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Computer codes for the analysis of marine steam condensers were coupled with a numerical optimization code to provide fully automated design.
The resulting programs are excellent design tools for the conceptual design of a condenser to meet specified constraints. Examples illustrate the versatility of this method.

Introduction

Background. In the last ten years, a revolution has swept the marine power plant industry that could result in the obsolescence of the marine steam power plant. When compared with the compact gas turbine, the massive size and weight of the marine steam power plant, which evolved in stride with the behemoth of the power industry, the stationary steam power plant, made this means of propulsion less desirable for naval vessels. Therefore, to keep steam propulsion competitive with marine gas turbine propulsion, it has become imperative for the naval engineering community to develop a more efficient, more compact, lighter weight steam power plant.

Advanced concepts must be explored in all areas of steam propulsion, such as pressurized boilers, super critical cycles, and compact marine condensers. Above all, overdesign by the use of unnecessary safety factors must be curtailed, and the minimum safe design must be identified and developed.

With the advent of the high speed digital computer, numerical methods of solving complex engineering problems are now possible. The case for utilizing an optimizing design scheme for the condenser portion of a steam propulsion plant can easily be made by re-emphasizing the fact that, in order for steam propulsion to remain a viable contender when the naval vessels of the late 1980's are designed, it must compete and succeed in an arena that is rapidly being dominated by gas turbines. All components of the naval ship's steam propulsion plant will have to be critically designed to ensure that the minimum design will still perform as and when required.

Objectives. The objectives of this work were to develop computer codes for marine condenser thermal design and to couple these codes with numerical optimization codes to provide a fully automated design capability. These design tools provide the naval architect and the naval engineer with the means to optimize size, weight, and cost of the marine steam propulsion plant for ships of the late 1980's. This can provide the optimum streamlined design of a steam plant, resulting in its reinstatement and continuance as a viable alternative to gas turbine propulsion.

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Marine Condenser Design Principles

The marine steam condenser is a shell and tube heat exchanger with steam condensing on the outside of tubes, the shell side, and cooling water flowing through the interior of the tubes, commonly referred to as the tube side of the condenser. A schematic representation of a marine steam condenser is shown in Fig. 1. The condenser is generally mounted underneath the power turbine. Steam from the turbine enters the condenser and flows down through thousands of horizontally mounted, cooled tubes. The condensate drops to the bottom of the condenser and is returned to the boiler to complete the power cycle. In any steam plant, air (and other non-condensible gases) finds its way into the system and collects in the low pressure regions of the condenser. This air is cooled and continuously removed from the condenser to avoid pressure buildup and loss of efficient heat transfer. An excellent description of condensers has been prepared by Sebald [1].

Naval condensers are presently designed following the standards developed by the Heat Exchange Institute [2, 3]. The technique involved is based on calculating a value of the overall heat transfer coefficient, U_c, as a function of the cooling water velocity, V, through the condenser tubes, the condenser tube wall thickness and material, the tube fouling characteristics, and the cooling water inlet (injection) temperature. Knowledge of these parameters and their associated correction factors leads to the simple formulation of U_c,

$$U_c = F_1F_2F_3\, C\sqrt{V}. \tag{1}$$

The correction factors F_1 (cooling water inlet temperature), F_2 (tube wall material), and F_3 (tube fouling) are tabulated in references [2] and [3]. In equation (1), the coefficient C has a weak dependence on tube outside diameter.

For a particular application, the designer must specify the desired characteristics for the condenser. For example, for a given set of input variables such as seawater velocity and inlet temperature, steam pressure and mass flow rate, as well as tube material and geometry, this technique can be used to calculate the required condenser surface area to transfer the specified heat load.

This procedure was coded in FORTRAN IV to provide an automated capability. In addition, a circular tube bundle

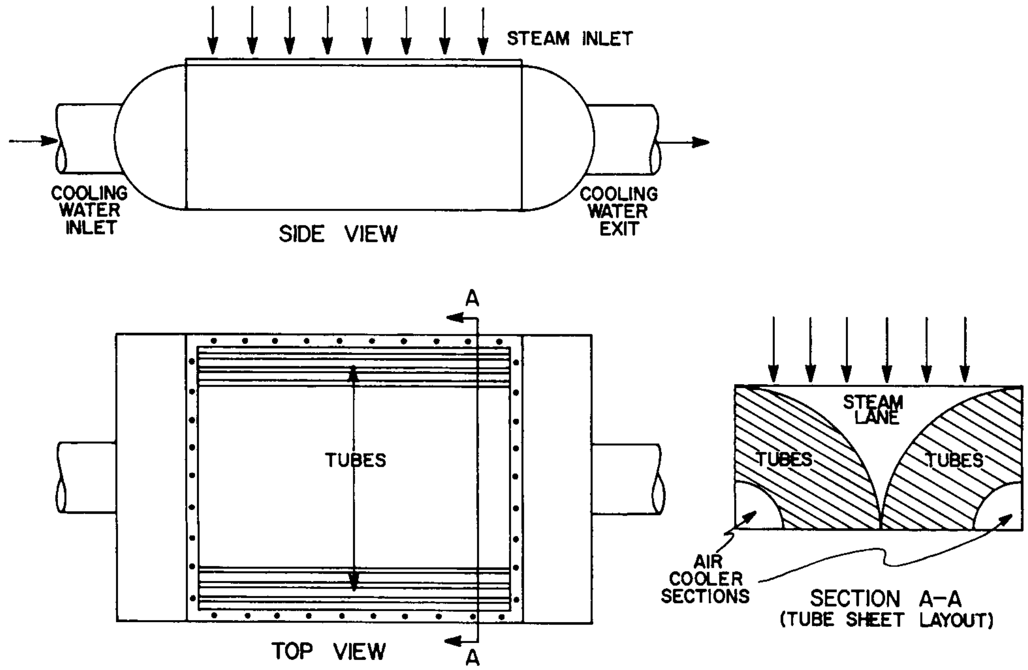

Fig. 1 Schematic Representation of a Marine Steam Condenser

Table 1 Input Parameters for Aircraft Carrier Condenser

TUBE INFORMATION		
TUBE OD, mm. 15.9	TUBE ID, mm. 13.4	TUBE WALL THICKNESS, mm 1.24
ABS. ROUGHNESS, m . . . . 1.52×10^{-6}	TUBE LENGTH, m 4.51	MAT'L CORR. FACTOR. . . 0.90
TUBE CLEAN. FACTOR. . . 0.85		MATERIAL. . . . . 90-10 CuNi
SEAWATER INFORMATION		
INJECTION TEMP.°C........24	SEA WATER VELOC. . m/sec..2.74	
INLET STEAM INFORMATION		
MASS FLOW RATE, KG/HR.........195,068		
TOTAL HEAT REJECTED, WATT..1.182×10^8		

Table 2 Comparison of Actual and Computer Parameters

	TECHNICAL MANUAL	HEI	ORCON1
Heat transfer area; m^2	1,487	1,496	1,487
Number of tubes	6,612	6,648	6,612
Bundle diameter; m	N/A	2.21	3.12
Heat rejected; WATTS	1.18×10^8	1.18×10^8	8.61×10^7
Overall heat transfer coefficient; $W/m^2 {}^\circ C$	3,606	3,617	2,862
Log mean temperature difference; °C	22.1	21.8	20.2
Seawater temperature rise; °C	11.1	11.1	8.0
Terminal temperature difference; °C	17.0	16.8	14.0
Seawater flow rate, ℓ/m	153,500	153,900	152,700
Tube-side pressure drop; meters of water column.	4.97	3.78	3.75
Exit steam vent rate as percentage of inlet steam	0	0	29

geometry and the cooling water pumping power requirement was calculated. The existing main propulsion condenser of an aircraft carrier was analyzed using input information, as shown in Table 1 [4]. The results from this analysis are shown in Table 2 in the column headed by HEI. Agreement, when compared to the actual data from the TECHNICAL MANUAL, is excellent.

In the late 1960's, engineers at the Oak Ridge National Laboratory developed a sophisticated condenser analysis computer code under contract to the Office of Saline Water. This code, called ORCON1 [5], was generated to aid in the analysis and parametric study of large, generally circular, steam condensers for use in large scale, multistage distillation plants for the production of potable water from seawater by the flash evaporation process. Much of ORCON1 was developed based on Eissenberg's research work [6] on the effects of condensate rain on the shell-side convective heat transfer coefficient. By considering the physics of condenser analysis, ORCON1 accounts for many secondary effects not directly considered in the Heat Exchange Institute standards.

Effects such as condensate inundation of the lower tubes in the condenser, non-condensible gas buildup, heat transfer enhancement, and high vapor velocity should all be considered in a proper condenser analysis. This is most desirable when dealing with unconventional condenser configurations. In addition, ORCON1 allows for *local* calculations within the condenser, rather than just overall performance, as obtained from the HEI method.

The program analyzes a single pass, circular or semi-circular condenser, shown schematically in Fig. 2. Steam flows on the shell-side of the tubes and variable salinity water flows on the tube-side. An optional, rectangular air cooler bundle is provided for, as well as elementary, shell-side baffles. The bundle is divided into 30 degree sectors and symmetry about the central axis may be employed to reduce computational effort. The tubes are placed on a 60 degree equilateral triangular pattern of concentric rows with the rows added from the outermost row to an inner void provided along the bundle's longitudinal axis. This serves as a collection header for noncondensable gases prior to passage through the air cooler, if specified. The steam is assumed to flow radially from the outside of the bundle to the central void.

It is expected that ORCON1 can be used to calculate a variety of secondary effects which the HEI method neglects, such as, tube-side and shell-side heat transfer enhancement. Additional information on this method can be found in reference [5].

Using a modified version of this code, to allow for fixed tube length and steam flow rate, the results shown as ORCON1 in Table 2 were obtained. In general, agreement between ORCON1 and the TECHNICAL MANUAL is unsatisfactory, presumably due to the conservative heat transfer coefficient on the shell-side of the condenser. The effect of this shell-side coefficient on ORCON1 results was indicated by Search [7].

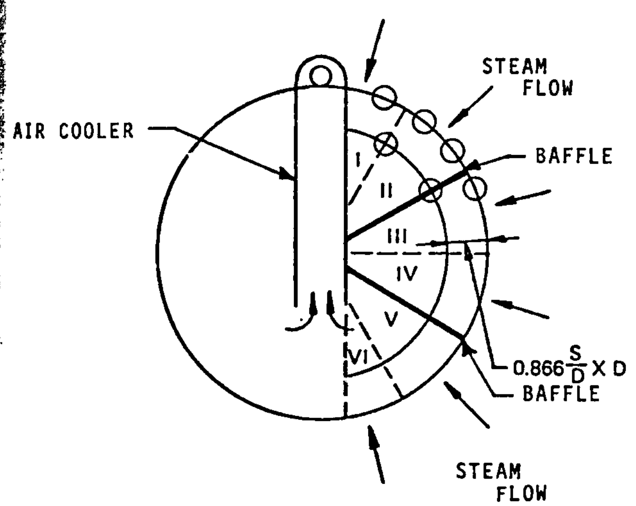

Fig. 2 Circular Condenser Bundle Designed by ORCON1

Additional difficulties were encountered when this code was coupled to the optimization program. For example, minor modifications in condenser geometry or heat transfer rates within the condenser often led to difficulties in computing logarithmic functions or in the solution of transcendental equations within ORCON1, resulting in program execution errors. Also, numerous design decisions were programmed into ORCON1. These internal decisions often conflict with design decisions best left to the optimization code. In short, the ORCON1 program was not developed to be coupled with an optimization program, and time did not permit modification of the program to produce a truly optimization oriented analysis code.

Numerical Optimization

Nearly all design problems require either the minimization or maximization of a parameter (or function). This parameter will be called the problem's objective function or design objective [8]. For the design to be acceptable, it must satisfy a set of design constraints. For example, if an engineer were designing a piping system to achieve the minimum in required pumping power, the minimum allowable flow delivered would be a meaningful constraint. Likewise, a constraint that required the inside diameter of the pipe to be less than the outside diameter of the pipe would be a necessity.

There are many optimization schemes available to the engineer. The various methods fall into three broad categories based on the type of problem to be solved: unconstrained minimization, and direct methods for solution of constrained problems [9]. An optimization program based on the last method was chosen for this research work.

Three basic definitions are required [10]:

Design Variables. Those parameters which the optimization program is permitted to change in order to improve the design.

Design Constraints. Any parameter which must not exceed specified bounds for the design to be acceptable. (Design constraints may be linear or nonlinear, implicit or explicit, but they must be functions of the design variables.)

Objective Function. The parameter which is going to be minimized or maximized during the optimization process. (The objective function may also be linear or nonlinear, implicit or explicit, and must be a function of the design variables). As can be seen readily by the definitions above, design constraints and objective functions are usually interchangeable.

Assuming that the optimization process requires the minimization of a particular objective function, the general optimization problem can be stated as:

Find the vector of design variables, $\bar{X}$, to

Minimize $F(\bar{X})$ subject to the constraints:

$$G_j(\bar{X}) \leq 0.0, j = 1,\text{NCON} \tag{2}$$

$$VLB_i \leq X_i \leq VUB_i, i = 1,\text{NDV}. \tag{3}$$

In this problem statement, $F(\bar{X})$ is the objective function, there are NDV design variables, and NCON constraints. VLB_i and VUB_i are the lower bounds and upper bounds, respectively, on the i-th design variable. If the inequality condition of equation (2) is violated, ($G_j(\bar{X}) > 0$), for any constraint, that constraint is said to be violated. If the equality condition is met, ($G_j(\bar{X}) = 0$), the constraint is active. If the inequality condition is met, ($G_j(\bar{X}) < 0$), the constraint is inactive. Because of the numerical problems involved in representing exact zero on a computer with a finite number of significant figures, the equality condition is represented by some small number, say $|G_j(\bar{X})| \leq 0.001$.

Any design which satisfies the inequalities of equations (2) and (3) is referred to as a feasible design. If a design violates any of these constraints, it is an infeasible design. The minimum feasible design is said to be optimal. Note that if it is desired to maximize the objective, this may be done by minimizing the negative of the objective.

An initial vector of design variables, $\bar{X}$, must be provided. There is no requirement that the initial $\bar{X}$ yield a feasible design. The optimization process then proceeds in an iterative fashion with the following iterative relationship:

$$\bar{X}^{q+1} = \bar{X}^q + \alpha^*\bar{S}^q \tag{4}$$

where q is the iteration number and α^* is the move parameter, a scalar, which defines the distance of travel in the direction of search $\bar{S}^q$.

The optimization program COPES/CONMIN [11, 12] is used to solve this problem. COPES is an acronym for *CO*ntrol *P*rogram for *E*ngineering *S*ynthesis, and CONMIN is an acronym for *CON*strained function *MIN*imization. CONMIN uses the Fletcher-Reeves algorithm [13] for locally unconstrained problems, and Zoutendijk's method of feasible directions [14, 15] for locally constrained problems. Modifications have been added to the algorithm such that if the initial design is infeasible, a feasible solution will be obtained with minimal increase in objective function.

The problem at hand may include equality constraints of the form

$$G_j(\bar{X}) = 0 \qquad j = 1,\text{NEQ} \tag{5}$$

where NEQ is the number of equality constraints. While the CONMIN program does not handle such constraints directly, they are easily included as penalty parameters. To achieve this, the objective function $F(\bar{X})$ is augmented to be

$$F'(\bar{X}) = F(\bar{X}) - K \sum_{j-1}^{\text{NEQ}} G_j(\bar{X}) \tag{6}$$

and the equality condition of equation (5) is treated as an inequality constraint

$$G_j(\bar{X}) \leq 0 \qquad j = 1,\text{NEQ}. \tag{7}$$

The parameter K is a positive constant approximately equal in magnitude to $F(\bar{X})$. The augmented objective $F'(\bar{X})$ is now minimized, making the $G_j(\bar{X})$ as large as possible, subject to the constraint of equation (7). This penalty function approach has been found to effectively satisfy the equality constraint while maintaining the rapid convergence characteristics of the CONMIN program. This is, in effect, a marriage of the exterior penalty function approach with the feasible directions method. A principle distinction is that here the penalty term in equation

Table 3 Input Parameters Used in OPCODE1

TUBE INFORMATION		
TUBE OD, mm.............15.9	TUBE ID, mm.......13.4	TUBE WALL THICKNESS, mm.....1.24
ABS. ROUGHNESS, m........ 1.52 x 10^{-6}	TUBE LENGTH, m.... 4.51	MATERIAL CORR. FACTOR......0.90
TUBE CLEAN. FACTOR....... 0.85	NR. OF PASSES..... 1	MATERIAL.............90-10 Cu-Ni
SEA WATER INFORMATION		
INJECTION TEMP.°C........24	SEA WATER VELOC. m/sec......2.74	
INLET STEAM INFORMATION		
AUX. STEAM, KG/HR.... 5,020	MAIN STEAM, KG/HR...190,000	TOTAL STEAM KG/HR........195,020
AUX. STEAM ENTHALPY CHANGE, KJ/KG........ 1,120.7	MAIN STEAM ENTHALPY CHANGE, KJ/KG........ 2,209.7	ADD'L. HEAT, WATTS.....1.459 x 10^5
SAT. TEMPERATURE,°C..... 51.7	SAT. PRESSURE, N/m^2..13,545	

Table 4 Initial Design by OPCODE1

TUBE INFORMATION		
TUBE OD, mm............15.9	TUBE ID, mm........13.4	TUBE WALL THICKNESS, mm.....1.24
ABS. ROUGHNESS, m....... 1.52 x 10^{-6}	TUBE LENGTH, m..... 4.51	MATERIAL CORR. FACTOR.........0.90
TUBE CLEAN. FACTOR...... 0.85	NR. OF PASSES...... 1	MATERIAL ....90-10 Cu-Ni
SEA WATER INFORMATION		
INJECTION TEMP.°C.......24	SEA WATER VELOC. m/sec....2.74	
INLET STEAM INFORMATION		
AUX. STEAM, KG/HR.... 5,020	MAIN STEAM, KG/HR...190,000	TOTAL STEAM KG/HR........195,020
AUX. STEAM ENTHALPY CHANGE, KJ/KG......... 1,120.7	MAIN STEAM ENTHALPY CHANGE, KJ/KG..........2,209.7	ADD'L. HEAT, WATTS.....1.459 x 10^5
SAT. TEMPERATURE,°C..... 51.7	SAT. PRESSURE, N/m^2..13,545	
BUNDLE INFORMATION		
BUNDLE DIAMETER, m....... 2.21	TUBE SHEET AREA, m^2..3.84	WATERBOX DEPTH, m..........1.11
VOLUME, m^3.............. 26.3	OVERALL LENGTH, m...6.72	TUBE PITCH/DIAMETER........1.60
AREA RATIO............... 0.342		
HEAT TRANSFER AND PUMPING INFORMATION		
HEAT REJECTED, WATTS.......1.182 x 10^8	OVERALL U, W/m^2 °C.... 3,618	LMTD, °C................ 21.9
S-W TEMPERATURE RISE,°C..11.0	PINCH POINT,°C............ 16.8	S-W FLOW RATE, m^3/sec.... 2.56
S-W FLOW RATE, ℓ/m....... 1.54 x 10^5	PUMPING POWER, Nm/sec..93,329	S-W PRESSURE DROP, m water 3.72
HEAT TRANSFER AREA, m^2...1,496	PUMPING POWER, KW..........93.3	

Table 5 Case One Optimization Results

TUBE INFORMATION		
TUBE OD, mm.............18.62	TUBE ID, mm..........17.45	TUBE WALL THICKNESS, mm...0.587
TUBE LENGTH, m.......... 3.947	NR. OF TUBES......8,860.6	MATERIAL CORR. FACTOR.....0.99
SEA WATER INFORMATION		
INJECTION TEMP, °C......24	SEA WATER VELOC. m/sec.... 1.216	
INLET STEAM INFORMATION		
AUX. STEAM, KG/HR....5,020	MAIN STEAM, KG/HR.....190,000	TOTAL STEAM, KG/HR........195,020
AUX. STEAM ENTHALPY CHANGE, KJ/KG......... 1,120.7	MAIN STEAM ENTHALPY CHANGE, KJ/KG..........2,209.7	ADD'L HEAT, WATTS........ 1.459 x 10^5
SAT. TEMPERATURE,°C.... 51.7	SAT. PRESSURE, N/m^2....13,545	
BUNDLE INFORMATION		
BUNDLE DIAMETER......... 2.997	TUBE SHEET AREA, m^2... 7.050	WATERBOX DEPTH, m........ 1.143
VOLUME, m^3............ 40.74	OVERALL LENGTH, m.... 6.233	TUBE PITCH/DIAMETER...... 1.625
AREA RATIO............. 0.342		
HEAT TRANSFER AND PUMPING INFORMATION		
HEAT REJECTED, WATTS.... 1.182 x 10^8	OVERALL U, W/m^2 °C......2,641	LMTD,°C..................21.9
S-W TEMP. RISE, °C..... 10.99	PINCH POINT, °C........16.84	S-W FLOW RATE, m^3/sec.... 2.575
S-W FLOW RATE, ℓ/m....... 1.542 x 10^5	PUMPING POWER, Nm/sec..14,685	S-W PRESSURE DROP, m Water 0.5822
HEAT TRANSFER AREA, m^2..2,047	PUMPING POWER, KW......14.7	

Table 6 Case Two Optimization Results

TUBE INFORMATION		
TUBE OD, mm............16.06	TUBE ID, mm...........14.86	TUBE WALL THICKNESS, mm....0.599
TUBE LENGTH, m..........4.304	NR. OF TUBES........6,788.5	MAT'L. CORR. FACTOR........0.99
SEA WATER INFORMATION		
INJECTION TEMP., °C.....24	SEA WATER VELOC.,m/sec..2.292	
INLET STEAM INFORMATION		
AUX. STEAM, KG/HR....5,020.	MAIN STEAM, KG/HR.....190,000	TOTAL STEAM, KG/HR......195,020
AUX. STEAM ENTHALPY CHANGE, KJ/KG........1,120.7	MAIN STEAM ENTHALPY CHANGE, KJ/KG..........2,209.7	ADD'L. HEAT, WATTS.... 1.459×10^5
SAT. TEMPERATURE, °C....51.7	SAT. PRESSURE, N/m^2....13,545	
BUNDLE INFORMATION		
BUNDLE DIAMETER, m.......2.23	TUBE SHEET AREA, m^2......3.91	WATERBOX DEPTH, m..........1.11
VOLUME, m^3..............25.95	OVERALL LENGTH, m.......6.54	TUBE PITCH/DIAMETER........1.60
AREA RATIO...............0.351		
HEAT TRANSFER AND PUMPING INFORMATION		
HEAT REJECTED, WATTS....1.182×10^8	OVERALL U, W/m^2 °C.....3,620	LMTD, °C..................22.2
S-W TEMP. RISE, °C......10.5	PINCH POINT, °C.........17.3	S-W FLOW RATE, m^3/sec......2.70
S-W FLOW RATE, ℓ/m......1.61×10^5	PUMPING POWER, Nm/sec..60,999	S-W PRESSURE DROP, m Water. 2.31
HEAT TRANSFER AREA, m^2.. 1,474	PUMPING POWER, KW......61.0	

Table 7 Case Three Optimization Results

TUBE INFORMATION		
TUBE OD, mm.......... 15.9	TUBE ID, mm........... 14.70	TUBE WALL THICKNESS, mm....0.589
TUBE LENGTH, m....... 4.34	NR. OF TUBES.........13,023.6	MAT'L CORR. FACTOR.........0.99
SEA WATER INFORMATION		
INJECTION TEMP., °C.. 24	SEA WATER VELOC., m/sec...2.19	
INLET STEAM INFORMATION		
AUX. STEAM, KG/HR....5,076	MAIN STEAM, KG/HR.....347,780	TOTAL STEAM, KG/HR.......352,856
AUX. STEAM ENTHALPY CHANGE, KJ/KG........1,120.7	MAIN STEAM ENTHALPY CHANGE, KJ/KG...........2,209.7	ADD'L. HEAT, WATTS..... 1.459×10^5
SAT. TEMPERATURE, °C.....51.7	SAT. PRESSURE, N/m^2....13,545	
BUNDLE INFORMATION		
BUNDLE DIAMETER, m.....3.02	TUBE SHEET AREA, m^2.......7.18	WATERBOX DEPTH, m..........1.143
VOLUME, m^3...........46.8	OVERALL LENGTH, m........6.62	TUBE PITCH/DIAMETER........1.585
AREA RATIO.............0.359		
HEAT TRANSFER AND PUMPING INFORMATION		
HEAT REJECTED, WATTS...2.149×10^8	OVERALL U, W/m^2 °C......3,486	LMTD, °C..................21.9
S-W TEMP. RISE, °C....10.99	PINCH POINT, °C.........16.84	S-W FLOW RATE, m^3/sec..... 4.684
S-W FLOW RATE, ℓ/m.... 2.806×10^5	PUMPING POWER, Nm/sec..93,231	S-W PRESSURE DROP, m Water. 2.03
HEAT TRANSFER AREA, m^2...2,819	PUMPING POWER, KW.......93.2	

(6) is defined over the entire design space, thus maintaining the slope continuity requirement at the constraint boundary. Using the present approach, it has not been found necessary to sequentially update the penalty parameter, K, as is required in more traditional penalty function techniques.

Design Examples

The HEI method was coupled with COPES/CONMIN to yield the automated design capability referred to hereafter as OPCODE1 (*OP*timized *CO*ndenser *DE*sign, Version 1). Similarly, the modified version of ORCON1, as mentioned earlier, was coupled to COPES/CONMIN to yield OPCODE2.

OPCODE1 was found to be a reliable design tool. As mentioned earlier, difficulties were encountered when using OPCODE2 as a design tool stemming from the complexities encountered when coupling a complex existing analysis program to an optimization program. Therefore, examples presented here will be limited to OPCODE1. Additional examples using OPCODE1, as well as examples using OPCODE2, may be found in Johnson [16].

Unless otherwise stated, the design variables considered and the associated side constraints were:

$$15.9 \leq \text{tube outside diameter (mm)} \leq 31.8$$

$$10.3 \leq \text{tube inside diameter (mm)} \leq 30.6$$

$$1.1 \leq \text{pitch/diameter ratio} \leq 3.0$$

$$0.305 \leq \text{tube length (m)} \leq 7.62$$

$$0.914 \leq \text{cooling water velocity (m/sec)} \leq 2.74$$

In a similar way, the design constraints and the associated upper and lower bounds were:

$$0.559 \leq \text{tube wall thickness (mm)} \leq 2.77$$

$$0.305 \leq \text{bundle diameter (m)} \leq 3.05$$

$$2.78 \leq \text{terminal temperature difference (°C)} \leq 19.4$$

$$2.78 \leq \text{seawater temperature rise (°C)} \leq 11.1$$

Case One - Minimization of Pumping Power at Constant Heat Load. The objective of this test case was to minimize the pumping power requirement for a specified heat load to the condenser. The input parameters are presented in Table 3, the initial design is presented in Table 4, and the results of the optimization are presented in Table 5.

These results show an 84 percent decrease in pumping power with a commensurate 37 percent increase in heat transfer surface area, a 55 percent increase in condenser volume, and a decrease of 27 percent in overall heat transfer coefficient. The log mean temperature difference remained constant.

The lower bound on tube wall thickness constraint and the upper bound on seawater temperature rise constraint were both active. No side constraints were active and no constraints were violated. The decrease in pumping power is impressive, but so are the increases in condenser dimensions with the implied increases in weight and cost.

To achieve the design presented here, the condenser was analyzed a total of 140 times (140 function evaluations), including those analyses required to compute gradient information by finite difference. The total computation time was 87 seconds on an IBM 360/67 Computer.

Case Two - Minimization of Pumping Power at Constant Volume and Heat Load. The objective of this case was to minimize pumping power while holding condenser volume and the heat rejected constant. Because equality constraints are not dealt with explicitly in CONMIN, the volume was included in the objective function:

$$\mathrm{OBJ} = \mathrm{POWER} - K(\mathrm{VOL}) \tag{8}$$

where K is a positive multiplier which acts as a penalty parameter to increase the volume. Volume was also included as a design constraint with an upper bound of 26.3 cubic meters as the target value. In this way, condenser volume was held constant to within 1.3 percent.

The input parameters are presented in Table 3, the initial design is presented in Table 4, and the results of the optimization are presented in Table 6.

The pumping power was reduced by 35 percent. The log mean temperature difference and the heat transfer area remained essentially constant, while the tube wall thickness decreased 52 percent, causing a ten percent increase in the tube material correction factor. The seawater velocity was reduced by 16 percent, however, and the overall heat transfer coefficient remained essentially constant. The volume constraint was active at the optimum. There were no other active constraints, no violated constraints, and no active side constraints. This design required 95 analyses and 60 seconds of computer time.

Case Three - Maximization of Heat Load at Constant Pumping Power. The objective of this case was to maximize the heat rejected while holding pumping power constant. In this case, the objective function was

$$\mathrm{OBJ} = \mathrm{QREJ} + K(\mathrm{POWER}) \tag{9}$$

where K is a positive multiplier which acts as a penalty parameter to increase the power. Power was also added as a design constraint with a target value of 93.3 kw as the upper constraint. In this way, the pumping power was constant within one percent. Turbine exhaust steam and auxiliary exhaust steam were both added as design variables to provide the driving force to increase the heat rejection rate.

The input parameters are presented in Table 3, the initial design is presented in Table 4, and the results of the optimization are presented in Table 7. These results show an 86 percent increase in the heat rejection rate, with an accompanying 88 percent increase in surface area. The log mean temperature difference remained constant and the overall heat transfer coefficient decreased by 3.7 percent. The condenser volume increased by 78 percent, and the number of tubes increased by 96 percent.

The upper constraint on pumping power was active, as desired, and the upper constraint on the tube sheet area ratio was active. There were no violated constraints and no active side constraints. This design required 168 analyses and 100 seconds of computer time.

It should be noted that the turbine exhaust steam rate increased by 83 percent, whereas the auxiliary exhaust steam rate increased only 1.0 percent. Inspection of the component parts of QREJ explains the dominance by the main steam rate.

The rejected heat was calculated by:

$$Q_{rej} = \dot{m}_{MS}\Delta h_{MS} + \dot{m}_{AS}\Delta h_{AS} + \text{heat from other sources} \tag{10}$$

where the subscript MS corresponds to main steam and the subscript AS corresponds to auxiliary steam. Heat from other sources was assumed constant during the optimization. The product $\dot{m}_{MS}\Delta h_{MS}$ was several orders of magnitude greater than the product $\dot{m}_{AS}\Delta h_{AS}$ and it therefore dominated the optimization process as a change in the $\dot{m}_{MS}$ design variable would yield a greater design improvement than would a like change in the $\dot{m}_{AS}$ design parameter.

In all probability, the $\dot{m}_{AS}$ design variable would not begin to make an appreciable move until the $\dot{m}_{MS}$ design variable approached its upper constraint. This could not occur as the $\dot{m}_{MS}$ design variable had an unbounded upper constraint.

No attempt was made to balance the flow of steam from the two sources to more evenly distribute the incoming heat load. Balancing could be accomplished by appropriately scaling the two mass flow rates so they would have approximately the same values.

Conclusions

The intent of this investigation was to couple a numerical optimization code with both a simple computer code for condenser design based on the HEI method and a more sophisticated code, ORCON1, and to test the programs developed with a variety of test cases to prove their viability. The resulting conclusions are summarized below:

1. OPCODE1 is an excellent design tool for the conceptual design of a condenser to meet specific constraints such as length, height, and weight.
2. OPCODE1 can optimize a variety of objective functions and will yield an optimum design on which expereince-based safety factors may be applied.

Recommendations

In addition to the insight that this investigation has given into the generation of automated design programs for condenser design, it has also generated an awareness of this investigation's shortcomings. Presented herein are recommendations for improving upon and furthering development of the OPCODE family of condenser design programs.

1. OPCODE1 should be expanded to include the capability for multi-pass condenser calculations, geometrical options, material costing equations, strength of material considerations, and an algorithm to compute steam velocity through the rows of tubes to ensure that the velocity remains realistic.
2. ORCON1 should be rewritten with the intent of applying optimization techniques to the new program. The new program should incorporate revised shell-side heat transfer coefficients, and should allow for heat transfer enhancement techniques. The revised ORCON1 should be coupled with COPES/CONMIN to provide a sophisticated condenser design code.

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