# ECONOMIC DESIGN AND OPERATION OF PROCESS HEAT EXCHANGE EQUIPMENT

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## Introduction

A thorough investigation of all equipment in a sugar factory covered by this title would result in a monumental publication. This relatively modest paper purports to draw attention to a few specific aspects of economy in juice heating. Considering the abundance of comprehensive articles on liquid-vapour and liquid-liquid heat exchangers appearing in chemical engineering journals it is surprising that the superficial treatment of this subject as shown by articles in sugar journals indicates that little application is made of this valuable fund of knowledge in the sugar industry. Although the empirical approach may be adequate for routine specification and control, the application of general chemical engineering techniques developed in this field would facilitate the attainment of a maximum operating economy and the optimum design of new equipment. It is hoped that this article will produce a stimulus to the application of established heat engineering techniques to the economic design and operation of juice heaters and heat exchange equipment in general in the sugar industry.

## Derivation of Heat Transfer Coefficients

In the general case of heating juice flowing inside a tube we are concerned with convectional heat transfer to a liquid under turbulent flow. Consider for example a juice velocity of only 3 ft. per sec. through a 1.5 in. i.d. tube. If the density is 65 lb per cu ft and the viscosity 0.5 cP, i.e. $0.5 \times 6.72 \times 10^{-4} = 3.36 \times 10^{-4}$

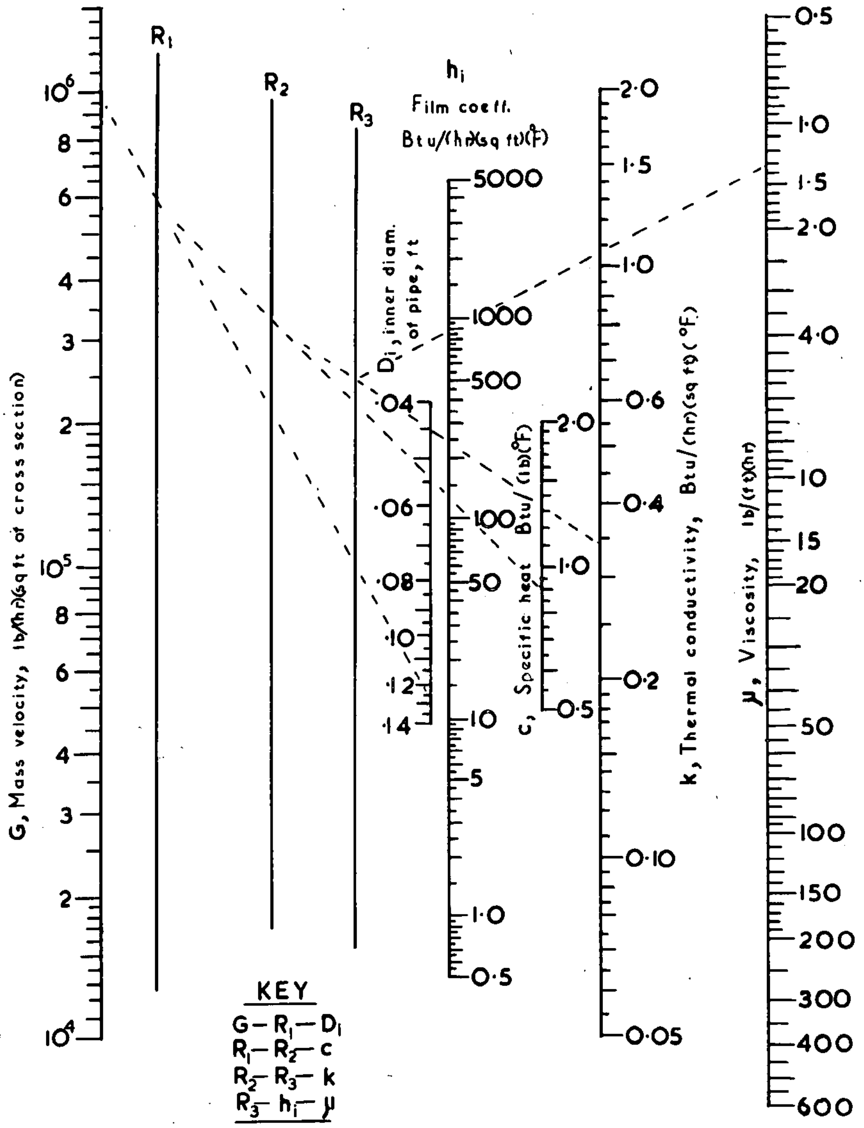


**FIGURE 1.** Nomograph for the determination of film coefficient inside tubes for turbulent flow of fluid[8].

lb/(ft)(sec), (taking conservative figures), then the value of the Reynolds number is:

$$N_{Re} = Du\rho/\mu = (1.5\times3\times65)/(12\times3.36\times10^{-4}) = \underline{\underline{7.25\times10^{4}}}$$

which is well over the laminar flow region (below a value of 2,100).

The overall heat transfer coefficient may be predicted from a knowledge of the physical conditions existing on either side of the tube wall but since it is dependent on a number of variables it is usual to estimate individual coefficients for the inner and outer pipe surfaces and to summate these, as discussed later.

*Liquid Film Coefficient Inside Tubes*

The fluid adjacent to the pipe wall is in laminar flow, hence heat transfer through this film is by conduction and the liquid film resistance will be dependent on the Reynolds number, as well as the thermal conductivity and specific heat of the fluid, i.e.

$$h_i = f(D, G, \mu, k, c)$$

By dimensional analysis it may be shown that

$$h_i D/k = f(DG/\mu,\ c\mu/k)$$

the three dimensionless groups being known respectively as, the Nusselt number ($N_{Nu}$), the Reynolds number ($N_{Re}$) and the Prandtl number ($N_{Pr}$), i.e.

$$N_{Nu} = f(N_{Re}, N_{Pr})$$

A considerable amount of research has resulted in the correlation

$$h_i D/k = 0.023(DG/\mu)^{0.80}(c\mu/k)^{\frac{1}{3}}$$

which holds for Reynolds numbers between 10,000 and 400,000 and Prandtl numbers between 0.7 and 120[18]. For liquids, this equation may be condensed to

$$h_i = 0.023G^{0.8}k^{\frac{2}{3}}c^{\frac{1}{3}}/D^{0.2}\mu^{0.47} \quad \ldots (1)$$

Equation (1) may be solved approximately by the use of the nomograph [3] in fig. 1.

Since $\mu$ decreases rapidly with an increase in temperature, the film coefficient increases and it is usual to calculate a mean coefficient for conditions prevailing at the mean temperature of the liquid in the exchanger. This is satisfactory for the case of low viscosity liquids where a small temperature difference prevails across the tube. However, in general it is necessary to estimate the actual wall temperature in contact with the heated fluid as discussed later.

Equation (1) shows that the liquid velocity is the most important factor determining the film coefficient inside the tubes. For example, if the liquid velocity was increased from 3 to 6 ft per sec (all other variables being constant) the film coefficient would increase, according to equation (1), by a factor of 1.74. Consequently, the liquid velocity should be as high as possible, the upper limit being economically dependent on the incremental cost of the exchanger and the pumping charges.[17]

*Outside Film coefficients*

In the sugar industry we are concerned with the condensation of the low pressure steam in the case of juice heaters and also in the transfer of heat through liquid films outside tubes in liquid-liquid heat exchangers. In the latter case, equation (1) may be used by substituting $D_e$ for D;

$$D_e = 4 \times \text{free area of cross section/perimeter}$$

which applies when the flow is parallel to the tubes and fully turbulent, i.e. $N_{Re} > 10{,}000$[21].

For steam-heated tubes, the installation may be either horizontal or vertical. For film condensation, Nusselt has developed the following equations [19]

Horizontal tubes: $$h_{oh} = 0.725\left(\frac{k^3_f\,\rho^2_f\,g\lambda}{N^{\frac{2}{3}}\Delta t_o D_o \mu_f}\right)^{\frac{1}{4}} \quad \ldots\ldots (2)$$

Vertical tubes: $$h_{ov} = 0.943\left(\frac{k^3_f\,\rho^2_f\,g\lambda}{\Delta t_o L \mu_f}\right)^{\frac{1}{4}} \quad \ldots\ldots (3)$$

which apply for $N_{Re}$ in the film of less than 2,100. In practice these equations are conservative by about 20 per cent due to the effect of ripple on the film.

Dropwise condensation would give higher values, but in general it is safest to assume film-type condensation for design purposes. When clean steam condenses on clean surfaces film-type condensation is always obtained.[15] The investigations of Osment *et al.* [20] have shown that overall heat transfer coefficients in surface condensers may be doubled by the injection of filming amines into the steam space to promote drop-type condensation.

It is interesting to note from equations (2) and (3) that the relative effectiveness of steam condensation rates for horizontal and vertical tubes is

$$\frac{h_{oh}}{h_{ov}} = \frac{0.725}{0.943}\left(\frac{L}{D_o N^{\frac{2}{3}}}\right)^{\frac{1}{4}} \quad \ldots\ldots (4)$$

Assuming that the tubes are 1.6 in outside diameter, 12 ft. long and 8 tubes are arranged in the average vertical stack, equation (4) indicates that the horizontal heater will have a 70 per cent greater condensing film coefficient than the vertical heater.

The factor N in equation (2) accounts for the effect of the accumulating condensate film around a vertical stack of horizontal tubes, the film coefficient diminishing for lower tubes. For this reason a staggered arrangement of the tubes would promote a higher film coefficient.[4]

*Film and Wall Temperatures*

In the case of turbulent liquid flow through tubes, the difference in temperature between the bulk of the liquid and the film in contact with the tube wall is often neglected particularly if the temperature difference across the wall is small. However, if correction is necessary then equation (1) is multiplied by

$$\phi = (\mu/\mu_w)^{0.14} \quad \ldots\ldots (5)$$

viscosity being the only variable which is significantly effected by temperature.

Estimation of the wall temperature may be achieved by a trial-and-error method using the equation[18]

$$\Delta t_i = \frac{1/h_i}{1/h_i + D_i/D_o h_o} \Delta t \quad \ldots\ldots \quad (6)$$

in which $h_i$ is estimated from equation (1) and $h_o$ from equation (2) or (3). For the preliminary estimation of $h_o$ the outer wall temperature is chosen midway between the bulk temperatures on either side of the wall.

The wall temperature is then obtained from

$$t_w = t + \Delta t_i \text{ for heating}$$
$$\text{or} \quad t_w = t - \Delta t_i \text{ for cooling}$$

In the case of condensing a vapour outside a tube, the condensate is normally under viscous flow and the temperature drop across the film is more significant. The mean film temperature is evaluated from [19]

$$t_f = t_s - 3(t_s - t_w)/4 \quad \ldots\ldots \quad (7)$$

The wall temperature is assumed initially and the value of the film coefficient, calculated by equation (2) or (3), is checked using equation (6).

For approximate working figures equation (1) is used without correction and the steam film coefficient may be determined from nomographs such as fig. 2.[25]

*Overall Heat Transfer Coefficients*

The overall heat transfer coefficient is compounded from the individual resistances due to the inside scale, the inside film, tube wall, outside film and outside scale as shown by equations (8) and (9). The overall coefficient may be based arbitrarily on either the inside or outside tube area but the chosen area should be stated. The outside area is the most usual choice.

$$U_o = \frac{1}{\frac{D_o}{D_i h_{di}} + \frac{D_o}{D_i h_i} + \frac{x_w}{k_m}\frac{D_o}{\bar{D}_L} + \frac{1}{h_o} + \frac{1}{h_{do}}} \quad \ldots \quad (8)$$

$$U_i = \frac{1}{\frac{1}{h_{di}} + \frac{1}{h_i} + \frac{x_w}{k_m}\frac{D_i}{\bar{D}_L} + \frac{D_i}{D_o h_o} + \frac{D_i}{D_o h_{do}}} \quad \ldots \quad (9)$$

In the above equations the diameter ratios correct the values of the individual coefficients to the selected area. In some cases one film coefficient may be considerably greater than any of the others so that the diameter correction has a small effect. In this case it is convenient to abbreviate the equation eliminating the diameters and to express the overall coefficient in terms of the tube-side area in contact with the highest resistance, i.e. lowest film coefficient.[18]

The coefficients $h_{di}$ and $h_{do}$ represent the fouling factors for the inner and outer tube surfaces, respectively. Their combined values may be determined by comparing the overall coefficients of the clean and scaled heaters. If however the outer wall is clean, the inside fouling factor may be calculated [14] from

$$1/h_d = 1/U_{od} - 1/U_{oo}$$

Another method of determining the fouling factor is by means of a Wilson plot, [1, 15] in which the reciprocal of U is plotted as a function of $u^{0.8}$ for both clean and fouled surfaces.

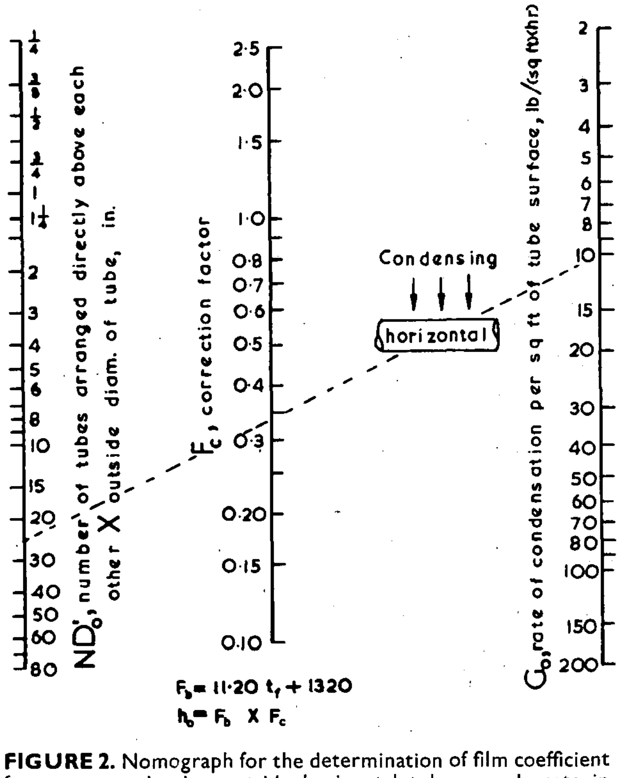


**FIGURE 2.** Nomograph for the determination of film coefficient for steam condensing outside horizontal tubes, condensate in laminar flow[25].

## Estimation of Coefficients in Practice

There is little information available on heat transfer coefficients of juice heaters under factory conditions in South Africa. For this reason, even fundamental questions such as the choice between vertical and horizontal heaters or the optimum juice velocity are often still a matter of controversy even after several decades of experience. In spite of this lack of practical information, many of the problems may be clarified by applying the standard chemical engineering techniques outlined in the previous section.

The author has determined $U_o$ on several local heaters and found rather low values of not more than 180 after being cleaned inside the tubes. One of these heaters will be used as an example of the application of the methods developed previously.

*Example*

The heater chosen for analysis is a horizontal tubular type with tubes arranged in a series of vertical stacks, 18 per stack on the average. The following data apply:

Brix of juice = 14.7°
Juice rate = 112 ton/hr
Heating range = 96° F to 192° F
Vapour satn. temp. = 218° F
Effective tube length = 11 ft $10\frac{1}{4}$ in
Inside tube diameter = 1.495 in
Outside tube diameter = 1.625 in
Total heating surface = 2,010 sq ft (based on o.d.)
Tube arrangement = square pitch, average 16 per stack
Tubes per pass = 18

*Heat Transfer Coefficient—Clean Tubes*
*Inner Film Coefficient:*

It may be assumed that flow is turbulent (as calculated earlier) hence the inner film coefficient may be estimated from equation (1). The mean juice temperature is

$$(96 + 192)/2 = 144° \text{ F or } 62° \text{ C}$$

Using the physical data in the appendix as an approximation:

$\mu = 0.65 \times 2.42$ lb/(ft)(hr)
$k = 0.346$ Btu/(ft)(hr)(°F)
$c = 0.92$ Btu/(lb)(°F)
$D_i = 1.495/12 = 0.125$ ft
Inside section $= 3.1416 \times (0.125)^2/4 = 0.0123$ sq ft/tube

$$G = 112 \times 2{,}000/(18 \times 0.0123)$$
$$= 1.012 \times 10^6 \text{ lb/(sq ft)(hr)}$$

From equation (1)

$$h_i = \frac{0.023(1.012 \times 10^6)^{0.8}(0.346)^{\frac{2}{3}}(0.92)^{\frac{1}{3}}}{(0.125)^{0.2}(1.573)^{0.47}}$$

$$= \underline{\underline{860 \text{ Btu/(hr)(sq ft)(°F)}}}$$

*Tube wall transfer rate:*

The tube wall coefficient may be determined as inferred from equation (8). Assuming 70–30 brass tubes:

$$k_m = 60 \text{ Btu/(hr)(sq ft)(°F/ft)}$$
$$x_w = (1.625–1.495)/12 = 0.108 \text{ sq ft}$$
$$k_m/x_w = \underline{\underline{5{,}556 \text{ Btu/(hr)(sq ft)(°F)}}}$$

*Steam side film coefficient:*

Generally it is preferable to use nomographs based on practical figures rather than the method discussed previously, the limitations of which have been pointed out. The nomograph by Stoever[25] fig. 2 may be applied for an initial estimate.

The condensing temperature is 218° F and

$\lambda = 966$ Btu/lb
$q = 112 \times 2{,}000 \times 0.92(192–96)$
$= 1.978 \times 10^7$ Btu/hr
$G_o = 1.978 \times 10^7/(966 \times 2{,}010)$
$= 10.19$ lb/(sq ft)(hr)
$ND'_o = 16 \times 1.625 = 26$

From fig. 2 (see dotted line example), the correction factor is 0.34. Assuming a wall temperature of (218 + 144)/2 = 181° F the mean condensate film temperature is, from equation (7)

$$t_f = 218–3(218–181)/4 = \underline{\underline{190° \text{ F}}}$$

and the corresponding base factor may be calculated from

$F_b = 11.2t_f + 1{,}320$ (see fig. 2)
$= 11.2 \times 190 + 1{,}320$
$= 3{,}460$

The film coefficient is calculated from

$h_o = F_b \times F_c$ (see fig. 2)
$= 3{,}460 \times 0.34$
$= \underline{\underline{1{,}180 \text{ Btu/(sq ft)(hr)(°F)}}}$

*Check on Wall Temperature:*

From equation (6)

$$\Delta t_i = \frac{1/860}{1/860 + 1.495/(1.625 \times 1{,}180)}(218-144) = \underline{\underline{44}}$$

and $t_w = 144 + 44 = \underline{\underline{188° \text{ F}}}$

From equation (7)

$$t_f = 218–3(218–188)/4 = \underline{\underline{186° \text{ F}}}$$

which is sufficiently close to the value assumed above.

*Corrected inside coefficient:*

Using equation (5) $h_i$ may be corrected to the wall temperature at which $\mu_w = 0.45$ cP and hence

$$h_i = 860\ (0.65/0.45)^{0.14} = \underline{\underline{906}}$$

This value is obtained approximately, following the example (dotted line) in fig. 1.

Checking again with equation (6)

$$\Delta t_i = \frac{1/906}{1/906 + 1.495/(1.625 \times 1{,}180)}\ (218–144)$$
$$= \underline{\underline{43° \text{ F}}}$$

Hence $h_i$ = $\underline{\underline{\text{906 is acceptable}}}$

*Check on outside film coefficient:*

In equation (2)

$k_f = 0.39$ Btu/(sq ft)(hr)(°F/ft)
$\rho_f = 60.2$ lb/cu ft
$g = 4.17 \times 10^8$ ft/hr²
$\lambda = 984$ Btu/lb
$N = 16$
$\Delta t_o = 218{-}188 = 30°$ F
$D_o = 1.625/12 = 0.135$ ft
$\mu_f = 0.32 \times 2.42$ lb/(ft)(hr)

$$h_{oh} = 0.725 \left( \frac{(0.39)^3 (60.2)^2\ 4.17 \times 10^8 \times 984}{(16)^{\frac{2}{3}} \times 30 \times 0.1354 \times 0.32 \times 2.42)} \right)^{\frac{1}{4}}$$
$$= 979$$

This figure is known to be conservative by 20 per cent. Increasing by 20 per cent:

$$h_{oh} = 979 \times 1.2 = \underline{\underline{1{,}175 \text{ Btu/(sq ft)(hr)(°F)}}}$$

which compares well with the previous figure.

*Overall coefficient—clean:*

In equation (8)

$\bar{D}_L = (D_o - D_i)/2.303 \log (D_o/D_i) = 0.1271$ ft

Substituting individual coefficients into equation (8)

$$U_{oo} = \cfrac{1}{\cfrac{1.625}{1.495 \times 906} + \cfrac{0.1258}{0.1271 \times 5{,}556} + \cfrac{1}{1{,}180}}$$
$$= \underline{\underline{450 \text{ Btu/(hr)(sq ft)(°F)}}}$$

*Effect of Fouling:*

From equation (8) it may be shown that if the tube wall becomes fouled

$$\frac{1}{U_{od}} = \frac{1}{U_{oo}} + \frac{1}{h_{di} D_i / D_o} + \frac{1}{h_{do}} \quad \ldots \textbf{(10)}$$

Measurement of $U_o$ in practice for this particular heater under the given operating conditions provided the value of $U_{od} = 157$.

Little information is available on fouling factors in cane juice heaters. The Sugar Research Institute, Mackay, [2] have conducted investigations on a pilot scale heater which, upon analysis, provided results in close agreement with $U_o = 450$ for a clean heater under the present conditions. The thermal conductivity of the scale was calculated as about 0.3 and after 100 hours operation the thickness of scale was about 0.006 inches.

Applying this information, it is possible to estimate approximately the effects of fouling. Assuming for example that the average scale thickness between cleanings was 0.005 inches, then the fouling factor would be

$$h_{di} = k/x = (0.3/0.005)12 = 720$$

and from equation (10)

$$U_{od} = \frac{1}{1/450 + 1.495/(1.625 \times 720)}$$
$$= \underline{\underline{287 \text{ Btu/(hr)(sq ft)(°F)}}}$$

This assumes no outside fouling. The Sugar Research Institute, Mackay,[2] observed on their pilot heater that the overall heat transfer coefficient decreased by as much as 30 per cent during a season due to fouling outside the tubes. The pilot heater was operated on factory exhaust steam. The heater examined in the present paper had been operating for a complete season, hence a similar degree of fouling could be expected. In the absence of any confirmatory data, if this is applied to the present case

$$U_{od} = 287 \times 0.7 = \underline{\underline{201 \text{ Btu/(hr)(sq ft)(°F)}}}$$

which is comparable with the actual measured figure of 157 and the outside fouling factor becomes

$$h_{do} = \frac{1}{1/201 - 1/287} = \underline{\underline{671 \text{ Btu/(hr)(sq ft)(°F)}}}$$

This value for $h_{do}$ is quite feasible even for a very thin film. Oil, for example, has a thermal conductivity as low as 0.07 so that a fouling factor of 671 could be accounted for by an oil film of thickness

$$\frac{0.07 \times 12}{671} = \underline{\underline{0.00125 \text{ inches}}}$$

In this connection it should be pointed out that the dropwise condensation promotion due to common oils is relatively inefficient (c.f. filming amines) and of short duration, particularly when other fouling compounds are present.[20]

The various heat transfer coefficients and fouling factors for the heater in question are summarised in Table 1.

TABLE I

*Horizontal Heater Coefficients*

| Coefficient | Btu/(hr)(sq ft)(°F) |
|---|---|
| $U_{oo}$ | 450 |
| $U_{od}$ (calc.) | 201 |
| $h_o$ | 1,180 |
| $h_{do}$ | 671 |
| $h_i$ | 906 |
| $h_{di}$ | 720 |
| $k_m/x_w$ | 5,556 |
| $U_{od}$ (measured) | 157 |

*Vertical vs. Horizontal Heaters*

Equation (4) indicates that, all other conditions being equal, the film coefficient for condensation in a horizontal heater will be greater than for a vertical heater provided that

$$\frac{0.725}{0.943} \left( \frac{L}{D_o N^{\frac{2}{3}}} \right)^{\frac{1}{4}} > 1 \quad \ldots\ldots \textbf{(11)}$$

Most tubes in cane juice heaters have $D_o = 1.625/12$ ft and $L = 12$ ft so that equation (11) would read

$$\frac{0.725}{0.943}\left(\frac{12 \times 12}{1.625\,N^{\frac{2}{3}}}\right)^{\frac{1}{4}} > 1$$

or N<172

Hence the condensing film coefficient for a horizontal heater is always greater than for a vertical heater. For the heater discussed above for example, N = 16 and from equation (4)

$$h_{oh}/h_{ov} = 1.486 \text{ or } h_{ov}/h_{oh} = 0.672$$

The condensing film coefficient (Table I) for a similar vertical heater would have been

$$h_{ov} = 1{,}180 \times 0.672 = 793$$

and the overall coefficient would have been (Table I)

$$U_{odv} = \frac{1}{1/201 - 1/1{,}180 + 1/793}$$
$$= \underline{\underline{185 \text{ Btu/(hr)(sq ft)(°F)}}}$$

which would require an increase of 9 per cent in heating surface. This is a very conservative example since the exchanger was poorly designed (square pitch) and heavily scaled. Had the tubes been staggered,[16] the number in a vertical row might have been reduced to eight. Using fig. 2, N = 8 hence $ND_o = 13$, $F_c = 0.42$ and $F_b = 3{,}460$. Hence $h_{oh} = 0.42 \times 3{,}460 = 1453$. The overall film coefficient would have been

$$U_{odh} = \frac{1}{1/201 - 1/1{,}180 + 1/1{,}453}$$
$$= \underline{\underline{208 \text{ Btu/(hr)(sq ft)(°F)}}}$$

and 13 per cent more heating surface would be required for a vertical heater. If, in addition, the heating surfaces were clean then

$$U_{ooh} = \frac{1}{1/450 - 1/1{,}180 + 1/1{,}453}$$
$$= \underline{\underline{485 \text{ Btu/(hr)(sq ft)(°F)}}}$$

Similarly $U_{oov} = \underline{\underline{379 \text{ Btu/(hr)(sq ft)(°F)}}}$

Thus 28 per cent more heating surface would be required for a clean vertical heater than for a clean horizontal heater with N = 8. The working value may range between 13 and 28 per cent, averaging about 20 per cent.

The existence of this difference between horizontal and vertical heaters cannot be disputed since it is based on calculations which have been substantiated by a large number of practical results from a wide field of application. Considering that overall coefficients are dependent on so many variables such as steam and juice properties, juice velocities, degree of inside and outside fouling, etc., it is not difficult to imagine why some sugar factory designers are unwilling to accept that this difference exists in practice.

A survey of costs per sq ft of heating surface for juice heaters from local suppliers has indicated that vertical heaters are normally about 5 per cent higher than horizontal heaters. This means that the total initial cost is 20 + 5 = 25 per cent higher for vertical heaters.

The average price of heaters is R6 per sq ft and for a 250 tch factory, with heating surface at 45 sq ft per tch,[11] the additional initial cost for correctly specified vertical heaters would be

$$\frac{25}{100} \times 6 \times 250 \times 45 = \underline{\underline{R16{,}875}}$$

To this must be added an additional 20 per cent on running costs.

This cost difference should be viewed in the light of convenience of *installation* and *operation* for the particular factory design. The choice of a vertical heater on either of these grounds is not necessarily based on *economy* and consequently falls beyond the scope of this paper.

**Economical Waste Heat Recovery**

A typical example of the recovery of waste heat in a sugar factory is the preheating of cane juice by means of evaporator vapours and condensates. A number of useful calculations has been presented by

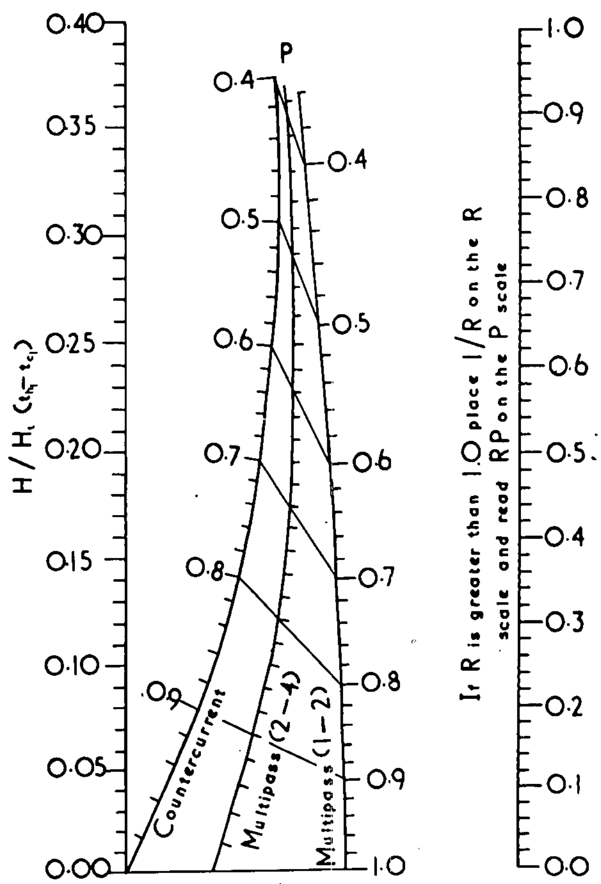


**FIGURE 3.** Nomograph for the evaluation of equations (13) and (14) in the economic optimization of waste heat recovery exchangers.[20] Copyright 1944 by the American Chemical Society and reprinted by permission of the copyright owner.

Happel[7] for determining the economic optimum heat recovery and these are discussed below.

*Recovery of Heat from Vapour or Exhaust Steam*

For the recovery of heat from steam or vapour at a constant temperature, $t_h$ to a liquid which is heated without vaporization from temperature $t_{c1}$ to $t_{c2}$

$$(t_h - t_{c2}) \text{ opt.} = H/H_t \qquad \ldots (12)$$
$$\text{where } H = 114rE/UY$$

This calculation assumes a knowledge of the total exchanger costs and marginal cost of steam production. The optimum value of $t_{c2}$ may then be determined. This equation applies also to the case of waste heat recovery from flue gases, as in a waste heat boiler, where the heated liquid temperature remains constant and the recovered heat is utilised in steam production.

*Recovery in Countercurrent Liquid-Liquid Exchangers*

For the recovery of heat from a hot liquid at temperature $t_{h1}$ to a cooler liquid at temperature $t_{c1}$ in a countercurrent (1–1) exchanger, the optimum final temperature $t_{h2}$ of the hot liquid may be determined from

$$[(1-P)(1-RP)] \text{ opt.} = \frac{H}{H_t(t_{h1} - t_{c1})} \qquad \ldots (13)$$

$$R = W_h C_h / W_c C_c$$
$$P = (t_{h1} - t_{h2})/(t_{h1} - t_{c1})$$
$$H = 114\, rE/UY$$

For any given problem the right hand side of the equation will be a constant and R will be fixed.

*Recovery in Multipass Exchangers*

Pre-heating of juice by condensates is commonly carried out in multipass exchangers of the 1–2 or 2–4 type as described by Webre.[27] For the case of a 1–2 type exchanger, the following equation applies in a similar manner to the previous expression

$$\left[1 - P(1 + R - \frac{RP}{2})\right] \text{opt.} = \frac{H}{H_t(t_{h1} - t_{c1})} \qquad \ldots (14)$$

For exchangers of the 2–4 type graphical differentiation is most convenient for the evaluation of P.

Ten Broeck[26] has presented a convenient nomograph for the evaluation of P for all three types of liquid-liquid exchangers. This nomograph is reproduced in fig. 3. The evaluation of $H_t$, the incremental cost of supplying heat, may present complications—it is composed of several elements. First there will be a saving resulting from decreased fuel consumption. The value of the heat saved may be determined from the price of fuel, its heating value and the expected furnace efficiency. The cost of supplying heat by a furnace will include the fixed charges on incremental cost of furnace as well as the fuel cost. Also the recovery of waste heat may reduce condensing and cooling costs.

**Economy by Control**

Since convection currents cause entrainment in clarifiers it is essential that the temperature of entering juice be stable. In the absence of proper control this is often achieved by superheating and flashing to constant temperature. The heat from flashed vapour is rarely recovered in spite of the fact that (e.g.) a 250 tch factory by maintaining 10° F of superheat in the juice would (if coal was being burnt) lose R7,200 per year in heat.*

Although the maintenance of 10° F is only necessary under conditions of very poor control there are cases where, due to excessive fluctuation in juice velocities and steam pressures even 10° F flash is insufficient to maintain a safe margin for occasional peak flow rates and resulting temperature drops below boiling. In such extreme cases automatic temperature control is not only a labour saving device but could be viewed as an economic advantage.

It should be mentioned that the maintenance of a small amount of flash is usually regarded as essential for the release of air from the juice and the acceleration of otherwise slow reactions but this discussion refers to excessive flash.

*Modes of Control*

*Conventional Control:* The normal method of control is to measure the outlet juice temperature and adjust the steam control valve to maintain the desired temperature. This usually requires a wide proportional band setting to maintain stability and hence reset response to correct the resulting offset due to load changes. When rapid changes in throughput occur the resulting short-term error can be corrected in part by the addition of derivative response.

*Condensate Throttling:* By throttling the condensate, a less responsive control action will be achieved but this system has the advantage of reduced initial cost. The behaviour of this type of system is difficult to predict.[22] It also assumes an oversized heating surface and is prone to the danger of excessive fouling on the steam side of the tubes if condensates are contaminated with oil, etc.

*Pressure-Cascade Control:* The most rapid recovery to load disturbances may be attained by cascading the output of a standard three-mode temperature controller into the set point of a proportional plus reset pressure controller. Changes in steam pressure are corrected directly by the pressure controller. Load changes are sensed rapidly by a change in shell pressure which is compensated by the pressure controller. The temperature control system senses the residual error and resets the pressure controller set point.

*Minimum Temperature Control:* In cases where more elaborate control is excluded due to cost, sharp downward peaks in the flashed juice temperature recording chart may be eliminated by the injection of

* The above amount was calculated assuming 4,600 hr/yr, 12,000 Btu/lb coal, a boiler efficiency of 70%, 20% recycle of filtrate on juice and 0.242c/lb coal.

higher pressure steam through a small Sarco type temperature regulator. As in the case of condensate throttling this system has some obvious disadvantages which may outweigh the low initial cost.

Whatever system is adopted the sizing of control valves and the design of thermometer probes and pockets should receive careful attention.

### Recent Trends in Heating Economy

Recent efforts to increase heater economy have been directed toward (a) more accurate optimization by the application of computers to relieve the tedium of design calculations (b) attempts to increase both inner and outer film coefficients and (c) the complete elimination of scaling.

#### *Optimum Design by Computer*

The design of a heater for optimum heat transfer, pressure drop and cost, entails accounting for so many variables simultaneously that the solution would generally require the comparison of costs for a considerable number of preliminary designs. For example, the heat transfer coefficient is a function of the liquid velocity which in turn influences the pressure drop. By the application of computer techniques both thermal and mechanical aspects may be considered simultaneously and by initially applying relatively empirical criteria, uneconomical designs may be eliminated at an early stage. Another advantage of computer methods is that it is possible to design an exchanger considerably more accurately than there is time to do by hand.

I.C.I. were recently faced with the design of a train of exchangers for the recovery of waste heat from gas to feedwater. A programme was developed capable of designing and costing exchangers for the full range of operating conditions. A typical design print-out is reproduced in fig. 4. A complete design providing all the data necessary for manufacture takes between five and ten seconds of the machine time.[12]

#### *Increasing Condensing Film Coefficients*

Considerable research has been conducted into the investigation of possible methods for the attainment of dropwise condensation. Osment *et al.*[20] conducted extensive tests on treated copper and brass tubes using various types of steam. Field tests using industrial steam showed the main cause of breakdown to be corrosion and oxidation of the metal surface rather than breakdown of the promoter film applied to promote dropwise condensation. Thiosilanes and xanthate compounds were most successful. After cleaning the tubes of a condenser by injection of 50 per cent hydrochloric acid followed by 50 per cent Teepol into the steam, 20 ml of a 1 per cent solution of thiosilane: $Si(SC_{12}H_{25})_4$ was injected to promote dropwise condensation. The test was continued with weekly injections of promoter and good dropwise condensation was achieved for one year. The amount of promoter used was 0.01 ppm on steam. The *overall* heat transfer coefficient ranged from 1,750 to 1,300 Btu/(hr)(sq ft)(°F).

| | | |
|---|---|---|
| GAS FLOW RATE | 120000.0000 | STANDARD CUBIC METRES PER HOUR |
| GAS BY-PASS FRACTION | 30.0000 | PER CENT |
| GAS INLET TEMPERATURE | 190.0000 | DEGREES CENTIGRADE |
| GAS OUTLET TEMPERATURE | 175.8902 | DEGREES CENTIGRADE |
| GAS INLET PRESSURE | 30.0000 | ATMOSPHERES |
| ALLOWABLE PRESSURE DROP | 3.0000 | POUNDS PER SQUARE INCH |
| ACTUAL PRESSURE DROP | 3.1114 | POUNDS PER SQUARE INCH |
| FEEDWATER INLET TEMPERATURE | 70.0000 | DEGREES CENTIGRADE |
| FEEDWATER FLOW RATE | 100.0000 | TONNES PER HOUR |
| FEEDWATER OUTLET TEMPERATURE | 140.0000 | DEGREES CENTIGRADE |
| FEEDWATER PRESSURE | 550.0000 | POUNDS PER SQUARE INCH GAUGE |
| MEAN HEAT TRANSFER COEFFICIENT | 439.8837 | BTU PER HOUR SQUARE FOOT DEGREE FAHRENHEIT |
| HEAT TRANSFER SURFACE AREA | 459.5006 | SQUARE FEET |
| AVERAGE GAS VELOCITY | 26.9181 | FEET PER SECOND |
| GAS INLET BRANCH PRESSURE DROP | 0.3000 | POUNDS PER SQUARE INCH |
| GAS OUTLET BRANCH PRESSURE DROP | 0.1500 | POUNDS PER SQUARE INCH |
| REYNOLDS NUMBER ON WATER SIDE | 40771.7852 | |
| WATER SIDE PRESSURE DROP | 0.1456 | POUNDS PER SQUARE INCH |
| NUMBER OF U-TUBES | 154.0000 | |
| TUBE PITCH | 1.5000 | EQUILATERAL TRIANGULAR |
| WATER STRAKE OUTSIDE DIAMETER | 32.3129 | INCHES |
| WATER STRAKE THICKNESS | 0.6833 | INCHES |
| GAS STRAKE OUTSIDE DIAMETER | 32.0000 | INCHES |
| GAS STRAKE THICKNESS | 0.6518 | INCHES |
| TUBEPLATE DIAMETER | 32.2129 | INCHES |
| TUBEPLATE THICKNESS | 2.3197 | INCHES |
| WATER TUBE DIAMETER | 1.0000 | INCHES |
| WATER TUBE WALL THICKNESS | 0.1090 | INCHES |
| STRAIGHT LENGTH OF TUBES | 5.6978 | FEET |
| AVERAGE U-TUBE LENGTH | 13.2129 | FEET |
| GAS INLET BRANCH BORE | 15.2543 | INCHES |
| GAS OUTLET BRANCH BORE | 12.8000 | INCHES |
| GAS END PLAT SMALLE RADIUS | 5.2916 | INCHES |
| GAS END PLATE LARGE RADIUS | 26.1783 | INCHES |
| GAS END PLATE THICKNESS | 0.7026 | INCHES |
| WATER END PLATE SMALL RADIUS | 5.4805 | INCHES |
| WATER END PLATE LARGE RADIUS | 26.5957 | INCHES |
| WATER END PLATE THICKNESS | 0.8689 | INCHES |
| BORE OF WATER BRANCHES | 5.1752 | INCHES |
| TOTAL NUMBER OF BAFFLES | 3.0000 | |
| No. of BAFFLES IN FIRST INTERVAL | 0.2692 | |
| BAFFLE SPACING | 2.9500 | FEET |
| LENGTH OF INTERVAL | 0.7940 | FEET |
| BAFFLE CUT-OFF (INNER) | 0.4516 | FEET |
| BAFFLE CUT-OFF (OUTER) | 0.7074 | FEET |
| No. OF BAFFLES IN SECOND INTERVAL | 0.5010 | |
| BAFFLE SPACING | 2.8958 | FEET |
| LENGTH OF INTERVAL | 1.4509 | FEET |
| BAFFLE CUT-OFF (INNER) | 0.4516 | FEET |
| BAFFLE CUT-OFF (OUTER) | 0.7074 | FEET |
| No. OF BAFFLES IN THIRD INTERVAL | 0.6857 | |
| BAFFLE SPACING | 2.8119 | FEET |
| LENGTH OF INTERVAL | 1.9281 | FEET |
| BAFFLE CUT-OFF (INNER) | 0.4516 | FEET |
| BAFFLE CUT-OFF (OUTER) | 0.7074 | FEET |
| No. OF BAFFLES IN FOURTH INTERVAL | 0.8250 | |
| BAFFLE SPACING | 2.7060 | FEET |
| LENGTH OF INTERVAL | 2.2324 | FEET |
| BAFFLE CUT-OFF (INNER) | 0.4516 | FEET |
| BAFFLE CUT-OFF (OUTER) | 0.7074 | FEET |
| No. OF BAFFLES IN FIFTH INTERVAL | 0.8329 | |
| BAFFLE SPACING | 2.9716 | FEET |
| LENGTH OF INTERVAL | 2.4752 | FEET |
| BAFFLE CUT-OFF (INNER) | 0.4516 | FEET |
| BAFFLE CUT-OFF (OUTER) | 0.7074 | FEET |
| No. OF BAFFLES IN SIXTH INTERVAL | 0.7672 | |
| BAFFLE SPACING | 2.8381 | FEET |
| LENGTH OF INTERVAL | 2.5150 | FEET |
| BAFFLE CUT-OFF (INNER) | 0.4516 | FEET |
| BAFFLE CUT-OFF (OUTER) | 0.7074 | FEET |
| OVERALL WEIGHT OF EXCHANGER | 2.1951 | TONS |
| MAXIMUM DIAMETER OF EXCHANGER | 2.6927 | FEET |
| OVERALL HEAT EXCHANGER LENGTH | 9.5558 | FEET |

**FIGURE 4.** Computer print-out for a typical heat exchanger design[12].

Complete dropwise condensation for a period of 2,000 hours has been attained by applying submicron films of paraxylene to chromium plated copper-nickle tubes. The use of submicron films of noble metals also shows promise.[5]

#### *Increasing Liquid Film Coefficients*

The stagnant liquid film inside heater tubes may be eliminated by the use of rotary scrapers. This is usually applied to the heating of viscous liquids in double walled exchangers although fluids of 0.5 to 25,000 cP are recorded. The rotating scraper maintains a thin highly agitated film in contact with the wall. Heat transfer coefficients are increased and over-heating eliminated. Hence this type of heater is suitable for thermally unstable liquids.

Direct contact liquid-liquid heat exchangers are receiving increasing interest. The Bradford Institute of Technology have surveyed recent developments.[6] The unit comprises basically a mixer-settler. If it was applied to juice heating, a suitable heating medium (e.g. a light oil) would be heated by steam and mixed directly with the juice. After decantation, the oil would be recirculated through the steam heater. The method is particularly suited to the heating of corrosive or highly fouling liquids.

**Discussion**

Considering that most of the heat consumed in a sugar factory is absorbed by process demands, the optimization of heat exchanger design and performance is important. An essential basis for the attainment of optimum conditions is a thorough knowledge of heat transfer coefficients and fouling factors under varying conditions. In spite of the fact that in other industries the optimization of heat exchangers has advanced to the stage of design selection by means of computers, in the sugar industry there are few factories where even records are kept of heat exchanger performance. This paucity of information precludes the accurate optimization of exchangers and often confuses the choice of alternative design conditions due to the unknown influence of operating variables.

In spite of the absence of practical data, the application of chemical engineering techniques has been shown to provide not only fairly reliable estimates of heat transfer coefficients but also to indicate the influence of the many variables upon which these coefficients depend. As an example, the overall heat transfer has been estimated from the physical properties of the juice, steam and pipe wall using the Nusselt method. Taking fouling factors from overseas tests, calculated estimates have been made with a fair degree of confidence.

It has been shown that the assumption of 30 per cent loss in overall coefficient to account for fouling outside the tubes would enable even closer agreement between the calculated and measured overall coefficients. This suggests that this loss of 30 per cent due to fouling outside the tubes after one year's operation (as measured overseas) could also occur in local heaters. For this reason chemical cleaning outside the tubes would probably allow for the installation of smaller heaters.

The Nusselt equation indicates that the most important single variable determining the inside (juice) film coefficient is the juice velocity. For this reason and to reduce scaling, the juice velocity should be maintained at the recommended value of 5 to 6 ft per sec[10]—unless sufficient information is available to show by means of economic balance calculations that the increased pumping costs prove the optimum economic velocity to be lower.

The dependence of the overall coefficient on the juice velocity for a typical fouled primary heater may be calculated using equation (1) and the data in table 1 as

$$U_{od} = \frac{1}{0.00388 + 1/(271u^{0.8})}$$

where u is the velocity of the liquid through the tubes in ft/per sec. Thus, for velocities of 3 and 6 ft per sec the respective overall coefficients would be 185 and 210 Btu/(hr)(sq ft)(°F) or an increase of 14 per cent. On the other hand, if the tubes were perfectly clean then the above equation would become

$$U_{oo} = \frac{1}{0.00112 + 1/(271u^{0.8})}$$

and for velocities of 3 and 6 ft per sec the corresponding overall coefficients would be 375 and 500 Btu/(hr) (sq ft)(°F) the increase being 33 per cent.

From the above calculation it is clear that the effect of juice velocity on heat transfer is only really appreciable when the heater is reasonably clean. An important inference from this conclusion is that only *clean* heaters have an appreciable amount of potential self regulation. It was shown earlier that only a small (13 per cent) difference exists between the coefficients of fouled vertical and horizontal heaters, the difference becoming significant (28 per cent) for clean heaters.

An important conclusion from the above is that specification of the maximum heating surface required to perform a given duty is quite simple provided sufficient safety margin is allowed so that the heater may operate when fully fouled and at low velocities. Such variables as: vertical or horizontal, high or low juice velocity, etc., may then be conveniently neglected and the heater manufacturer may justly claim that his heaters are equal in performance to any others on the market. However, by taking this line of least resistance it is quite possible that the resulting oversized heaters are operated at a relative economic loss. Furthermore, the tendency would be to allow the accumulation of an abnormal degree of fouling (particularly outside the tubes) before cleaning. This in turn would result in reduced controlability of the juice temperature. Since the majority of local heaters have necessarily been installed without a substantial basis of practical data on heat transfer coefficients and a knowledge of the effect of juice velocity and fouling, it may be assumed that they are generally designed with a generous margin of safety. There is, therefore, every reason to believe that the initiation of a programme for the tabulation and correlation of relevant data would facilitate the reduction of juice heater costs and the elimination of such anomalies as the fruitless operation of heavily fouled heaters at excessive velocities.

Regarding the recovery of waste heat, the very existence of the economic relationships expressed by equations (12), (13) and (14) indicates that recovery of heat is economical up to a point and thereafter the cost of recovery increases beyond the marginal steam cost. In South Africa, maintenance costs are relatively low and consequently the economic recovery limit may be higher than in other countries. For this reason the optimum recovery point must be determined from a knowledge of local conditions and not based on empirical data from overseas. This again would necessitate a more elaborate system for process data retrieval. The resulting rationalisation of the design and operation would logically lead to a significant reduction in production costs.

## Summary and Conclusions

It has been shown that the application of general chemical engineering techniques to heat transfer problems associated with the sugar industry provides data which agree with practical experience. Detailed calculations based on these techniques have shown that many interesting conclusions may be drawn regarding the most economical design and operating conditions for heat exchangers.

It is suggested that in the absence of both detailed practical performance data and calculated estimates, heat exchangers are necessarily oversized to allow a margin of safety for the unknown effect of numerous operating variables. To substantiate this remark it has been calculated that the heater used as an example in this report had a clean heat transfer coefficient of 450 Btu/(hr)(sq ft)(°F) but due to fouling (inside and outside) that determined by measurement was only 157. In spite of this, the required juice temperature was still attained.

Such heavy scaling has been shown to detract from the self regulation of the heater. For example, an increase of 3 to 6 ft per sec in juice velocity results in 33 and 14 per cent increase in overall coefficient for the clean and fouled heater, respectively.

Calculations relating to horizontal and vertical heaters have shown that 28 per cent additional heating surface would be required on a vertical heater when clean but only 13 per cent when fouled, since the outside film coefficient becomes less significant as fouling factors increase. Fouling masks the effect of operating variables and hence it is easier to design for fouled performance than for optimum performance.

Calculations have been presented for determining the economic limit of waste heat recovery. A certain degree of control is necessary for the maintenance of optimum economic conditions. The economy of heat transfer equipment is under constant investigation as shown by the abundance of literature. Various methods are being tested for increasing both inner and outer film coefficients and eliminating scaling. The design of heat exchangers for accurate optimum conditions is now processed by programmed computers in ten seconds. In order that the sugar industry take full advantage of recent developments towards increased heat transfer economy it is essential that a system be established for the retrieval of operating data to provide the basis for optimum design and operation.

## Nomenclature

B Concentration of sugar solution, degrees brix

c Specific heat, Btu/(lb)(°F); $c_c$, for cold fluid; $c_h$ for hot fluid

D Diameter of pipe, ft; $D_i$, inside diameter; $D_o$ outside diameter; D′, diameter in inches; $\bar{D}_L$ logarithmic mean

E Incremental heat exchanger cost, R/sq ft

G Mass Velocity, lb/(sq ft)(hr), through tube cross-section; $G_o$, based on outside tube area for condensation

g Acceleration of gravity, ft/(hr)$^2$

H 114rE/UY; $H_t$, total cost of supply incremental heat, R/MMBtu

h Individual or film heat-transfer coefficient, Btu/(sq ft)(hr)(°F); $h_i$, inside tube; $h_o$, outside tube; $h_h$, for horizontal tube; $h_v$, for vertical tube; $h_d$, for fouling factor; $h_{oo}$ outside clean tube

k Thermal conductivity, Btu(ft)/(sq ft)(hr)(°F); $k_f$, at mean film temperature; $k_m$, of tube wall

L Length of tube, ft

m Maintenance and repair allowance costs, fraction per year

N Average number of tubes in a vertical stack directly in line for horizontal heater

P $(t_{h1} - t_{h2})/(t_{h1} - t_{c1})$, fractional approach of hot fluid temperature difference to difference in entering temperatures of the two fluids

q Heat transfer rate, Btu/hr

R $W_hC_h/W_cC_c$

r $(m + 1/T_m)$, fraction/yr — fraction of total annual charges on cost/sq ft of surface to allow for maintenance, depreciation and acceptable minimum profit

t Temperature, °F; $t_c$, of cold (heated) fluid; $t_h$, of hot (heating) fluid; $t_1$, at inlet; $t_2$ at outlet; $t_f$, mean film temperature; $t_w$, of tube wall; $t_s$, steam condensing temperature

$T_m$ Maximum acceptable payout time before taxes and depreciation, years

U Overall heat transfer coefficient, Btu/(sq ft)(hr)(°F); $U_o$, based on outside tube area; $U_i$, based on inside area; $U_d$, for fouled heater; $U_{oo}$, for clean heater based on outer surface

u Fluid velocity through tubes, ft/sec

W Flow rate of fluid, lb/hr; $W_h$, for heating fluid; $W_c$, for heated fluid

x Thickness of resistance to heat flow, ft; $x_w$, of the tube wall

Y Equipment on steam time, fraction/yr

*Greek Letters*

$\Delta t$ Overall temperature drop, $t_h - t_c$, °F; $\Delta t_i$ between wall and fluid outside tube

$\lambda$ Latent heat of condensation, Btu/lb

$\mu$ Viscosity, lb/(ft)(hr); $\mu_f$, at mean film temperature; $\mu_w$, at wall temperature

$\rho$ Density, lb/cu ft; $\rho_f$, at mean film temperature

$\phi$ Ratio $(\mu/\mu_w)^{0.14}$ to correct for wall temperature

*Dimensionless Groups*

$N_{Nu}$ = Nusselt number, hD/k

$N_{Pr}$ = Prandtl number, $c\mu/k$

$N_{Re}$ = Reynolds number, $Du\rho/\mu = DG/\mu$

**Bibliography**


1 Anon., "Heat Transfer Coefficients and Scaling in Juice Heaters", Research Inst. Mackay Tech. Rep. No. 19, 1954, 4.

2 *Ibid.*, p 10–11.

3 Brown, G. G., "Unit Operations", 1st ed., p 442, Wiley, New York, 1950.

4 Chapman, A. J., "Heat Transfer", 1st ed., p 290–291, Macmillan, New York, 1960.

5 Erf and Thelen, "Dropwise Condensation", paper presented to First International Symposium on Water Desalination, Oct.–1965, U.S.A.

6 Hanson, C. and Ingham, J., "Direct Contact Liquid-Liquid Heat Exchange", Br. Chem. Eng. **10** (6) 1965, 391–399.

7 Happel, J., "Chemical Process Economics" 1st ed., p 189–191, Wiley, New York, 1958.

8 Honig, P., "Principles of Sugar Technology", Part 1, 1st ed., p 27, Elsevier, Neth., 1953.

9 Hugot, E., "Handbook of Cane Sugar Engineering", 1st (English) ed., p 313, Elsevier, Neth., 1960.

10 *Ibid.*, p 315.

11 *Ibid.*, p 320.

12 Kirkley, D. W. and Momtchiloff, I. N., "Heat Exchanger Design Programme for Optimization Studies", Br. Chem. Eng., **10** (1) 1965, 31–33.

13 Landt, H. E., "Zur, Viscositat von reinen Zuckerlösungen", Zucker, **6** (23) 1953, 558–563.

14 McAdams, W. H., "Heat Transmission", 3rd ed., p 188, McGraw-Hill, New York, 1954.

15 *Ibid.*, p 345–347.

16 *Ibid.*, p 418.

17 *Ibid.*, p 430.

18 McCabe, W. L., and Smith, J. C., "Unit Operations of Chemical Engineering," 1st ed., p 438–444, McGraw-Hill, New York, 1956.

19 *Ibid.*, p 473–477.

20 Osment, B. D. J., Tudor, D., Spiers, R. M. M., and Rugman, W., "Promoters for the Dropwise Condensation of Steam", Trans. Instn. Chem. Engrs. **40** (161) 1962, 152–160.

21 Perry, J. H., "Chemical Engineers Handbook", 4th ed., Chap 10, p 14, McGraw-Hill, New York, 1963.

22 *Ibid.*, p 22–105 to 22–107.

23 Pidoux, G., "Formal Zur Berechnung der Viscositat im Bereich," Zucker, **14** (20)1 961, 523–532.

24 Ryley, J. T., "Controlled Film Processing", Ind. Chem., **38** (448) 1962, 311–319.

25 Stoever, H. J., "Heat Transfer—Conduction, Radiation and Convection", Chem. and Met. Eng., **51** (5) 1944, 98–107.

25 Ten Broeck, H., "Economic Selection of Exchanger Designs", Ind. Eng. Chem., **36** (1) 1964, 64–67.

27 Webre, A. L., "Duplex Unit Juice Heaters", Sugar y Azucar, **56** (1) 1961, 35–37 and 58.


**Appendix**

*Viscosity of Sucrose Solutions*

The viscosity of pure sucrose solutions at various temperatures and concentrations may be represented according to Pidoux[23]) by a series of linear plots on a logarithmic ordinate and special temperature as abscissa. Consequently, only two values of viscosity for one concentration are required to determine all values at various temperatures from the plot. Such a plot is shown in fig. 5 for which the data of Pidoux[23]) and Landt[13]) were used. The temperature coefficient $\phi$ whose values correspond to the temperatures (t) on the abscissa is calculated from

$$\phi = t \times 10^3/(t + 273.16)^2$$

*Thermal Conductivity of Sucrose Solutions*

The thermal conductivity of sucrose solutions at various temperatures and concentrations has been tabulated by Honig.[8]) However, due to the presence of several obvious errors and the fact that extreme accuracy is unnecessary for industrial scale calculations, a linear multiple regression analysis was carried out in order to express the thermal conductivity (k) in terms of concentration (B) and temperature (t). The small inaccuracy incurred due to the slight non-linearity of the t *vs.* k relationship is not significant for industrial calculations. The value of k in Btu(ft)/(sq ft)(hr)(°F) may be calculated from t in °F and B in weight per cent from:

$$k = 3.61 \times 10^{-4}\,t - 1.96 \times 10^{-3}\,B + 0.322$$

*Specific Heat of Sucrose Solutions*

The specific heat changes very little with temperature variations. For industrial calculations in sugar factories Hugot has proposed the following formula for the calculation of specific heats of sugar liquors (c) in Btu/(lb)(°F) from the brix (B):

$$c = 1 - 0.006\,B$$

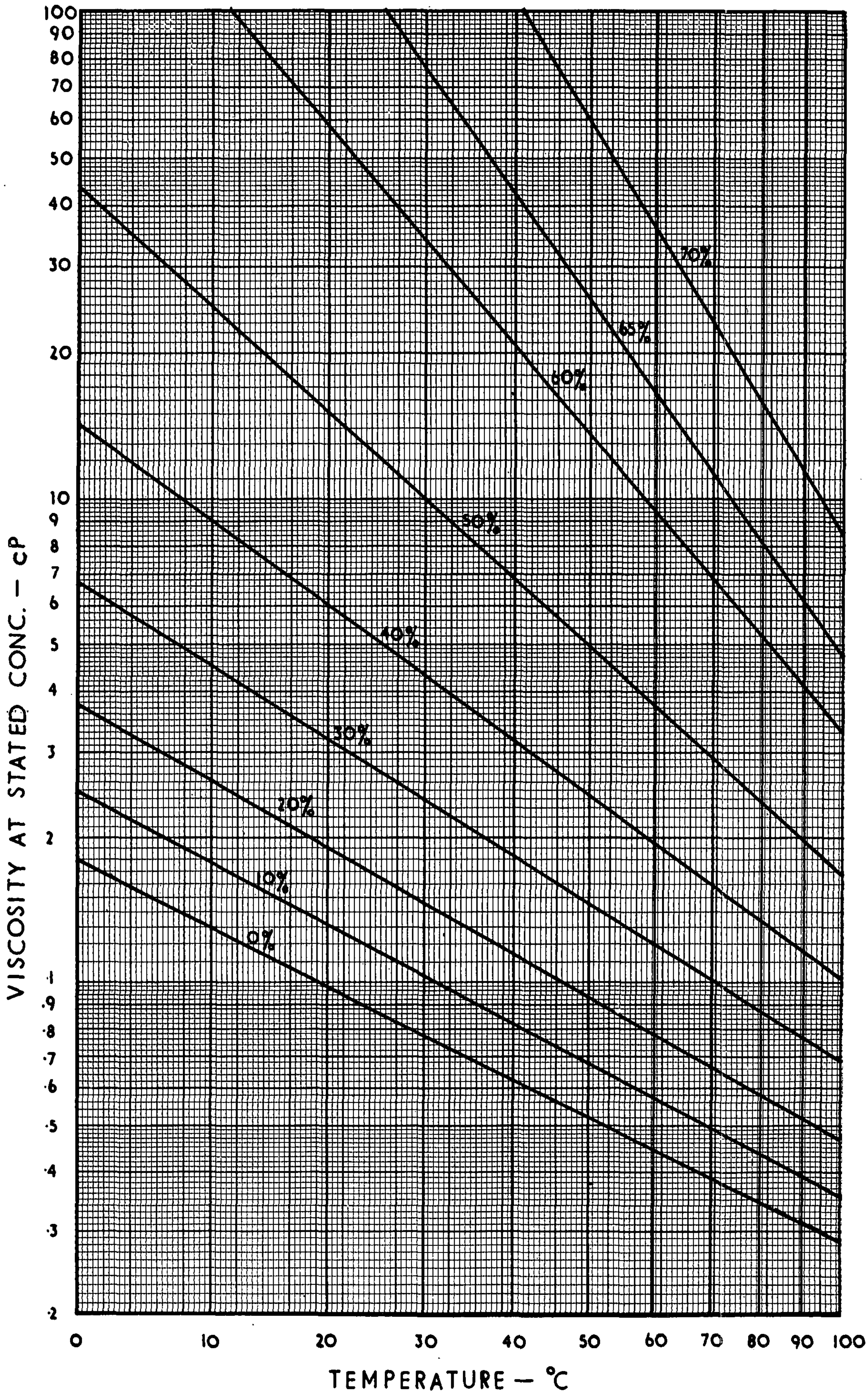


**FIGURE 5.** Graph for the evaluation of sucrose solution viscosities at various temperatures and concentrations[23].

**Mr. Hulett:** What effect do stripping plates on the tubes have on vertical juice heaters?

**Mr. Buchanan:** I have no figures from practical measurements, but obviously the condensate accumulation as it runs down the tubes will be reduced and hence the condensate film resistance will be reduced by the use of stripping plates.

There is a calculation to show that for the outside film coefficient of vertical and horizontal tubes to be equal

$$h_{ov} = \frac{793 \times (12)^{\frac{1}{4}}}{L^{\frac{1}{4}}} = 1,453$$

$$\text{whence } L — 1.066$$

This indicates that for a vertical heater to be equal in efficiency to a horizontal unit, stripping plates would be required at intervals of one foot along the length of the tubes.

**Mr. Wagner:** Is it actually possible to control juice temperature by controlling the condensate flow? We have tried unsuccessfully to use this type of control at Pongola. Can you cite an example of the application of this system?

**Mr. Buchanan:** As stated in this paper, this type of control is unstable and is not regarded as good practice. I think Mr. Gunn may provide a further practical example.

**Mr. Gunn:** We have not had satisfactory results from juice temperature control by condensate throttling controllers. I would like to add that I feel that the calculations for the factors in this paper appear more academic than practical. After all, a heater installed in a factory must be able to maintain temperatures for a week's run under the fouling conditions.

**Mr. Buchanan:** You are quite right in that a heater must be designed to cope with fouling over a certain period, however, many heaters are allowed to scale so badly and still attain the required temperatures that one wonders if they have not been oversized. It is the turn-around time between cleanings that I am questioning and I feel that some benefit could be derived by investigating the economic optimum cleaning frequency against the installed heating surface.

Concerning the practical aspect of the empirical calculations for individual and overall heat transfers coefficient I disagree entirely that these are of academic interest only. I have pointed out their limitations but one of the purposes of this paper has been to show by calculation that these coefficients compare well with measured data in practice. The empirical formulae are based on practical data from a wide field and are used for design purposes in the absence of such practical data. These formulae provide an essential basis for the prediction of coefficients under different operating conditions. In the absence of specific performance data from practical measurements these formulae provide the only means for resolving controversies regarding vertical and horizontal heaters, etc.

**Mr. Young:** Were the figures for thermal conductivity of sucrose solutions taken from Honig?

**Mr. Buchanan:** They were taken from a table in Honig's book and were subjected to a multiple regression analysis in order to provide the formula relating thermal conductivity at different temperature and concentration levels.

Some of his figures were inaccurate, possibly due to printing errors, and the formula eliminates these errors.