PRELIMINARY DESIGN AND INTEGRATION PROCEDURES FOR GAS TURBINE INTERCOOLERS ON NAVAL COMBATANTS

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(ABSTRACT)

The methodology used in analyzing the feasibility of installing direct and indirect intercooling systems on naval gas turbines is presented. The indirect system is comprised of two types of heat exchangers; an air to ethylene glycol, plate fin heat exchanger, and an ethylene glycol to seawater shell and tube heat exchanger. The direct system utilizes an air to seawater shell and tube heat exchanger. The analysis requires, as input, air mass flow rates, compressor efficiencies and pressure ratios. The output, based on given environmental constraints and an assumed overall intercooler effectiveness, provides mass flow rates of seawater and ethylene glycol, heat exchanger effectiveness and size, intermediate fluid temperatures, and air and seawater outlet temperatures. The output provides preliminary data for specific heat exchanger design and pump and piping selections.

ACKNOWLEDGMENTS

The author wishes to acknowledge the significant contributions made by Mr. John Purnell for his size and weight analysis of intercooling systems employing an intermediate fluid loop, and to Mr. John Wohlgemuth for his efforts on the size and weight analysis of direct intercooling systems, both of whom are in the Engines Branch, DTNSRDC.

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NOMENCLATURE

A	Surface Area, [ft^2]
BTU	British Thermal Unit
C	Heat Capacity, [BTU/s-R]
C_{MAX}	Heat Capacity, largest in system, [BTU/s-R]
C_{MIN}	Heat Capacity, smallest in system, [BTU/s-R]
C_{PA}	Specific Heat of Air at Constant Pressure, [BTU/lbm-R]
C_{PE}	Specific Heat of Ethylene Glycol, [BTU/lbm-R]
C_{PSW}	Specific Heat of seawater, [BTU/lbm-R]
E_{OA}	Effectiveness of Overall System
E_{PF}	Effectiveness of Plate-Fin Heat Exchanger
E_{ST}	Effectiveness of Shell-and-Tube Heat Exchanger
LMTD	Log Mean Temperature Difference
M_A	Mass Flow Rate of Air, [lbm/s]
M_E	Mass Flow Rate of Ethylene Glycol, [lbm/s]
M_{SW}	Mass Flow Rate of Seawater, [lbm/s]
nc	Compressor Efficiency
NTU	Number of Heat Transfer Units of a Heat Exchanger
P	Pressure, [lbf/in^2]
Q	Rate of Heat Transfer, [BTU/s]
R	Degrees Rankine
s	Time in Seconds
T_{AMB}	Ambient Temperature, [R]
T_{A1}	Temperature, Intercooler Inlet Air, [R]
T_{A2}	Temperature, Intercooler Outlet Air, [R]

T_{CI} Temperature, Inlet Of Cold Fluid, [R]

T_{CO} Temperature, Outlet Of Cold Fluid, [R]

T_{EC} Temperature, Minimum Ethylene Glycol, [R]

T_{EH} Temperature, Maximum Ethylene Glycol, [R]

T_{HI} Temperature, Inlet Of Hot Fluid, [R]

T_{HO} Temperature, Outlet Of Hot Fluid, [R]

T_{SW1} Temperature, Seawater Inlet, [R]

T_{SW2} Temperature, Seawater outlet, [R]

ΔT_{lm} Log Mean Temperature Difference, [R]

ΔT_m Arithmetic Mean Temperature, [R]

U Overall Conductance for Heat Transfer, [BTU/ft^2-s-R]

1,2,...State Points

INTRODUCTION

The U.S. Navy is committed to the development and maintenance of efficient gas turbines for main propulsion on its surface fleet in the 3000 to 10,000 ton displacement range. There have been numerous attempts to "add on" to existing gas turbine systems, the Rankine Cycle Energy Recovery program, RACER, being the most recent example. The work described in this paper, in contrast, aims to promote the adaptation of modern simple cycle, aircraft derivative, gas turbine engines to the intercooled and regenerated cycle. The use of proven turbomachinery and its aerodynamic and thermodynamic characteristics will result in an engine that may not have to be significantly altered to be compatible with the addition of heat exchangers. That does not imply that the level of technology will not be greater than that used in engines currently in the fleet, but rather, prudent design practices will be used in deriving an engine suitable for marine use from the latest aero-engine.

Although the entire engine system works in an intercooled and regenerated cycle, only the intercooler system is discussed in this thesis. More specifically, the evolutionary process used to determine the size, capacity, and shipboard impact of the intercooler system on a modern naval combatant is presented. The discussion entails both new, or forward fit, designs of proposed ships and possible back fit designs on existing ships. In a back fit design, the existing main propulsion engine system will be entirely removed and replaced by the intercooled and regenerated engine system. Also included is a comparison of an indirect heat transfer, or intermediate loop system, with a direct intercooler system. The intermediate loop intercooler system is examined because of the potential to use a smaller heat exchanger external to the gas turbine module. Despite the additional fluid system and its associated equipment, a smaller heat exchanger is in general easier to place in a ship. In fact, the heat exchanger and intermediate fluid pump can be positioned away from the gas turbine, to effect a better machinery arrangement, with no decrease in the performance of the engine. The actual intercooler in this case, an air to intermediate fluid heat exchanger, is mounted on the engine and enclosed in the gas turbine module. This is done to minimize the air side pressure losses. The penalty for locating the intermediate fluid to seawater heat exchanger away from the gas turbine is the need for longer and therefore heavier pipe systems. The gas turbine's overall pressure ratio, the pressure ratio split, and shipboard interfaces determine which system is superior for a particular application. The design assumptions and results are presented throughout the text. The results are plotted to illustrate the trends that lead to the final conclusions. The plotted results are also a pictorial of the evolutionary process required to obtain a "workable" preliminary design.

INTERCOOLER DESIGN METHODS AND CONSTRAINTS

One of the most basic and perhaps obvious design constraints for shipboard intercoolers is a limitation on the deck area and height available. Therefore, one of the initial design goals is to minimize the required heat transfer area. The first system examined is an intermediate loop intercooler system. The system is comprised of a shell-and-tube heat exchanger for the intermediate fluid to seawater loop and a plate-fin heat exchanger for the compressed air to intermediate fluid loop. The intermediate fluid chosen for this analysis is an ethylene glycol solution. The ethylene glycol solution was chosen for its good heat transfer characteristics and because it does not promote corrosion. The specific heat, C_{PE}, of the solution can approach 1.0 (BTU/lbm-R) at low ethylene glycol concentrations. The significance of this is realized numerous times within the analysis.

Attempts to minimize the size of the ethylene glycol to seawater heat exchanger in the intermediate loop system resulted in a limit on the mass flow rates of seawater and ethylene glycol. The flow rate of seawater is limited to 2000 gallons per minute for each engine, and the flow rate of the ethylene glycol solution may not exceed that of the seawater. The details of this decision are explained in the next section of this thesis. The assumptions and their justifications are presented in the following text. The specific heat of seawater, C_{PSW}, is assumed to be 1.0 (BTU/lbm-R). The percent solution by weight of ethylene glycol is 20 percent. Higher concentrations of ethylene glycol could not be used without a significant increase in viscosity. This would lead to a reduction in the Reynolds number and heat transfer coefficient. The mass flow rate of the fluid would have to be increased to enable the heat exchanger to reach the desired level of heat transfer effectiveness. The specific heat of the ethylene glycol solution is expected to be in a range of between 0.95 and 0.98 (BTU/lbm-R) at the temperatures it will be exposed to. These assumptions and constraints placed on the two working fluids lead to the conclusion that the heat capacity of seawater, C_{SW}, will always be greater than that of the ethylene glycol solution, C_E, where the heat capacity is defined as the product of mass flow rate and specific heat. Since the mass flow rate of the low pressure compressor discharge air and its associated temperatures are known, the specific heat at constant pressure of air, C_{PA}, can be determined. C_{PA} is calculated at an average value of the air inlet and outlet temperatures, and is approximately 0.24 (BTU/lbm-R). Using these estimated values for the specific heats of the working fluids the following relationship between the heat capacities of seawater, ethylene glycol, and air can now be established:

$$C_{SW} > C_E > C_A. \tag{1}$$

In accordance with reference [1] the effectiveness of a heat exchanger is defined as

$$E = \frac{C_{HOT}(T_{HI} - T_{HO})}{C_{MIN}(T_{HI} - T_{CI})} \tag{2}$$

where C_{HOT} and C_{MIN} are the heat capacities of the hot fluid and of the fluid with the smallest capacity, respectively. An examination of equation (1) reveals that the capacity of the low pressure compressor discharge air is the smallest of the 3 fluids, and also equal to C_{HOT}. Therefore, the capacity ratio in equation (2) will cancel. The capacity ratio C_{HOT}/C_{MIN} for the ethylene glycol to seawater system is also equal to 1.0. As a result, the overall effectiveness can be defined as

$$E_{OA} = \frac{T_{A1} - T_{A2}}{T_{A1} - T_{SW1}} . \tag{3}$$

In order to establish heat exchanger size trends, two methods of analysis are compared, the LMTD and NTU methods. The LMTD method is used in a strictly counterflow heat exchanger. The LMTD of the ethylene glycol to seawater heat exchanger is defined as

$$\Delta T_{lm} = \frac{(T_{EH}-T_{SW2})-(T_{EC}-T_{SW1})}{\ln\left[\frac{(T_{EH}-T_{SW2})}{(T_{EC}-T_{SW1})}\right]} . \qquad (4)$$

For multipass heat exchangers the mean temperature is found by applying a correction factor to the log mean temperature difference. The correction factor is a function of both the capacity ratio, C_{COLD}/C_{HOT}, and the heat exchanger effectiveness. Because of the complexity of the equations used to determine the correction factor, the use of the LMTD method is frequently avoided in multipass applications. Another reason for avoiding the LMTD method is that many times it requires an iterative solution. Nevertheless, using the definition of the log mean temperature difference as a preface, a heat exchanger size trend can be developed if the overall effectiveness is assumed to be a given design parameter. The overall rate of heat transfer, Q, can be calculated from the equation

$$Q = M * C_P(T_{HOT}-T_{COLD}) \qquad (5)$$

where on the air side of the heat exchanger, T_{HOT} equals the air inlet temperature, T_{A1}, T_{COLD} is the air outlet temperature, T_{A2}, C_P is the specific heat at constant pressure of air, C_{PA}, and M is the mass flow rate of air within the compressor. M and T_{A1} can be obtained either through a cycle analysis or from the engine manufacturer. The value of T_{A2} is determined from equation (3) because the effectiveness and the seawater inlet temperature are specified in the design criteria, and the air inlet temperature is known as mentioned above. The specific heat at constant pressure is, as mentioned earlier, approximated using the arithmetic mean between T_{A1} and T_{A2}. With the overall rate of heat transfer known from equation (5), and the design criteria supplied in Appendix A, any of the fluid temperatures can be determined. Using the LMTD, equation (4), the general equation for the rate of heat transfer becomes,

$$Q = UA\Delta T_{lm}. \tag{6}$$

From equation (6) trends can be developed that show the changes in heat transfer area, A, with respect to changes in various fluid temperatures for counterflow heat exchangers. In equation (6) the quantity U, the overall heat transfer coefficient, while not specifically known, is assumed nearly constant as suggested in reference [1]. The size trend is thus based on the quantity UA rather than A alone.

The alternative to using the LMTD method and its associated correction factors for multipass heat exchangers is the NTU method, where NTU, the number of heat transfer units, is defined as

$$NTU = \frac{UA}{C_{MIN}} \tag{7}$$

In general, the NTU method appears to be superior to the LMTD method in that no correction factors are required, thus simplifying the procedure. Additionally, there is no need to iterate on an unknown temperature as is often the case with the LMTD method. A more detailed comparison of the two methods is found in reference [1]. In light of the discussion above, only the NTU method is used in the following analysis for both the shell-and-tube and plate-fin heat exchanger.

For the analysis of the shell-and-tube heat exchanger the expression relating the effectiveness and NTU is

$$E = \frac{2(1-Z)}{(1+R)(1-Z)+\sqrt{(1+R^2)}(1+Z)} \tag{8}$$

where $R=C_{MIN}/C_{MAX}$, $Z=e^{-\Gamma}$, and $\Gamma=NTU\sqrt{(1+R^2)}$. This equation is specifically for multipass, overall counterflow heat exchangers with the shell side mixed. This specific equation is used because it is the best representation of the anticipated configuration for an indirect intercooler. Solving equation (8) for NTU:

$$NTU = \frac{-\ln\left[\frac{P-1}{P+1}\right]}{Y} \qquad (8a)$$

where $P = \frac{2-EX}{EY}$, $X=1+R$, $Y=\sqrt{(1+R^2)}$, and E is effectiveness.

Analogous to the LMTD method, the overall heat transfer coefficient, U, is not specifically known but is assumed to be constant. Therefore, a trend can be developed relating the effectiveness and the heat transfer area in term of the product UA. For the shell-and-tube heat exchanger the quantities C_{MIN} and C_{MAX} are the heat capacities of the ethylene glycol solution and seawater, respectively. The effectiveness is determined from equation (2). The analysis for the plate-fin heat exchanger is very similar to that of the shell-and-tube type in that the quantity NTU, from equation (7), is unknown and a relationship between the heat exchanger effectiveness and NTU must be developed. For a counterflow plate-fin heat exchanger, the appropriate relationship, as suggested in reference [1], is

$$E = \frac{1-B}{1-RB} \qquad (9)$$

Solving for NTU:

$$NTU=\frac{-\ln\left[\dfrac{E-1}{ER-1}\right]}{1-R} \tag{9a}$$

where $B=e^{-NTU(1-R)}$. For the plate-fin heat exchanger, the quantities C_{MIN} and C_{MAX} are the heat capacities of air and ethylene glycol solution, respectively. The heat exchanger effectiveness is calculated from equation (2). Once again, the product UA is used to compare heat transfer areas at various values of effectiveness. The heat exchanger effectiveness is thus a vital part of this analysis. In the intermediate loop intercooler system, 3 different heat exchanger effectiveness equations are used. They are, the overall effectiveness, and the effectiveness for the plate-fin and shell-and-tube heat exchangers. Recall that the overall effectiveness was defined by equation (3), in terms of the variables used in this thesis. The plate-fin and shell-and-tube effectiveness equations relating the effectiveness and the appropriate fluid temperatures will be presented in a similar manner.

Earlier design studies have shown significant improvements in gas turbine performance in an overall heat exchanger effectiveness range between 80 and 90 percent. By specifying an overall effectiveness, equation (3) yields the air outlet temperature, T_{A2}. Using the data in Appendix A, an intercooler system with an overall effectiveness of 80 percent results in an air outlet temperature of approximately 614°R. Raising the effectiveness decreases the air outlet temperature, thus decreasing the work consumed in the high pressure compressor, but not without penalty. In addition to the increased heat transfer area required, a major penalty is the increased probability of air side condensation at some operating points. Depending on the amount of condensate being entrained in the air flow it can be a serious problem in terms of high pressure compressor blade erosion. Bulk condensation should therefore be avoided whenever possible and practical. In order to determine when condensation is likely to occur the saturation temperature of the combustion air entering the high pressure compressor must be determined. By using the given ambient conditions and the low pressure compressor pressure ratio, the saturation temperature of the combustion air can be determined with the use of the steam tables.

The ambient conditions used in this thesis are 100°F and 100 percent humidity. A relative humidity of 100 percent is specified only for the purpose of examining condensation potential. This condition is too severe in terms of heat exchanger sizes and fluid flow rates for the entire engine cycle. Although condensation is detrimental to the engine, its occurrence is expected to be infrequent. Therefore, the design of the intercooler must not be dictated by concerns regarding condensation alone. Additional studies are required to determine a representitive ambient condition for a large range of operating conditions. For this analysis, the design is based on the conditions listed in Appendix A. Thus, the saturation temperature of the combustion air, while operating at full power, is approximately $613^{\circ}R$. Using equation (3) with the data provided and T_{A2} equal to $614^{\circ}R$, to avoid condensation, results in an effectiveness of 80 percent. It can now be concluded that at an overall effectiveness of 80 percent or lower, bulk condensation is avoided in the engine while operating at full power. The overall effectiveness is therefore fixed at 80 percent for the remainder of this analysis.

The analysis of the intermediate loop is now directed towards finding the "best mix" between the effectiveness of the plate-fin and shell-and-tube heat exchangers, while maintaining an overall effectiveness of 80 percent. The effectiveness of the plate-fin heat exchanger is defined as

$$E_{PF} = \frac{T_{A1} - T_{A2}}{T_{A1} - T_{EC}}. \qquad (10)$$

It is only a function of the cold ethylene glycol temperature T_{EC} because the air temperatures are already fixed as discussed earlier. The shell-and-tube effectiveness,

$$E_{ST}=\frac{T_{EH}-T_{EC}}{T_{EH}-T_{SW1}} \qquad (11)$$

is a function of the hot and cold ethylene glycol temperatures. An upper limit is placed upon the hot ethylene glycol temperature based on the solution's boiling point. This reduces the dependency of the shell-and-tube effectiveness on the hot ethylene glycol temperature. The boiling point of a 20 percent solution at 1 atmosphere is approximately 680°R. To allow for uncertainties, the upper limit established for the hot ethylene glycol in this analysis is 660°R. This affords a reasonable margin to account for temperature gradients in the heat exchangers. In addition, the ethylene glycol loop is pressurized, which will also enhance the margin set for boiling. The cold ethylene glycol temperature also has its practical limits. It can not exceed the temperature of the hot ethylene glycol, or drop below the seawater inlet temperature. Increasing the cold ethylene glycol temperature above the seawater inlet temperature causes a reduction in the shell-and-tube heat exchanger effectiveness and an increase in the plate-fin heat exchanger effectiveness. This implies similar trends in the size of the respective heat exchangers. Because there is a practical limitation on the effectiveness of a heat exchanger, an upper limit of approximately 92 percent was placed on the plate-fin heat exchanger. Although this is relatively high, it is attainable with current technology. Applying equation (10) it is found that the cold ethylene glycol temperature must be approximately 590^{o}R to establish an effectiveness of 92 percent.

$$T_{EC} = T_{A1} - \left[\frac{T_{A1} - T_{A2}}{E_{PF}}\right]$$

$$= 890 - \left[\frac{890-614}{.92}\right]$$

$$= 590\ ^{o}R$$

The air inlet and outlet temperatures are given from previous discussion. A cold ethylene glycol temperature of 590^{o}R reduces the effectiveness of the shell-and-tube heat exchanger to approximately 61 percent.

$$E_{ST} = \left[\frac{T_{EH} - T_{EC}}{T_{EH} - T_{SW1}}\right]$$

$$= \frac{660-590}{660-545}$$

$$= 0.609$$

An effectiveness of this magnitude is well within current technology and will be shown to have a positive effect with respect to shipboard arrangements.

One of the remaining concerns in this system is the seawater outlet temperature, T_{SW2}. In order to maintain a clean heat transfer surface on the tube side of the heat exchanger, the seawater temperature must remain below the calcium carbonate scaling temperature. Scaling can occur at a seawater temperature of approximately $165^{\circ}F$. $125^{\circ}F$ was selected as the upper limit on the seawater outlet temperature. Limiting the seawater outlet temperature to $125^{\circ}F$ allows a $40^{\circ}F$ temperature margin for temperature gradients within the heat exchanger. This not only alleviates the problem of scaling, but also places a calculated limit on the seawater mass flow rate. As seen in equation (5), if the rate of heat transfer, the inlet temperature, and the specific heat of seawater are all known, the mass flow rate depends only on the outlet temperature. Note that once Q has been determined from the engine air data it has the same value for the ethylene glycol and seawater loops also.

A complete set of design parameters has now been specified for the indirect intercooler system. A similar procedure is followed in the development of a set of design parameters for the direct intercooler system. The direct system is comprised of a shell-and-tube heat exchanger in which the low pressure compressor discharge air is on the shell side, and the seawater is within the tubes. The design goal remains to minimize the required heat transfer area. The NTU method, as previously outlined for a multipass counter flow, mixed shell side heat exchanger, is used to establish a trend of the size of the heat transfer area. The overall rate of heat transfer, Q, is calculated from equation (5) where T_{HOT} and T_{COLD} are the inlet and outlet air temperatures, respectively. The specific heat, C_p, is that of air at an average temperature between the inlet and outlet temperatures. The mass flow rate of air, M, is provided by the engine manufacturer or by cycle analysis.

The overall system performance requirements on the direct system are similar to the intermediate loop system in that an overall effectiveness of 80 percent is desired. The direct system, in contrast to the intermediate loop system, requires only one heat exchanger. Therefore, the overall effectiveness, as defined in equation (3), is equivalent to the effectiveness of the shell-and-tube heat exchanger. The NTU method is used to make required heat transfer area comparisons with respect to effectiveness as shown in equations (8), and (8a). Although increasing the effectiveness above 80 percent will enhance gas turbine performance, the analysis shows that the size of the heat exchanger also increases significantly. Therefore, the maximum effectiveness of the direct system is also set at 80 percent. The following section examines the impact that the size of the heat exchanger has on the ship design. Equation (2) shows that if the same heat capacity criteria is used in the direct system as was used in the intermediate loop analysis, an effectiveness of 80 percent will again result in an air outlet temperature, T_{HO}, of approximately 614^oR.

$$T_{HO} = T_{HI} - E(T_{HI} - T_{CI})$$
$$= 890 - 0.80(890 - 545)$$
$$= 614^oR$$

To reiterate, the same considerations and restrictions regarding condensation that were discussed in the intermediate loop system apply to the direct system.

Although both systems cause a pressure loss on the air side of the respective heat exchangers, the shell-and-tube heat exchanger used in the direct system will cause a greater one. This is due to the greater number of passes that the air must make and to the pitch to diameter ratio of the tubes in order to achieve the desired performance. A greater number of passes means that the air has a more tortuous path to follow, thus more energy input is required for the air to complete the flow path. The pitch to diameter ratio determines the available space between adjacent tubes. If the space between the tubes is decreased, implying a smaller pitch to diameter ratio assuming a constant diameter, more energy input is required to maintain the same air flow through the heat exchanger. Additionally, the manifolds used to direct the air from the engine to the heat exchanger and back to the engine cause additional losses. These losses are a result of the severe flow path bends associated with the manifolds. These and other factors have an influence on the pressure drop of the system. Studies have shown that the pressure drop is a relatively strong function of the length of the tube bundle, the number of passes, and the pitch to diameter ratio. As a result of this dependency, the current shell-and-tube heat exchanger design has 2 shell side passes and a single tube side pass. The length of the tube bundle and pitch to diameter ratio is discussed in greater detail in the next section of this thesis.

SHIP INTEGRATION AND INSTALLATION

In this section several different configurations of a main propulsion engine with an intercooler system are examined to determine their impact on a ship's machinery arrangement. The previous section discussed the intermediate loop and direct systems separately, but because both systems have heat exchangers external to the gas turbine module, they are combined in this section. Figure (1a) shows a typical indirect intercooler system in relation to the high and low pressure compressors, and the various flow paths of air, ethylene glycol, and seawater. Similarly, the direct system of intercooling is shown in figure (1b).

As in many ship design problems, the design considerations are based on arrangement limitations, while the arrangement is a reflection of the design parameters and goals. These aspects of the design will be discussed, both for new ship installation and back fit capability.

In both systems the type of heat exchanger chosen for the seawater loop is shell-and-tube with the seawater on the tube side. The direct system has air on the shell side, while the intermediate loop system has ethylene glycol on the shell side. Therefore, the direct system requires only one cooling fluid system, a seawater system. Figures (2) and (3) show two possible arrangements for a direct system. In figure (2), one heat exchanger per engine is used, while

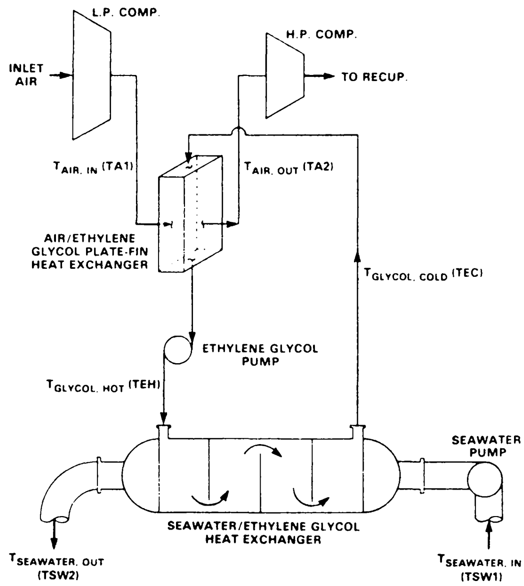


Fig. 1a. Indirect configuration.

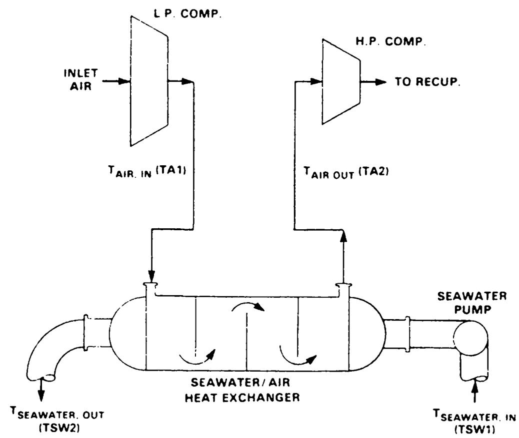


Fig. 1b. Direct configuration.

Fig. 1 Gas Turbine Intercooler System Schematic

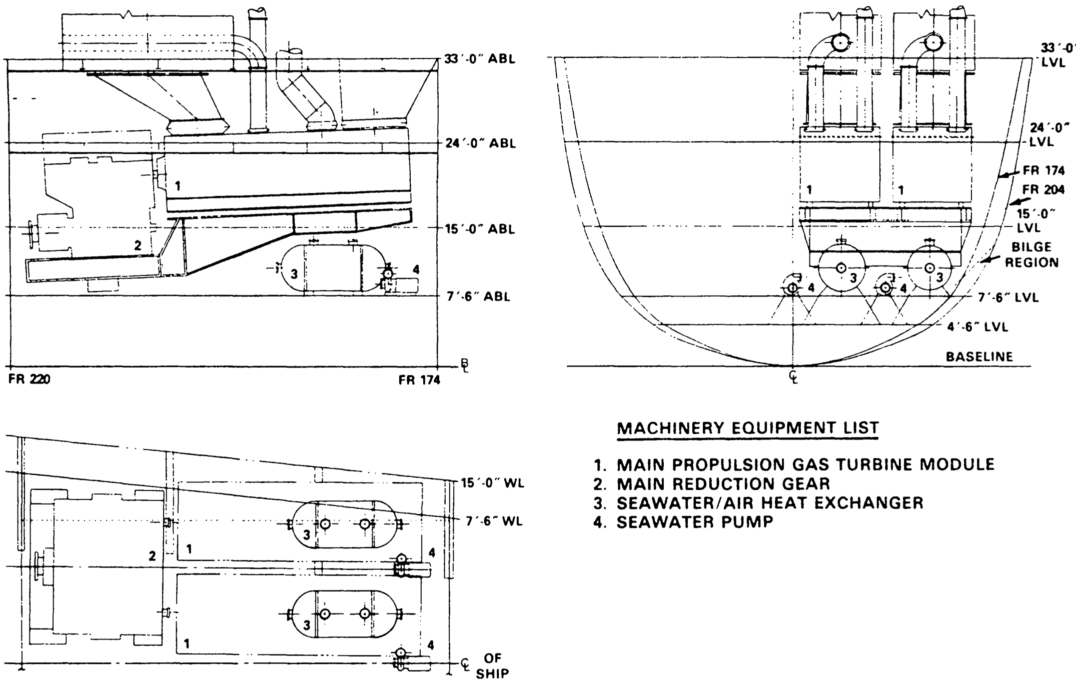


Fig. 2 Direct Intercooler System Arrangement, One Heat Exchanger Per Engine (Destroyer)

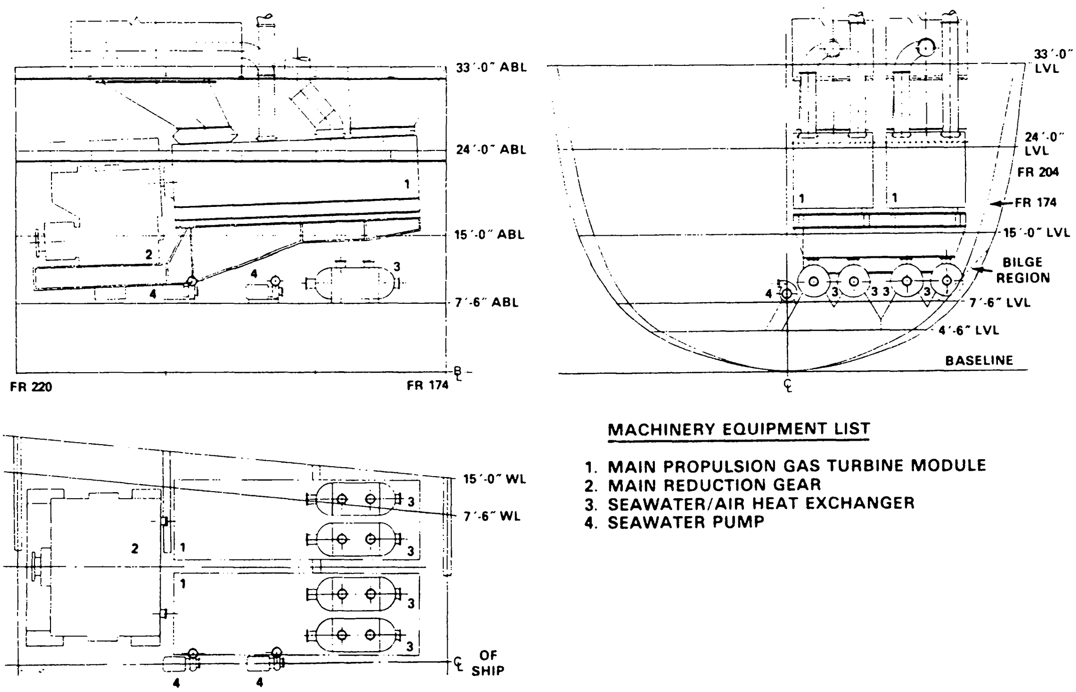


Fig. 3 Direct Intercooler System Arrangement, Two Heat Exchangers Per Engine (Destroyer)

figure (3) shows 2 heat exchangers per engine. One seawater pump is required for each heat exchanger. If the intercooling is split into two parallel shell-and-tube heat exchangers per engine, the number of seawater pumps is still expected to remain at one per engine, as shown in figure (3). One reason for splitting the intercooling into two heat exchangers in a parallel circuit is to reduce the diameter of the heat exchanger, thus making it easier to install under the gas turbine module. The seawater pumps in each system have the necessary piping system to provide seawater coolant to either engine's intercooler. With this capability, either pump can be shut down while keeping one engine on-line. In an effort to minimize the impact the heat exchangers will have on the machinery arrangement, the goal is to install them below the gas turbine module, without interfering with the foundation for the module or protruding below the tank top level. This may not be possible in every installation. It is also desirable to be able to use these systems to back fit existing ships with a relatively small impact on the current machinery arrangement. An examination of the ship classes that are currently being considered as back fit candidates, the DD 963, and CG 47, shows that the foundations and sub bases for the current gas turbine modules will have to be redesigned. The piping for the prairie and masker systems, bleed air system, and starting air system will also need to be redesigned and relocated. For future ships, the size and arrangement limitations of the intercooled engines will be known, therefore they can be designed accordingly.

Figures (4) and (5) are similar arrangement sketches to figures (2) and (3) except that the intermediate loop systems are shown. Here we see, in both the destroyer and frigate cases, two pumps per engine installation. One pump services the seawater cooling loop, and the other the ethylene glycol, because the intermediate loop intercooling system uses two fluid systems for heat removal. Due to the additional pump in the intermediate loop system, the direct system appears to effect an easier arrangement. In some cases this may be true, but a trade off between a large diameter heat exchanger and an extra pump with its associated piping, must be considered. There are numerous factors that must be examined before a choice can be made. One of these factors is the location of the gas turbine modules with respect to the tank top level and the hull. In some current installations, the modules are very close to the turn of the bilge and to the deep transverse web frames. In such a case a large diameter heat exchanger may be quite difficult, if not impossible, to install. The "tightness factor" of the machinery space must also be considered. The "tightness factor" is the ratio of the total deck area of all the equipment in a compartment and the total deck area of that compartment. A typical machinery arrangement

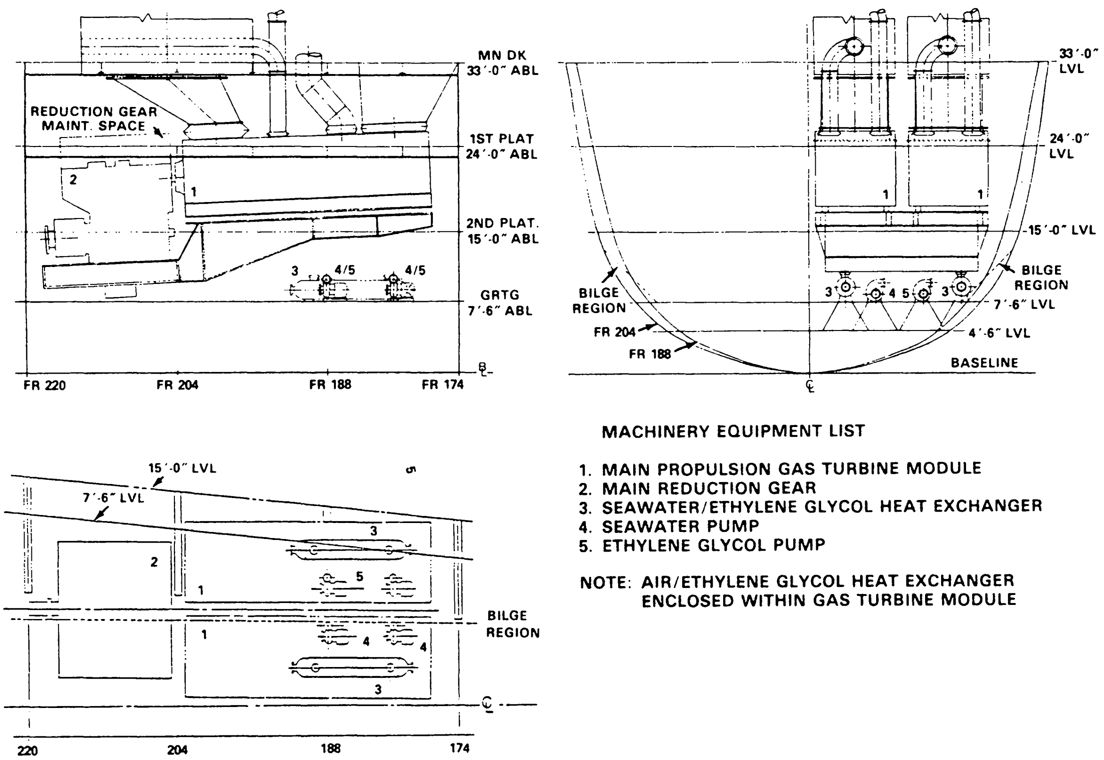
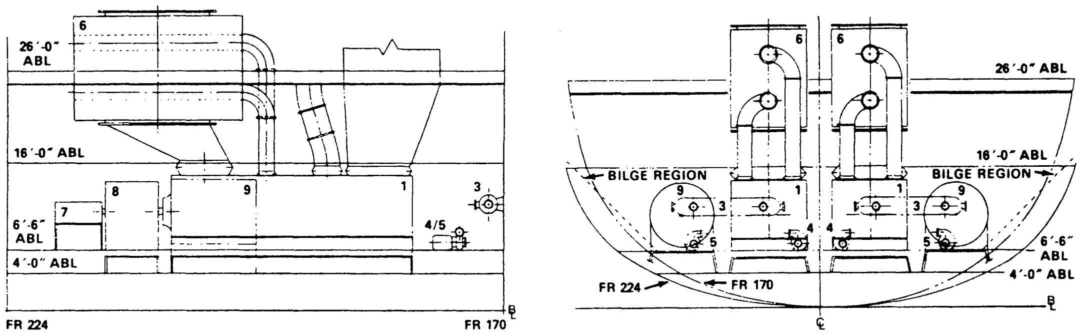
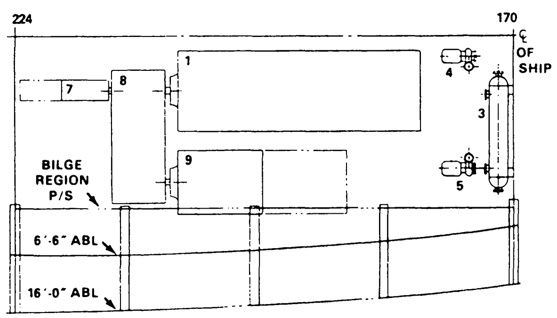


Fig. 4 Indirect Intercooler System Arrangement, (Destroyer)





MACHINERY EQUIPMENT LIST

1. MAIN PROPULSION GAS TURBINE MODULE
2.
3. SEAWATER/ETHYLENE GLYCOL HEAT EXCHANGER
4. SEAWATER PUMP
5. ETHYLENE GLYCOL PUMP
6. MAIN PROPULSION GAS TURBINE RECUPERATOR
7. SHIP SERVICE GENERATOR SET
8. COMBINING GEAR
9. MAIN PROPULSION AC SYNCHRONOUS GENERATOR SET

NOTE: AIR/ETHYLENE GLYCOL HEAT EXCHANGER ENCLOSED WITHIN GAS TURBINE MODULE

Fig. 5 Indirect Intercooler System Arrangement, (Frigate) "tightness factor" is, according to reference [2], approximately 0.30. For those ships at or above this value, an additional set of pumps may cause significant arrangement problems. As this brief discussion shows, there are still a sufficient number of machinery arrangement unknowns to warrant more detailed arrangement studies before the final selection of the intercooler systems is made.

Regardless of which intercooler system is chosen, the maximum allowable length of the heat exchanger from flange to flange must be established. Since the present U.S. Navy design criteria for a new engine states that it shall not exceed the current LM 2500 module footprint, a length not to exceed 50 percent of the module is stipulated as a heat exchanger design goal. Therefore, the heat exchanger length is limited to approximately 10 to 12 feet. This design limitation will enable relatively easy access to the tubes of the heat exchanger for maintenance and cleaning. In addition to the size of the heat exchangers the size of the pumps must also be considered. Therefore, in either system, the direct or the intermediate loop, a limit on the flow rates of both the seawater and ethylene glycol is set at 2000 gallons per minute. This limits the size of the pumps and associated piping. The size of the heat exchanger is a weak function of flow rate. That is, lowering the flow rates will cause only a small increase in the size of the heat exchanger. For example, the diameter of the direct system's heat exchanger decreases by one inch when the flow rate is doubled, while the length of the heat exchanger is held constant. The same trend is also expected for the intermediate loop system. The real advantage of restricting the flow rate is limiting the size of the pipe required.

For the intercooler systems currently under study, 8 and 6 inch schedule 40 pipes will be used for the seawater and ethylene glycol loops, respectively. These pipes will accommodate the anticipated flow rates of the seawater and ethylene glycol loops, while maintaining acceptable velocities at the inlet and exit of the heat exchangers. The weights of the respective piping systems when charged are approximately 50 and 32 pounds per linear foot. Because of the weight involved, it is desirable to keep the length of piping to a minimum. Placing the heat exchanger closer to the seawater inlet and pump would lengthen the piping on the air side but reduce the length of the heavier seawater piping. For the direct system, a trade off study must be conducted to examine the effects of a longer flow path on the air side with the associated increases in the pressure loss, versus the reduction in length and weight of the seawater piping. An important restriction to the trade off study is the requirement that the seawater pump be located close to the centerline of the ship to meet U.S. naval ship design specifications. There is no problem of air side pressure loss in the intermediate loop system due to the location of the shell-and-tube heat exchanger. The cooling of the air takes place in a plate-fin heat exchanger contained in or on the gas turbine module. A trade off study similar to that done for the direct system will determine the optimum arrangement for the seawater and ethylene glycol loops.

A review of figures (3) through (5), reveals that the outboard engine presents the greatest difficulty for the heat exchanger arrangement. In the direct system, the heat exchanger may need to be located close to the gas turbine module to minimize pressure losses in the air flow path. This will lengthen the seawater piping required. In the intermediate loop system the ethylene glycol loop will increase in length as the length of the seawater loop is decreased, (and vice versa), depending on the location of the heat exchanger. Because the seawater piping is larger and heavier, the initial goal regarding the intermediate loop system is to shorten the seawater loop. Regardless of which intercooler system is chosen, the outboard engine remains the most difficult to arrange. It is assumed in each of the cases described above that the seawater pump is located such that it will meet the U.S. naval ship design specifications. Figures (3) through (5) also indicate that the maximum acceptable diameter of the heat exchanger is determined by the space available under the outboard gas turbine module. In many cases, as noted earlier, the outboard engine is located very close to the hull, and/or deep transverse structural web frames. This may in fact limit, if not prohibit, the option of installing intercooled engine systems on existing ships without an extensive redesign of hull and foundation structures. There is still a considerable amount of intercooler system design work to be done, therefore no definite conclusions can, or should be, made at this time regarding intercooled engine installations on existing ships.

An important element of every U.S. Navy design is the need for machinery that can be easily maintained. The length limitation imposed on the shell-and-tube heat exchanger, for example, is the result of the maintainability requirement. The shell-and-tube heat exchanger length is limited, as discussed earlier, to no more than 50 percent of the gas turbine module length. If the heat exchanger is located under the gas turbine module, as expected, this length will make it possible for the ship's crew to remove the heads of the heat exchanger for inspection and/or cleaning. Additionally, sufficient space will be provided around the heat exchanger for tube bundle removal when the need arises.

The location of the various pumps used in the system must also be determined in such a manner as to permit inspection, maintenance, or removal, with relative ease. The seawater system is a relatively simple system in that the only machinery required is the pump. A plate strainer with a mesh of not greater than 1/2 inch is proposed for this system. This type of strainer is located in the sea chest, and thus has no impact on the machinery arrangement. Valves and piping make up the balance of the system. The valves may require motor controllers in various locations of the system, and will have to be incorporated in a detailed arrangement of the piping. In general, the seawater system is a low maintenance item as well as the closed ethylene glycol loop. Because the ethylene glycol loop is a closed system, no strainer is needed in the loop. One of the few requirements for the ethylene glycol loop is that there be a means of supplying makeup solution to the system in the event of a leak. One reliable solution is to incorporate a head/replenishment tank in the loop to keep a positive head on the pump, and to supply makeup ethylene glycol solution. Once again, there will be valve motor controllers in the pipe system that must be included in the arrangement of the pipes, pumps, and heat exchangers.

In the previous section, the problems of condensation and scaling in the heat exchangers were discussed and methods were proposed to avoid their occurrence. If any water condenses out of the low pressure discharge air, it is likely to occur at the exit of the heat exchanger. These water particles must be kept from entering the high pressure compressor inlet stage. In the intermediate loop system, a trap and drain system may be inappropriate on the plate-fin heat exchanger because of the plate-fin design. A means for removing the condensate must therefore be accomplished in the already constrained air ducting. One possible solution for the intermediate loop system is to meter the flow rate of the ethylene glycol, as a function of the ambient conditions, to avoid condensation. In the direct system, a moisture separator may be a possible solution to control the entrainment of water into the high pressure compressor. It is not uncommon for the tubes of a shell-and-tube seawater heat exchanger to "sweat" when operating in a high relative humidity. Large droplets, which are common in "sweating" conditions, are relatively easy to separate from the flow path, albeit at the expense of a higher pressure loss, and greater volume and weight. These are some of the most current measures being studied to control the initiation of condensation, and protect the high pressure compressor in the event of condensation.

While condensation control is a concern, it is estimated, by way of a climatic survey of the world, reference [3], that such conditions may exist only 1 to 2 days a year. Therefore, elaborate methods to control or eliminate condensation should not be the driving force in the design process.

The problem of calcium carbonate scaling can also have an effect on the arrangement of the heat exchangers. Although the seawater outlet temperature has been limited to

125°F to avoid scaling, some hot spots may exist within the heat exchangers, and localized scaling may occur. If scaling does occur the heat exchanger should be located such that there is relatively easy access to the tube bundle. Therefore, a maintenance area should be allocated for the ends of the heat exchanger to enable the tubes to be inspected and cleaned or removed. This reinforces the need to limit the length of the heat exchanger to no more than 50 percent of the length of the gas turbine module.

DISCUSSION OF RESULTS

The design day criteria for this analysis, as noted earlier, is an ambient temperature of 100°F, 100 percent relative humidity, and 85°F seawater. Based on this information and the design parameters presented in the previous section, an overall heat exchanger effectiveness of 80 percent was chosen for this study. The initial reason for this choice was the intent to minimize the size of the intercooler system without a large gas turbine performance penalty and, to a lesser degree, to control the occurrence of condensation in the heat exchangers. These parameter values resulted in a lower limit on the intercooler outlet air temperature. The shell-and-tube heat exchanger effectiveness for the direct intercooler system is equivalent to the overall effectiveness, equation (3). Since the air and seawater inlet temperatures, T_{A1} and T_{SW1}, are known, only the intercooler outlet air temperature, T_{A2}, is affected by the effectiveness. In the intermediate loop system, the effectiveness of both the plate-fin and shell-and-tube heat exchangers are limited by the overall effectiveness of 80 percent. Using the known air inlet temperature, T_{A1}, and the cold ethylene glycol temperature, T_{EC}, equation (10) yields a plate-fin effectiveness of approximately 92 percent. The effectiveness of the shell-and-tube heat exchanger in the intermediate loop system that results in an overall effectiveness of 80 percent is therefore fixed at about 61 percent. The advantage of this conservative shell-and-tube heat exchanger effectiveness lies in a weight and volume savings. The method for determining the temperatures in equation (11), (the effectiveness of the shell-and-tube heat exchanger), was discussed in the previous section. Prefaced with the information above, figure (6) is presented to illustrate the effects of the seawater mass flow rate and ethylene glycol fluid temperatures on the exit temperature of the air.

Figure (6) is a result of a series of calculations in which the hot ethylene glycol temperature, T_{EH}, remains constant at $660^{o}R$, and the seawater outlet temperature, T_{SW2}, is at or near its limit of $125^{o}F$. The computer code listed in Appendix B was used to perform the calculations where the overall effectiveness was varied from 70 percent to 90 percent and the cold ethylene glycol temperature, T_{EC}, varied from $550^{o}R$ to $590^{o}R$ in 10 degree increments. The objective of this data set was to show the large range in overall effectiveness at various intermediate fluid temperatures and the corresponding minimum seawater mass flow rate. The solid line with a negative slope in figures (6a) and (6b) is the seawater outlet temperature limit of $125^{o}F$ at the appropriate seawater mass flow rate. Any data points above and to the right of this line will satisfy the seawater scaling criteria.

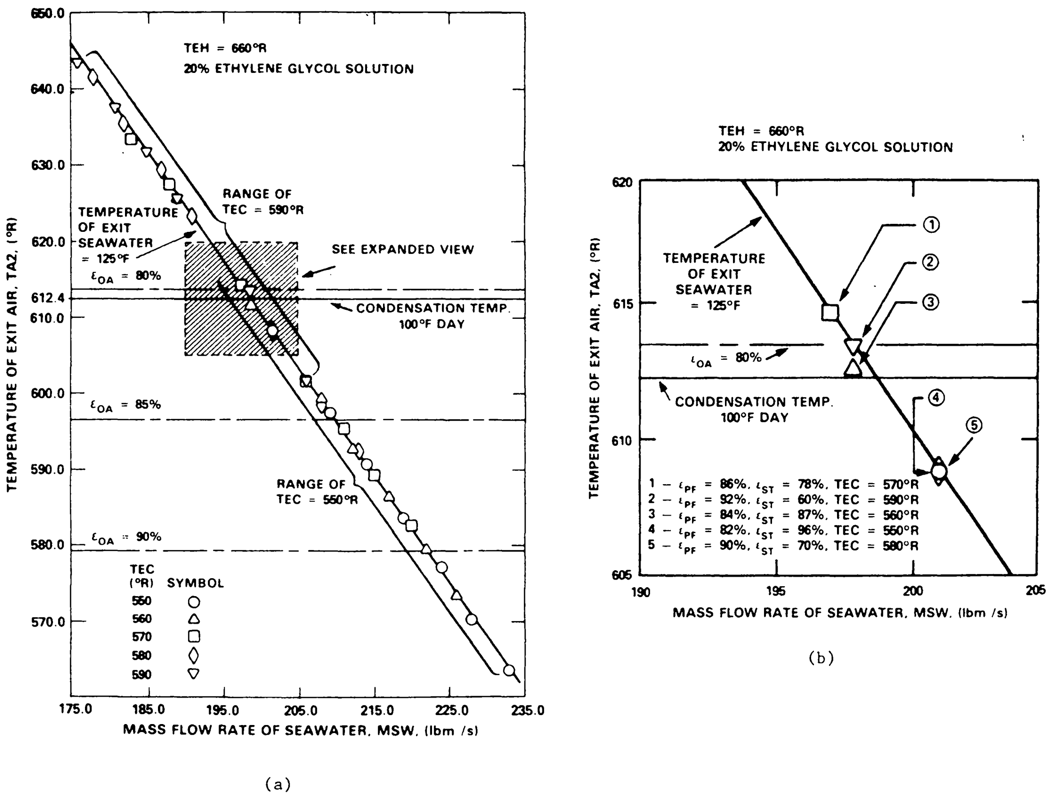


Fig. 6 Exit Air Temperature And Seawater Mass Flow Rate As A Function Of Overall Effectiveness And Ethylene Glycol Temperatures

Figure (6) shows that because the cold ethylene glycol temperature is not a function of the overall effectiveness, a range of cold ethylene glycol temperatures, between 550^{O}R and 590^{O}R satisfies the criterion for an overall effectiveness of approximately 80 percent. When the hot ethylene glycol temperature and the overall effectiveness are held constant, the cold ethylene glycol temperature determines the effectiveness of both the shell-and-tube and plate-fin heat exchangers. Therefore, for each value of the cold ethylene glycol temperature there is a unique plate-fin and shell-and-tube effectiveness.

In both the direct and intermediate loop systems the seawater outlet temperature is a direct function of the seawater mass flow rate. This relationship can be shown using equation (5) with the following known data: Q, the rate of heat transfer, is approximately 7900 (BTU/s), the specific heat, C_P, of seawater is assumed to be 1.0 (BTU/lbm-R), T_{COLD}, the seawater inlet temperature is 545^{O}R, and T_{HOT}, the seawater outlet temperature is limited to 585^{O}R. Solving for M, the mass flow rate of seawater, it can be shown that the minimum mass flow rate occurs at the maximum seawater outlet temperature, 585^{O}R. Note that the rate of heat transfer has been determined by substituting the value of T_{A2}, from equation (3) at an effectiveness of 80 percent, into equation (5) as T_{COLD}, with M, C_P, and T_{HOT} obtained from cycle analysis or supplied by the engine manufacturer. Because one of the stated objectives of the intercooler system is to minimize the impact it has on the machinery arrangement of a ship, a relatively low seawater mass flow rate is desired. This was discussed in detail in the previous section. Figure (6) shows that in order to obtain an overall effectiveness of approximately 80 percent, the seawater mass flow rate must be approximately 200 (lbm/s) for the criteria discussed earlier.

Figure (7) is constructed to show the relationship between the shell-and-tube and plate-fin heat exchangers and the seawater mass flow rate and the ethylene glycol temperatures. The lines labled shell-and-tube effectiveness are horizontal because the hot ethylene glycol temperature is fixed at 660^{o}R and the seawater inlet temperature at 545^{o}R. Therefore, regardless of the mass flow rate of the seawater, the effectiveness of the shell-and-tube heat exchanger remains the same for a constant value of the cold ethylene glycol temperature. The plate-fin effectiveness is dependant on the cold ethylene glycol temperature and the air outlet temperature. The air outlet temperature is a function of the overall effectiveness as noted earlier. Therefore, for the criteria described in this thesis, each data point on the curves labeled as plate-fin effectiveness represent a separate overall effectiveness. Only the line of overall effectiveness equal to 80 percent is shown in figure (7). This vertical line shows that there is a large range of in-

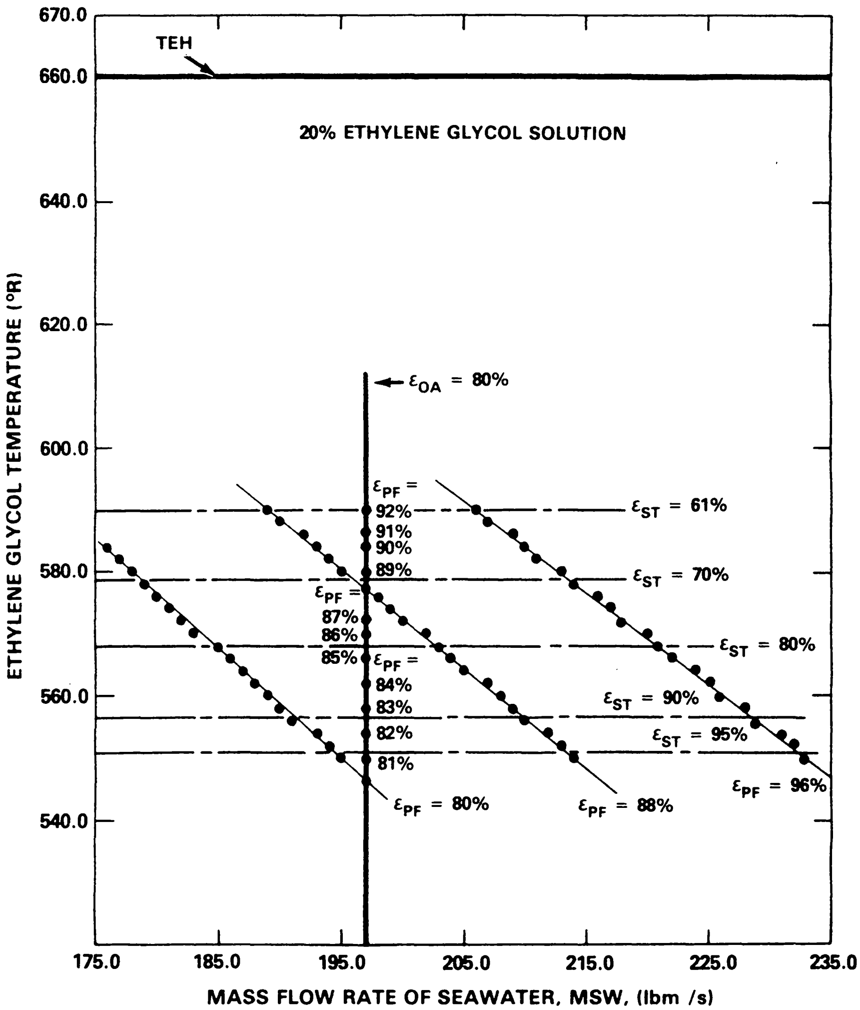


Fig. 7 Plate-Fin And Shell-And-Tube Heat Exchanger Effectiveness As A Function Of Ethylene Glycol Temperatures And Seawater Mass Flow Rate

dividual heat exchanger effectiveness values to meet that condition. Noting this fact, it is desirable to find the "optimum" balance between the effectiveness of the shell-and-tube and plate-fin heat exchangers. The trends that lead to the selection of the "optimum" balance are illustrated in figure (8). Equations (5), (10), and (11) were used to generate the data for figure (8). From equation (10), it is shown that as the cold ethylene glycol temperature increases while the intercooler inlet and outlet air temperatures remain constant, the plate-fin effectiveness increases. The resulting increase in the plate-fin effectiveness translates to as much as a 24 percent increase in required heat transfer area. A similar trend for the shell-and-tube heat exchanger is shown in figure (8), using equation (11). Because the hot ethylene glycol and seawater inlet temperatures are fixed, the cold ethylene glycol temperature directly affects the shell-and-tube effectiveness. Thus, as the cold ethylene glycol temperature increases, the shell-and-tube effectiveness decreases. The resultant decrease in effectiveness produces a decrease in required heat transfer area, for the shell-and-tube heat exchanger, by as much as 56 percent.

Figure (8) also shows, using equation (11), that as the cold ethylene glycol temperature varies from $550^{o}R$ to $590^{o}R$, the plate-fin effectiveness rises by approximately 12 percentage points, from 80 to 92 percent while the overall ef-

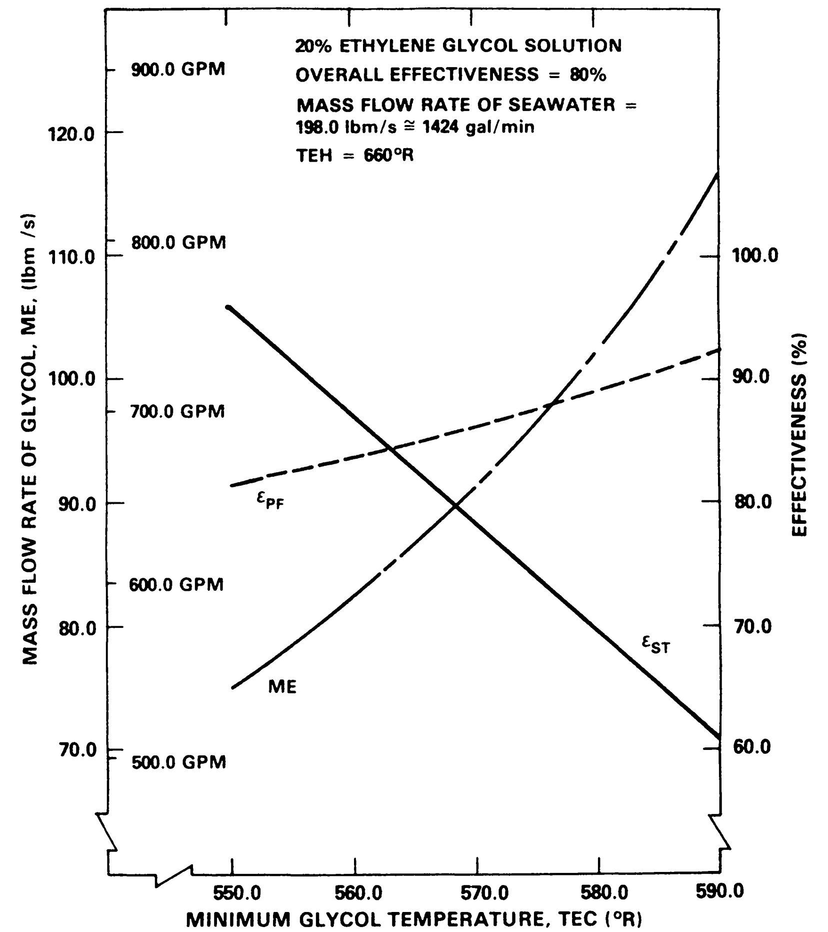


Fig. 8 Ethylene Glycol Mass Flow Rate And Heat Exchanger Effectiveness As A Function Of The Minimum Ethylene Glycol Temperature

fectiveness is constant at 80 percent. The effectiveness of the shell-and-tube heat exchanger for the same conditions drops even more dramatically by 35 percentage points, from 96 to 61 percent. A review of equations (10) and (11) shows that if the overall effectiveness is fixed at 80 percent, the only variable is the cold ethylene glycol temperature. The importance of the cold ethylene glycol temperature is that it controls the mass flow rate of the ethylene glycol solution. Equation (5) illustrates this fact, where the rate of heat transfer, Q, and the hot ethylene glycol temperature, T_{HOT} in this case, are fixed for the system, and the specific heat of a 20 percent solution of ethylene glycol varies only slightly in the expected range of temperatures. Therefore, by maximizing the temperature difference in the ethylene glycol solution, the mass flow rate of the ethylene glycol solution is minimized. For the direct system, both the seawater inlet and outlet temperatures are fixed design parameters, and the rate of heat transfer depends entirely on the overall effectiveness chosen. Once the rate of heat transfer is determined, the seawater mass flow rate is determined according to equation (5). The only change that will influence the seawater mass flow rate in the direct system is the lowering of the seawater inlet temperature, but this would result in a drop in overall effectiveness below 80 percent if all other temperatures remain constant.

Figure (9), derived from the data shown in Appendix A and earlier design studies, shows the effect of the cold ethylene glycol temperature on the weight of the shell-and-tube heat exchanger, and pump. It is important to note that the trends shown agree with those in figures (6) through (8). More specifically, the required heat transfer surface area of the shell-and-tube heat exchanger is reduced at higher values of the cold ethylene glycol temperature.

The results of the size and weight versus pitch to diameter ratio study for the direct intercooler system, are shown in figure (10). Each data point shown satisfies the requirement for the overall effectiveness at a reasonable preliminary design allowable pressure drop. For this study, it appears that there is a range of choices in the selection of a direct system heat exchanger. Pitch to diameter ratios between 1.7 to 1.875 are all attractive choices. The two data points on either end of the curves are excluded from further consideration mainly due to the large weight associated with them. Additionally, the shell diameter at a pitch to diameter ratio of 2.0 is expected to be too large for shipboard installations.

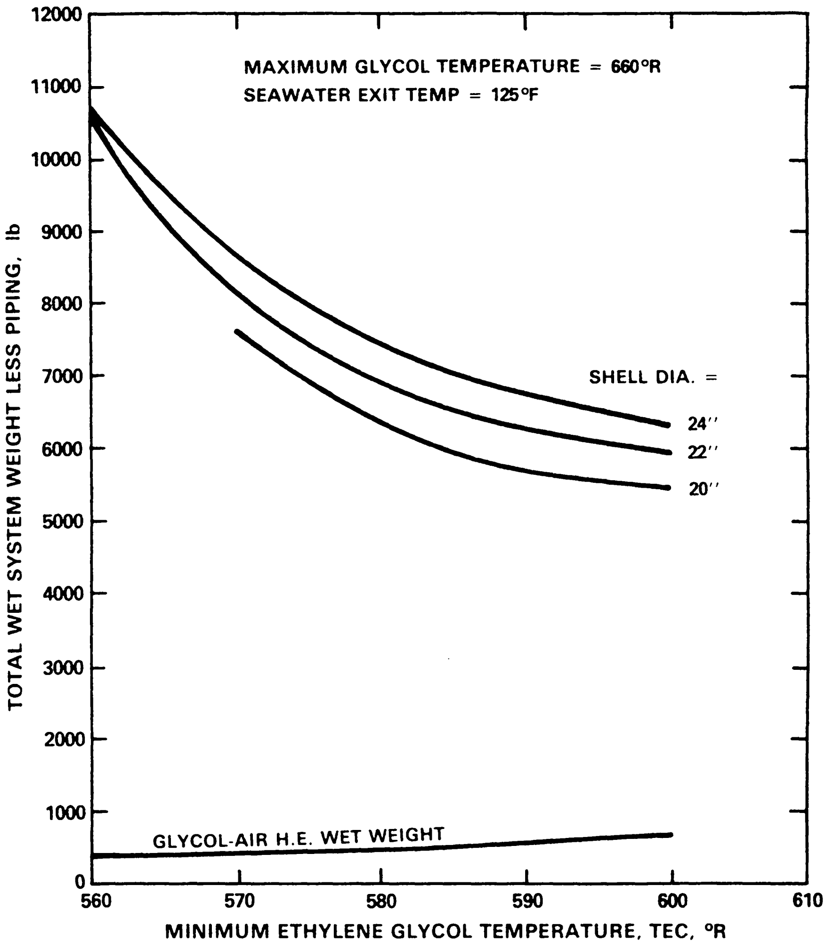


Fig. 9 Heat Exchanger Weights At Various Shell Diameters And Minimum Ethylene Glycol Temperatures

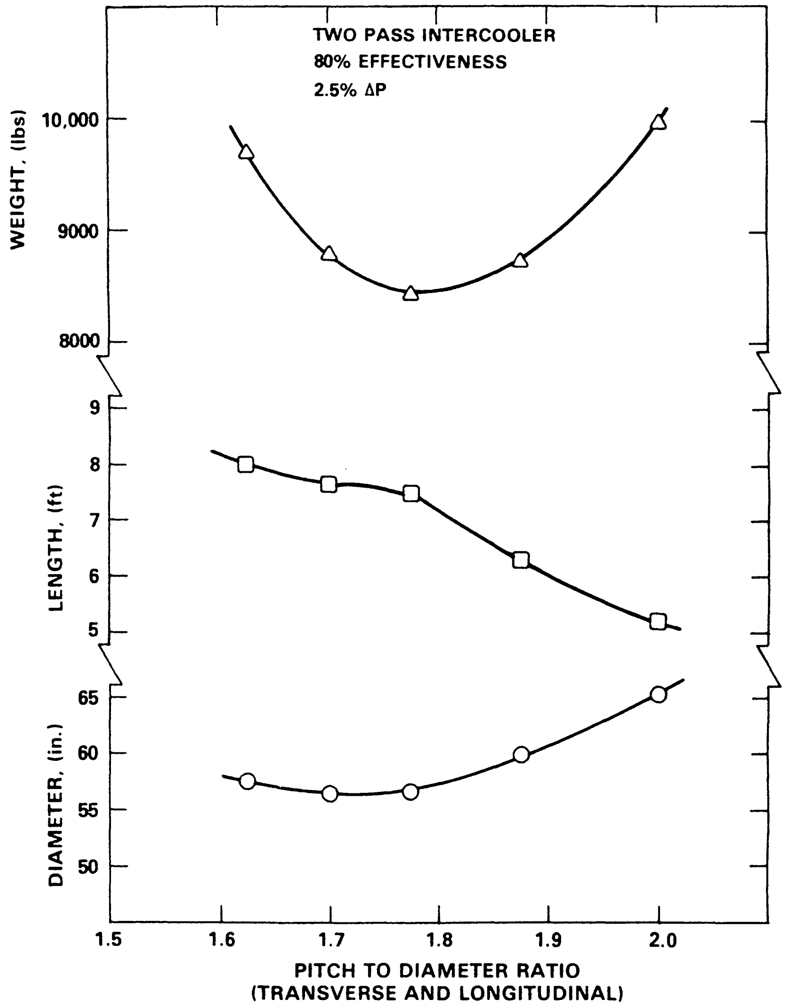


Fig. 10 Shell-And-Tube Heat Exchanger Weight, Length, And Diameter As A Function Of The Pitch To Diameter Ratio

REMARKS AND CONCLUSIONS

The analysis and material presented here is only a small part of the total gas turbine intercooler system design process. Nevertheless, it has proven to be a useful tool in reducing the number of trial iterations necessary in sizing an intercooler system. Equally important, the design trends that lead to a satisfactory design have been established. These trends will aid the detail designer in choosing a final system configuration and the level of performance of the system.

Currently, this analysis is a preface to heat exchanger codes that deal with specific geometry and flow path characteristics. Future plans are to link this method to the appropriate heat exchanger design codes to further reduce the design iteration process. A Fortran listing of a preliminary performance and sizing code used in this analysis is included in Appendix B.

The results based on this analysis lead the designer towards the following:

- minimum shell-and-tube heat exchanger size and weight;
- maintaining a minimum cold ethylene glycol temperature to prevent air side condensation; and

- minimizing the pumping capacities required, ie.,mass flow rates, for seawater and ethylene glycol.

The comparison shown in table (1) illustrates the similarities and differences between the direct and intermediate loop intercooler systems on a per engine basis.

A preliminary assessment of the overall ship systems impact can be made from the data shown. The comparison includes only the major equipment within each system, but is sufficient to indicate an overall trend that is not expected to change as the system becomes more defined. Currently, the intermediate loop system is nearly 250 percent more dense than the direct system, as shown in table(1). If a direct system using 2 seawater to air heat exchangers, each with a volume of approximately 110 cubic feet is used, the density disparity decreases to 210 percent. Although the intermediate loop system is more dense, there are more components in the system, therefore implying increased maintenance and support.

Additionally, the increased number of pumps and their associated pipe systems create a more difficult machinery arrangement design. Each system, the direct and indirect, has the ability to transfer sufficient amounts of heat from the engine air flow to the seawater, but the indirect system must use two fluid loops. According to the observations

TABLE 1: SYSTEM COMPARISON

(on a single engine basis)

Direct Intercooler System		Intermediate Loop Intercooler System	
WEIGHT:			
SEAWATER TO AIR HEX	8750 lb	ETHYLENE GLYCOL TO AIR HEX	800 lb
SEAWATER PIPING	2500 lb	SEAWATER TO ETHYLENE GLYCOL HEX	6500 lb
SEAWATER PUMP	1200 lb	ETHYLENE GLYCOL PIPING	1600 lb
MISC. VALVES, FLANGES	1300 lb	ETHYLENE GLYCOL PUMP	1000 lb
		SEAWATER PIPING	2500 lb
		SEAWATER PUMP	1200 lb
		ETHYLENE GLYCOL STORAGE	450 lb
		MISC. VALVES, FLANGES	1800 lb
TOTAL	13750 lb	TOTAL	15850 lb
VOLUME:			
SEAWATER TO AIR HEX	293 FT^3	*ETHYLENE GLYCOL TO AIR HEX	-----
SEAWATER PIPING (50 FT)	25 FT^3	SEAWATER TO ETHYLENE GLYCOL HEX	52 FT^3
SEAWATER PUMP	30 FT^3	ETHYLENE GLYCOL PIPING	15 FT^3
		ETHYLENE GLYCOL PUMP	30 FT^3
		SEAWATER PIPING (50 FT)	25 FT^3
		SEAWATER PUMP	30 FT^3
		ETHYLENE GLYCOL STORAGE	9 FT^3
TOTAL	348 FT^3	TOTAL	161 FT^3
FLUID FLOW RATES:			
SEAWATER	1500 GPM	SEAWATER	1500 GPM
		ETHYLENE GLYCOL	850 GPM
EQUIPMENT DIMENSIONS: (LWH)			
SEAWATER TO AIR HEX	11'-9" X 5'-0"DIA	*ETHYLENE GLYCOL TO AIR HEX	
SEAWATER PUMP	4'-0" X 2'-6" X 3'-0"	SEAWATER TO ETHYLENE GLYCOL HEX	13'-0" X 2'-0"DIA
SEAWATER PIPING	8" SCHEDULE 40	ETHYLENE GLYCOL PIPING	6" SCHEDULE 40
		ETHYLENE GLYCOL PUMP	4'-0" X 2'-6" X 3'-0"
		SEAWATER PIPING	8" SCHEDULE 40
		SEAWATER PUMP	4'-0" X 2'-6" X 3'-0"
		ETHYLENE GLYCOL STORAGE	1'-6" X 2'-0" X 3'-0"

* ENCLOSED WITHIN GAS TURBINE MODULE

above, and the supporting data in this thesis, it appears that the direct intercooler system is superior. There is a significant reduction in overall machinery components resulting in lower maintenance requirements, and increased flexibility in machinery arrangements.

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APPENDIX A

The following is a listing of the ambient conditions assumed in the analysis , and the various temperature ranges and limits used. Also listed is selected full power engine data, based on figure (11).

T_{AMB} = 560.^{o}R

P_{AMB} = 14.7 PSIA

T_{A1} = 890^{o}R (Approximate)

T_{SW1} = 545.^{o}R

T_{SW2} = 585.^{o}R

T_{EC} = 550. - 590.^{o}R

T_{EH} = 660.^{o}R

M_A = 120 lbm/sec (Approximate)

P_2/P_1 = 4.2 (Approximate LP compressor ratio)

E_{OA} = 0.80

nc = 0.86

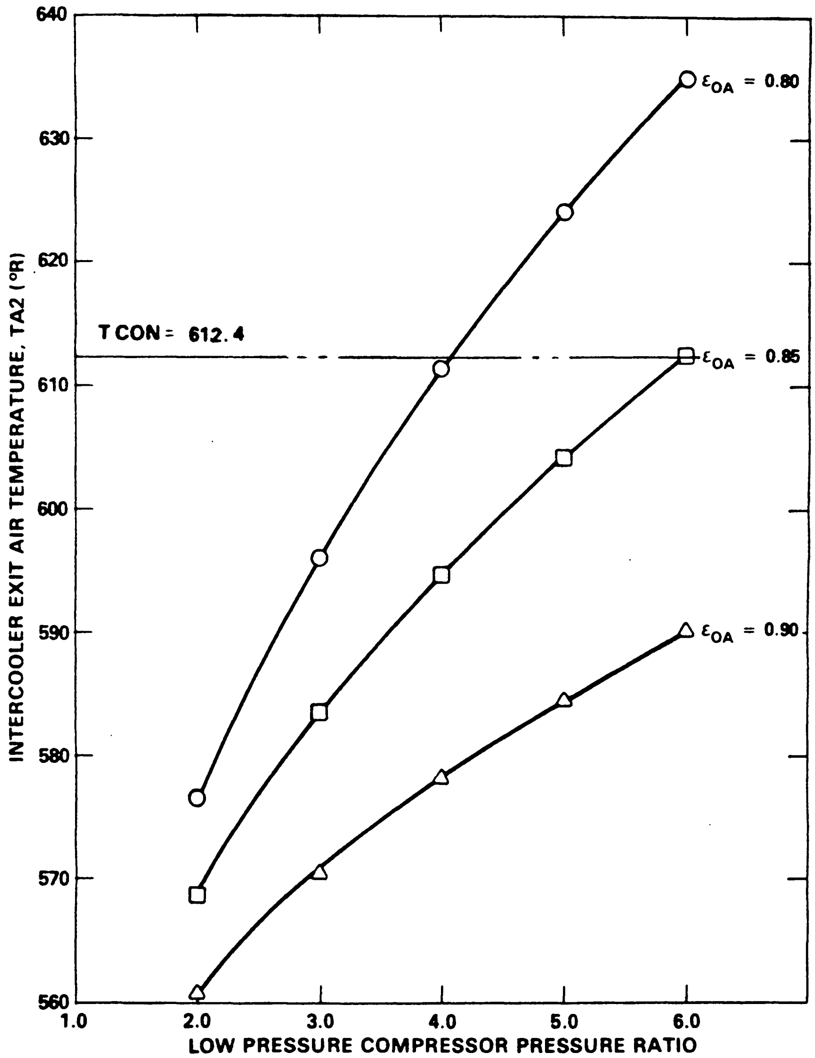

Fig. 11 Intercooler Exit Air Temperature As A Function Of The Overall Effectiveness And Low Pressure Compressor Compression Ratio

APPENDIX B

PROGRAM LISTING

```
C     PROGRAM HEXLOOP IS DESIGNED TO DETERMINE THE RANGE OF MASS
C     FLOW RATES ACCEPTABLE IN AN ICR ENGINE, AND SHIP ARRANGEMENT.
      IMPLICIT REAL*8 (A-H,O-Z)
      REAL*8 MA,ME,MSW,MSWMAX,TAI(100),K
      INTEGER DUM
      OPEN (UNIT=6,NAME='LOOP.OUT',TYPE='NEW')
C
C     INPUT INLET CONDITIONS
C
      PRINT*,'INPUT THE SEAWATER TEMPERATURE, DEG R'
      READ*,TSW1
      PRINT*,'INPUT THE AMBIENT AIR TEMPERATURE, DEG R'
      READ*,TAMB
      PRINT*,'INPUT THE LP COMPRESSOR PRESSURE RATIO'
      READ*,PR
      PRINT*,'INPUT THE COMPRESSOR EFFICIENCY IF KNOWN, 0.0 TO 1.0'
      PRINT*,'IF NOT KNOWN ENTER 0.0'
      READ*,EFFC
      PRINT*,'INPUT THE MASS FLOW RATE OF AIR INTO THE INTERCOOLER,lbm/
     &sec'
      READ*,MA
      PRINT*,'INPUT THE HIGHEST INTERMEDIATE FLUID TEMPERATURE
     & DESIRED, DEG R'
      READ*,TEH
      PRINT*,'INPUT THE DESIRED OVER-ALL EFFECTIVENESS, 0.0 TO 1.0'
     &HIGHTEST AND LOWEST VALUE, AND THE STEPSIZE.
      READ*,EOH,EOL,EOS
      PRINT*,'INPUT THE HIGHEST VALUE OF SEAWATER MASS FLOW RATE DESIRED
     &, (LBM/S), AND THE STEP DESIRED'
      READ*,MSWMAX,MSWI
      PRINT*,'INPUT THE RANGE, AND INCREMENT OF TEMPERATURE DIFFERENCE
     & BETWEEN TSW1,AND TEC, DEG R, SEPARATED BY COMMAS'
      READ*,DELT1,DELT2,DDELT
      PRINT*,'INPUT THE SEAWATER OUTLET TEMPERATURE LIMIT, DEG R'
      READ*,TSWL
      PRINT*,'INPUT THE CONDENSATION TEMP FOR THIS PERFORMANCE CASE,
     & DEG R'
      READ*,TCON
      PRINT*,'DO YOU WANT TO DETERMINE THE RANGE OF SEAWATER MASS FLOW
     & RATES?, ENTER 1 FOR YES OR 2 FOR NO'
      READ*,DUM
      PRINT*,'IS THE INTERMEDIATE FLUID ETHYLENE GLYCOL, DIETHYLENE
     & GLYCOL, OR TRIETHYLENE GLYCOL? ENTER 1,2,OR 3'
      READ*,MN
      IF(MN .NE. 1) GO TO 5
      PRINT*,'INPUT THE PERCENTAGE OF ETHYLENE GLYCOL SOLUTION, .2, .4,
     &.6, .8, 1.0'
      READ*,PSOL
C
C***************BEGIN PERFORMANCE CALCULATIONS************************
C
    5 WRITE(6,100)
      TOL=0.001
      R=0.06855618
C
C  R IS THE GAS CONSTANT IN (BTU/LBM-R)
C  BEGIN ITERATIVE SOLUTION OF THE COMPRESSOR EXIT TEMPERATURE,
C  ISENTROPIC THE INITIAL VALUE OF K-1/K WHERE K=Cp/Cv IS 0.28
C  K=1.40
C
      GAMMA=0.28
      DO 1 J=1,100,1
      TAI(J)=(PR**GAMMA)*TAMB
      IF (J.EQ.1) GO TO 2
      IF (TAI(J) .GT. TAI(J-1)-TOL .AND. TAI(J) .LT. TAI(J-1)+TOL)
     &GO TO 4
    2 CALL TRANSPA(TAI(J),CPA)
      K=CPA/(CPA-R)
      GAMMA=(K-1)/K
    1 END DO
    4 IF(EFFC .NE.0.0) GO TO 800
      EFFC=0.86
  800 TA1=((TAMB*EFFC)+(TAI(J)-TAMB))/EFFC
      WRITE(6,101) TSW1,TA1,MA
C
C     ********TA1 IS THE INTERCOOLER INLET TEMPERATURE***********
C
      DO 3 DELT=DELT1,DELT2,DDELT
```

```
      WRITE (6,499) TCON
  499 FORMAT(1X,'THE CONDENSATION TEMPERATURE IS ',F7.2)
      WRITE(6,500) DELT
  500 FORMAT(1X,'THE TEMPERATURE DIFFERENCE BETWEEN Tseawater,in
     & AND Tglycol,cold =',F7.2)
      TEC=TSW1+DELT
  202   WRITE(6,102)
        WRITE(6,103) TEC,TEH
C
C***********SHELL AND TUBE EFFECTIVENESS*******************
C
          DO 200 EO=EOL,EOH,EOS
          EC=(TEH-TEC)/(TEH-TSW1)
          PRINT*,'EC=',EC
          TA2=TA1-(EO*(TA1-TSW1))
C
C***********PLATE FIN EFFECTIVENESS************************
C
          EH=(TA1-TA2)/(TA1-TEC)
          WRITE(6,104)
          PRINT*,'EH=',EH
          DT=TCON-TA2
          IF (EH .LT. 1.0) GO TO 20
          GO TO 3
   20     QA=CPA*(TA1-TA2)
          QDOT=QA*MA
          TAVGG=(TEC+TEH)*0.5
          CALL TRANSPG(MN,TAVGG,CPE,PSOL)
          QE=CPE*(TEH-TEC)
          ME=QDOT/QE
          CE=ME*CPE
          QDE=CE*(TEH-TEC)
          IF (QDE .LT. (QDOT-1.0) .OR. QDE .GT. (QDOT+1.0)) GO TO 3
          CALL MASSFLO(MA,MSW,QA,QDOT,TSW1,CE,TEH,TSW2,DUM,QDE,QDSW,
     &TSWL,MSWMAX,MSWI,EC,EH,EO,ME,TA2)
          IF(TA2 .GE. TCON) GO TO 300
          WRITE(6,498)DT
  498     FORMAT(1X,'*****THE BULK AIR TEMP. OUT IS ',F6.3,' DEGREES
     & BELOW THE CONDENSATION POINT*****')
  300     WRITE(6,107)
          WRITE(6,108)CPA,CPE,QDE,QDSW
  107     FORMAT(1X,'CPA (B/LBM R)',5X,'CPE (B/LBM R)',3X,'QDE
     & (B/SEC)',3X,'QDSW (B/SEC)')
  108     FORMAT(5X,F6.4,11X,F6.4,10X,F7.2,7X,F7.2)
          WRITE(6,109)
  109 FORMAT('*******************************************************
     &**************************************************************
     &**********',/)
  104 FORMAT(4X,'EFF.ST',10X,'EFF.PF',11X,'EFF.OA',4X,'MDOT EG (LBM/S)'
     &,1X,'MDOT SW (LBM/S)',7X,'QDOT (B/S)',5X,'TSW,OUT (R)',3X,
     &'TAIR,OUT (R)')
  100  FORMAT(1X,'Tseawater,in(R)',4X,'Tair,in(R)',4X,'Mair(lbm/sec)')
  101   FORMAT(5X,F6.2,11X,F6.2,10X,F6.2)
  102   FORMAT(/,1X,'Tcold glycol(R)',3X,'Thot glycol(R)')
  103   FORMAT(5X,F6.2,11X,F6.2,/)
  200 END DO
    3 END DO
      STOP
      END
C
C
C     SUBROUTINE MASSFLO DETERMINES THE MASS FLOW RATES OF THE AIR
C     SEAWATER, AND ETHYLENE GLYCOL SOLUTION IN (lbm/sec).
C
      SUBROUTINE MASSFLO(MA,MSW,QA,QDOT,TSW1,CE,TEH,TSW2,DUM,QDE,QDSW,
     &TSWL,MSWMAX,MSWI,EC,EH,EO,ME,TA2)
      IMPLICIT REAL*8 (A-H,O-Z)
      REAL*8 MA,ME,MSW,MSWMAX
      INTEGER DUM
      I=0
C     ASSUME A VALUE OF 500(lbm/sec) FOR SEAWATER MASS FLOW RATE
      MSW=500.0
C     ASSUME A VALUE OF 1.0(B/lbm-R) FOR SEAWATER SPECIFIC HEAT
      CPSW=1.0
  100 TSW2=(QDOT/(MSW*CPSW))+TSW1
      IF (CE .LT. CSW .AND. TSW2 .LT. TEH) GO TO 110
    6 MSW=MSW+1.0
      I=I+1
      CSW=MSW*CPSW
      TSW2=(QDOT/CSW)+TSW1
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    IF (CE .LT. CSW .AND. TSW2 .LT. TEH) GO TO 7
    GO TO 6
  7 IF (TSW2 .GT. TSWL) GO TO 6
110 IF (I .EQ. 0) GO TO 120
    IF (DUM .EQ. 1) GO TO 5
    QDSW=CSW*(TSW2-TSW1)
    GO TO 3
  5 L=0
      DO 1 MSW=MSW,MSWMAX,MSWI
      CSW=MSW*CPSW
      TSW2=(QDOT/CSW)+TSW1
      QDSW=CSW*(TSW2-TSW1)
      IF (CE .LT. CSW) GO TO 2
      GO TO 1
  2   IF (QDSW .LT.(QDOT-10.0) .OR. QDSW .GT.(QDOT+10.0)) GO TO 1
      IF (TSW2 .GT. TSWL) GO TO 1
      L=L+1
      IF(L .GT. 1) GO TO 10
  3   WRITE(6,105) EC,EH,EO,ME,MSW,QDOT,TSW2,TA2
105   FORMAT(4X,F6.3,10X,F6.3,11X,F6.3,9X,F7.2,9X,F7.2,11X,F7.2,
     &9X,F6.2,8X,F6.2)
        GO TO 1
   10   WRITE(6,106) MSW,TSW2
  106   FORMAT(68X,F7.2,27X,F6.2)
    1 END DO
      RETURN
  120 MSW= MSW-10.0
      GO TO 100
      END
C
C
C
C     SUBROUTINE TRANSPA DETERMINES THE SPECIFIC HEAT OF AIR (B/lbm-R)
C
      SUBROUTINE TRANSPA(T,CPA)
      IMPLICIT REAL*8 (A-H,O-Z)
      DATA A1,A2,A3,A4,A5,A6,A7,A8/1.0115540E-25,-1.4526770E-21,
     &7.6215767E-18,-1.5128259E-14,-6.7178376E-12,6.5519486E-08,
     &-5.1536879E-05,2.5020051E-01/
      IF (T .LT. 500. .OR. T .GT. 2000.) GO TO 999
      CPA=((((((A1*T+A2)*T+A3)*T+A4)*T+A5)*T+A6)*T+A7)*T+A8
      RETURN
  999 WRITE(6,200)T
  200 FORMAT(10X,'TRANSPA INPUT OUT OF RANGE',5X,'TEMP=',F8.2)
      RETURN
      END
C
C
C
C     SUBROUTINE TRANSPG DETERMINES THE SPECIFIC HEAT OF THE ETHYLENE
C     GLYCOL SOLUTION (B/lbm-R)
C
      SUBROUTINE TRANSPG(MN,T,CPE,PSOL)
      IMPLICIT REAL*8 (A-H,O-Z)
      IF (MN .EQ. 1 .AND. T .GT. 900. .OR. T .LT. 480.) GO TO 100
      IF (MN .EQ. 1 .AND. T .GT. 800. .OR. T .LT. 480.) GO TO 100
      IF (MN .EQ. 2 .AND. T .GT. 740. .OR. T .LT. 500.) GO TO 100
      IF (MN .EQ. 3 .AND. T .GT. 740. .OR. T .LT. 500.) GO TO 100
      CALL TABXY(MN,T,CPE,PSOL)
      RETURN
  100 WRITE(6,101) T
  101 FORMAT(10X,'TRANSPG INPUT OUT OF RANGE',5X,'TEMP=',F8.2)
      RETURN
      END
C
C
C
C     SUBROUTINE TABXY READS FROM THE DATA BLOCK AND INTERPOLATES
C     TO FIND THE VALUE OF SPECIFIC HEAT FOR THE ETHYLENE GLYCOL
C     SOLUTION (B/lbm-R)
C
      SUBROUTINE TABXY(MN,Y,Z2,X)
      IMPLICIT REAL*8 (A-H,O-Z)
      INTEGER XBLK,XF,YBLK,YF
      DIMENSION EGPROP(25,52)
      COMMON/MAPS/NX,N1Y,N2Y,N3Y
      COMMON/ARRAY/EGPROP
      IF (MN-1)200,30,40
   30 XBLK=1
      YBLK=2
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      XF=NX
      YF=N1Y
      J2S=2
      GO TO 250
   40 IF (MN-2)200,50,60
   50 XBLK=1
      YBLK=25
      XF=NX
      YF=N2Y
      J2S=25
      GO TO 250
   60 IF (MN-3)200,70,200
   70 XBLK=1
      YBLK=39
      XF=NX
      YF=N3Y
      J2S=39
      GO TO 250
  200 WRITE(6,220)MN
  220 FORMAT(2X,'MAP NUMBER ,MN, FAIL ALL COMPONENT TESTS IN TABXY','MN
     &=',I2)
      RETURN
  250 IF (X-EGPROP(XF,XBLK))300,280,280
  280 I=XF
      IHI=XF
      ILO=XF
      GO TO 380
  300 IF (EGPROP(1,XBLK)-X)340,320,320
  320 I=1
      IHI=1
      ILO=1
      GO TO 380
  340 DO 360 I=1,XF
      IF (X-EGPROP(I,XBLK))370,370,360
  360 CONTINUE
  370 IHI=I
      ILO=I-1
  380 IF (Y-EGPROP(YF,YBLK))420,400,400
  400 J=Y
      JHI=JLO
      JHI=YF
      JLO=YF
      GO TO 460
  420 IF(EGPROP(1,YBLK)-Y)540,440,440
  440 J=1
      JHI=1
      JLO=1
  460 IF (IHI-ILO)500,480,500
  480 M=I
      N=J2S+J
      Z2=EGPROP(M,N)
      RETURN
  500 C=(X-EGPROP(ILO,XBLK))/(EGPROP(IHI,XBLK)-EGPROP(ILO,XBLK))
      N=J2S+J
      Z2=EGPROP(ILO,N)+C*(EGPROP(IHI,N)-EGPROP(ILO,N))
      RETURN
  540 DO 560 J=1,YF
      IF (Y-EGPROP(J,YBLK))580,580,560
  560 CONTINUE
  580 JHI=J
      JLO=J-1
      IF (IHI-ILO)620,600,620
  600 C=(Y-EGPROP(JLO,YBLK))/(EGPROP(JHI,YBLK)-EGPROP(JLO,YBLK))
      M=I
      NLO=J2S+JLO
      NHI=J2S+JHI
      Z2=EGPROP(M,NLO)+C*(EGPROP(M,NHI)-EGPROP(M,NLO))
      RETURN
  620 CX=(X-EGPROP(ILO,XBLK))/(EGPROP(IHI,XBLK)-EGPROP(ILO,XBLK))
      CY=(Y-EGPROP(JLO,YBLK))/(EGPROP(JHI,YBLK)-EGPROP(JLO,YBLK))
      MLO=ILO
      MHI=IHI
      NLO=J2S+JLO
      NHI=J2S+JHI
      A2=EGPROP(MLO,NLO)+CX*(EGPROP(MHI,NLO)-EGPROP(MLO,NLO))
      B2=EGPROP(MLO,NHI)+CX*(EGPROP(MHI,NHI)-EGPROP(MLO,NHI))
      Z2=A2+CY*(B2-A2)
      RETURN
      END
C
C
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```
C
C     DATA BLOCK FOR THE PROPERTIES OF ETHYLENE GLYCOL, DIETHYLENE
C     GLYCOL, AND TRIETHYLENE GLYCOL.
C
      BLOCK DATA ETHGLY
      IMPLICIT REAL*8 (A-H,O-Z)
      DIMENSION CP1BLK(25,22),CP2BLK(25,13),CP3BLK(25,13),TMP1BLK(25),
     &TMP2BLK(25),TMP3BLK(25),PSOLBLK(25)
      DIMENSION EGPROP(25,52)
      COMMON/MAPS/NX,N1Y,N2Y,N3Y
      COMMON/ARRAY/EGPROP
      EQUIVALENCE (EGPROP(1,1),PSOLBLK(1)),(EGPROP(1,2),TMP1BLK(1))
      EQUIVALENCE (EGPROP(1,3),CP1BLK(1,1))
      EQUIVALENCE (EGPROP(1,25),TMP2BLK(1))
      EQUIVALENCE (EGPROP(1,26),CP2BLK(1,1))
      EQUIVALENCE (EGPROP(1,39),TMP3BLK(1))
      EQUIVALENCE (EGPROP(1,40),CP3BLK(1,1))
C********************PROPERTIES OF EHTYLENE GLYCOL********************
C     NX:NUMBER OF PERCENT SOLUTION VALUES
C     N1Y:NUMBER OF TEMPERATURE VALUES FOR ETHYLENE GLYCOL
      DATA NX,N1Y/ 5,22 /
      DATA (PSOLBLK(I),I=1,25)/0.20,0.40,0.60,0.80,1.0,20*0./
      DATA (TMP1BLK(I),I=1,25)/
     &480.,500.,520.,540.,560.,580.,600.,620.,640.,660.,680.,700.,
     &720.,740.,760.,780.,800.,820.,840.,860.,880.,900.,3*0./
      DATA ((CP1BLK(I,J),J=1,22),I=1,25)/
     &0.9313,0.9377,0.9421,0.9418,0.9451,0.9499,0.9571,0.9631,0.9697,
     &0.9751,0.9797,0.9831,0.9871,0.9911,0.9951,0.9991,1.0030,5*0.,
     &0.8052,0.8187,0.8317,0.8438,0.8552,0.8658,0.8756,0.8846,0.8916,
     &0.9002,0.9096,0.9206,0.9296,0.9377,0.9452,0.9518,0.9576,5*0.,
     &0.7002,0.7168,0.7337,0.7499,0.7653,0.7799,0.7937,0.8067,0.8188,
     &0.8302,0.8408,0.8506,0.8596,0.8678,0.8752,0.8818,0.8876,5*0.,
     &0.6152,0.6322,0.6481,0.6613,0.6753,0.6893,0.7037,0.7167,0.7282,
     &0.7403,0.7522,0.7650,0.7750,0.7834,0.7901,0.7953,0.7989,5*0.,
     &0.5306,0.5438,0.5565,0.5671,0.5803,0.5951,0.6135,0.6264,0.6346,
     &0.6452,0.6566,0.6689,0.6819,0.6957,0.7103,0.7257,0.7419,445*0./
C
C*********************PROPERTIES OF DIETHYLENE GLYCOL*****************
C     N2Y:NUMBER OF TEMPERTURE VALUES FOR DIETHYLENE GLYCOL
      DATA NX,N2Y/ 5,13 /
      DATA (TMP2BLK(I),I=1,25)/
     &500.,520.,540.,560.,580.,600.,620.,640.,660.,680.,700.,
     &720.,740.,12*0./
      DATA ((CP2BLK(I,J),J=1,13),I=1,25)/
     &0.9298,0.9305,0.9392,0.9351,0.9385,0.9427,0.9477,0.9541,0.9601,
     &0.9661,0.9729,0.9769,0.9792,
     &0.8312,0.8395,0.8502,0.8602,0.8702,0.8798,0.8908,0.9039,0.9152,
     &0.9258,0.9356,0.9446,0.9528,
     &0.7242,0.7362,0.7477,0.7603,0.7737,0.7883,0.8023,0.8163,0.8303,
     &0.8443,0.8583,0.8723,0.8863,
     &0.5343,0.5455,0.5522,0.5602,0.5682,0.5754,0.5854,0.5988,0.6102,
     &0.6208,0.6302,0.6402,0.6502,
     &0.6292,0.6412,0.6532,0.6652,0.6772,0.6889,0.7019,0.7175,0.7302,
     &0.7414,0.7494,0.7615,0.7751,260*0./
C
C*********************PROPERTIES OF TRIETHYLENE GLYCOL****************
C     N3Y:NUMBER OF TEMPERATURE VALUES FOR TRIETHYLENE GLYCOL
      DATA NX,N3Y/ 5,13 /
      DATA (TMP3BLK(I),I=1,25)/
     &500.,520.,540.,560.,580.,600.,620.,640.,660.,680.,700.,
     &720.,740.,12*0./
      DATA ((CP3BLK(I,J),J=1,13),I=1,25)/
     &0.9363,0.9344,0.9380,0.9400,0.9420,0.9437,0.9467,0.9505,0.9551,
     &0.9605,0.9675,0.9725,0.9767,
     &0.8317,0.8385,0.8435,0.8502,0.8576,0.8662,0.8742,0.8816,0.8902,
     &0.8996,0.9110,0.9190,0.9253,
     &0.7243,0.7363,0.7494,0.7602,0.7694,0.7746,0.7867,0.8067,0.8202,
     &0.8305,0.8352,0.8455,0.8571,
     &0.6033,0.6173,0.6319,0.6453,0.6578,0.6685,0.6825,0.6999,0.7153,
     &0.7299,0.7437,0.7566,0.7688,
     &0.4807,0.4909,0.5099,0.5253,0.5399,0.5537,0.5667,0.5777,0.5903,
     &0.6037,0.6183,0.6323,0.6463,260*0./
      END
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