THE UNIVERSITY OF MICHIGAN

INDUSTRY PROGRAM OF THE COLLEGE OF ENGINEERING

ECONOMIC DESIGN OF HEAT TRANSFER EQUIPMENT

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May, 1962

IP-565

UR0191

ACKNOWLEDGMENTS

The Computing Center of the University of Michigan is gratefully acknowledged for generously providing computer time so that this paper might be prepared. The author is also indebted to the Industry Program of the College of Engineering, The University of Michigan, for providing copies of this paper.

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INTRODUCTION

Heat exchangers, condensers, and reboilers, constitute a major capital investment and operating expense in most chemical plants and refineries. Rising costs and increased competition, therefore, are forcing companies to design and select heat transfer equipment in a more judicious manner. The design and selection process has been greatly improved by the availability and use of modern digital computers. Computers have not only relieved engineers from repetitious and laborious calculations but have permitted more elaborate and detailed analysis. Many trial designs and alternate approaches can now be considered using the computer at a fraction of the cost and time required for an engineer to do a single simple design. The engineer can thus be freed from the routine calculation and can be used more effectively in preparing data for the computer and in analyzing the computer results to obtain the most economical design.

The economic design of heat transfer equipment is complicated by both the large number of independent variables and the non-linearity of the equations describing heat transfer and pressure drop. A portion of the total number of variables involved is fixed by the physical limitations of the problem. In a condenser, which will be discussed in this paper, the following variables will be assumed fixed in the problem statement associated with the design: condensing load, inlet pressure, inlet saturated vapor temperature, type of tube side fluid, tube side fluid temperature, fouling rates, allowable pressure drops and materials of construction. A number of variables are classified as being freely choosable. Such variables are tube side fluid flow rate or tube side fluid velocity, tube dimensions, number of tube passes, tube length, tube arrangement, and tube type. When values are assigned to the free variables, the remaining variables encountered in the heat transfer and pressure drop equations become fixed. Such variables are the number of tubes, total heat transfer surface area, shell size, logarithmic mean temperature difference, tube side liquid flow rate (if velocity was fixed) or tube side fluid velocity (if flow rate was fixed) and all the heat transfer coefficients and pressure drops. To obtain the most economical condenser, the freely choosable variables must be selected in such a way that the sum of the resulting operating and fixed capital expenses on a yearly basis is a minimum. Finding the set of variables giving the minimum cost can be accomplished by search techniques using a digital computer.

SEARCH METHODS

Methods for finding the maximum or minimum value of analytic functions are well known. A common technique is the method of steepest ascent[1] or gradient method which assumes that the derivatives of the response function (cost) are continuous and that the independent variables have the same dimensions. For complicated systems procedures utilizing the calculus of variation may be employed. Such methods cannot be used, however, in the heat transfer design problem because the variables cannot be arranged to form an analytic function. The tube dimensions and number of tube passes can only have discrete values. Furthermore, if the design is in accordance with the "Standards of

Tubular Exchanger Manufacturers Association",[2] the tube length has fixed values and the tube pitch has a minimum value. The search methods used, therefore, must be satisfactory for the types of systems encountered. There are three such methods which can be easily used in conjunction with digital computers. They are the factorial method, the univariate method, and the random method.[3,4]

The factorial method is characterized by selecting the trial values of the free variables at points of a grid in factor space. Figure 1 exemplified a 3 x 3 factorial design in which only two of the free variables, water velocity and tube length are allowed to vary. The remaining variables are preset at some arbitrarily selected values. The trial values of velocity and tube length are selected to provide an even coverage of the selected region to be searched. The maximum search region or factor space is bounded in Figure 1 by the maximum and minimum tube lengths in one direction and by the minimum and maximum allowable water velocities in the other direction. In the 3 x 3 factorial design nine trial designs are required. The trial design giving the minimum cost is said to be the optimum design. As the number of free variables considered is increased, the number of trial designs increases rapidly. For example, if three tube lengths, three water velocities, two pass arrangements, and two tube diameters are considered, there would be 3 x 3 x 2 x 2 or 36 trials needed.

In the univariate method, as the name implies, the effect of changes in one variable at a time are examined. The variables to be studied are arranged in some order. Initial estimates are made for all the variables except for the first variable to be studied. Trial designs

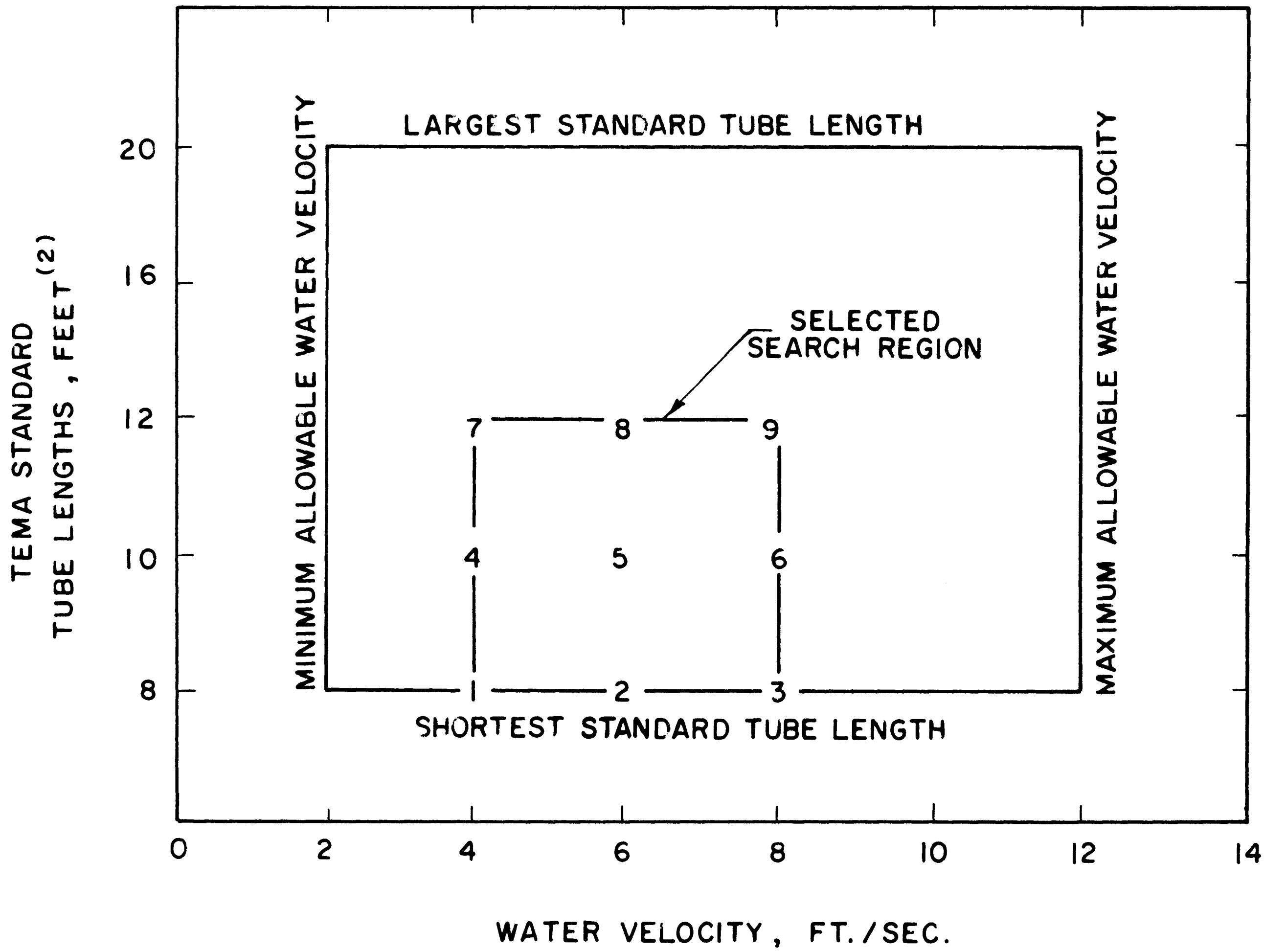

Figure 1. Factorial Search Pattern Where Only Tube Length and Tube Side Water Velocity are Considered as Variables.

are made at several values of the first variable until the best value is obtained with all the other variables being held fixed. Using the best value of the first variable studied, the second variable is studied in a similar fashion until the best value of that variable is obtained. The method is extended until all the free variables have been examined individually. The process can then be repeated in its entirety or repeated with a reduced number of variables until the desired level of attainment is reached. Figures 2 and 3 show typical univariate schemes. Figure 2 represents the first level of search and Figure 3 represents the second level of search utilizing the information obtained in the first level.

A third procedure is the random method. This method is characterized by selecting the values of the free variables for each trial design in a random fashion. The set of randomly selected variables giving the minimum cost heat exchanger is said to be the optimum. Figure 4 shows the random trial points for a heat transfer system in which only velocity and tube length are allowed to vary. The random method can be modified by specifying values for certain variables and by selecting the remaining variables in a random fashion. This is called a stratified random method and is useful when the "best" values for certain variables are known by prior knowledge or by previous search results.

The random methods have one advantage over the other methods. The number of trial designs required for this method is independent of the number of freely choosable variables. This does not necessarily mean that fewer trials are required to attain a so-called optimum design,

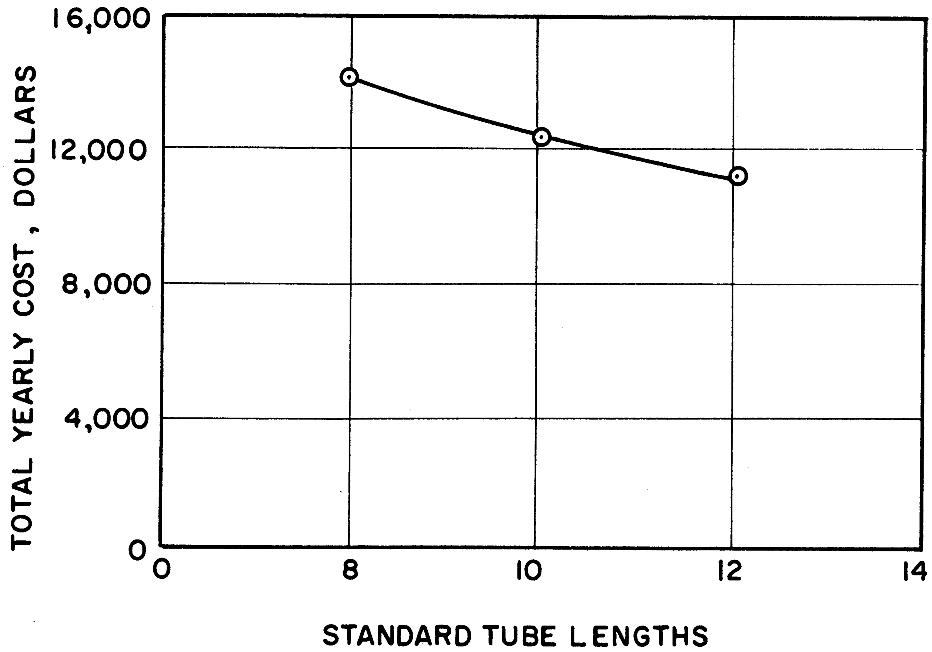

Figure 2. Univariate Search - First Level; 3/4 Inch Tubes, 2 Tube Pass, 8 ft./sec. Water Velocity.

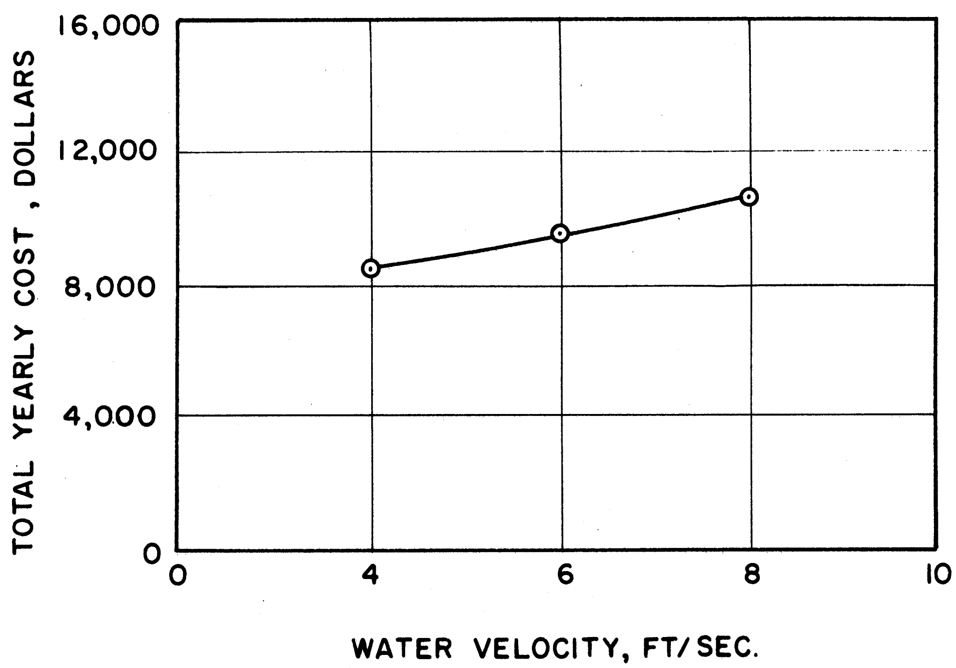

Figure 3. Univariate Search - Second Level; 3/4 Inch Tubes, 2 Tube Pass, 12 ft. Tube Length.

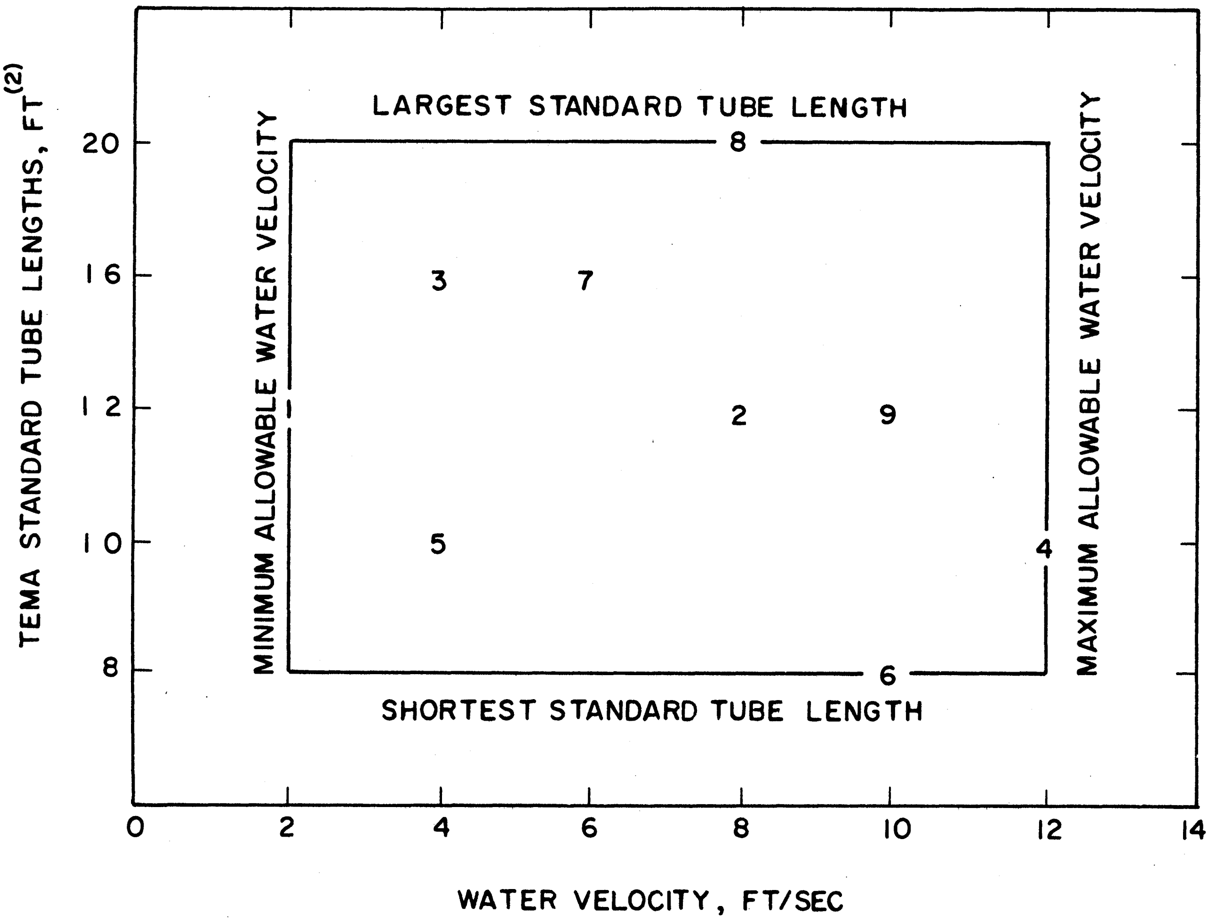

Figure 4. Random Search Pattern Where Only Tube Length and Tube Side Water Velocity are Considered as Variables.

but it does provide some measure of coverage of the factor space to be searched with a reasonable number of trials.

ECONOMICS

In the process of the search for the optimum heat exchanger for a particular service by a digital computer or by ordinary means, the cost of each designed unit must be determined. This, obviously, is one of the most important features of the program, because the resultant optimum is no better than the economic data upon which it is based. Slight errors or differences in the data can drastically change the designed physical description and operating conditions of the so-called optimum heat exchanger. Cooling water costs are particularly critical. Even with small differences in cost per gallon, the final design is greatly altered because of the large amounts of cooling water required on a yearly basis.

There appears to be two basic approaches to estimating heat exchanger costs. In the first method the unit is examined in its entirety and placed in a very definite category.(5) The cost of the heat exchanger is then determined by finding a cost graph or plot which is valid for units having the physical description of the designed heat exchanger. For example, suppose that in the course of the search for the optimum design, a floating head exchanger operating at 150 lb./sq.in. and 500°F is selected with 20 ft., 3/4 in. O.D. tubes on a 15/16 in. triangular pitch. In order to determine the cost of the exchanger it is necessary to have the information contained in a graph such as Figure 5. Figure 5 contains the necessary physical restrains so that

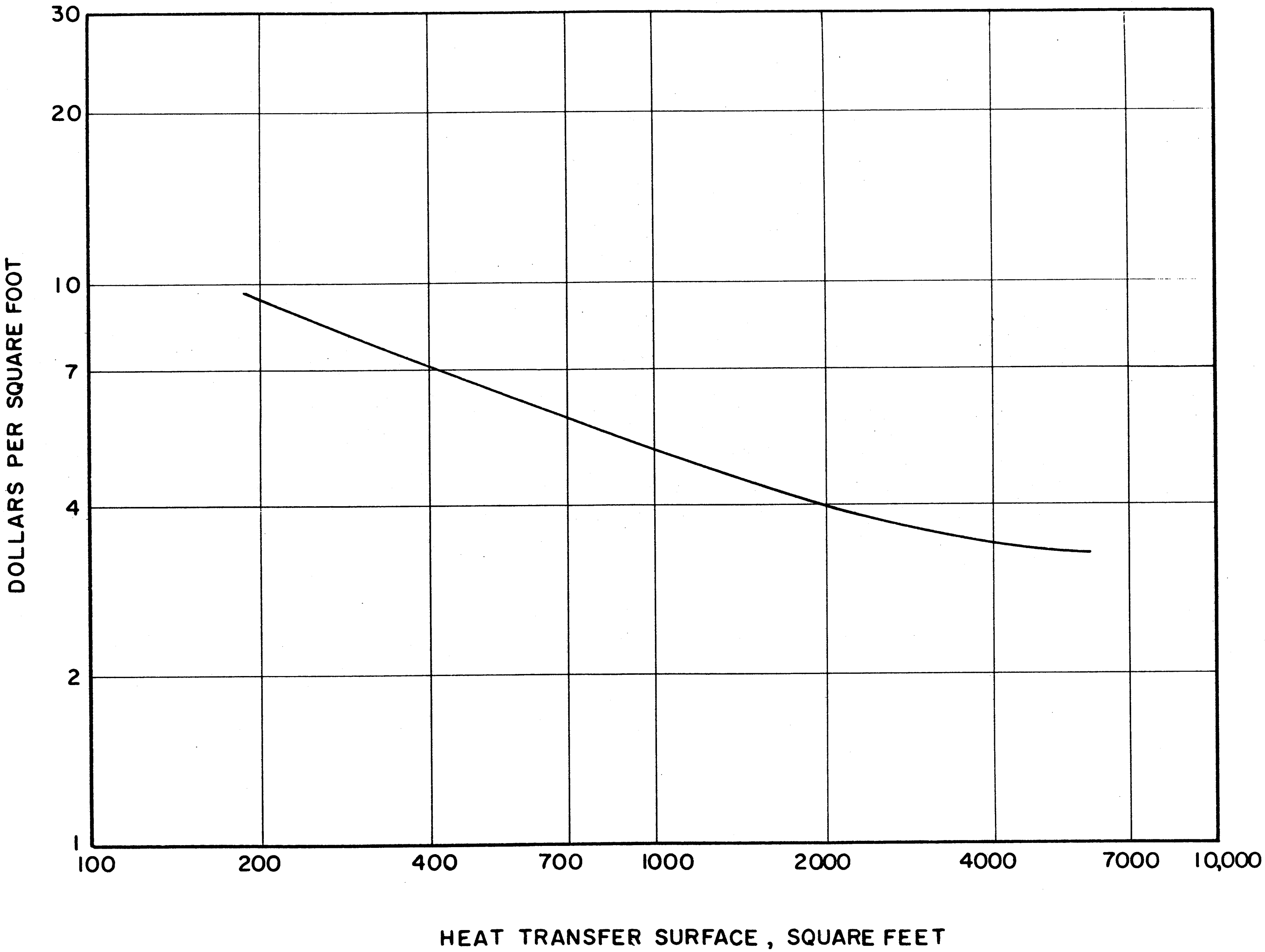

Figure 5. Heat Exchanger Costs as a Function of Heat Transfer Surface for All-Steel 20 Foot Bundles Containing 3/4 in. Tubes on a 15/16 in. Triangular Pitch. The operating Conditions are 150 lb./sq.in. and 500°F.(5)

it is quite accurate (in that no interpolation or extrapolation is required) for the heat exchanger pricing.

To utilize the first method of approach with a digital computer a sufficient number of graphs, similar to Figure 5, are selected for use. Each graph is then reduced to a mathematical function which the computer can evaluate. The nature of Figure 5 suggests that a reasonable function might be a simple polynomial in terms of the logarithm of heat transfer surface (in square feet) as is shown in Equation (1).

$$\ln(\$/\mathrm{sq.ft.}) = A(i) + B(i)\ln(\mathrm{sq.ft.}) + c(i)[\ln(\mathrm{sq.ft.})]^2 + \ldots x(i)[\ln(\mathrm{sq.ft.})]^n \quad (1)$$

The total cost of the exchanger is then equal to the product of the cost per square foot times the number of square feet of surface required.

$$\mathrm{cost} = (\text{heat transfer surface, sq.ft.}) \times \exp.(\ln(\$/\mathrm{sq.ft.})) \quad (2)$$

The coefficients for Equation (1) can be obtained by any of several techniques for finding the coefficients of a nth order polynomial in a least mean square sense. For the curve in Figure 5,

$$\mathrm{cost/sq.ft.} = \exp.[6.14888 - 0.98756 \ln \mathrm{sq.ft.} + 0.047995(\ln \mathrm{sq.ft.})^2] \quad (3)$$

Usually a second, third or fourth order polynomial is sufficient. For use with the computer each function and variable is assigned a name (a storage location in the computer memory) so that it can be used in arithemetic operations.

In the computer program the required physical dimensions are calculated for the proposed operating conditions. The computer then searches for the cost function which most nearly matches the designed unit. If, for example, thirty categories of exchangers are described by equations similar to Equation (3), the function whose restrains most nearly match those of the designed unit is selected. The search or selection process for the cost function is accomplished using logical statements. The resulting action taken by the computer in response to a logical statement depends upon the answer to a number of yes-no or true-false questions concerning the input data. Once the proper cost function is selected it is a simple matter to evaluate the cost of the heat exchanger.

A modification of the system can provide additional flexibility and accuracy where a limited number of cost functions are used. Of the restrains specified in Figure 5, it is not likely that the tube material and tube length of the designed exchanger will match those specified in Figure 5, whereas the remaining restrains are common to a wide class of applications. Equation (2) is first modified by subtracting the cost of the steel tubes from the total heat exchanger cost and then adding the cost of the replacement tubes. This gives,

$$\text{total cost} = \text{cost of all steel exchanger} - \text{steel tube costs} + \text{replacement tube cost} \qquad (4)$$

The cost of the all steel exchanger is calculated from Equation (2). The costs of the steel and replacement tubes can be calculated from,

$$\text{tube cost} = [\text{base cost/lb.} + \text{exp.}(-0.93 \times \text{total tube wt.})] \times \text{tube wt.} \qquad (5)$$

In Equation (5), the exponential term is included to account for the variation in cost with the size of the order. A different equation, based on the cost per square foot, could also be used since there is a definite relationship between tube weight and heat transfer surface. Further modification of Equation (2) can be made to account for different length bundles. Equation (4) is then written as,

$$\text{total cost} = (\text{cost of all steel exchanger} - \text{steel tube costs} + \text{replacement tube cost}) \times \text{length factor} \qquad (6)$$

These modifications greatly enhance the utility of Figure 5. Having reduced the number of restrains by two, the number of designs which can be satisfactorily represented by a single cost curve is increased from 1 to 35 (when seven types of alloys and five tube lengths are considered). A typical computer flow chart is shown in Figure 6.

The shortcoming of both the unmodified and modified forms is the difficulty of adequately covering all designs with a limited number of cost curves. This is especially a problem when materials other than steel are used for the construction of the tube sheets, shell, channel, and floating head. A cost curve, which is valid for the dimensions of the unit but not for the materials of construction, can be in error by as much as 100 per cent in determining the heat exchanger cost. Further modification of the technique, although possible, is not easily accomplished and is probably not warranted. The predicament, however, suggests a second method of treating the economic data.

In the second method one views the heat exchanger as being built from several standard parts which can be put together to form a complete

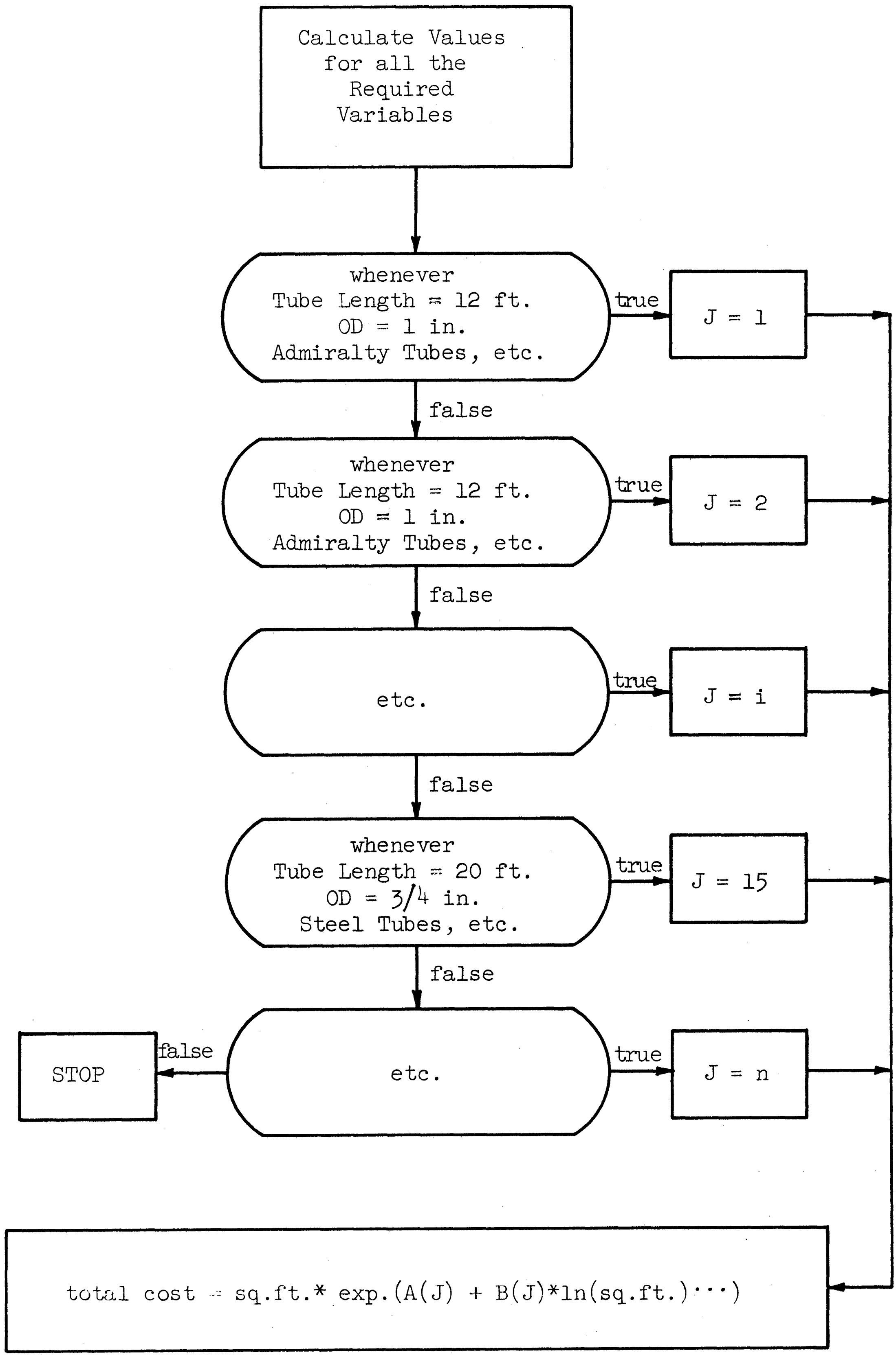

Figure 6. Computer Flow Chart to Determine the Correct Cost Function to be Used in Evaluating the Heat Exchanger Cost.

unit. The cost of each part is figured individually taking into account the materials of construction and size. This method is characteristic of the way a heat exchanger fabricator might price his product or set a bid price on a specified exchanger. A version of this method has been presented by Sieder and Elliott(6) who tabulated the costs of individual component parts as a function of the shell size, working pressures, and materials of construction. The costs of the component parts (i.e., tube sheets, shell and cover, and channel and floating head cover) were expressed as a per cent of the cost of an all steel heat exchanger having the same shell size. Table I gives the above cost information for a 450 lb./sq.in. working pressure. The cost of an all-steel heat exchanger was presented in graphical form with the cost per square foot of heat transfer surface given as a function of the shell size with parameters of working pressure and tube diameter. Figure 7 presents the cost of an all-steel heat exchanger containing 16 foot, 3/4 inch diameter tubes on a square pitch as a function of shell size and TEMA Class R(2) standard working pressures. When the standard working pressures of the shell side and tube side are not the same, multipliers are used to correct the cost percentages given in the tables similar to Table I. To account for the extra expense when tubes other than steel tubes are used, Sieder and Elliott gave a table of the extra cost of alloy tubes in dollars per square foot.

As an example of this method, suppose the cost of a heat exchanger having the following description is desired:

shell size	32 inch
length	16 feet

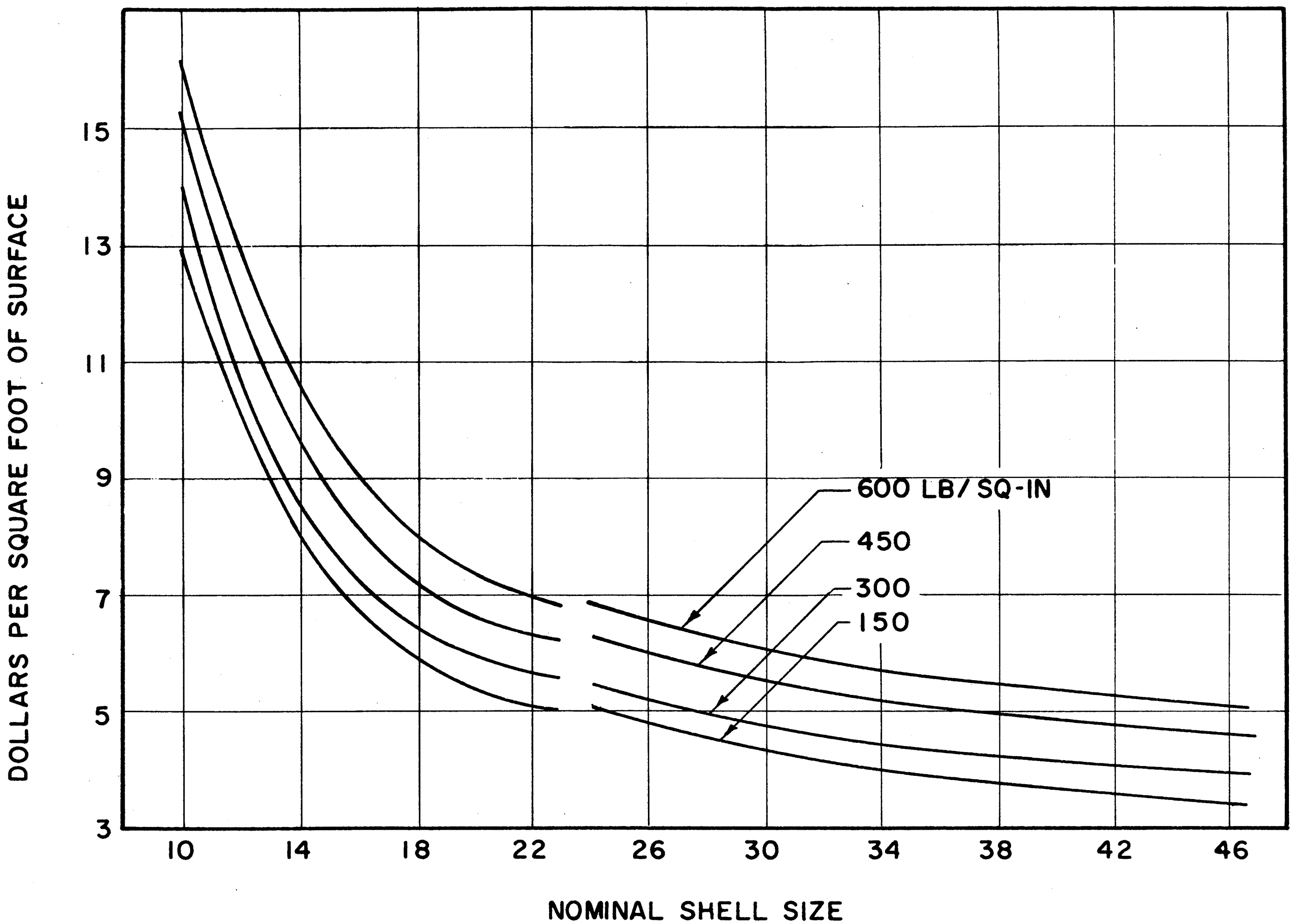

Figure 7. Cost Per Square Foot, TEMA Class R, All-Steel Heat Exchangers With 3/4 inch tubes.(6)

TABLE I

EXTRA FOR ALLOY CONSTRUCTION IN PERCENT OF ALL-STEEL HEAT EXCHANGER PRICE, 450 lb. WORKING PRESSURE.(6)

	12	13	16	18	20	22	24	27	30	33	36	39	42	Multipliers for Mixed Pressures 150	300
All Steel Heat Exchanger	100	100	100	100	100	100	100	100	100	100	100	100	100		
Naval Rolled Tube Sheets & Baffles	14	17	19	21	22	22	22	23	23	24	25	26	27		
Monel Tube Sheets & Baffles	24	31	35	37	39	39	40	41	41	42	42	42	43		
$1\frac{1}{4}$ Cr-$\frac{1}{2}$ Moly Tube Sheets & Baffles	6	7	7	7	8	8	8	9	10	10	11	11	12		
4-6 Percent Cr-$\frac{1}{2}$ Moly Tube Sheets & Baffles	19	22	24	25	26	26	26	25	26	26	26	27	27		
11/13 Cr Tube Sheets & Baffles	21	24	26	27	27	27	27	27	27	28	28	28	29		
304 Stainless Tube Sheets & Baffles	22	27	29	30	31	31	31	31	30	30	30	31	31		
Monel Shell & Cover	51	55	57	57	56	56	55	54	52	50	48	47	46	.81	.89
$1\frac{1}{4}$ Cr-$\frac{1}{2}$ Moly Shell & Cover	23	25	26	27	27	27	26	23	22	21	19	19	18	.81	.89
4-6 Percent Cr-$\frac{1}{2}$ Moly Shell & Cover	31	34	37	38	38	37	36	34	32	30	28	27	26	.81	.89
11/13 Chrome Shell & Cover	33	36	37	39	39	39	38	36	34	31	30	28	28	.81	.89
304 Stainless Shell & Cover	36	39	41	41	41	40	39	37	35	33	31	30	30	.81	.89
Monel Channel & Floating Head Cover	42	42	43	42	40	39	37	36	33	32	31	30	30	.88	.95
$1\frac{1}{4}$ Cr-$\frac{1}{2}$ Moly Channel & Floating Head Cover	25	26	26	25	24	23	23	22	22	21	21	20	20	.88	.95
4-6 Percent Cr-$\frac{1}{2}$ Moly Channel & Floating Head Cover	38	39	39	39	37	35	33	31	29	27	26	25	25	.88	.95
11/13 Cr Channel & Floating Head Cover	38	40	40	40	37	36	34	32	30	28	27	26	26	.88	.95
304 Stainless Channel & Floating Head Cover	39	40	40	39	38	36	35	33	31	29	28	27	26	.88	.95

tubes	3/4 inch x 14 BWG
shell working pressure	450 lb./sq.in.
tube working pressure	300 lb./sq.in.
area	2255 sq.ft.
tube material	4 - 6% chrome
tube sheets and baffles	11 - 13% chrome
shell and cover	11 - 13% chrome
channel	4 - 6% chrome

From Figure 7, the price per square foot for a 32 inch unit with a working pressure of 450 lb./sq.in. is $5.26 per square foot. From Table I, the component part percentage costs are:

all steel heat exchanger	100%
chrome tube sheets and baffles	28%
chrome shell and cover	31%
chrome channel 0.95 x 27	26%
	185%

The extra tube cost is $2.35 per square foot. The heat exchanger cost per square foot is then,

$$(\$5.26/\text{sq.ft.})(1.85) + \$2.35/\text{sq.ft.} = \$12.10/\text{sq.ft.}$$

The total unit cost is,

$$(\$12.10/\text{sq.ft.})(2255 \text{ sq.ft.}) = \$27,300.00$$

The method is most accurate when the considered heat exchangers have 16 foot tubes on a square pitch. It is not nearly as accurate for units having different tube lengths, pitches and arrangements. This is an inherent weakness in the procedure. The cost of a 20 foot unit would not cost two-thirds more than a comparable unit 12 feet long. This is because the cost of the construction, engineering, channel, floating head, cover, and tube sheets would be nearly identical for both units while the tube and shell expense would be approximately proportional to length. For this reason the total unit costs of a 20 foot unit should be only slightly higher than for a comparable 12 foot exchanger. The use of correction factors for tube length and tube pitch or the use of additional cost curves would greatly increase the accuracy of the system. An even greater accuracy could be obtained by calculating the all-steel heat exchanger cost on the basis of the shell size and the square feet of 16 foot steel tubes that could be placed into the calculated shell size on a square pitch. The 16 foot steel tube and shell costs would then be subtracted and the correct length alloy tube and shell costs added. In this manner a minimum of error would be encountered.

To utilize the data of Sieder and Elliott in a digital computer program two alternate approaches are possible. The data can either be reduced to a system of nth order polynomials obtained by least square means or be stored as tables in the computer. Use of the first approach, which was discussed earlier, would require numerous equations but would also require less computer storage than storing the data _per se_. Where the available computer memory is large, as in the IBM 709 digital computer, storage of the data in tabular form is more efficient. The nature of the data suggests that they should be stored as a two dimensional array with one index of the array representing the shell size and the second index representing the materials of construction or working pressure. By specifying values of both indices, the desired value of the function can be obtained from memory. Once the percentage costs for component parts are obtained, the method of solving the problem with a digital computer is, therefore, identical to the method used previously in the example problem.

In the computer program written to do the actual condenser design a brief routine was included to calculate the cost of the designed unit. When preparing the routine it was found that the data of Sieder and Elliott were most adaptable to computer storage and use if they were first grouped according to component parts rather than working pressure. Table II gives all the data on tube sheets. As can be seen from the table, the index, VTS, accounts for both the tube sheet material and working pressure. When VTS = 1, the tube sheets are admiralty and the standard working pressure is 150 lb./sq.in. When VTS = 13, the tube sheets are again admiralty, but the standard working pressure is 450 lb./sq.in. The index, U, corresponds to the shell size. When U = 11, the shell size is 24 inches; when U = 17, the shell size = 36 inches. To obtain the cost of the tube sheets as a per cent of an all steel heat exchanger, the indices, VTS and U, of the assigned storage variable name, TSAB(VTS, U), are specified. The datum stored in that location can then be used in calculating the price of the unit. For example

$$TSAB(16,17) = 26$$

In similar manner the shell and cover data were stored in an array called SAC(VSC, U), and the channel and floating head data were stored

TABLE II

COMPUTER VARIABLE TSAB (VTS,U). EXTRA FOR ALLOY CONSTRUCTION OF TUBE SHEETS AND BAFFLES IN PER CENT OF ALL-STEEL HEAT EXCHANGER COST

nom. shell size-in.		5	6	8	10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40	43	45	48
	U	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
	VTS																						
150 lb/sq.in. adm.	1	5	7	9	11	13	15	18	20	21	21	21	21	21	21	21	21	21	21	22	22	22	22
150 lb/sq.in. monel	2	10	13	16	19	23	31	34	36	37	38	39	39	39	39	39	39	39	38	38	38	38	38
150 lb/sq.in. 1 1/4 Cr	3	4	5	6	6	6	6	7	7	7	8	8	8	8	9	9	9	9	9	9	9	9	9
150 lb/sq.in. 4 Cr	4	8	11	14	17	19	22	24	25	26	26	26	25	25	25	25	26	24	24	24	24	24	24
150 lb/sq.in. 11-13 Cr	5	9	13	16	19	21	23	25	26	27	27	27	27	27	26	26	26	26	26	26	26	26	26
150 lb/sq.in. 304	6	10	13	16	19	22	26	28	30	30	31	31	31	30	30	30	30	29	29	29	29	29	29
300 lb/sq.in. adm.	7	6	8	10	12	14	17	19	21	22	22	22	22	22	22	22	23	24	24	24	25	25	25
300 lb/sq.in. monel	8	11	14	17	20	24	31	35	37	39	39	40	40	40	41	41	41	41	41	41	42	42	42
300 lb/sq.in. 1 1/4 Cr	9	4	5	6	6	6	7	7	7	8	8	8	8	8	9	9	10	10	10	10	11	11	11
300 lb/sq.in. 4 Cr	10	8	11	14	17	19	22	24	25	26	26	26	25	25	25	25	25	26	26	26	26	26	26
300 lb/sq.in. 11-13 Cr	11	9	13	16	19	21	24	26	27	27	27	27	27	27	27	27	27	27	27	27	28	28	28
300 lb/sq.in. 304	12	10	13	16	19	22	27	29	30	31	31	31	31	31	30	30	30	30	31	31	31	31	31
450 lb/sq.in. adm.	13	6	8	10	12	14	17	19	21	22	22	22	22	23	23	23	24	25	25	26	27	27	27
450 lb/sq.in. monel	14	11	14	17	20	24	31	35	37	39	39	40	40	41	41	42	42	42	42	42	43	43	43
450 lb/sq.in. 1 1/4 Cr	15	4	5	6	6	6	7	7	7	8	8	8	9	9	10	10	10	11	11	11	12	12	12
450 lb/sq.in. 4 Cr	16	8	11	14	17	19	22	24	25	26	26	26	26	25	26	26	26	26	26	27	27	27	26
450 lb/sq.in. 11-13 Cr	17	9	13	16	19	21	24	26	27	27	27	27	27	27	27	27	28	28	28	28	29	29	29
450 lb/sq.in. 304	18	10	13	16	19	22	27	29	30	31	31	31	31	31	30	30	30	30	30	31	31	31	31
600 lb/sq.in. adm.	19	6	8	10	12	14	17	19	21	22	22	22	22	22	23	23	24	25	25	26	27	27	27
600 lb/sq.in. monel	20	11	14	17	20	23	21	24	25	36	37	39	39	40	41	41	42	43	43	43	43	43	43
600 lb/sq.in. 1 1/4 Cr	21	4	5	6	6	6	6	7	7	7	7	7	8	8	9	9	10	10	10	11	12	12	12
600 lb/sq.in. 4 Cr	22	8	11	14	17	19	21	23	24	24	24	25	25	25	25	25	25	26	27	27	28	28	28
600 lb/sq.in. 11-13 Cr	23	8	11	14	17	19	20	22	24	25	26	26	26	26	27	27	27	28	29	29	30	30	30
600 lb/sq.in. 304	24	9	13	16	19	21	25	27	29	29	29	29	29	30	30	30	30	31	32	33	33	33	33

TABLE III

COMPUTER VARIABLE BACST (VTCST, U). COST IN DOLLARS PER SQUARE FOOT FOR TEMA CLASS R ALL-STEEL HEAT EXCHANGERS

nom. shell size-in.		5	6	8	10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40	42	45	48
	U	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
	VTCST																						
3/4 inch tubes 150 lb/sq.in.	1	24.00	20.00	16.00	13.00	10.00	8.05	6.70	5.88	5.40	5.10	5.15	4.80	4.55	4.30	4.15	4.00	3.90	3.80	3.70	3.52	3.49	3.30
3/4 inch tubes 300 lb/sq.in.	2	25.00	21.00	17.00	14.00	10.70	8.63	7.30	6.42	5.98	5.68	5.51	5.2[illegible]	4.98	4.75	4.60	4.46	4.32	3.25	4.18	4.05	4.00	3.75
3/4 inch tubes 450 lb/sq.in.	3	26.00	22.00	18.00	15.10	11.90	9.68	8.15	7.20	6.68	6.30	6.00	6.00	5.70	5.50	5.32	5.20	5.05	4.95	4.90	4.72	4.65	4.40
3/4 inch tubes 600 lb/sq.in.	4	28.00	24.00	20.00	16.20	12.90	10.68	9.10	8.03	7.22	6.94	6.90	6.60	6.30	6.10	5.90	5.75	5.60	5.50	5.40	5.25	5.15	5.00
1 inch tube 150 lb/sq.in.	5	27.00	23.00	19.00	15.40	11.90	9.30	7.55	6.50	5.80	5.48	5.40	5.10	4.80	4.55	4.40	4.20	4.10	4.00	3.90	3.80	3.70	3.60
1 inch tube 300 lb/sq.in.	6	28.00	24.00	20.00	16.20	12.50	9.85	8.08	6.95	6.30	5.95	5.95	5.60	5.34	5.10	4.90	4.75	4.63	4.52	4.47	4.35	4.25	4.10
1 inch tube 450 lb/sq.in.	7	29.00	25.00	21.00	17.30	14.00	11.00	9.05	7.82	7.18	6.80	6.90	6.55	6.25	6.00	5.80	5.62	5.50	5.40	5.27	5.15	5.10	4.90
1 inch tube 600 lb/sq.in.	8	30.00	26.00	22.00	18.20	15.10	12.15	10.05	8.67	7.93	7.60	7.70	7.33	7.00	6.70	6.50	6.30	6.18	6.00	5.86	5.70	5.60	5.50

in an array called CAFH(VCFH, U). The data for steel heat exchanger costs were also stored. Tube cost data, Table III, were obtained from Figure 7 and a like figure valid for 1 inch diameter tubes, and stored in an array called BACST(VTCST, U). As can be seen in the table when VTCST = 1, the data are for standard 150 lb./sq.in. units having 3/4 inch diameter tubes and when VTCST = 6, the data are for standard 300 lb./sq.in. units having 1 inch diameter tubes. The multiplying factors used in heat exchangers having mixed pressures were also stored in arrays. Table IV gives the multiplying factors for the shell and cover. In the array called MUSAC(INDEXB, INDEXC), INDEXB refers to the standard working pressure required for the shell and cover and INDEXC refers to the maximum standard working pressure required.

TABLE IV

COMPUTER VARIABLE MUSAC (INDEXB, INDEXC)
(Multiplying Factors for Mixed Pressures)

INDEXC	1	2	3	4
INDEXB				
1	1.00	0.92	0.81	0.75
2	0.00	1.00	0.89	0.83
3	0.00	0.00	1.00	0.92
4	0.00	0.00	0.00	1.00

The completed version of the computer routine used to calculate the heat exchanger cost and the other pertinent economic information is shown in Figure 8. The expressions are given in the MAD language (Michigan Algorithm Decoder). This language (and a compiler to translate

```
WHENEVER STWPT .G. STWPS
MAXWP = STWPT
OTHERWISE
MAXWP = STWPS
END OF CONDITIONAL
INDEX = 6*(MAXWP/150 -1)
VTS = INDEX + KTS
VSC = INDEX + KSC
VCFH = INDEX + KCFH
INDEXA = STWPT/150
INDEXB = STWPS/150
INDEXC = MAXWP/150
WHENEVER OD .G. 0.750
INDEXD = 4
OTHERWISE
INDEXD = 0
END OF CONDITIONAL
VTCST = INDEXD + INDEXC
CSTPSF = BACST(VTCST,U)*(100.0 + TSAB(VTS,U) + SAC(VSC,U)*
1MUSAC(INDEXB,INDEXC) + CAFH(VCFH,U)*MUCFH(INDEXA,INDEXC))
1/100.0
AOTST = 3.1416*OD*TUBEL*XMAXT/12.0
TWTS = XMAXT*TUBEL*WTFTS
STUCST = (BASES + 61.0*EXP.(-0.93*TWTS))*TWTS
TWT=X*TUBEL*WTFT
TUBCST=(BASE+61*EXP.(-0.93*TWT))*TWT
CONCST = CSTPSF*AOTST*CSTIND + (TUBCST - STUCST)*TCSTI
INSTCT = (CONCST*INSTPC)/100.0
INCOCT = CONCST + INSTCT
YRCAP = INCOCT/AMORD
MAINT = AOTST*MCSTF
YROPER = POWCST + FLUCST + MAINT
YRTOT = YRCAP + YROPER
```

Figure 8. Computer Routine to Calculate the Economic Information Required in the Search Analysis. The Expressions are in the MAD Language (Michigan Algorithm Decoder).

programs written in the MAD language into machine-language routines for execution on the IBM-704-709-7090 computers) was developed and is used extensively at The University of Michigan with the IBM 709 digital computer. The names of the variables are, in general, self-explanatory. The only symbol needing special explanation is the .G. which stands for "greater than". In statement 1. the standard working pressure for the tube side, STWPT, is compared to that on the shell side, STWPS. When the tube side pressure is greater, the maximum working pressure, MAXWP, is set equal to STWPT. Otherwise, MAXWP is set equal to STWPS. Statement 6. is used in calculating the array indices. As seen in Table II, the data for tube sheets are stored according to materials of construction. The first six values of VTS refer to different materials at a working pressure of 150 lb./sq.in.; the second six values refer to the same materials at a working pressure of 300 lb./sq.in., etc. The value of INDEX, therefore, fixes the general location in the array. In statement 7., the exact value of VTS is fixed by adding to the value of INDEX the value of KTS which stands for the type material used. For example, when KTS is 2, monel tube sheets are used and when KTS is 6, stainless steel tube sheets are used. KSC stands for the shell and cover material and KCFH stands for the channel and floating head material. Statements 10., 11., and 12. are used to calculate the indices required in the multiplying factors. Statements 13. - 18. are used to determine the index VTCST required in the array containing the all steel heat exchanger costs, Table III. INDEXD is set equal to 4 when the tube diameter is greater than 3/4 inches and to 0 when the diameter is equal to or less than 3/4 inches. This fixes the general region within the array. The variable INDEXC fixes the working pressure within the general region and therefore fixes the value of the index VTCST in statement 18. In statement 19. the cost per square foot of heat transfer surface is calculated as was done in the example problem. In statement 20. the total area of plain steel tubes is calculated based upon the maximum number of tubes that can be placed in the shell. The steel tube weight is calculated, (statement 21.), and the steel tube coat computed (statement 22.). The form of the equation in statement 22. has already been discussed. The total tube weight and tube cost for alloy tubes is determined in statements 23. and 24. In statement 25. the condenser cost is found by first determining the cost of the desired unit having steel tubes, adding the cost of alloy tubes and subtracting the cost of the steel tubes. CSTIND and TCSTI refer to cost indices. The installation cost is calculated in statement 26. and the total installed cost given in statement 27. The yearly capital expense is then found by dividing the installed cost by the amortization rate. Yearly operating costs are calculated in statement 30. by adding the costs of electric power, tube side fluid (water), and maintenance, and the yearly total cost is calculated by adding the yearly fixed capital expense and the yearly operating costs.

EXAMPLE PROBLEM

To illustrate the feasibility of the three search methods described, an example problem was solved using each method. The problem considered was to design a 60,000 lb./hr. n-propanol condenser for a recommended process such that the yearly cost (operating and fixed capital expense) is a minimum. The design conditions were as follows:

condensing load	60,000lb./hr.
inlet saturated vapor temperature	220°F
maximum inlet water temperature	80°F
winter inlet water temperature	50°F
maximum exit water temperature	140°F
allowable shell side pressure drop	3.0 lb./sq.in.
allowable tube side pressure drop	10.0 lb./sq.in.
water cost	0.001¢/gal.
electricity cost	$0.008/kw.hr.

Certain additional restrictions were imposed to reduce the number of required trials. They were:

tube material	admiralty
tube sheets	admiralty
shell and cover	1 1/4 per cent chrome
floating head cover	4-6 per cent chrome
tube arrangement	square
tube diameter	3/4 in.
tube pitch	1 in.

All the trial designs were done on an IBM 709 computer at the University of Michigan using a condenser program prepared by the author. After reading in the design conditions, economic data, additional restrictions, tube length, tube passes and water velocity, the computer calculates the minimum number of tubes required to condense the designated vapor load under summer conditions. The number of tubes required invariably is less than the number of tubes that could be placed into a standard shell size. The tubes are then fitted into the bottom portion of the shell on the specified pitch until all the required tubes are placed in the shell. The shell side pressure drop is calculated based on the vapor flowing down across the tubes. It was assumed that the top portion of the shell would be sufficiently open to permit unrestricted flow of the vapor at the top of the condenser. If the shell side pressure drop affects the condensing temperature, the logarithmic temperature is corrected and is used to obtain a more accurate solution. Finally, the desired economic information is calculated based on the economic input data. The water cost is based on the summer requirements.

In the second part of the computer program, the calculated standard shell size is filled completely with tubes and the shell side pressure drop is determined based on the baffled flow of the vapor. The pressure drop is used to obtain the "exit" vapor temperature which is used in calculating the logarithmic temperature difference. Since there are more tubes than required, the water velocity is reduced until the summer operating conditions are satisfied. In similar fashion, the water requirements are found for the winter operating conditions. The yearly water cost is based on an average of the summer and winter requirements. (Water costs were based on actual usage rather than on maximum demand.) The desired economic information is again calculated and printed. The net result is that two complete designs are completed for each set of input data.

Sixteen trials were allotted to each search method. Eight trials were used to evaluate plain tube designs and eight were used to evaluate finned tube designs. The results of the investigation are given in Tables V, VI, and VII

Only tube length, water velocity, and tube passes were considered as free variables because of the limited number of trials considered. In the factorial method two values of each variable were tried. As can be seen from Table V, the minimum cost plain tube design is $6886/yr. when a 2 pass, 8 ft. tube bundle is used with a 4 ft./sec. water velocity. A finned tube unit proved to be somewhat better. The best finned tube design would require $6125/yr. and would have a 2 pass, 10 ft. bundle with a water velocity of 10 ft./sec. As can be further noted, many of the trial designs were only partially satisfactory because the maximum allowable exit water temperature was exceeded. In other cases the design was totally unsuccessful because either the maximum exit water temperature was exceeded or the maximum allowable tube side pressure drop was exceeded. These results indicate that only a limited region of satisfactory operation exists. This is especially true for finned tube units. Even though they are less expensive to purchase and operate, the factor space of satisfactory operation for a finned tube condenser is even more restricted than for a plain tube condenser.

The results of the univariate method are given in Table VI. More successful plain tube designs were obtained with this method but the best design was inferior to the best design obtained by the factorial method. There were no completely satisfactory finned tube condenser designs.

TABLE V

TRIAL DESIGNS AND CALCULATED RESULTS FOR A FACTORIAL SEARCH PATTERN FOR THE DESIGN OF A n-PROPANOL CONDENSER

Plain Tube Trial Designs

	Trial 1	Trial 2	Trial 3	Trial 4	Trial 5	Trial 6	Trial 7	Trial 8
tube length-ft	8.0	8.0	8.0	8.0	12.0	12.0	12.0	12.0
water velocity-ft./sec.	4.0	8.0	4.0	8.0	4.0	8.0	4.0	8.0
tube passes	2	2	4	4	2	2	4	4
total yearly cost-$/yr.	6886	8705	----	----	----			
comments			excessive exit water temperature	excessive exit water temperature	excessive exit water temperature	excessive exit water temperature	excessive exit water temperature	excessive tube side pressure drop
cost - $/yr. (based on partially filled tube bundle and summer water requirements)	9643	14627	7208	9657	8217	11780	----	----

Finned Tube Trial Designs

	Trial 1	Trial 2	Trial 3	Trial 4	Trial 5	Trial 6	Trial 7	Trial 8
tube length - ft.	8.0	8.0	8.0	8.0	12.0	12.0	12.0	12.0
water velocity-ft./sec.	10.0	14.0	10.0	14.0	10.0	14.0	10.0	14.0
tube passes	2	2	4	4	2	2	4	4
total yearly cost-$/yr.	6125	----	----	----	----			
comments		excessive tube side pressure drop	excessive exit water temp.	excessive tube side pressure drop	excessive exit water temperature	excessive tube side pressure drop	excessive exit water temperature	excessive exit water temperature
cost - $/yr. (based on partially filled tube bundle and summer water requirements)	7444	----	----	----	----	----	----	----

TABLE VI

TRIAL DESIGNS AND CALCULATED RESULTS FOR AN UNIVARIATE SEARCH PATTERN FOR THE DESIGN OF A n-PROPANOL CONDENSER

Plain Tube Trial Designs

	Trial 1	Trial 2	Trial 3	Trial 4	Trial 5	Trial 6	Trial 7	Trial 8
tube length-ft.	10.0	10.0	8.0	12.0	16.0	10.0	10.0	10.0
water velocity-ft./sec.	6.0	6.0	6.0	6.0	6.0	3.0	4.5	7.5
tube passes	2	4	2	2	2	2	2	2
total yearly cost-$/yr.	7447	----	8753	----	----	----	7470	7470
comments		excessive exit water temperature		excessive exit water temperature	excessive exit water temperature	excessive exit water temperature		
cost-$/yr. (based on partially filled tube bundle and summer water requirements)	10711	7772	11908	9941	8687	7974	9147	12343

Finned Tube Trial Designs

	Trial 1	Trial 2	Trial 3	Trial 4	Trial 5	Trial 6	Trial 7	Trial 8
tube length-ft.	10.0	10.0	8.0	12.0	16.0	10.0	10.0	10.0
water velocity-ft./sec.	12.0	12.0	12.0	12.0	12.0	8.0	9.0	10.0
tube passes	2	4	2	2	2	2	2	2
total yearly cost-$/yr.	----	----	----	----	----	----	----	----
comments	excessive tube side pressure drop				excessive exit water temperature			excessive exit water temperature
cost-$/yr. (based on partially filled tube bundle and summer water requirements)	----	----	----	----	----	----	6629	7017

TABLE VII

TRIAL DESIGNS AND CALCULATED RESULTS FOR A RANDOM SEARCH PATTERN FOR THE DESIGN OF A n-PROPANOL CONDENSER

Plain Tube Trial Designs

	Trial 1	Trial 2	Trial 3	Trial 4	Trial 5	Trial 6	Trial 7	Trial 8
tube length-ft.	12.0	8.0	8.0	16.0	16.0	8.0	12.0	10.0
water velocity-ft./sec.	7.0	7.0	5.0	4.0	10.0	5.0	5.0	6.0
tube passes	2	2	4	2	4	2	4	4
total yearly cost-$/yr.	----	8772	----	----	----	8726	----	----
comments	excessive exit water temperature		excessive exit water temperature	excessive exit water temperature	excessive tube side drop pressure		excessive exit water temperature	excessive exit water temperature
cost-$/yr. (based on partially filled tube bundle and summer water requirements)	10846	13233	7520	8073	----	10615	----	7772

Finned Tube Trial Designs

	Trial 1	Trial 2	Trial 3	Trial 4	Trial 5	Trial 6	Trial 7	Trial 8
tube length-ft.	12.0	10.0	8.0	8.0	16.0	10.0	12.0	12.0
water velocity-ft./sec.	12.0	6.0	18.0	16.0	6.0	12.0	18.0	16.0
tube passes	4	2	4	4	2	2	2	2
total yearly cost-$/yr.	----	----	----	----	----	----	----	----
comments		excessive exit water temperature	excessive tube side pressure drop	excessive tube side pressure drop	excessive exit water temperature	excessive tube side pressure drop	excessive tube side pressure drop	excessive tube side pressure drop
cost-$/yr. (based on partially filled tube bundle and summer water requirements)	----	----	----	----	----	----	----	----

The random method had the poorest performance as shown in Table 7. The best plain tube design obtained was approximately $2000/yr., higher than for the best design obtained using the factorial method. There were no satisfactory finned tube condenser designs.

The total computer time required to process all the trial designs was 10.7 minutes at a cost of $70.00. This is about $1.50/trial or about $24.00/method if both plain and finned tube condensers are considered.

CONCLUSIONS

It is difficult to make any definite conclusions from the results of one problem, but it appears obvious that the techniques can be used to advantage with a considerable saving in both time and labor. Certainly more trials should be used once the desired search method is selected. There seems to be little doubt that a less expensive design could have been achieved if a second level search had been carried out with both the factorial and univariate methods. There is a limit to the number of trials that can be made, however, before the computer expense exceeds the savings in finding a better design. The best results will probably be obtained by using 8-10 trials to broadly search the factor space. The results of the first level of search would then be used to closely search a more limited region to obtain the best design conditions. Too few initial trials could be uninformative while too many trials would not be economical because of the large number of unsuccessful trials that would be obtained.

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