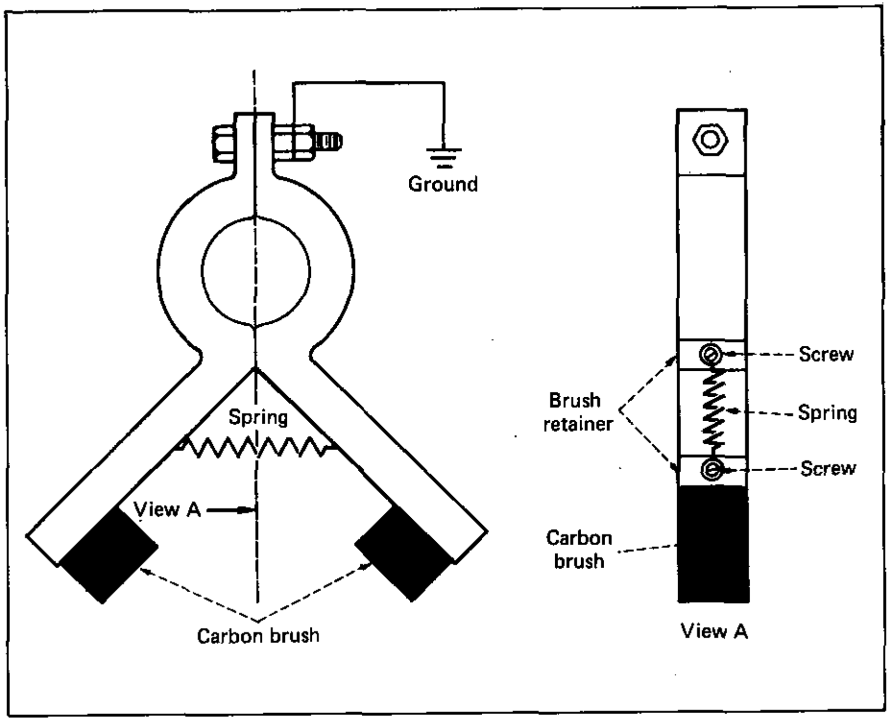

Figure 1 — Typical brush assembly for removing current

Locating and detecting shaft currents

Shaft currents can be located by finding equipment damage, which occurs where resistance is encountered. Look for pits and grooves and temperature "hot spots" on the surface of the material, all of which can be caused by sparks.

A simple method for detecting shaft currents is to touch

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*Indian Farmers Fertiliser Coorperative Ltd. (IFFCO), 26 Nehru Place, New Delhi-110 019, India.

one end of a grounded wire to the rotor and observe the spark. The voltage can be measured with an oscilloscope and brushes located at different places on the machine. (Voltage cannot be measured correctly with a voltmeter because of internal resistance.)

Sometimes current flows within closed seals or bearings and is difficult to detect. This current can be measured by placing a noninductive resistor of about 1 ohm between the brush and ground, and measuring the voltage drop across the resistor with an oscilloscope. The current can then be calculated from:

$$I = V/R$$

where I = current in amps, V = voltage drop, and R = resistance in ohms.

Allowable levels of magnetism

To minimize the generation of shaft currents, the following maximum levels of residual magnetism are recommended:

- 3 gauss or less for bearings, seals, journals, collars, gears and other surfaces in contact with lubricating oils.
- 6 gauss or less for all other bearings, seals, journals, collars and gears.
- 8 gauss for casing, pipes, etc., away from surfaces in contact with lubricating oils.

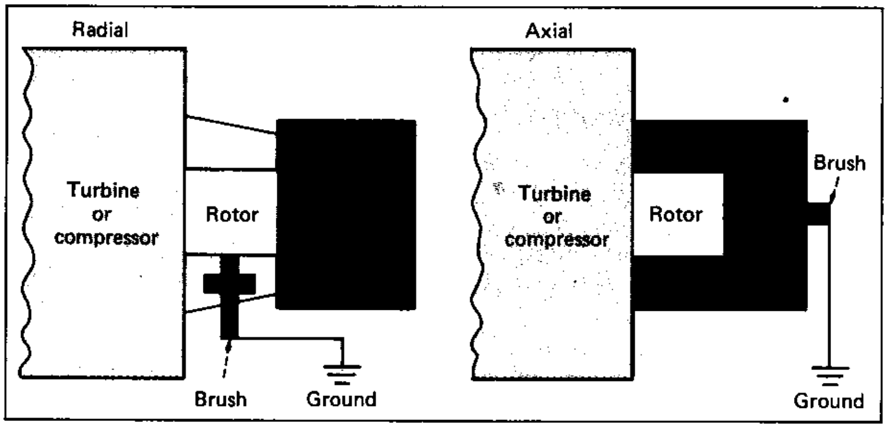

Figure 2 — Brushes can be installed on the rotor either radially or axially

These levels are for fully disassembled equipment in free air, as measured by a calibrated Hall transducer.

Brushes control magnetism and shaft currents

Equipment should be demagnetized periodically — e.g., during scheduled maintenance downtime — according to the manufacturer's instructions. This is a time-consuming process, and requires disassembly and reassembly of the unit.

Magnetism can be minimized by applying brushes to the shaft to drain away electrostatic charges and to remove electrostatic current from around the bearing and seal surfaces. Fig. 1 illustrates a typical brush arrangement.

First, one must know the locations of the shaft currents. Brushes are then positioned so that the current can travel the path of least resistance to leave the unit without any sparking. They can be placed either axially or radially, as shown in Fig. 2. Axial brushes are effective when shaft currents are generated at the extreme end of a turbomachinery set, such as at the end of a compressor-turbine train. Radial brushes can be used in a turbine where the shaft is projected between the casing and bearing housing, and, if possible, should be mounted on the shaft inside the bearing housing.

Brushes must be made of a material with sufficient chemical resistance to withstand the fluid being handled and the temperatures that the brushes will encounter. Carbon, graphite and graphite-containing materials are common. The brush should be soft, so that during operation it does not cause shaft wear. And it should always be in contact with the shaft so that sparking does not occur.

The grounding arrangement required will be different for different installations, and will depend on where and how currents are generated. It is advisable to have an electrical expert determine the best setup.

Current and voltage should be measured periodically. If the magnetic field increases to above the permissible limit, demagnetization should be done as soon as possible.

Preventive measures

The following procedures are recommended for existing plants:

1. Keep spare parts free from residual magnetism, and check them periodically.
2. Conduct welding near turbomachinery and piping very carefullly. The ground clamps should be connected to the metal piece that is to be welded, and welding cable should not cross any part of the turbomachine.
3. Demagnetize any part after detecting magnetic field above the recommended limit.

For new installations:

1. Ensure that all parts are completely free from residual magnetism during manufacturing and assembly.
2. Install permanent brushes to route currents around sensitive equipment parts.
3. Install sensing devices to continually monitor shaft currents and internal magnetic fields.

Correctly position jam nuts

J. P. Reznicek*

Jam nuts are found on many pieces of process equipment. As shown in Fig. 1, such nuts are used to introduce additional frictional forces, to hold a screw adjustment when the loading on the screw is negligible. Indeed, wherever a gap should be secured or a given position regularly maintained, such nuts should be considered. However, they offer no advantages for normal bolt connections where there is no need for adjustments, since merely tightening a single, regular nut has the same effect.

It is a common misconception that a jam nut always sits on top, or behind, the load-bearing nut (as in a. in the Fig.). However, this is correct only when a screw is in compression. When a screw is in tension (b. in Fig.), the jam nut should be placed underneath, or in front of, the load-bearing nut.

When a screw having two nuts with jam forces between them is drawn in greater detail (see the inset), it becomes clear that tension forces can be carried from nut to bolt by nut No. 1 only, and compression forces by nut No. 2 only.

The total load for the load-carrying nut is:

$$F_T = F_J + F_L \tag{1}$$

and that for the jam nut is:

$$F_T = F_J - F_L \tag{2}$$

where F_J is the jam force, and F_L is the acting load.

Considering the above equations, it is natural to make the jam nut thinner than the load-carrying nut. However, this creates a pitfall if the jam nut is not properly positioned: if overloaded, it can damage the screw thread, and thus jeopardize the function of the piece of equipment.

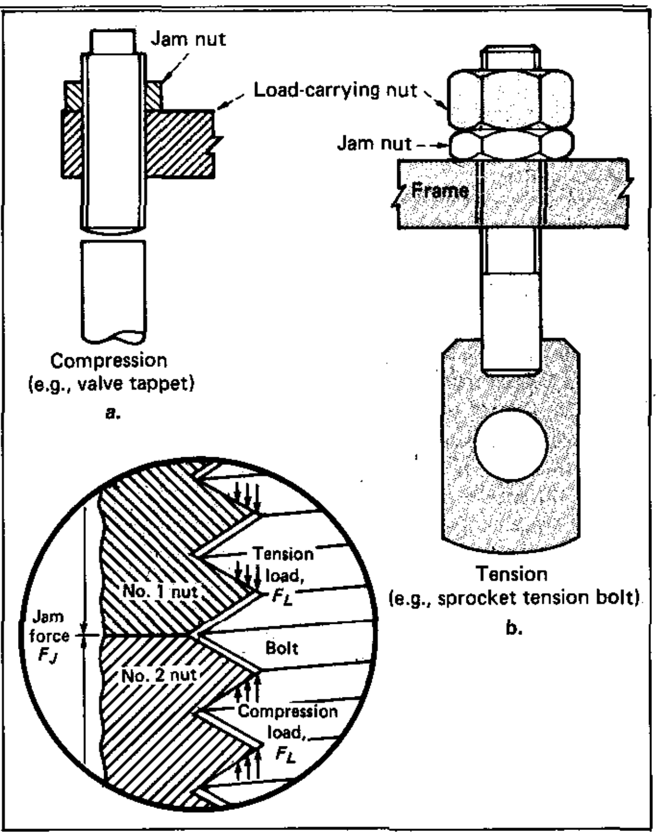

Figure — Appropriate position of jam nut depends upon type of load

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*164 Bellefontaine Street, Scarborough, Ont., M1S 4E6, Canada

Quickly estimate steam turbine power

K. Kuppuraj*

Back-pressure, or condensing, extraction steam turbine/alternator systems are widely used in the chemical process industries to meet electrical-power and process-steam requirements. These turbines employ superheated steam at high temperatures and pressures.

The two-part nomograph presented here estimates the electrical power generated at the alternator terminals. If the electrical power is known, the extracted or condensed steam flow in the turbine can be estimated. Estimation is quick and accurate; it does not require aids such as the steam tables, calculators or personal computers.

The nomograph solves the following equation:

$$P = \frac{W(h_i - h_o)\eta}{3{,}414 \times 10^6}$$

where: P = electrical power generated at alternator terminals, MW.

W = Steam flow through steam turbine, lb/h.

h_i = Steam enthalpy at steam turbine inlet, Btu/lb.

h_o = Steam enthalpy at steam turbine exit, Btu/lb.

η = Combined overall efficiency of both steam turbine and alternator.

Fig. 1 is used to generate a factor, which is used in Fig. 2 to solve the equation.

Example I

In a process plant an energy-conservation engineer estimated that there was a potential to generate 150,000 lb/h of superheated steam at 400 psi and 750°F. The plant needs the same quantity of steam at 20 psi for process uses. (Note: all pressures are absolute.) If the matching back-pressure

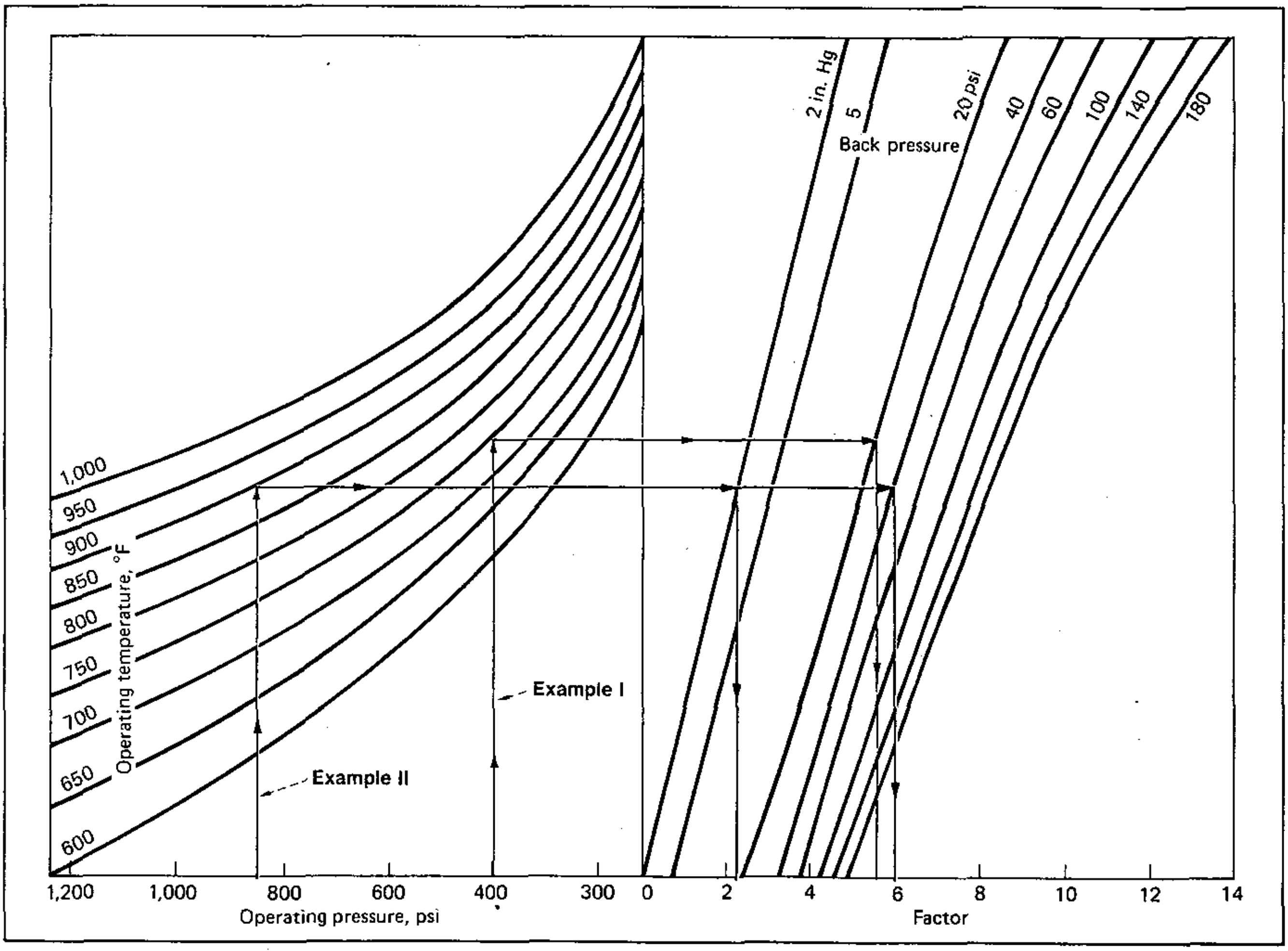

Figure 1 — Use this graph to determine the factor for a problem; the factor is used in Fig. 2 to yield the result

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*Industrial Power Products, Bharat Heavy Electricals Ltd., Tiruchirapalli, 620 014, India.

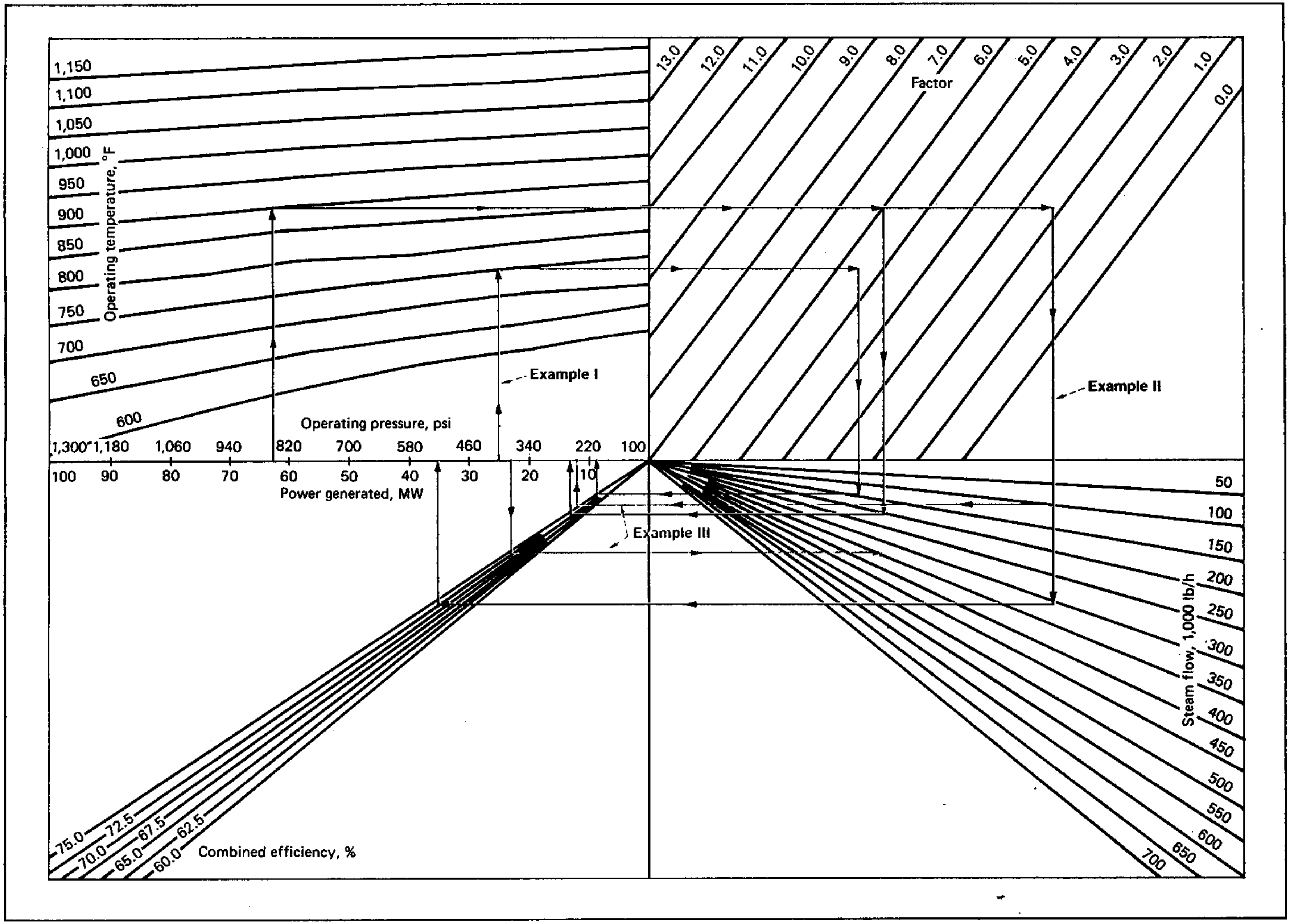


Figure 2 — This graph, along with the factor determined in Fig. 1, is used to relate electrical power to steam properties

steam turbine and alternator operates at 70% combined efficiency, how much electrical power could be expected at the alternator terminals?

Solution: In Fig. 1, draw a line connecting 400 psi to 750°F; proceed to 20 psi to read a factor of 5.5.

Using Fig. 2, plot from same operating pressure and temperature, to cut the line on the right top part at 5.5 and proceed downward to a steam flow of 150,000 lb/h. Proceed left to a combined efficiency of 70% and read the power generated at the alternator terminals as 8.5 MW.

Example II

In a condensing-extraction steam turbine used in a process plant, steam at 850 psi and 900°F is expanded to 2 in. Hg back-pressure, and process steam is extracted at 40 psi. The turbine operates at a combined efficiency of 72.5%. If 300,000 lb/h of steam is condensed and 200,000 lb/h of steam is extracted from the turbine, what is the total power developed at the alternator terminals?

Solution: Plot in Fig. 1, for the operating steam pressure of 850 psi, operating temperature of 900°F, and back pressures of 2 in. Hg and 40 psi; read the operating factors as 2.2 and 6.0, respectively.

In Fig. 2, plot, as previously, for the new values as shown in the nomograph to read the power generated as 35 MW and 13 MW, respectively. The total power generated at the terminals is 48 MW.

Example III

In Example II, if 100,000 lb/h of steam is condensed and a total power of 35 MW is generated, what will be the extracted steam quantity?

Solution: In Fig. 2, plot for 100,000 lb/h of condensed steam flow, and read the power developed as 12 MW. So, balance power developed due to extracted steam is 23 MW.

Plot backwards from 23 MW, cutting a combined efficiency of 72.5% and proceeding to the right. This line hits the extension of the extracted-steam line projected earlier from a factor of 6.0 where the extracted steam is 340,000 lb/h.

Electrical circuits in hazardous locations

All equipment for hazardous locations is not alike. Some may be better suited to corrosive environments.

R. A. Pankowski, Allen-Bradley Co.

☐ Throughout the chemical process industries, chemical engineers become involved in the design, building and operation of electrical systems that must operate in hazardous locations.

Of course, the engineer should specify items of equipment that have been designed for the proper Class, Group and Division (see Table I) as specified in the National Electrical Code (NEC). However, careful consideration must be given to the environment in which this equipment will operate. Caustics, acids and other corrosive agents can ruin electrical contact surfaces, and destroy exposed components in a short time.

Such damage can be minimized by thoughtful plant layout and by the use of electrical equipment, enclosures and switching gear that is suited to the corrosive environment as well as to the hazardous location. Without adequate provision for the environment, downtime and premature replacement of equipment can prove to be very expensive.

Planning for circuit reliability

Planning for circuit reliability should begin in the original plant design. Whenever possible, electrical specifications should not be decided upon until the physical plant layout is determined. Planning to locate as many control components as possible in areas unlikely to receive heavy contamination can significantly reduce the potential for future failures and the necessity of premature replacements. Personnel charged with the maintenance and repair of circuits and equipment in existing plants should also be consulted, since their experience will lead to selection of equipment more suitable to the environments and will reduce maintenance and repair of circuits, enclosures and operators. Such repair tends to be costly, particularly in any Division 1 environment.

A prudent engineer who is charged with equipment design for a hazardous location (either for a new plant or for a plant expansion) should first consider the split between Division 1 and Division 2. Essentially, in Division 1 situations the hazard will normally be present; in Division 2, the hazard will be present only in cases of malfunction or other emergency condition.

Hazardous locations—A short summary

Class I locations

Division 1 locations are those places where ignitable concentrations of flammable gases or vapors exist under normal conditions, or may frequently exist because of leakage or owing to maintenance operations, or where malfunctions may release ignitable vapors and simultaneously cause failure of electrical equipment.

Division 2 locations are those where flammable liquids or gases are present but are normally confined and can escape only through accident or abnormal operation. Also included are areas where ignitable concentrations of gases or vapors are normally prevented by mechanical ventilation, but might become hazardous because of failure or abnormal operation of the equipment. A third Division 2 situation is an area adjacent to a Division 1 location, where ignitable concentrations of gas or vapor might occasionally be communicated (unless approved means of preventing such communication are in effect).

Class II locations

Divisions 1 and 2. These are locations analogous to Class I, Divisions 1 and 2, except that *combustible dust* may be present rather than ignitable gases or vapors.

Class III locations

These are locations that are hazardous because of the presence of easily ignitable fibers or flyings, but where such fibers or flyings are not likely to be in suspension in the air in quantities sufficient to produce ignitable mixtures.

Division 1 locations are those in which easily ignitable fibers or materials producing combustible flyings are handled, manufactured or used.

Division 2 locations are those in which easily ignitable fibers are stored or handled (except in process of manufacture).

Typical hazardous locations

Class I
- Petroleum refining facilities
- Petroleum distribution points
- Petrochemical plants
- Dip tanks containing flammable or combustible liquids
- Plants manufacturing organic coatings
- Spray finishing areas (residue must be considered)
- Solvent extraction plants
- Plants manufacturing or using pyroxylin

Class II
- Manufacture and storage of magnesium
- Manufacture and storage of starch
- Fireworks manufacture and storage
- Flour and feed mills
- Areas for packaging and handling of pulverized sugar, and cocoa
- Facilities for the manufacture of magnesium and aluminum powder
- Spice grinding plants

Class III
- Rayon, cotton, and other textile mills
- Combustible fiber manufacturing and processing plants

The engineer should also consider what changes in the Code or state-of-the-art will allow an installation that will last longer, be more reliable, and reduce both initial installation costs and maintenance expense. For example, manufacturers of equipment for hazardous locations do not always spell out the materials used in their pieces of equipment. A number of manufacturers produce pushbutton stations for Division 1 and Division 2. In examining their specifications, questions that should be asked include:

- With what external coatings are the raw castings treated before painting?
- Of what kind of stainless steel are the operating shafts in pushbuttons made—Type 302 or Type 316? Indeed, are they stainless steel at all?
- What features have been included in the actuating means of the pushbutton to protect against entry of contaminants into the enclosure?
- Is the piece of equipment actually UL Listed, or is it merely equipment that "complies with UL Standards"?

Changes in the Code

Reviewing changes in the National Electrical Code helps to keep the design engineer up-to-date with product offerings and technologies (which change on a regular basis). For example, Article 501-5 of the 1981 edition of the Code spells out, in depth, the requirements for seals and drainage on stations in Division 1 locations. It should be noted that a change in this 1981 edition, in the form of an exception, is listed for conduit runs of 1½ in. and smaller: "Apparatus which may produce arcs or sparks need not be sealed if the current interrupting contacts are enclosed within a chamber hermetically sealed against the entrance of gases or vapors."

This exception in the Code offers potential savings by the elimination of costly external seal fittings, and the use of a hermetically sealed switch, which is not only more cost-effective, but which provides added circuit reliability as well. Sealed switch devices are available that are UL Listed for Division 1 and Division 2 hazardous locations. In a Division 1 application, it may be possible to eliminate the external seal fitting, and also gain greater contact reliability. In a Division 2 installation, the hermetically sealed switch also allows the installer to use equipment further tailored to the environment, such as a NEMA Type 4X corrosion-resistant enclosure.

By keeping abreast of manufacturers' current offerings in components and enclosures, and studying the specific application before selecting materials, engineers can design electrical systems using state-of-the-art technology. They can be confident that they are taking steps to optimize circuit protection, minimize downtime, and lower costs, and that they are addressing the hazardous-location division properly before embarking on a new plant or expansion layout.

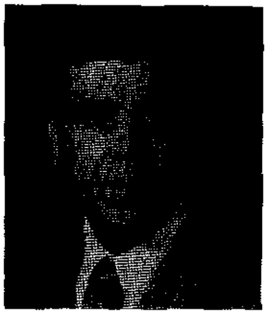
Some specific strategies

Here are some strategies for reducing downtime in hazardous locations:

- Where possible, isolate control components from sources of contamination and corrosion at the time of original plant design.
- Review, using data from existing plants, the repair and replacement of components, noting the mode of failure and the influence of the environment on the failure. Specify equipment to reduce or eliminate these failures.
- Review hazardous locations for possible reclassification from Division 1 to Division 2, if the Code, your company, and insurance rules permit.
- Study the application before specifying materials. Do not rely on outdated specifications. State-of-the-art is constantly changing.
- Specify corrosion-resistant materials that are compatible with the environment for applications where circuits cannot be isolated from corrosive atmospheres. Premature replacement is costly.
- Consider factory-sealed or hermetically sealed equipment.
- Specify sealed switches for applications where internal components may be subjected to airborne corrosive particles and fumes.
- Select equipment that is truly UL Listed, not just "complies with UL Standards."

***Roy V. Hughson**, Editor*

The author



Richard A. Pankowski is Marketing Supervisor, Product Marketing Dept., Industrial Control Div., Allen-Bradley Co., 1201 South Second St., Milwaukee, WI 53204; telephone (414) 671-2000. He holds an A.A.S. from Milwaukee Institute of Technology, and a B.S. in mechanical engineering from Marquette University.

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