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OPTIMIZATIONS OF GEOTHERMAL CYCLE SHELL AND TUBE EXCHANGERS OF VARIOUS CONFIGURATIONS WITH VARIABLE FLUID PROPERTIES AND SITE SPECIFIC FOULING

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ABSTRACT

A new heat exchanger design program, SIZEHX, is described. This program allows single step multiparameter cost optimizations on single phase or supercritical exchanger arrays with variable properties and arbitrary fouling for a multitude of matrix configurations and fluids. SIZEHX uses a simplified form of Tinker's method for characterization of shell side performance; the Starling modified BWR equation for thermodynamic properties of hydrocarbons; and transport properties developed by NBS. The report will include results of four parameter cost optimizations on exchangers for specific geothermal applications. The relative mix of capital cost, pumping cost, and brine cost ($/Btu) are determined for geothermal exchangers illustrating the invariant nature of the optimal cost distribution for fixed unit costs.

INTRODUCTION

Little information exists concerning general strategies for the optimization of heat exchangers. A few simple cases have appeared[1,2,3] which have one or more of the following drawbacks: (1) solutions are based on constant properties, (2) wall and fouling resistances are ignored or treated as constant, and (3) shell side pressure drop is based on ideal, pure cross-flow, friction data. Leakage (or bypass) factors must be assumed apriori.

In geothermal applications, specifically supercritical binary power plants using light hydrocarbons as secondary fluids in a Rankine cycle, the forgoing exchanger optimization schemes are seriously limited. For example, on even the most easily exploited, moderate temperature, low salinity (<20,000 ppm TDS) resources (1) brine scaling rates can change as much as a factor of thirty over the temperature and velocity range of economic interest,[4] and (2) the working fluid specific heat, density, thermal conductivity, and viscosity can change by factors of about three or more in the primary heat exchanger. These result in a markedly nonlinear overall heat transfer coefficient distribution. Because the working fluid in most supercritical Rankine cycles operates just slightly above the critical pressure, assumptions of linearity in these properties between terminal conditions can lead to significant errors. In addition, more complex fluids; i.e., mixtures of light hydrocarbons are being seriously considered[4] as "solutions" to problems of reservoir temperature decline for binary cycle power plants.

On Heber brine, the fouling resistance dominates at the exchanger cold end, and thus can have a *decisive* influence on the brine injection temperature and therefore the cycle design energy cost and resource utilization efficiency. Increasing the "design fouling factor" increases the optimum injection temperature, and therefore it must be prudently "selected" to ensure economically feasible designs. Obviously, one needs site specific scaling data to make good system design decisions. In addition, because exchangers are traditionally purchased based on *customer* specified terminal temperatures and fouling factors, the geothermal customer needs efficient heat exchanger design aids. A problem in many cases is that suitable codes are proprietary. Another problem is which code to use. A *third* problem is that the optimum injection temperature depends also on exchanger <u>cost</u> and these costs are not very well known.

A common dilemma arises when the customer goes out for preliminary exchanger quotes. It is the rule rather than the exception to not only receive quotes at a variety of unit costs, but also a variety of configurations all for the same *specified* service, duty, terminal temperatures, and fouling factors! On detailed comparison of the various proposals, it is also not uncommon to find that the design velocities and pressure drops are all quite different leading to differences in subsequent operating costs. How is the choice to be made? How significant are the differences? Unfortunately, even if the significance of the differences are not known, the customer usually has only two choices - he must either (1) accept the low bid (which might be one with the greatest pumping power requirements, thus compromising the cycle net power) or (2) throw them all out and go back to the drawing board.

Recent geothermal research activities, then, have revived a multitude of interesting and complicated problems in the ostensibly well known area of heat exchanger design. Because the exchangers in geothermal binary cycles represent the order of 30% of the total capital cost, it is imperative

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they be built with a minimum of over-design in both capital cost (area) and operating cost (pressure drop) while maximizing brine utilization efficiency. The problem then becomes a complex mix of trade-offs between several competing influences. Several optimizable parameters can be investigated to determine their sensitivity on the design objective. The purpose of this study is to explore some of the trade-offs in heat exchanger selection for geothermal binary cycle power plants.

THE GEOTHM/SIZEHX PROGRAM

We have developed a new heat exchanger design routine, SIZEHX (Ref. 5, Sec. 5) which can be used for exchanger conceptual design and optimization at an enhanced level of understanding. The SIZEHX code was developed to complement the GEOTHM[5] heat and mass balance routines and take advantage of GEOTHM's powerful multiparameter optimization[6] capabilities.

The SIZEHX code uses relatively complex computational algorithms with none of the previously mentioned simplifying assumptions with regard to constancy of properties and scaling. We have developed a practical cost optimization strategy which can be applied to geothermal exchangers to determine optimum terminal conditions, velocities, and design coefficients at the conceptual level prior to requesting exchanger quotes. Space does not permit a description of the SIZEHX code here; however, relatively complete documentation[5] is available.

COMPLETE CYCLE CHARACTERIZATION

Previous studies have shown[6] that to determine the overall optimum, or minimum energy cost, design for an assumed conversion process on an assumed resource, the complete geothermal plant and field must be characterized and optimized. However, to perform these complete cycle optimizations, one needs subsystem costs and overall design heat transfer coefficients as input. The design process, then, is iterative in nature, and at various points in time, the process designer must get updated heat exchanger and other subsystem quotes to be sure he is currently assuming realistic overall heat transfer coefficients and unit costs.

ISOLATING THE HEAT EXCHANGER FOR SUBSYSTEM OPTIMIZATION

For the purposes of this heat exchanger study, then, assume that preliminary overall process economic calculations have been done and that we can simply single out the primary heater of a supercritical isobutane binary cycle by fixing the resource temperature and working fluid states. We want to investigate in detail the effects of exchanger geometry, cost per unit area, and design fouling factor on total annual exchanger operating costs to establish design U's and optimum exit brine temperatures.

Specifically, we suggest that investigating the above effects on single tube pass exchangers assuming single segmental baffles will provide typical general performance for any well designed commercial exchanger.

We define as "optimum" an exchanger for which the following function has been minimized:

$$X_{TOT} = X_A + X_{Pi} + X_{Po} + X_{UF} \quad (\$/\text{Btu}) \qquad (1)$$

where

X_A	is the heat exchanger total capital investment	($/Btu)
X_{Pi}	is the cost of tube side pumping power	($/Btu)
X_{Po}	is the cost of shell side pumping power	($/Btu)
X_{UF}	is the utility fluid (brine) cost	($/Btu)

It is important to note here that we do not fix tube length, number of tubes, or velocities with SIZEHX. We specify tube diameter and mass rate, and the former are determined by zone heat balances and momentum balances over the exchanger length to satisfy the specified pressure drops.[5] These then are subsequently optimized along with the pinch point in accordance with the specified minimum total cost objective, Equation (1).

ASSUMPTIONS FOR THIS STUDY

For the purposes of this study, we have assumed the resource temperature is fixed at 455.37°K (360°F), the isobutane mass flow rate is known, and the inlet temperature and pressure and exit temperature are fixed at values established by Ben Holt.[4] These are:

$$\begin{aligned} T_{in} &= 319.26^\circ K\ (115^\circ F) \\ P_{in} &= 45.83 \text{ Bars } (650 \text{ psig}) \qquad (2) \\ T_{out} &= 422.04^\circ K\ (300^\circ F) \end{aligned}$$

In this study, all cases have assumed a 4/2 series/parallel exchanger array with external plumbing cost and external plumbing pressure drop ignored. We have normalized the brine cost ($/Btu) to values recently established by Holt[4] for the same brine flow rate and exchanger duty. We also assume the same tube side design fouling factors established by Holt from tests conducted by San Diego Gas and Electric, SDGE,[7] on brine at the Heber Resource. For carbon steel tubes, Holt's design fouling factor was characterized by (Ref. 4, p. 29):

$$\begin{aligned} R_f &= .0001\ ^\circ F/\text{Btu/hrft}^2 \text{ for } 270^\circ F<T_B<360^\circ F \\ R_f &= .0011 \quad \text{"} \quad \text{"} \quad 176^\circ F<T_B<270^\circ F \qquad (3) \\ R_f &= .0033 \quad \text{"} \quad \text{"} \quad 148^\circ F<T_B<176^\circ F \end{aligned}$$

Any dependence of brine velocity on the design fouling factor was ignored herein, but SIZEHX could easily deal with this if it were factored in.

All cases assume 1.77 and 1.71 for the exchanger direct and indirect cost factors, and a fixed charge rate of 20% to establish the exchanger annual capital cost. For pumping power, we have assumed plant power at 35.2 mills/kwh[4] with pump adiabatic efficiencies of 80% and motor efficiencies of 95%. In order to investigate exchanger geometry with a minimum of constraints, we assume tube and shell side pressure drops and brine flow rate "are material" and explore a variety of configurations and costs with the following as optimizable parameters:

1. The heat exchanger pinch point delta T (K°)
2. The tube side pressure drop (BARS) (4)
3. The shell side pressure drop (BARS)
4. The tube outside diameter (m)

All cases assume bare 16 gauge carbon steel tubes (no external surface). All cases assume an equilateral triangular tube matrix with a tube pitch to diameter ratio, s/d_o, equal to 1.25. The shell diameter to baffle spacing ratio, Ds/ℓ, is varied between:

$$1.0 \leq Ds/\ell \leq 5.0$$

and for reasonable corresponding baffle cut to diameter ratios, H/Ds, we adopt values suggested by A.P. Fraas (Ref. 8, p. 154).

RESULTS OF SIZEHX HEAT EXCHANGER OPTIMIZATIONS

Table 1 and Figures 1 and 2 are the results of our preliminary optimizations of primary heat exchangers for a supercritical isobutane binary cycle on a Heberlike resource.

Input values in Table 1 were selected to explore the effects of (1) shell side geometry and tube size, (2) exchanger cost per unit area and (3) tube side cleanliness. All cases except Case 6 assume Holt's design fouling factor. Cases 1, 2, and 3 cover a broad range of baffle cuts and spacings all at a fixed $6.00 per square foot and .75 in O.D. tube diameter. Cases 2, 4, and 5 explore a range of tube O.D.'s from the calculated optimum, 0.438 inch O.D., to 1.0 inch O.D. all for the same baffle cut and spacing ratios and cost per square foot. Cases 2, 7, and 8 illustrate the effect of increasing exchanger cost per square foot for fixed baffle cut and spacing ratio and tube O.D. Finally, cases 2 and 6 illustrate the influence of tube side fouling for the same fixed baffle cut and spacing ratio, tube diameter, and cost.

Table 1 shows that matrix geometry and tube O.D. have *virtually no effect* on the optimum brine exit temperature. This is determined by (1) brine cost, (2) capital cost, and (3) fouling factor alone for the assumed working fluid states. Another interesting point is that the optimum tube side and shell side pressure drops are virtually the same (cases 1,2,3) for the same tube diameter and cost per square foot *regardless* of baffle spacing ratios (for the Ds/ℓ and H/Ds pairs assumed here). The larger baffle spacings and cuts achieve higher overall heat transfer coefficients and fewer tubes through higher optimum brine velocities and simply result in shorter, more practical, shells. The total cost per Btu, X_{TOT}, decreases with increasing baffle spacing primarily by reducing the exchanger capital cost fraction, X_A/X_{TOT}.

In addition, as the cost per square foot increases (cases 2, 7, and 8), the optimum pinch point (mean delta T) and the tube side and shell side pressure drop increase obtaining higher economically justified design overall heat transfer coefficients and even fewer tubes through increased velocities, resulting in even shorter shells. As the cost per square foot increases, the capital cost fraction X_A/X_{TOT} increases, of course, but the optimum brine cost fraction X_{UF}/X_{TOT} decreases.

It should be pointed out, however, that although Case 1 (at $Ds/\ell = 1.0$) is shown as the lowest total (fouled) cost, this "design" might have to be thrown out for structural reasons - the shell inside diameter is about 45 inches here making the unsupported tube length to diameter ratio about 90/0.75 = 120. (The SIZEHX code currently performs no tube natural frequency calculations or other obviously important structural checks).

It should also be pointed out that our simplistic fixed unit cost assumption, dollars per square foot of *matrix surface*, can be misleading for significantly different numbers of tubes (drilling and rolling costs), significantly different lengths (baffle cost), and for significantly different shell diameters (shell cost).

For all the .75 inch O.D. tube, design fouling factor cases shown here, the number of tubes is between 1523 and 1674 and the shell diameters range from 45.7 in. diameter to 47.8 in. diameter (both relatively small ranges). For the .438 in. O.D. "optimum tube diameter" case, Case 4, however, the number of tubes is 6687 and the shell is about 55.5 in. O.D. For the 1.0 in. tube case, the number of tubes is 837 and the shell is 45.5 in. in diameter.

From the forgoing, then, the case with the most "optimistic" capital cost here is clearly Case 4 which is only of academic interest any way from the point of view of scale removal.

CONCLUSIONS

A new shell and tube heat exchanger design routine SIZEHX[5] has been used with simple cost assumptions to demonstrate the influence of matrix geometry, tube diameter, fouling, and cost per square foot on the optimum total annual cost, injection temperature, and brine velocities on supercritical geothermal exchangers optimized at a fixed brine cost.

The brine cost is found to dominate all other considerations in geothermal heat exchangers. A 2% reduction in X_{TOT} can achieve a $4 M savings over the life of each 50 MWe (net) geothermal binary power plant at 1976 brine costs in the primary exchanger/field subsystem alone.

The SIZEHX/GEOTHM[5] program has performed complex multiparameter optimizations believed to be impractical with previously documented codes. A *specified* design fouling factor was used in this work; however, SIZEHX is capable of determining the *optimum design fouling factor*, or optimum cleaning frequency, given the cleaning costs and the cost of plant-down time, assuming constant scaling rates.

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TABLE 1 SIZEHX Optimized Supercritical (1/1) Exchanger Designs (4/2 series/parallel arrays)

Sample Case	No. of Optim. Param-eters	Assumed Input Values			- CALCULATED OPTIMUM RESULTS - Optimizable Parameters													
		Ds/ℓ	H/Ds	$/ft^2	Δt_{pp} (C°)	ΔP_i (Bars)	ΔP_o (Bars)	d_o (in.)	Δt Mean (C°)	U_{ol} (B/hrft^2°F)	V_{brine} (max) (ft/sec)	Tube length (ft)	Brine exit temp. (°F)	X_A/X_{TOT} (%)	X_{pu}/X_{TOT} (%)	X_{uf}/X_{TOT} (%)	X_{TOT} X10^6 ($/Btu)	F/Q (%)
1	3	1.0	0.46	6.00	13.454	1.973	0.791	0.75 *	17.02	271.1	4.891	66.8	150.5	.093	.0180	.889	.632	.1105
2 Base-line	3	2.0	0.25	6.00	13.486	1.998	0.758	0.75 *	17.05	238.6	4.795	74.4	150.5	.104	.0176	.878	.640	.1092
3	3	5.0	0.16	6.00	13.46	1.991	0.796	0.75 *	17.03	202.1	4.637	85.0	150.6	.121	.0164	.862	.652	.1114
4	4	2.0	0.25	6.00	13.651	1.906	0.598	0.438	17.18	247.5	4.676	29.3	150.5	.100	.0156	.884	.635	.0961
5	3	2.0	0.25	6.00	13.443	1.967	0.803	1.00 *	17.01	223.5	4.745	115.	150.6	.111	.0178	.872	.645	.1111
6 "clean" ≠	3	2.0	0.25	6.00	9.598	2.305	0.865	0.75 *	13.12	315.2	5.052	80.9	139.6	.107	.0205	.873	.613	.1218
7	3	2.0	0.25	12.00	14.02	2.208	1.266	0.75 *	17.68	263.0	5.156	69.1	154.1	.166	.0218	.812	.703	.1486
8	3	2.0	0.25	18.00	16.27	2.224	1.272	0.75 *	20.04	273.6	5.379	59.3	161.8	.196	.0207	.783	.757	.1521

All above except Case 6 assume the step function fouling factor from Ref. 4 on tube side and zero fouling factor on shell side.
≠ Assumes .0001 °F/Btu/hr ft^2 fouling factor on tube side - virtually clean.
* Specified parameter for these runs.

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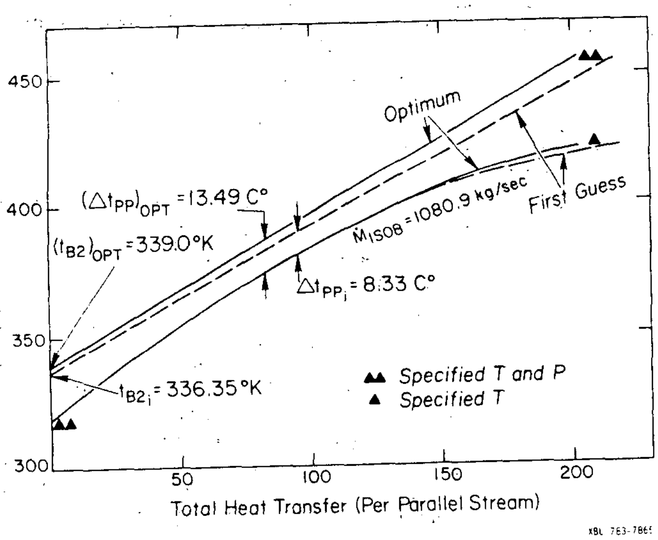

FIGURE 1 Typical T/Q plot (Case 2) illustrating change of brine temperature profile and pinch point shift in SIZEHX optimization.

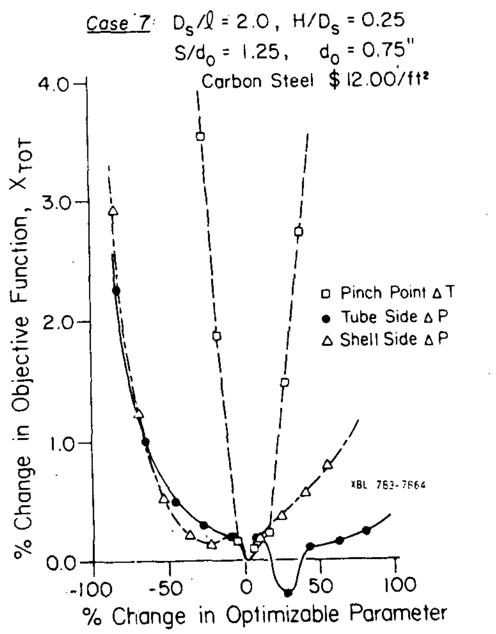

FIGURE 2 Typical sensitivity plot from 3 parameter optimization (Case 7). Note the strong influence of pinch point (brine cost) on the total cost, X_{TOT} ($/Btu).

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