Title

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Publication Date
1978-06-01

MASTER

To be presented at the Energy Technology
Conference and Exhibition, Huston, Texas,
November 5-9, 1978; and to appear in EMERGING
ENERGY TECHNOLOGIES, W. B. Bradley, Editor,
American Society of Mechanical Engineers

LBL-7067
Conf-781112--3

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June 1978

Prepared for the U. S. Department of Energy
under Contract W-7405-ENG-48

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Heat Exchanger Design - "Why Guess a Design Fouling Factor When it Can Be Optimized?"

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Abstract

A new general surface heat exchanger design relationship is derived that *uniquely* relates the optimum design fouling resistance and the optimum design heat transfer coefficient with the ratio of cleaning cost to capital plus operating costs, at the optimum design condition. Implementation of this simple result to practical problems in design, however, requires numerical techniques. A new shell and tube heat exchanger design program, SIZEHX, is applied to a problem of current interest to confirm the derivation. SIZEHX can cost effectively perform single-step, *multiparameter cost optimizations* on single phase or supercritical exchanger arrays with variable fluid properties and arbitrary linear fouling for single-pass, segmentally baffled shell-and-tube configurations for a variety of fluid pairs, including hydrocarbon mixtures. The economic influence of several general design parameters on a geothermal exchanger are presented in the form of 3-D computer generated plots.

[1] Member ASME

Nomenclature

a = constant, the linear rate of increase of fouling resistance with time, $(Wyr/m^2K)^{-1}$

A = area, m^2

b = constant, residual fouling resistance after cleaning, $(W/m^2K)^{-1}$

C_A = installed cost per unit outside area, $\$/m^2$

C_{CL} = cost per cleanout per unit inside surface area, $\$/m^2$

C_{DT} = process downtime cost per cleanout per unit inside surface area, $\$/m^2$

d = tube diameter, m

D = shell diameter, m

ff_{ij} = local tube side fouling factor, $(W/m^2K)^{-1}$

FCR = fixed charge rate, $(yr)^{-1}$

H = baffle window height, m

ℓ = tube length between baffles, m

L = length of each tube, m

$\dot{m}$ = mass flow rate, kg/s

N_D = design cleaning frequency or number of exchanger cleanouts per year, $(yr)^{-1}$

N_s = number of shells in series, in a series-parallel network

N_p = number of parallel shell streams

N_T = number of tubes per shell

P.F. = matrix packing factor, dimensionless

P = pressure, MPa

ΔP = pressure drop, MPa

Q_D = exchanger duty, W

R_D = design fouling resistance, $(W/m^2K)^{-1}$

$R(\Theta)$ = total thermal resistance at time Θ due to fouling, $(W/m^2K)^{-1}$

S = tube pitch, m

$\Delta t\bar{m}$ = overall mean temperature difference, K deg.

Δt_{pp} = pinch point temperature difference, K deg.

T = temperature, K

T_{RES} = resource temperature, K

T_{TURB} = turbine inlet temperature, K

T.D.S. = total dissolved solids

U_C = clean overall heat transfer coefficient, W/m^2K

U_D = design overall heat transfer coefficient, W/m^2K

$U_{\Theta=0}$ = the overall heat transfer coefficient under clean startup conditions, W/m^2K

U_Θ = overall heat transfer coefficient at time Θ after cleaning, W/m^2K

V = velocity, m/s

X_A = exchanger annual capital investment, $/Btu and $/yr

X_{CL} = annual cost of exchanger cleaning ($X_{CL} \equiv X_{CLC} + X_{CLDT}$), $/Btu and $/yr

X_{CLC} = annual cost of cleaning the exchanger including the cost of labor and chemicals but not the process downtime, $/Btu and $/yr

X_{CLDT} = annual cost of process downtime chargeable to the exchanger for cleaning, $/Btu and $/yr

X_{Pi} = annual cost of tube-side pumping power, $/Btu and $/yr

X_{Po} = annual cost of shell-side pumping power, \$/Btu and \$/yr

X_{TOT} = total annual heat supply costs, \$/Btu

$X_{TOT'}$ = exchanger total annual cost (excluding the brine cost), \$/yr

X_{UF} = annual brine cost, \$/Btu

ε_{CL} = cleaning effectiveness, dimensionless

Θ_D = design operating time between cleanouts, yr

Subscripts

D = design

e = exit (i.e., T_{ie} is the tube side exit temperature, K)

i = inside

j = local zone value (i.e., T_{Bij} = brine local bulk temperature, K)

o = outside

OPT = optimum

S = shell

Θ = time, yr

Objective

The objective of the report is to develop and quantify a general relationship between exchanger and process economics and exchanger design parameters for a heat exchange step in an energy conversion process.

We will show that at the optimum economic condition, a simple product function of the design overall heat transfer coefficient, U_D, and the design fouling resistance, R_D, is uniquely related to the exchanger and process economic assumptions. That is, for fixed assumptions regarding exchanger unit capital costs, process energy costs, exchanger unit cleaning costs, and the cost penalties of process downtime, *neither* the design U factor *nor* the design fouling factor are *arbitrary* (1) exchanger design variables.

In addition, through a numerical example on the primary heaters of a proposed geothermal binary cycle power plant (2) with a complex brine scaling model, we will determine the dimensionless sensitivities of the total cost of the plant heat supply to various optimizable parameters of the exchanger design. These sensitivities are displayed via computer-generated 3-D plots.

Introduction

Heat exchanger design is an extremely complex part of overall process economic design. With the recent escalation of fuel costs relative to equipment capital costs, and the large fraction of heat exchange cost in low-temperature, Rankine cycle processes such as geothermal (3), the economic evaluation of heat exchange subsystems becomes a critical part of overall process economics. At current and projected energy costs, it frequently will not suffice to "optimize" a heat exchanger with regard to simply the sum of the exchanger annual capital investment and the pumping power cost. For geothermal power plants with inherently low cycle efficiencies, for example, the cost of process heat and the penalty of process downtime are much larger, respectively, than exchanger annual capital investment and pumping power cost.

A case in point is the design of the primary heaters for a geothermal binary cycle power plant (2) on a moderate temperature (182 C), low salinity (≈15,000 ppm T.D.S.) resource. For a 50 MWe (net) plant with isobutane as the secondary working fluid operating at an 85% capacity factor, we find that the sum of the exchanger annual capital investment, X_A, and tube side and shell side pumping power costs, X_{Pi} and X_{Po}, is only about 13% of the total heat supply cost, X_{TOT}. X_{TOT} includes the exchanger annual capital investment, pumping power cost, brine cost, X_{UF}, and the cost of exchanger cleaning and downtime, X_{CLC} and X_{CLDT}. The brine cost dominates all other costs by about a factor of 4 (4).

Furthermore, for this process, the sensitivity of the total heat supply cost, X_{TOT}, to exchanger pinch-point temperature difference and cleaning frequency at the optimum condition, are both significantly greater than the sensitivity of X_{TOT} to the exchanger tube-side or shell-side velocities (or pressure drops). It is extremely important therefore that the pinch point and cleaning frequency be accurately determined.

Designers should remember that the two most important exchanger economic variables, the exchanger terminal temperature specification (or mass flow ratio) and the design fouling factor are normally *specified* to the exchanger manufacturer when requesting quotations. This places critical *exchanger* economic decisions squarely on the *process* designer.

After developing a general relationship for the optimum cleaning frequency for fouling, which is assumed to be a linear function of time, but an arbitrary function of temperature (or position), we will illustrate the ability of the SIZEHX code to "home in" on the optimum cleaning frequency in a single-step, *four parameter optimization* on the conceptual design of a geothermal exchanger.

Optimum Design Cleaning Frequency

In this section we develop a relationship for the optimum cleaning frequency (i.e., optimum design fouling factor) of an exchanger for which a linear fouling rate can be assumed.

Figure 1 is a plot of the decay with time of the overall heat transfer coefficient for a small, 4-tube heat exchanger array tested by the San Diego Gas and Electric Company (SDGE) on geothermal brine at Heber, California (5). It is obvious from this plot and others in Ref. 5 that a linear fouling model is quite adequate for brine at the Heber resource for low velocities. After 500 hours of testing, there was no indication of asymptotic fouling behavior (6,7) for this brine on carbon steel tubes at low velocities. The measured fouling rate is also a function of brine temperature, but this complexity need not be included in the derivation that follows. Subsequent numerical calculations include the temperature dependence, however, and will be presented as general verification of the analytical result.

Under the above and similar circumstances the overall heat transfer coefficient, U_{Θ}, based on outside tube area, can be characterized by

$$\frac{1}{U_{\Theta}} = \frac{1}{U_{\Theta=0}} + R(\Theta), \tag{1}$$

where $U_{\Theta=0}$ is the overall heat transfer coefficient under clean startup conditions, and $R(\Theta)$ is the total thermal resistance at time Θ due to fouling.

If we define the design overall heat transfer coefficient, U_D, as the overall heat transfer coefficient (based on outside area) when the fouling resistance is R_D, U_D can be related to the clean coefficient, U_C, for the given exchanger through

$$\frac{1}{U_D} = \frac{1}{U_C} + R_D = \frac{1}{U_C} + b + a\Theta_D = \frac{1}{U_C} + b + \frac{a}{N_D}, \tag{2}$$

where Θ_D is the design operating time between cleanouts, b is the residual fouling resistance after cleaning, a is the rate of increase of fouling resistance with time, and N_D is the assumed design cleaning frequency.

<u>Objective Function</u>

For fixed terminal temperatures (i.e., brine cost immaterial) the exchanger total annual cost, $X_{TOT'}$, (excluding the brine cost) is the objective function to be minimized. This can be characterized by

$$\begin{aligned} X_{TOT'} &= X_A + X_{Pi} + X_{Po} + X_{CL} \\ &\equiv X_A + X_{Pi} + X_{Po} + X_{CLC} + X_{CLDT} \end{aligned} \qquad (\$/yr) \tag{3}$$

where X_A is the exchanger annualized capital investment,

X_{Pi} is the annual cost of tube-side pumping power,

X_{Po} is the annual cost of shell-side pumping power,

X_{CLC} is the annual cost of cleaning the exchanger including the cost of labor and chemicals but not process downtime,

and X_{CLDT} is the annual cost of *process* downtime *chargeable to the exchanger* for cleaning.

With the above definitions, Eq. (3) can be rewritten as

$$\frac{X_{TOT}'}{A_o} = C_A \cdot FCR\left[1 + \left(\frac{X_{Pi} + X_{Po}}{X_A}\right)\right] + N_D\left(C_{CL} + C_{DT}\right) , \qquad (3a)$$

where A_o is the exchanger outside surface area, C_A is the installed cost per unit outside surface area, FCR is the fixed charge rate, C_{CL} is the cleaning cost per cleanout per unit inside surface area, C_{DT} is the process downtime cost per cleanout per unit inside surface area, and N_D is the number of exchanger cleanouts per year. Expressing Eq. (3a) in terms of exchanger duty, Q_D, and mean temperature difference, $\Delta\bar{tm}$, gives

$$U_D\left(\frac{\Delta\bar{tm}}{Q_D}\right) X_{TOT}' = C_A \cdot FCR\left[1 + \left(\frac{X_{Pi} + X_{Po}}{X_A}\right)\right] + N_D\left(C_{CL} + C_{DT}\right) . \qquad (3b)$$

Substitution of Eq. (2) into Eq. (3b) yields

$$\left(\frac{\Delta\bar{tm}}{Q_D}\right) X_{TOT}' = \left[\frac{1}{U_C} + b + \frac{a}{N_D}\right]$$
$$\left\{C_A \cdot FCR\left[1 + \left(\frac{X_{Pi} + X_{Po}}{X_A}\right)\right] + N_D\left(C_{CL} + C_{DT}\right)\right\} . \qquad (3c)$$

The optimum cleaning frequency, $N_{D_{OPT}}$, for minimum exchanger total annual cost can now be determined from Eq. (3c) by partial differentiation with respect to N_D and setting the result equal to zero. This differentiation implies that quantities such as X_{Pi}, X_{Po}, and X_A in Eq. (3c) are *independent* of the cleaning frequency, N_D. However, Figure 1 shows that the rate of fouling, a, (Eq. (2)) can, in general, be a function of velocity, and, therefore, X_{Pi} may not be totally independent of N_D.

If we assume here that the rate of fouling at low velocity is *independent* of velocity, the result is;

$$\left(N_D\right)^2_{OPT} = \frac{aU_C C_A \cdot FCR\left[1 + \left(\frac{X_{Pi} + X_{Po}}{X_A}\right)\right]}{\left(1 + b\,U_C\right)\left(C_{CL} + C_{DT}\right)} . \tag{4}$$

Equation (4) expresses the optimum design cleaning frequency $\left(N_D\right)_{OPT}$ in terms of the initial clean coefficient, U_C; the linear fouling rate, a; the residual resistance, b; the unit economic factors C_A, C_{CL}, and C_{DT}; the fixed charge rate, FCR; and the ratio of the exchanger annual operating costs exclusive of cleaning to the annualized capital investment. Equation (4) is *similar* but not identical to that obtained by Mueller (8) and recently reported by Neill (9). While Eq. (4) is quite useful, a more fundamental result can be obtained. For example, if we define cleaning effectiveness, ε_{CL}, by

$$\varepsilon_{CL} = (R_D - b)/R_D = 1 - b/R_D , \tag{5}$$

then

$$R_D = \frac{a}{\varepsilon_{CL} N_D} , \tag{5a}$$

and recalling from Eq. (2) that

$$\frac{1}{U_C} = \frac{1}{U_D} - R_D = (1 - U_D R_D)/U_D , \tag{2a}$$

then with Eqs. (5a) and 2a), the first group on the right-hand side of Eq. (4) becomes

$$\frac{aU_C}{1 + bU_C} = \frac{N_D \varepsilon_{CL} U_D R_D}{1 - \varepsilon_{CL} U_D R_D} . \tag{6}$$

In addition, recall that

$$\frac{C_A \cdot FCR}{\left(C_{CL} + C_{DT}\right)} \equiv \frac{C_A \cdot FCR \cdot A_o N_D}{\left(C_{CL} + C_{DT}\right) \cdot A_i \cdot \left(\frac{d_o}{d_i}\right) \cdot N_D} \equiv N_D \left(\frac{X_A}{X_{CL}}\right) \tag{7}$$

Now, substituting Eqs. (6) and (7) into Eq. (4) for $N_D \equiv \left(N_D\right)_{OPT}$, rearranging, and *assuming* X_A, X_{Pi}, and X_{Po} are *optimum* values, we obtain

$$\left[\frac{X_{CL}}{X_A + X_{Pi} + X_{Po}}\right]_{OPT} = \left[\frac{\varepsilon_{CL} U_D R_D}{1 - \varepsilon_{CL} U_D R_D}\right]_{OPT} \tag{4a}$$

Finally, if we invoke Eq. (5a) and further simplify, the result is

$$\left[\frac{X_{CL}}{X_A + X_{Pi} + X_{Po} + X_{CL}}\right]_{OPT} = \varepsilon_{CL} \left(U_D R_D\right)_{OPT}$$

$$= \left[\frac{aU_D}{N_D}\right]_{OPT} . \tag{4b}$$

We can see from Eq. (4b) that *at the optimum design condition*, the optimum design overall heat transfer coefficient and the optimum design fouling resistance are *uniquely determined* for a given set of cleaning cost, capital cost, and process energy cost assumptions.

In physical/economic terms, equation (4b) simply states that at the optimum design condition, the ratio of the cleaning cost to total cost (excluding the brine cost) is equal to the ratio of the fouling resistance to the total (series) resistance.

This general dimensionlsss relationship of explicit design parameters reconciles, in an extremely simple form, *numerous* underlying detail design parameters that contribute to an overall optimum economic design.

This underlying detail will typically include:

1. Temperature and density dependent thermodynamic and transport properties,
2. possible change of phase,
3. consideration of flow regimes,
4. chemistry, heat, mass, and momentum flux influences on fouling, including corrosion,
5. complex exchanger configurations and materials,
6. process economic forces.

Because of the foregoing complexity, the practical application of Eq. (4b) in design obviously requires numerical techniques. In order to *satisfy* Eq. (4b) in new situations, several independent design parameters generally must be manipulated. Multiparameter optimization capabilities become a *practical necessity.*

It should be obvious that the exchanger computer model must be capable of simulating these *real devices* to better than first-order accuracy.

We believe Eq. (4b) is a general result that can be used to select reasonable $U_D R_D$ products for exchangers in process design at the conceptual level when critical economic decisions are made and plant capacity factors are selected.

Equation (4b) could also be used to update or expand the range of typical (suggested) fouling resistances in Section 9 of the TEMA Standards (10) using measured U factors and economic data from currently operating industrial plants.

Recent Developments of the GEOTHM Code

We have developed a new shell-and-tube heat exchanger design program, SIZEHX (11). When used with the powerful multiparameter optimization capabilities of the GEOTHM code (12-14), SIZEHX allows the simultaneous determination of the exchanger optimum terminal temperatures, cleaning frequency, tube diameter, and tube-side and shell-side pressure drops. Therefore, optimum values of X_{CL}, X_A, X_{Pi}, X_{Po} (and X_{UF}) can be computed to satisfy (4b) in the general case where terminal temperatures are not all known. SIZEHX can be used for the conceptual design of single phase and supercritical single pass exchanger arrays with single segmental baffles (4).

SIZEHX is a new zoned exchanger routine, with efficient convergence algorithms allowing cost effective computations on N_s/N_p series/parallel arrays for a variety of fluid pairs, including fluid mixtures with variable properties and arbitrary linear fouling rate models. Conventional, simple Colburn type correlations are used for heat transfer with zone-local coefficients and dimensionless groups.

SIZEHX incorporates a simplified form of Tinker's method (15) for characterization of shell-side performance--an LBL extension (11) of the work of A.P. Fraas (16); Starling's modified BWR equation (17) for thermodynamic properties of hydrocarbon fluids; Keenan and Keyes' equation for water (18); and transport properties developed by H.J.M. Hanley of NBS (19). Optimizations utilize the nonlinear programming methods of Davidon, Fletcher, and Powell (20) applied in GEOTHM by Pines and Green (12).

Selecting an Example for Numerical Calculations

We now seek to verify Eq. (4b) using the multiparameter optimization features of SIZEHX, choosing a suitably complex heat exchanger example of current, general interest. We have selected the 425 MWt primary brine-to-isobutane exchanger array from conceptual design studies (2) for a proposed 50 MWe (net) geothermal binary cycle power plant on the Heber resource.

For this brine the fouling rate is temperature and velocity dependent and at low velocities the fouling resistance changes by about a factor of 30 over the temperature range of economic interest.

This example is appropriate because

1. The fouling, though moderate for geothermal systems, is relatively well known and linear, yet complex.
2. The isobutane thermodynamic and transport properties vary over wide limits due to close operating proximity to the critical point.

3. The cycle process states have been optimized.
4. The contractor's exchanger conceptual design is documented, manufacturers quotes have been obtained, and economic assumptions are known (2).

This example is also interesting because, although the shell side film is *generally* controlling, tube side fouling is *locally* controlling at the cold brine exit end.

Approach

To perform the economic optimizations on the *primary heat exchanger subset* of the complete binary cycle of Ref. 2, we use the same approach as that reported in Ref. 4. We assume that preliminary overall process economic calculations have determined near-optimum *working fluid temperatures* and that the *primary heater* can be "singled out" for subsystem optimization by fixing the resource (well head) temperature and the working fluid states.

In this example, however, rather than specifying the working fluid *upstream pressure*, we specify the primary exchanger working fluid *downstream* pressure (and temperature). This fixes the turbine states for fixed condenser inlet state and the cycle *gross* power output for specified working fluid mass flow rate. This newer procedure makes the cycle net energy and cost normalization calculations more direct.

Assumed Fixed Process States

For this study the inlet brine (simulated as pure H_2O) state is fixed at 455.37 K (360 F) and 1.06 MPa (139 psig). The isobutane (shell side) mass flow rate is fixed at 1028 kg/sec (8.160×10^6 lb/hr). The isobutane temperatures and pressures are fixed at:

T_{in} = 319.26 K (115 F)

T_{out} = 422.04 K (300 F)

P_{out} = 4.137 MPa (600 psia)

The foregoing values from Ref. 2 have been found to be very near optimum for the contractor's assumed process costs (Ref. 14).

Tube Side Fouling Factor Model

The tube side fouling factor distribution used in Ref. 2 was based on linear fouling and a 1-year cleaning frequency. Whether or not the velocity dependence on the fouling resistance indicated in Ref. 5 was incorporated into the design fouling factor model assumed in Ref. 2 is unknown. For this study, then, the tube side fouling factor distribution assumed is given simply by (Ref. 2)

$$
\begin{aligned}
ff_{ij} &= 1.761 \times 10^{-5}/N_D \text{ for } 405.4\text{ K} \leqslant T_{B_{ij}} \leqslant 455.4\text{ K} \\
ff_{ij} &= 1.937 \times 10^{-4}/N_D \text{ for } 353.2\text{ K} \leqslant T_{B_{ij}} \leqslant 405.4\text{ K} \qquad (8) \\
ff_{ij} &= 5.812 \times 10^{-4}/N_D \text{ for } 337.6\text{ K} \leqslant T_{B_{ij}} \leqslant 353.2\text{ K} ,
\end{aligned}
$$

where N_D is the design cleaning frequency in cleanouts per year and $T_{B_{ij}}$ is the "brine" local bulk temperature. Any dependence of ff_i on velocity is ignored. The cleaning effectiveness, Eq. (5), was assumed equal to 0.90, in these calculations, and the shell side fouling factor was assumed equal to zero.

Objective Function for Numerical Calculations

In the *numerical calculations* we define as "optimum" an exchanger for which the total annual cost, given by the following function, has been minimized:

$$X_{TOT} = X_A + X_{Pi} + X_{Po} + X_{CLC} + X_{CLDT} + X_{UF} \quad (\$/\text{Btu}) \tag{9}$$

Note that this is different from Eq. (3) in that, for generality, the brine cost, X_{UF}, has been included, and the costing is now done with SIZEHX on a dollar per Btu basis. With X_{UF} included the brine exit temperature is optimizable.

Optimizable Parameters

For the SIZEHX computations, the matrix geometry is fixed, and the tube outside diameter in all cases is 0.01905 m (0.75 in.). The following were specified as optimizable parameters:

1. The heat exchanger pinch point, Δt_{pp}. (K deg.)
2. The cleaning frequency, N_D. $(yr)^{-1}$
3. The tube side pressure drop, ΔP_i. (MPa)
4. The shell side pressure drop, ΔP_o. (MPa)

Economic Assumptions

(a) Capital Costs. The exchanger purchased cost used herein has been normalized to the $\$64.6/m^2$ ($\$6.00/ft^2$) indicated in Ref. 2 for the same shell side and tube side maximum pressures.

For computing the exchanger installed annualized capital investment, we used the factored estimate method described by Milora and Testor (Ref. 3) and assumed a direct cost factor (installation, instrumentation, piping, insulation, foundations, fireproofing, controls, etc.) of 1.77, an indirect cost factor (engineering and legal fees, contingency, overhead and escalation and environmental impact) of 1.71, and a fixed charge rate of 0.15.

(b) Utility Fluid Costs. The brine pricing method used in Ref. 2 was based on the cost of services approach in accordance with generally accepted practice in the oil industry. The 1976 price of Heber brine thus computed for the 50 MWe (net) binary plant was \$0.606/kJ (\$0.574 per million Btu). For our calculations, the brine cost has been normalized to the foregoing for the same brine flow rate and exchanger duty, Q_D.

(c) Cleaning Costs. The exchanger cleaning costs consist of labor and materials for the actual cleaning plus the cost of process downtime. These computations assume \$1.80/m^2 (\$0.167/ft^2) of inside surface for labor and materials (i.e., chemicals).

The cost of process downtime assumes the following: For one exchanger cleanout per year, the plant capacity factor is 85% and the value of process electrical energy is 35.2 mills/kWh (Ref. 2). For any other cleaning frequency, N_D, a new plant capacity factor was computed assuming 3.0 days of plant downtime per additional cleaning with a linearly adjusted process energy cost.

Note that this is not strictly correct in that the cost of a *plant* shutdown has been "charged" only to *the primary exchanger*, whereas other process equipment (not considered here) obviously requires periodic maintenance. Perhaps a better method would be to multiply X_{CLDT} in Eq. (3) or C_{DT} in Eqs. (3a), (3b), (3c), (4) and (7) by some fraction, β, which would represent the fractional cost of the subject exchanger to the cost of all equipment requiring maintenance at frequency, N_D.

(d) Pumping Power Costs. Pumping power costs herein have been normalized to the 35.2 mills/kWh previously mentioned from Ref. 2 for the same plant capital cost, process net energy (fixed gross energy), and capacity factor. We have assumed the adiabatic efficiency of the pumps was 80% and the mechanical efficiency of the motors was 95%.

Exchanger Configuration Assumptions

Because SIZEHX can currently only model shell side performance for counter-currently disposed, single segmental baffle configurations (Refs. 4, 11), we have assumed fixed values for the shell (inside) diameter to baffle spacing ratio, D_s/ℓ, and the corresponding baffle cut ratio, H/D_s, suggested by Fraas (Ref. 16).

The graphical results presented assume 1.905×10^{-2}m O.D. $\times$ 1.65 $\times 10^{-3}$m wall tubing (0.75 in. O.D. $\times$ 16 ga. wall tubing) and

$D_s/\ell = 1.0$

$H/D_s = 0.46$

$S/d_o = 1.25$ (equilateral triangular array).

The above baffle spacing ratio leads to excessive unsupported tube lengths for the subject exchanger except possibly for large diameter shells with a "no tubes in the window" bundle configuration. The significance of this is discussed later.

Graphical Results of Exchanger Design Optimization

Figure 2 is a sensitivity plot illustrating that Eq. (4a) and, therefore, Eq. (4b), are satisfied at the optimum cleaning frequency for this example, even though a complex step function fouling factor distribution, Eq. (8), was used as input, *and* the optimization included the pinch point delta T and tube and shell side pressure drops as optimizable parameters. The reader should note here that the powerful multiparameter optimization features of the SIZEHX/GEOTHM code have been used to *verify* Eq. (4b) for this complex example whereas, for tractibility in the *derivation* of Eq. (4b), we had to *assume* that X_A, X_{Pi}, and X_{Po} were optimum values. With other, arbitrarily selected, and widely different constant fouling rates and cleaning effectivenesses, the SIZEHX/GEOTHM optimization routines *repeatedly* converge upon minimum X_{TOT} with $U_D R_D$ consistent with Eq. (4b). We are confident that Eq. (4b) is general for linear fouling, and that similar simple, general results would be obtained for more complex (i.e., asymptotic) fouling models (Refs. 6,7).

Figure 3 is another sensitivity plot that clearly shows the relative importance of accurately specifying the optimum pinch point delta T (terminal temperatures) and cleaning frequency (design fouling factor), compared to the optimum tube and shell side pressure drops. This plot also shows in this case, for the four optimizable parameters chosen, it is a *little* safer to err on the high side--i.e., a higher pinch point (mean temperature difference), more frequent cleaning (or a *lower* design fouling factor) and higher allowable pressure drops (velocities).

Figures 4, 5, 6, and 7 clearly illustrate the total exchanger-brine subsystem *economic design space*. To produce these plots, an 8 × 8 array of exchanger designs was computed with the SIZEHX/GEOTHM code for wide ranges of two alternately selected values of the four optimizable parameters with the other two fixed at their previously determined optimum values.

In Figure 4, for example, $X_{TOT} = X_A + X_{Pi} + X_{Po} + X_{CL} + X_{UF}$ is plotted as a function of the tube side pressure drop and the shell side with the pitch point temperature difference fixed at its optimum value of 9.527 K and the cleaning frequency fixed at its optimum value of 0.968 cleanouts per year.

From Figure 4 we can see that the X_{TOT}-pressure drop surface is virtually flat with four shallow local minima (see Figure 5). Depending upon which one of these minima the optimizer converges upon (which in this instance depends on first guess and allowed step size), dictates in some measure the resulting L/D of the shells. For a minimum X_{TOT} at low tube side ΔP, for example, the shell will be relatively large in diameter and short. At a higher "optimum" tube side ΔP and the *same* matrix proportions, conversely, the shell will have fewer tubes and will be longer.

Therefore, Figure 4 and the corresponding contour plot, Figure 5, *clearly* show that if the terminal temperatures and design fouling factor are properly specified, current exchanger purchasing practices are acceptable. That is, allowing manufacturers reasonably wide latitude (consistent with process economics) for varying controlling side pressure drops or fluid velocities (within acceptable TEMA structural limits) for fixed D_s/ℓ and H/D_s is quite reasonable from a total energy cost point of view. Even though pumping power costs, X_{Pi} and X_{Po}, rise with increasing velocities, the total annual cost per Btu (*for fixed matrix proportions and baffle cut*) remains virtually constant due to a corresponding increase in U_D and reduction in annual capital cost, X_A. It follows, then, that even though the area requirements quoted by various proposers may be different (for the *same matrix* proportions and baffle cut) because of differences in manufacturers' design philosophy or selection criteria, the *total annual cost* to the purchaser will be relatively unaffected.

However, a *quite different* picture is obtained if the optimum pinch point and design fouling factor are not specified. In Figure 6 X_{TOT} is plotted (note the scale change) as a function of pinch point delta T and cleaning frequency, N_D. The corresponding contour plot is Figure 7. From these plots it is obvious that with either a poorly specified pinch point (terminal temperatures) or cleaning frequency (design fouling factor) the cost penalties can be quite severe. Even for the low, 1976 brine costs assumed here (Ref. 2), a very modest 2% increase in X_{TOT} above the minimum represents a $3.6M increase in operating cost of the brine-primary heater subsystem alone over the 25-year life of each of these 50 MWe geothermal power plants (see, for example, Figure 3).

Perhaps one of the most interesting surfaces in this series is a plot of the primary exchanger array overall heat transfer coefficient, U_D, as a function of tube side and shell side pressure drop. This is shown in Figure 8. In this plot the pinch point temperature difference, Δt_{pp}, and the cleaning frequency, N_D, have been fixed at their previously computed optimum values. Note how the shell side fluid (supercritical Isobutane) controls the designs. The local "ridges and humps" on this surface are the result of the radically varying thermodynamic (specific heat) and transport properties of Isobutane in the near critical region. These U factor peaks (local reductions in capital cost) correspond to the four shallow local minima previously indicated in Figure 4 and Figure 5. It is particularly important to note (Table 1, Example 1) that the minimum cost design (X_{TOT} minimized) selected is not constrained with a maximum U factor specification.

Configuration Details of SIZEHX Optimum Designs

The previous plots have illustrated the general economic design space for single segmental baffle exchangers with $D_s/\ell = 1.0$ and $H/D_s = 0.46$ in a geothermal binary cycle power plant. However, this matrix configuration, Example 1, turns out to be structurally marginal because of the large unsupported tube lengths.

In a recent SIZEHX documentation study (21), other, more appropriate, baffle spacings were investigated, but time did not permit the inclusion of plotted results here. Table 1 is a list of computed optimum design details for Example 1 and another example from Ref. 21 for $D_s/\ell = 2.0$. Costing assumptions are the same for both cases.

It is interesting to note that the SIZEHX computed optimum designs agree very well, in general, with what is stated and can be deduced from information in Ref. 2, except for U_D (and therefore A_o). The SIZEHX optimum design U_D for Example 2 is 30% lower than that stated in Ref. 2 and cannot be "driven" up to the stated value, "about 1419 W/m^2K (250 $Btu/hrft^2F$)," even if the pressure drops are set at the stated allowable process limits.

This 30% lower U_D should be no surprise--we've assumed a particular single segmental baffle configuration. The contractor may be assuming one of the newer multiple segmental or rod-baffle types which ostensibly achieve significantly higher heat transfer per unit pumping power.

If the above is true, the contractor has selected a heat exchanger overall design that is very near optimum.

Conclusions

A general functional relationship has been developed for linear fouling that describes in economic terms a unique *linear* relationship between the overall design heat transfer coefficient and the design fouling factor at the optimum condition. This relationship defines a necessary condition for an optimum surface heat exchanger design.

Some features of a new exchanger conceptual design aid, SIZEHX, were described. The state-of-the-art multiparameter optimization capabilities of the SIZEHX/GEOTHM code were used on a complex, general example of current interest to verify the derived results.

Graphical results presented illustrate the *general economic design space* for the primary heaters of a geothermal power plant and the cost sensitivity for four important exchanger independent variables.

The minimum cost of cleaning (fouling) for this geothermal application is found to be about 15% (Table 1, Example 2) of the exchanger capital plus operating costs but only about 2.2% of the total heat supply cost at 1976 brine prices--that is, exchanger fouling should not discourage the development of geothermal binary plants with surface exchangers on the Heber-like resources.

Incorrect pinch point temperature difference (terminal temperatures) or fouling factor specification, in general, however, can lead to costly exchanger over-design or inadequate performance. The potential for *significant savings* in the fabrication of future heat exchangers exists using the design aids and physical/economic principles described herein. However, little will be accomplished unless the sad state of affairs with regard to fouling characterization (6,7) is improved.

Acknowledgments

The authors wish to acknowledge the efforts of Lt. J.D. Williams, USAF (formerly with LBL) in the development of SIZEHX, L.F. Silvester for providing the thermodynamic and transport properties and the 3-D plots herein, K.F. Mirk for his continued leadership, and the Electric Power Research Institute for its permission to reproduce from EPRI 376 and to quote from EPRI-ER 301. This work was funded by the Division of Geothermal Energy of the U.S. Department of Energy.

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Figure Captions

Figure 1: Overall heat transfer coefficient vs. time for brine at the Heber resource (Ref. 5). Graph reproduced by permission of EPRI. The shell side fluid is distilled water.

Figure 2: Comparison of exchanger optimum cleaning frequency determined by SIZEHX and given by Eq. (4a). Example 1: $D_s/\ell = 1.0$, $H/D_s = 0.46$, $S/d_o = 1.25$, $d_o = 1.905 \times 10^{-2}$ m, carbon steel.

Figure 3: Sensitivity plot illustrating relative importance of pinch point temperature difference and cleaning frequency compared to pressure drop for the primary heater of a geothermal binary cycle power plant. Example 1: $D_s/\ell = 1.0$, $H/D_s = 0.46$, $S/d_o = 1.25$, $d_o = 1.905 \times 10^{-2}$ m.

Figure 4: Influence of tube side and shell side pressure drop on total heat supply cost, X_{TOT}. $D_s/\ell = 1.0$, $H/D_s = 0.46$, $S/d_o = 1.25$, $d_o = 1.905 \times 10^{-2}$ m, 1 Btu = 1.055×10^3 J.

Figure 5: Contour plot corresponding to Figure 4. Note the existence of 4 local minima and the insensitivity of X_{TOT} to tube and shell side pressure drop. The pinch point temperature difference and the cleaning frequency, N_D, have been set at computed optimum values of 9.53 K and 0.97 y^{-1} for plotting Figures 4 and 5.

Figure 6: Influence of pinch point temperature difference and design cleaning frequency on total heat supply cost, X_{TOT}. $D_s/\ell = 1.0$, $H/D_s = 0.46$, $s/d_o = 1.25$, $d_o = 1.905 \times 10^{-2}$ m (0.75 in.), 1 Btu = 1.055×10^3 J.

Figure 7: Contour plot corresponding to Figure 6. The tube side and shell side pressure drops have been set at computed optimum values of 0.033 MPa (4.79 psi) and 0.121 MPa (17.5 psi), respectively, for plotting Figures 6 and 7.

Figure 8. Overall heat transfer coefficient, U_D, as a function of tube side and shell side pressure drop. The pinch point temperature difference, Δt_{pp}, and cleaning frequency, N_D, have been set at their previously computed optimum values. Note how the shell side fluid (supercritical isobutane) controls the design.

Table 1: Effect of matrix configuration on SIZEHX optimum designs.

Variable		Example 1	Example 2	Notes
Documentation: Run (date)		POPE 001 (6/5/78)	POPE 017 (6/12/78)	
Input data				(1)
N_s, N_p		4,2	4,2	(2)
Matrix configuration		single segmental	single segmental	(3)
D_s/ℓ, H/D_s, S/d_o		1.0 0.46 1.25	2.0 0.25 1.25	(3)
P.F.		0.90	0.915	(3)
ε_{CL}		0.90	1.00	(3)
$\dot{m}_o$ (total)	(kg/s)	1028.	1028.	(2)
Computed Data				
Q_D	(MW)	425.96	425.96	
$\Delta t\bar{m}$	(K deg.)	13.96	14.62	
U_D	(W/m^2K)	1221.0	1001.2	
$R_D \times 10^4$	$(W/m^2K)^{-1}$	1.474	1.5114	
A_o (total)	(m^2)	24,985.	29,094.	
N_D	$(yr)^{-1}$	0.968	0.937	(4)
$\dot{m}_i$ (total)	(kg/s)	867.24	875.69	
T_{ie}	(K)	340.50	341.63	
Δt_{pp}	(K deg.)	9.527	10.084	(4)
V_i (max)	(m/s)	0.871	0.671	
L	(m)	18.06	16.04	
N_T		2889	3789	
D_o	(m)	1.618	1.835	
ΔP_i	(MPa)	0.033	0.039	(4,7,8)
ΔP_o	(MPa)	0.120	0.149	(4,7)
$X_{CL}/(X_A+X_{Pi}+X_{Po}+X_{CL})$		0.1637	0.1447	(6)
$X_{TOT} \times 10^6$	(\$/Btu)	0.659	0.678	(5,9)

(1) See text for "given inputs" from Ref. 2 and multitude of other assumptions including costing.

(2) From Ref. 2.

Table 1: (continued -- notes)

(3) Assumed values, see text.

(4) Optimizable parameter for these runs.

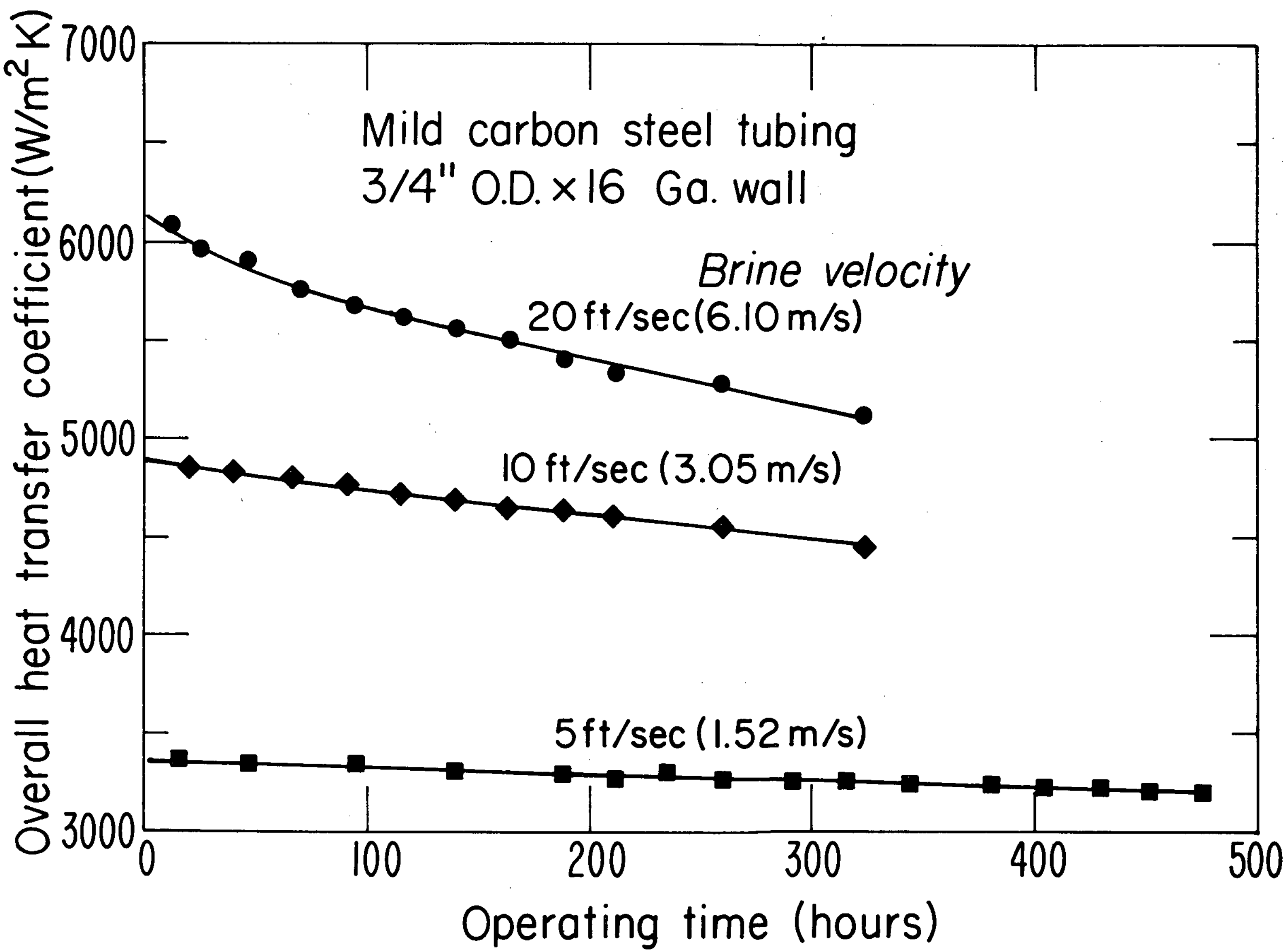
(5) Objective function for these runs.

(6) See Eq. (4b).

(7) Flange to flange; i.e., pressure drop of connecting plumbing ignored.

(8) Thickness of scale (fouling) ignored (Ref. 5).

(9) 1 Btu = 1.055×10^3 J.



XBL 786-2579A

Fig. 1

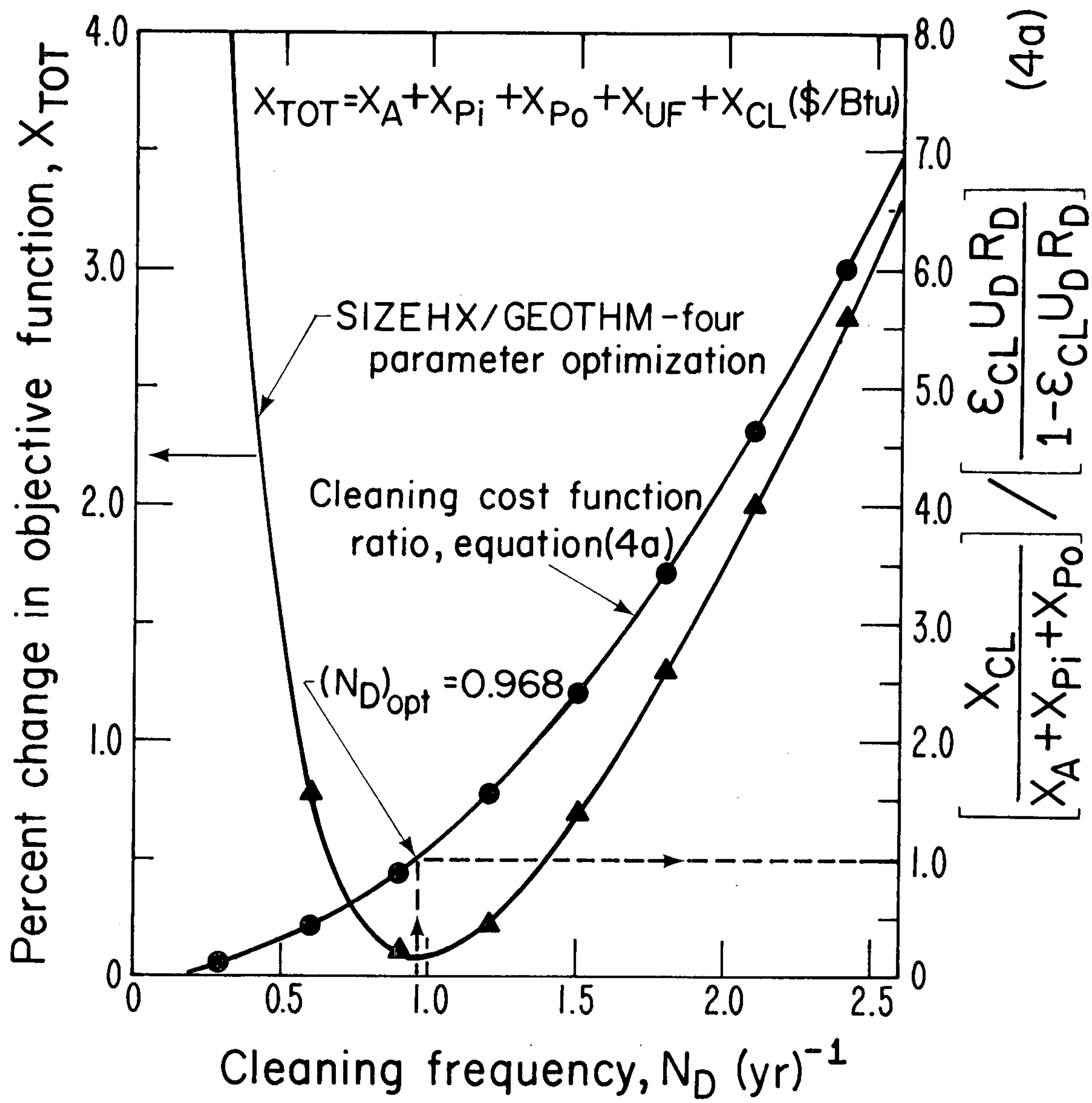


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Fig. 2

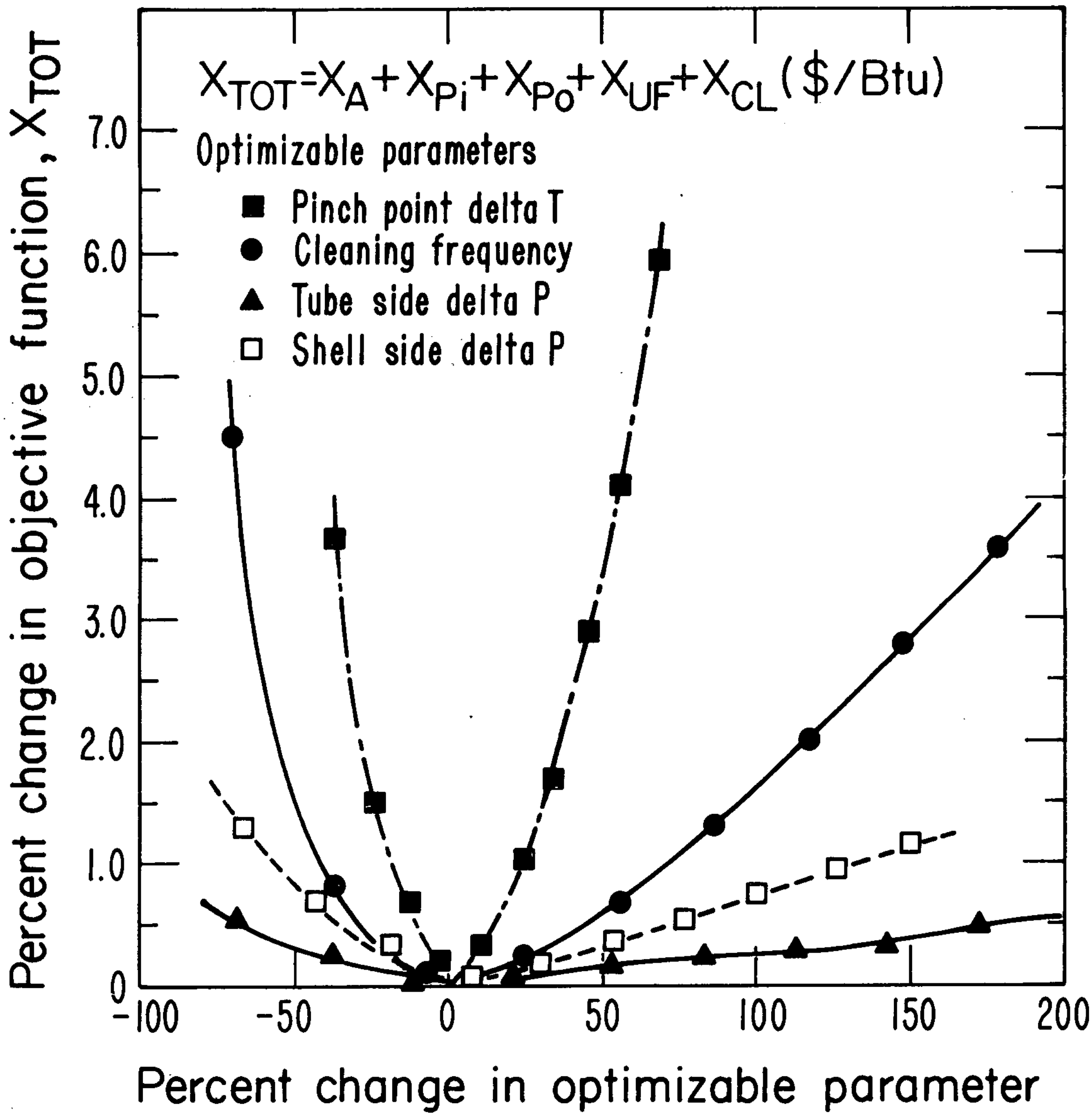

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Fig. 3

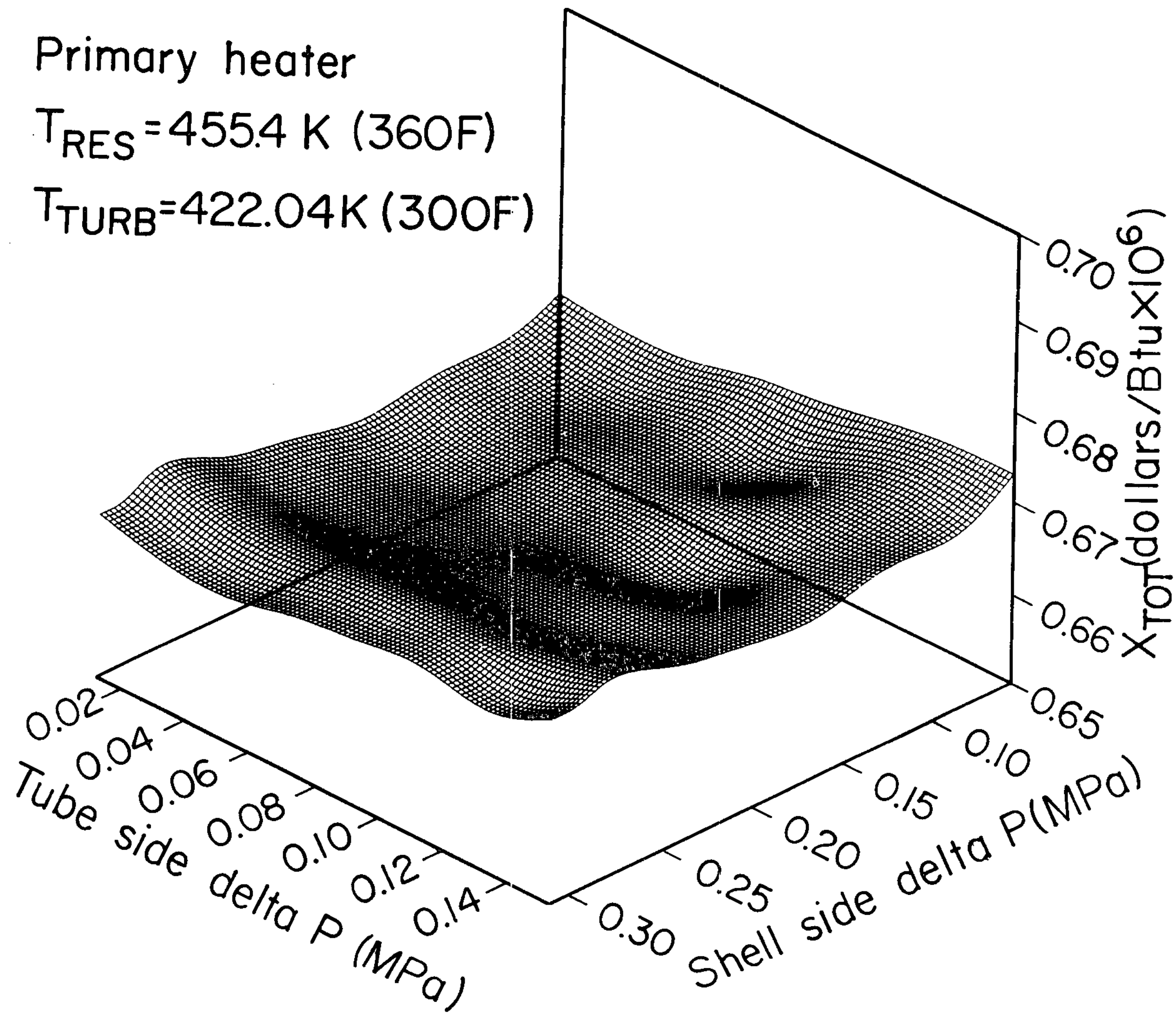

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Fig. 4

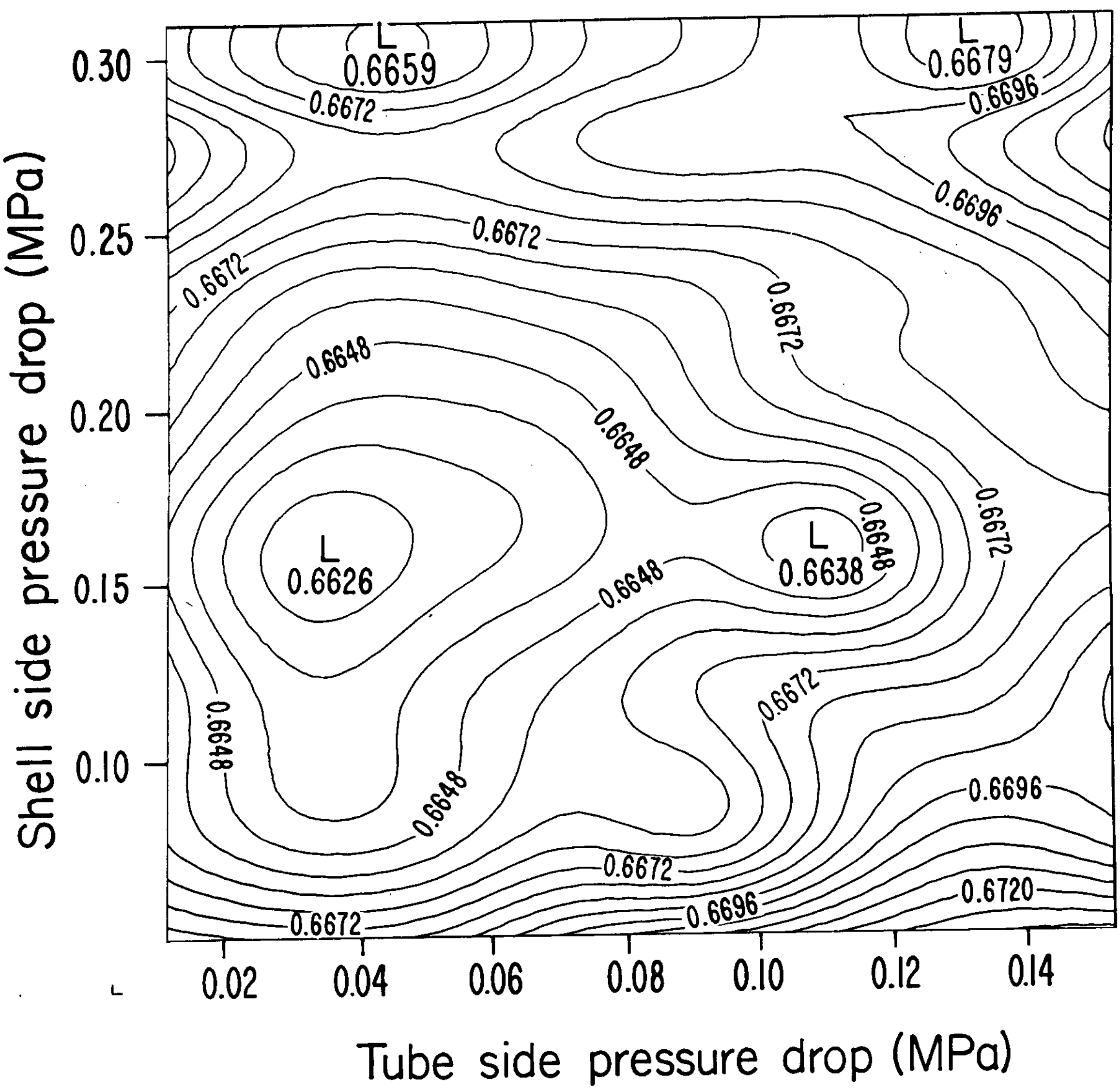

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Fig. 5

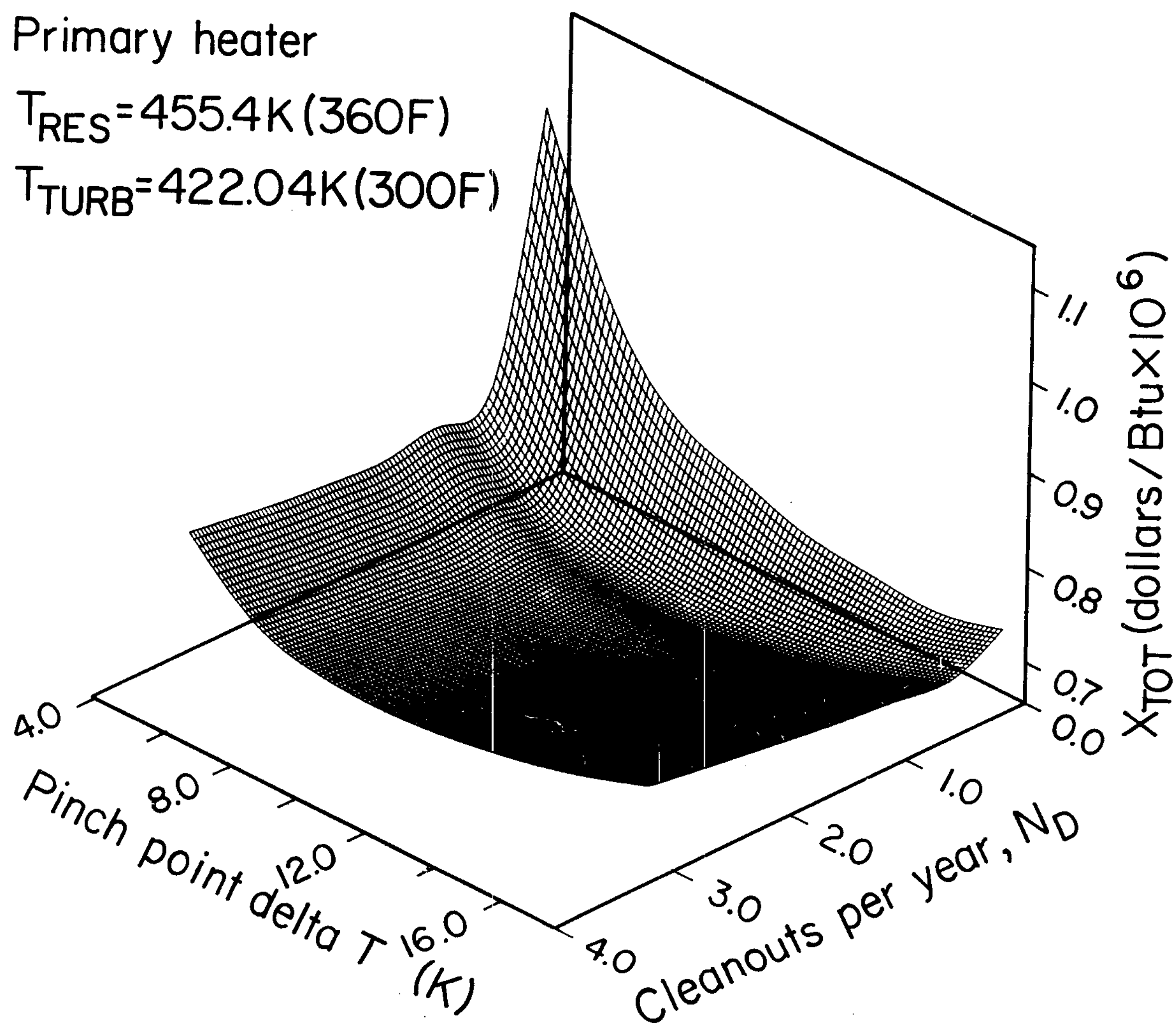

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Fig. 6

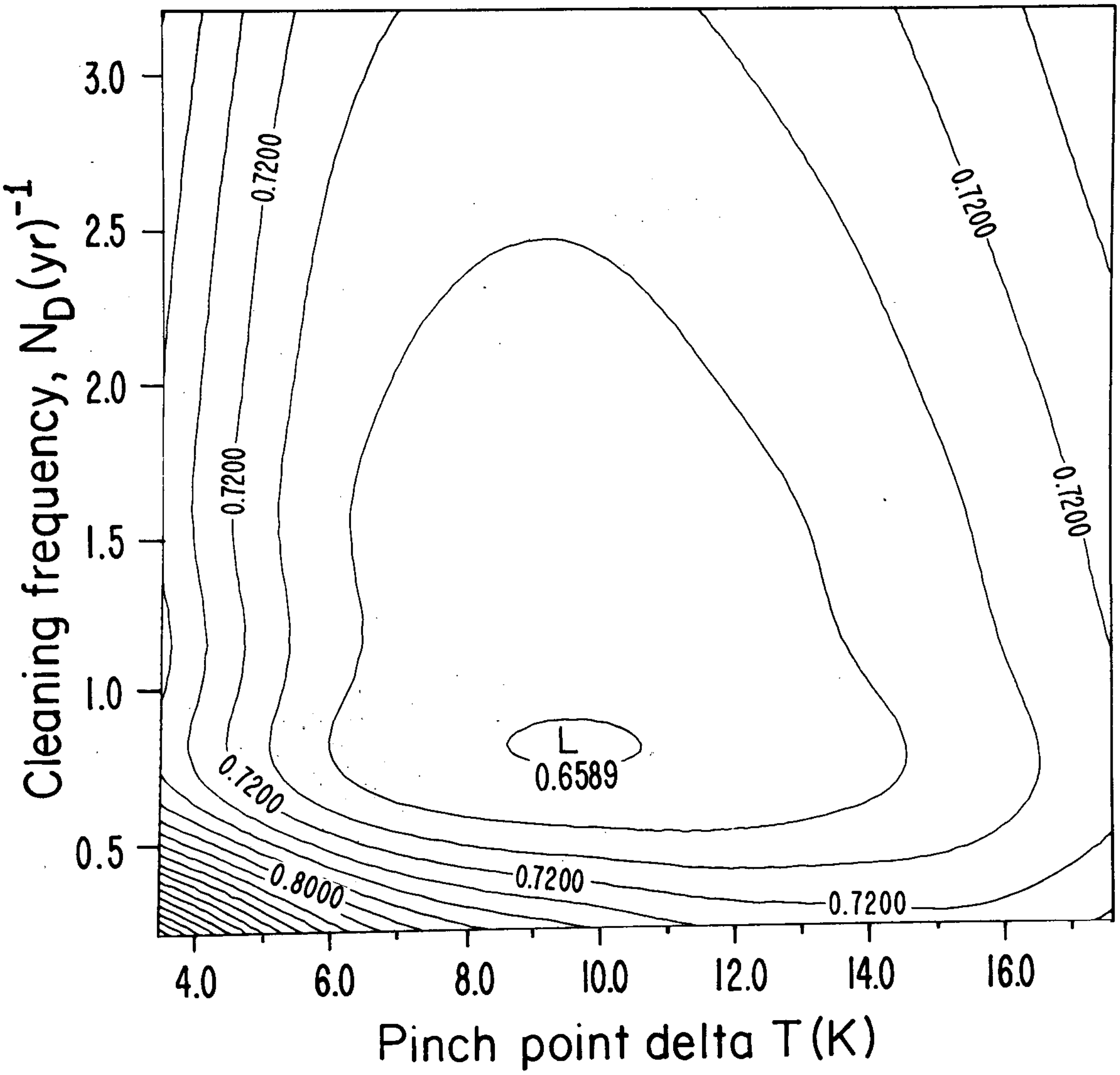

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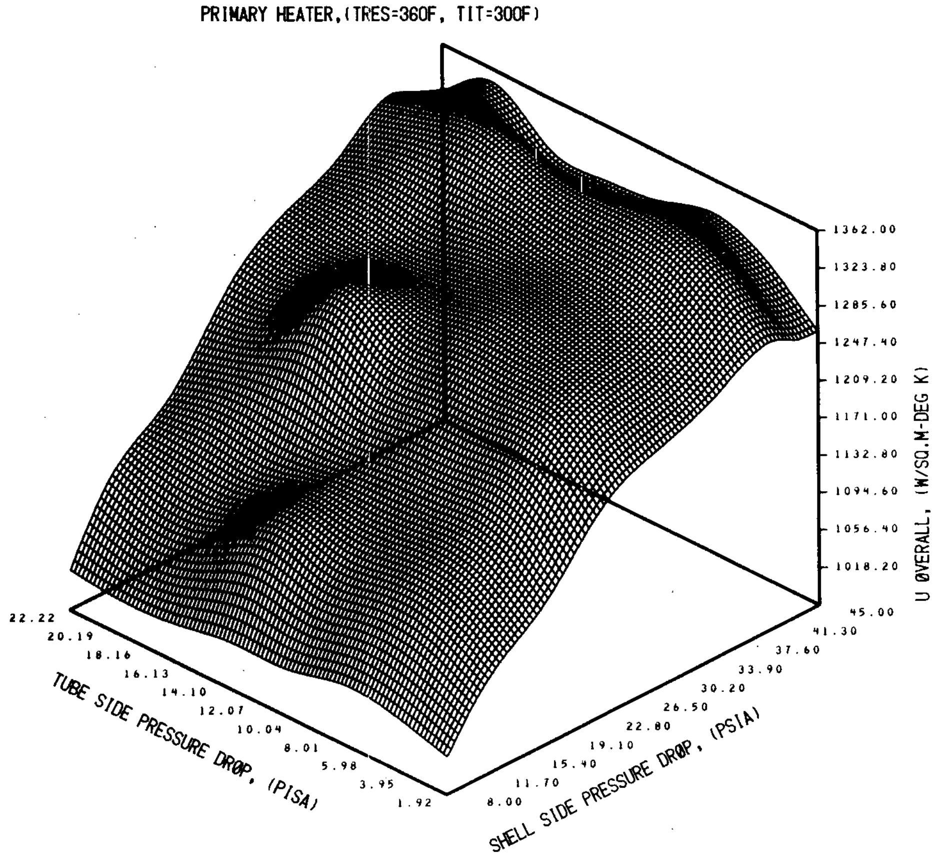
Fig. 7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1.92
TUBE SIDE PRESSURE DRØP, (PISA)
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26.50
22.80
19.10
15.40
11.70
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SHELL SIDE PRESSURE DRØP, (PSIA)

Fig. 8

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