A COMPUTER MODEL FOR THE HEAT LOSSES
IN THE EXHAUST MANIFOLD OF AN
INTERNAL COMBUSTION ENGINE

by

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A model was developed for the exhaust manifold of an internal combustion engine for use in a computer simulation program. Previous experimental work that developed heat transfer correlations for the port is combined with other general empirical correlations for turbulent pipe flow to give an overall heat transfer model for the manifold. The model developed has the advantages of being applicable to engines of varying numbers of cylinders and displacement while remaining simple. Correlations are also included to accept different geometries and materials for the manifold piping. The model has been applied to a turbocharged diesel simulation program being developed at the Massachusetts Institute of Technology. The model was validated by comparing predicted performance to test results from a specially instrumented turbocharged engine with thermocouples located along one section of the exhaust manifold. The results of a study done into the effects of exhaust manifold insulation on engine performance are presented. Examples of the output for this program, including both heat transfer and pressure drop data for various exhaust manifold designs, are included with this thesis.

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The computer code printed in Volume II is the result of the efforts of many people. They have been recognized in the subroutines where their work is represented.

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CHAPTER 1.

INTRODUCTION

In the past decade there has been a strong increase in interest in computer simulation of internal combustion engine operation and performance. This interest has been due to the increased sophistication of today's engines where it is necessary to minimize pollutants and maximize economy. The rapidly increasing power of computers to carry out large calculations has aided this effort. The rational behind computer simulation programs is that if they can be written to accurately predict the performance of existing engines, then they can eventually be applied to study possible design modifications of internal combustion engines without building the engine. As a design tool, a computer simulation program can be used to predict the potential benefit, if any, that a given engine modification will have on the emissions or performance of an internal combustion engine.

One promising design modification to internal combustion engines that is currently of interest is lower heat rejection through the use of ceramic materials. Since ceramic materials can withstand much higher operating temperatures than metals, the use of ceramics in engines can be very attractive. An engine computer simulation code is a valuable tool for studying the usefulness of ceramics in internal combustion engines. The results can be used to indicate where ceramics would be most effective in improving engine performance and just as importantly to help define those areas where the use of ceramics would not provide any improvement in engine performance.

A simulation program is comprised of models of each component of an
engine which predicts performance over a range of conditions. These models are linked in the simulation to calculate the expected performance of the engine as a whole. The more accurately the components of the engine are represented, the better a simulation program is able to predict actual engine performance.

In the simulation of turbocharged engines, one important component of the engine which needs to be modelled is the exhaust manifold. The exhaust manifold of the engine is especially difficult to model because of its complex shape and flow characteristics. It is important to be able to quantify the losses in the manifold, due to heat transfer and flow friction and to determine the energy available at the inlet to the turbine for the extraction of usable work. Since low heat rejection engines are expected to be turbocharged, it is important that a simulation code for these engines include an accurate model for the exhaust manifold.

Previous models for exhaust manifolds used in computer simulation programs were often limited to either accounting for heat transfer or pressure fluctuations in the manifold but not both. Often, the manifold was modelled as a single plenum with uniform properties. With such a lumped parameter model it is not possible to account for the variation in losses along the length of the manifold due to the change in conditions with respect to position in the manifold. In these simulations, it was usually necessary to define the wall temperature along the manifold a priori. In order for the simulation to be useful to predict the performance of engines yet to be built, it is necessary to predict the wall temperatures that would result for a given exhaust manifold shape and wall composition.
The immediate purpose of the present study was to develop a model for inclusion in an overall engine simulation currently being written at MIT. The primary goal of this engine simulation is to simulate the operation of a turbocompounded diesel engine. Given the engine size, materials of construction, and the engine operating speed and load the program provides an output that predicts the performance of the entire system. In this simulation, all conditions are assumed to be steady-state.

The simulation is expected to be especially useful in predicting the performance gains or losses associated with changes in material such as the use of ceramic linings or changes in components such as different turbochargers. The simulation has been in operation since 1984 and the current thrust has been to upgrade the heat transfer models used. The original model of the exhaust manifold used was a simple plenum with no means of predicting the inside wall temperature of the ducts. The present study was initiated to provide additional insight with respect to the gases flowing through the manifold and to provide additional flexibility in specifying the exhaust manifold of the engine to be simulated. A schematic of the engine layout that the simulation program is based on is shown on Figure 1.1.

Recently, turbochargers have become a common addition to medium size diesel engines. In order to study the performance of a given engine with a turbocharger, it is important to be able to predict the flow losses that would occur in the exhaust manifold. By using a suitable model for the manifold, it is possible to optimize the performance of a given engine configuration before building the engine and proceeding with test block experiments. Whereas in the past a model for the
exhaust manifold was not of much interest, the use of turbocharging and the possibility of building low heat rejection engines, makes such a model more relevant today.

Another new source of interest in studying the exhaust gases as they flow through the exhaust manifold has been the increasing effort to minimize pollutants from internal combustion engines. In studying the emissions of an engine, it is important to know what happens as the gases flow through the manifold. Chemical reactions can still occur after the gases leave the cylinder of the engine if the temperature remains hot enough. To predict these reactions, it is necessary to be able to determine the expected temperature of the gases as they flow through the exhaust manifold. While the chemical reaction of the exhaust gases in the exhaust manifold has not been part of the current study, the model developed here and the gas temperature predictions could be useful in the study of these reactions.

Experimental work done in the area of exhaust manifold losses has not been very extensive. Rush [1] conducted an experimental study on the heat losses in the port section of a spark ignition engine. By measuring the heat added to the cooling water flowing around the exhaust port of a spark-ignition engine, Rush was able to quantify the average heat transfer in the port. Hires and Pochmara [2] conducted an analytical study for comparison with the results of Rush. By using an electrical resistance analogy, Hires and Pochmara were able to expand on Rush's work and study potential decrease in heat transfer from insulating the exhaust port.

Malchow, Sorenson, and Buckius [3] provided experimental data of heat transfer in a straight section of an exhaust manifold. They found
that the measured heat transfer coefficient followed the same Reynolds number relationship as is used for fully developed turbulent pipe flow but that measured values were about a factor of two greater. Robertson [4] was interested in the transient response of the temperatures in the exhaust manifold on start-up of a cold engine. He used turbulent pipe flow correlations similar to those used in the present model but did not include any special consideration for heat transfer in the exhaust port.

The disadvantage with each of these studies is that they only consider heat transferred as an average rate over the cycle of the engine. They do not address the cyclic aspect of the flow through the exhaust manifold. In an operating internal combustion engine the gas temperature and the heat transfer vary considerably over the complete engine cycle. For the current study it was considered important to follow these variations over the operating cycle of the engine.

Other work, concerning the loss of availability as gases flow between the cylinder and the exhaust manifold was reported by Watson [5] and by Primus [6]. Watson discussed the variation of the flow pattern of the exhaust gases as the exhaust valve opens and how the losses are affected by the opening of the exhaust valve. Primus concentrated on a Second Law of Thermodynamics analysis of the flow through the exhaust valve and the manifold. Turbine performance is strongly dependent on the available energy of the exhaust gases at the turbine inlet. Primus demonstrated how the flow losses incurred as the gases flow through the exhaust valve and exhaust manifold result in the loss of availability. Although he did not include heat transfer in his study, Primus' work provides useful insight into losses due to pressure drop in an exhaust manifold.
Caton [7,8] did both an analytical development and experimental work on the transient nature of heat transfer in the exhaust port of a gasoline engine. By using a fast time response device to measure the exhaust temperature in the exhaust port, Caton was able to approximate a set of relationships for the heat transfer coefficient in the exhaust port as a function of crank angle and exhaust flow rate. His results provided a basis for the heat transfer correlations used for the port in this study.

There were two fundamental areas that were addressed in formulating the current model. The first was how to represent the complex shape of an exhaust manifold in a manner suitable for input into a computer program while still taking into consideration the various design parameters that will affect the performance of the manifold. It was considered important to provide for both shape factors such as overall dimensions and the number of bends and to provide for construction with different materials. The second area addressed was how to model the loss mechanisms in the manifold in ways that account for cyclical variations of the flow and variations of the flow properties along the length of the manifold. For example, it was required that variation in the manifold wall temperature be allowed as the port wall temperature is normally much different than the wall temperature further along the manifold. This difference results in a substantial change in the heat flux from the gases to the walls of the manifold along the length of the manifold.

The shape of the manifold was modelled as representative sections connected in series (see Chapter 2). As in the main program, calculations are only carried out for those sections pertinent to a
single cylinder. The temperature, pressure, and fuel fraction are calculated at each crank angle during the engine cycle for each zone separately and the results are echoed with the appropriate phase shift for the other sections of the manifold. By providing for multiple zones, it was possible to account for the different flow characteristics in each section and predict the resulting heat transfer losses with greater accuracy.

To calculate the heat transfer, the model developed in this thesis uses a combination of empirical correlations for turbulent flow in pipes and exhaust ports. The analytical solution for steady-state heat transfer through cylindrical composite walls is used to predict the inside surface temperature of the manifold walls. A finite difference scheme is used to calculate the transient wall temperatures.

Following the development of the model for the exhaust manifold, the necessary programming changes were made in the existing code for this engine simulation. Once the model was inserted in the larger program, it was validated using data obtained from an experiment conducted by Cummins Engine in which time averaged temperature data were taken at four locations along the exhaust manifold of an 8 cylinder turbocharged diesel engine. Following this validation, the model was calibrated to match test data with a 6 cylinder turbocompounded diesel engine also from Cummins. With the turbocompounded engine as a base case, series of runs were done with different degrees of insulation in the exhaust manifold and in the engine cylinders to assess the effects of the changes on the performance of the whole engine.

This thesis is divided into three main sections. The first deals with the development of the model. The second section covers the heat
transfer, pressure loss and thermodynamic equations used. The final section discusses the results of parametric studies carried out once the model was successfully inserted into the program. Included as Appendices are a listing of the program and examples of the input and output files.
CHAPTER 2.
GEOMETRICAL MODEL

In most engine simulation programs of this type, the exhaust manifold is treated as a simple control volume [8]. A simple plenum is used with the exhaust gases from each cylinder flowing directly into the volume and mixing instantaneously with the all of the gases in the manifold i.e. the plenum is assumed to be perfectly mixed. This approach has two primary disadvantages. First, with a single plenum, the exhaust gases from a given cylinder are diluted by the gases in the manifold in a manner that is not representative of a real manifold. In an actual manifold individual runners separate the gases from different cylinders during much of the flow path between the exhaust valve and the turbine. The artificial dilution results in damping the temperature peaks that occur in the manifold and at the turbine inlet. Thus the gas temperature fluctuations at the inlet to the turbine of an engine would be substantially greater than those predicted by a simple plenum.

The second disadvantage of a common plenum model is that it is not possible to follow the change in gas temperature as the gases flow through the exhaust manifold, mix with exhaust gases from other cylinders at each section, and loose heat to the surroundings. With a common control volume, only one temperature is used to represent the temperature of the gases in the exhaust manifold. Again, this is not a good representation of a real exhaust manifold where the gas temperature varies along the length of the run between the exhaust valve and the turbine inlet.
To avoid these disadvantages with a single plenum model, in this work the exhaust manifold is divided into separate sub-control volumes connected in series. The manifold is considered to be a composite of ports, runners, and a common plenum section. The port section is taken to be that portion of the exhaust manifold volume contained within the head of the engine i.e. cooled by the water jacket. The runners represent the sections of the manifold outside the head where the gases from one cylinder flow without mixing with any other gases or mix with the gases from only a few other cylinders. And the plenum is defined as the volume where the gases from all of the engine cylinders are mixed before entering the turbine. The properties of an additional control volume are calculated in this model. This control volume is composed of all of the sub-control volumes that make up the exhaust manifold. This volume has properties that represent the average properties of the separate sub-control volumes. The average manifold control volume is used to determine the change in pressure in the manifold based on the overall mass balance. In this manner, the heat transfer and gas temperatures along the exhaust manifold are calculated using the sub-control volumes and the manifold pressure is calculated based on an overall average volume that has the same heat transfer as the sum of the heat transfer for the sub-control volumes. Figure 2.1 shows a schematic of the most simple form of this model.

This exhaust manifold model only calculates the properties of the ports and runners for one master port and representative runners. The port and runner information for the other cylinders in the engine is assumed to mirror the master port and runners but phased appropriately in time. For this purpose, information about the master sections is
stored in high-speed memory and is retrieved with the appropriate phase shift to determine what was occurring in other ports and runners at any given time during the cycle. Inherent in this model is the assumption that the sections of the exhaust manifold can be divided up in such a way that each cylinder of the engine has the same effective manifold configuration. While this approximation is not strictly true in an actual engine, the error made in representing some manifold sections as longer than in actuality is assumed to be counter-balanced by the representation of other sections as shorter. The net result is that the overall surface area, cross-sectional area and volume can be matched with that of a real manifold. This model also meets the requirement that it match the overall level of detail in the existing code to which it is being added.

In order to be able to adjust the exhaust manifold for different engine configurations, the exhaust manifold input was set up to be flexible. The number of sub-control volumes used to represent the manifold is an input variable. For the two engines modelled, three and four sub-control volumes where used. The program in its final form accepts up to six manifold volumes but can be made to accept more sections by redimensioning arrays and reformatting the output. Also specified as input, are the number of pipes from other cylinders that come together at the inlet to a given control volume (see for example, Figure 7.3 of the set up for an 8 cylinder engine). In general, the product of the number of pipes coming into all of the sub-control volumes must be equal to the number of cylinders in the engine. To further generalize the system, the user must specify the phase shift of the other pipes coming into each joint based on the engine firing order.
The port may contain either one or two exhaust valves. In either case only one control volume is used for what has been defined as the port. Figure 2.2 shows the layout of a typical head with two exhaust valves per cylinder. In the case of an engine with two valves per cylinder, the volume and surface area are calculated from the sum of the sections of the port before the exhaust streams come together and the common section downstream. The average cross-sectional area is weighted based on surface area of the individual sections and the surface area of the common section. Rush [1] found that changing the cross-sectional area of the port had almost no affect on the port heat transfer. Apparently, the counter-balance between the increase in heat flux due to the higher gas velocities and the decrease in heat transfer surface area for a decrease in port flow area was such that the two effects cancelled each other out. The wall construction in terms of thickness and thermal properties is assumed to be the same for all parts of the port.

The heat transfer from the exhaust gases in the port to the exhaust valve(s) is not included in the port heat transfer. The reason for this omission is that the heat transfer to the exhaust valves from the port is negligible with respect to the rest of the heat transfer to the port walls. In a normally cooled engine, the reason for the relatively low heat transfer to the valves is twofold. First, the exhaust valves represent less than 10% the inside surface area of the port. Second, exhaust valve temperatures for diesel engines are typically 700-900K [10]. The port wall temperature for a water cooled engine is about 440-450K (this is substantiated by the results of this study). For port gas temperatures that are typically in the range of 800-900K, the temperature drop between the port gases and the valve is approximately a
factor of three to four less than the temperature drop between the gases and the cooled port walls. For the same heat transfer coefficient, the effect of the smaller area and the lower temperature drop results in the heat transfer to the exhaust valves of 2.5 - 3% the heat transfer to the rest of the port. On an engine basis, the port heat transfer to the exhaust valves is even less significant. In an insulated engine, the valve heat transfer would be a higher percentage of the port heat transfer. However, because of the area ratio between the rest of the port and the exhaust valve(s) and the decreasing heat transfer due to insulation, the valve heat transfer in the port is assumed to be negligible. The exhaust valves are implicitly included in the cylinder heat transfer as part of the cylinder head area.
CHAPTER 3.
MANIFOLD HEAT TRANSFER

The engine exhaust system is composed of the exhaust ports and the exhaust piping up to the inlet of the turbocharger. The flow in this part of the engine is highly irregular with periods of very high mass flow and other periods of almost no mass flow. Consequently, the heat transfer coefficient from the exhaust gases to the manifold wall varies substantially over the period of one cycle. Since the exhaust gases exit from the engine at temperatures ranging from 750 to 1000K and wall temperatures are as low as 400K, the heat transfer from the gases to the surroundings can be significant. In order to evaluate the heat transfer from the exhaust gases to the manifold walls, empirical heat transfer relations are applied. These relations are used in conjunction with the manifold thermal conductivity and the ambient boundary conditions are used to determine the heat transfer rate for the exhaust.

To determine the local heat transfer, the exhaust manifold is divided into sections connected in series. Each section is considered separately. Within each zone the gas temperature, the heat transfer coefficient, and the inside wall surface temperature are assumed to be uniform. Both the gas temperature and heat transfer coefficient are allowed to vary with time. This model allows the inside wall surface temperature to be taken as either constant with time or varying with time. The method for the calculation of the wall temperature is the same as was developed by Assanis [11] for the cylinder heat transfer in the same program. Starting with an initial guess for the steady-state wall temperature, the new value for the wall temperature is calculated
at the end of each cycle based on the heat transfer over the previous cycle. Once the program heat transfer and mass flow have converged using steady-state wall temperatures, a transient calculation is done if it has been specified by the user. The transient solution is superimposed on the steady-state solution to give the wall temperature profile as a function of time.

The heat transfer coefficient for the exhaust gases is determined using empirical correlations applied to the flow rate for each particular section. For the port section, the results of Caton [7] are used. For the other sections of the manifold, turbulent pipe flow correlations are used. The mass flow used in these correlations is the instantaneous average of the mass flow into and out of the section being analyzed.

3.1 Port Heat Transfer:

The heat transfer in the exhaust port is highly unsteady. When the exhaust valve first comes open, a high velocity jet of high temperature gases sets up recirculation zones in the port [5] that result in a higher heat transfer coefficient than the overall mass flow in the port would indicate. When the exhaust valve is fully open, the flow resembles turbulent pipe flow and the heat transfer coefficient is related to pipe flow relationships. Then, as the exhaust valve closes, there is another period when a narrow jet of gases affects the heat transfer by again setting up recirculation zones. The period the valve closed is much longer than the period when it is open. During the closed period the mass flow rate approaches zero and a correspondingly low heat transfer coefficient exists.
In order to quantify the heat transfer in the exhaust port, the results of Caton [7] are applied. Based on experiments done with fast response fine wire temperature measurements of the port gas temperature in a spark ignition engine, Caton arrived at the following correlations for the heat transfer for various phases of the exhaust process:

Valve opening phase (L/D < 0.2):

\[ \text{Nu} = 0.4 \text{Re}^{0.6} \]  
(3-1)

Valve open phase (L/D > 0.2):

\[ \text{Nu} = 0.023 \ C_{R} \ C_{EE} \text{Re}^{0.8} \text{Pr}^{0.3} \]  
(3-2)

Valve closing phase (L/D < 0.2):

\[ \text{Nu} = 0.5 \text{Re}^{0.5} \]  
(3-3)

Valve closed phase:

\[ \text{Nu} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} \]  
(3-4)

where

- D = valve diameter
- L = valve lift
- Nu = Nusselt Number = \( \frac{hD}{k} \)
- Re\( j \) = jet Reynolds Number = \( \frac{V_{j}D}{v} \)
- V\( j \) = velocity of flow through exhaust valve
- \( v \) = dynamic viscosity of flow
Re = pipe Reynolds number = \frac{VD}{v}

V = pipe flow velocity

C_R = correction factor for surface roughness

C_{EE} = correction factor for entrance effects

Two modifications were made to these results for the present study. The correction factor for surface roughness has not been included. The effects of surface roughness on heat transfer and the reason for excluding them in the present study are discussed in the next section. Also, during the period when the valve is closed, the mass flow is based on the instantaneous mass flow rate between sections instead of on the average flow over the complete cycle as was done by Caton. For a turbocharged engine, this flow is not zero in general because gas flow is induced in inactive sections of the exhaust manifold by the pressure pulsations produced when other cylinders exhaust. Using the instantaneous value of the mass flow rate throughout the cycle more closely approximates the process that actually occurs in an engine. It should be noted that this approach is not applicable to a naturally aspirated engine (such as Caton's) since in such engines the mass flow during periods when the valve is closed approaches zero.

The correction factor for entrance effects, C_{EE}' during the periods when the exhaust valve is fully open is the same as that used for the other sections of the manifold and is discussed in the following section.
3.2 Other Sections of the Manifold:

For the other sections of the exhaust manifold downstream of the exhaust port, turbulent pipe flow correlations are applied. The heat transfer coefficient for fully developed flow in a straight pipe, can be found using the Dittus-Boelter equation [12]:

\[ \text{Nu} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} \]  \hspace{1cm} (3-5)

In an exhaust manifold of an engine not all of the qualifying conditions for this equation apply. In general the flow is not fully developed and the pipe is not straight. There are other factors as well that can cause a deviation from the heat transfer predicted by this equation. They are the variation of the gas properties with temperature, the surface roughness, and viscous heating. For an accurate heat transfer model each of these possible factors should be considered and the base equation adjusted to account for their influence if it is necessary. Each of these effects was considered separately.

3.2.1 Entrance Effects

The heat transfer enhancement due to entrance effects was the subject of a study by Boelter, Young, and Iversen [13]. They did a series of experiments measuring the heat transfer coefficient under different flow conditions at a tube entrance. They found that the local heat transfer coefficient is raised by a factor of up to three at the inlet of the tube. Both the magnitude of the intensification factor and rate at which it decays to unity were strongly dependent on the flow characteristics at the tube entrance (e.g. fully developed velocity distribution vs. uniform velocity distribution, or a sharp bend at the
entrance vs. a straight pipe). Figure 3.1 shows some of their results. Fitting a curve to the experimental results for a tube with an elbow at the entrance, yields the following equation for the local heat transfer coefficient:

\[
C_{EE} = \frac{\frac{\text{Nu}_x}{\text{Nu}_\infty}}{2.2 \ (x/D) ^{-0.3}} \quad 3 < x/D < 14
\]

(3-6)

where
- \( x \) = entrance length from the inlet of the tube
- \( D \) = tube diameter
- \( \text{Nu}_x \) = local Nusselt number
- \( \text{Nu}_\infty \) = Nusselt number for fully developed flow

In order to find the average increase in heat transfer for a given section, equation 3-6 is integrated over the length of the section and divided by its length.

\[
C_{EE} = \frac{\text{Nu}_{ave}}{\text{Nu}_\infty} = \frac{1}{L_2 - L_1} \int_{L_1}^{L_2} [2.2 \ (x/D)^{-0.3}] \ dx
\]

or,

\[
C_{EE} = 3.1 \ D^{0.3} \frac{(L_2)^{0.7} - (L_1)^{0.7}}{L_2 - L_1}
\]

(3-8)

where \( L_1 \) = distance from the exhaust valve to the inlet of the section
L_2 = distance from the exhaust valve to the outlet of the section.

One disadvantage with using Boelter's results is that the experimental data was only taken up to Reynolds numbers of 55,000. In the engine exhaust, peak film Reynolds numbers of up to 400,000 are expected. Deissler [14,15] did analytical work that corresponded to Boelter's results but extended the range up to 200,000 and found that the influence of entrance effects was in general insensitive to Reynolds numbers effects. The insensitivity of the correlation to the Reynolds number and the relatively short time that the Reynolds number is above the range investigated justifies the use of the correlation for the exhaust flow.

3.2.2 Effect of Curved Pipe

A bend in a pipe with turbulent flow passing through it tends to increase the heat transfer coefficient from the fluid to the pipe wall because of a decrease in the boundary layer thickness in the vicinity of the bend. This effect tends to decrease as the Reynolds number and the overall level of turbulence increases. Hausen [16] recommends using the following correlation for the heat transfer enhancement due to bends in a tube:

$$ C_{BP} = \frac{Nu_{BP}}{Nu_{S}} = \left( 1 + \frac{21}{Re^{0.14} \cdot d / D} \right) $$  \hspace{1cm} (3-9)

where

- $C_{BP}$ = correction factor for pipe bends
- $Nu_{BP}$ = Nusselt number for bent pipe
- $Nu_{S}$ = Nusselt number for straight pipe
d = tube diameter
D = bend diameter

This correlation has the advantage over other correlations given in the literature [17,18,19] because it combines the results of the others to extend over a wide range of Reynolds numbers.

The bent pipe correlation is applied to the sections of the manifold downstream of the port. In general, these sections may have both straight and bent portions. In order to account for bends without over compensating, the heat transfer enhancement due to a bend in a manifold section is weighted according to the percentage of the pipe section that is bent. In the code, the portion of the section that is bent is calculated from the radius of the bend and the total number of degrees of the bend in a given section. This approach in contrast to the results of Ede [20] who found that the heat transfer coefficient is increased for some length downstream of the bend. However, he also found that the downstream length affected decreases with increasing Reynolds number. In the range of Reynolds number for the exhaust manifold, the portion of the pipe affected downstream of the bend is neglected.

3.2.3 Effect of Variable Gas Properties

With large temperature differentials between the bulk gas temperature and the wall temperature, the effect of the natural variation in the gas properties can be a significant factor in the heat transfer calculations. Both the viscosity and the thermal conductivity of a gas vary with the temperature raised to the 0.68 power [15] and the density varies inversely with the temperature. The influence of temperature on these properties can affect the heat transfer when a large temperature drop exists between the gas bulk temperature and the wall temperature. The specific heat and the Prandtl number are not significantly affected by temperature.
There are two conflicting methods commonly used to account for the
temperature dependence of these properties. One involves correcting the
Nusselt number derived using bulk temperature properties. The correction
factor is expressed as the ratio of the wall temperature to the bulk
temperature raised to a power. The exponent of the correction factor is
determined by the geometry and the type of flow. The second method consists
of evaluating the equations for heat transfer using the properties found at
some reference temperature that accounts for the variation in such a way that
the properties can be considered constant.

In order to use the first method, the appropriate exponent for the
temperature ratio is needed. Experimental tests involving the cooling of
gases under forced convection are rare but some evidence exists demonstrating
that for cooling of gases, the effect of property variation is nil for wall
temperature to bulk temperature ratios greater than .25 [21]. On this basis
Kays and Perkins recommend that the exponent for the correction factor for
property variation with temperature for heat extraction from gases should be
zero (ie. no correction is necessary).

The reference temperature method is used the most often to account for
temperature dependent properties. In their analytical treatment, Deissler and
Eian [14] recommend using a reference temperature based on a weighted average
between the bulk and wall temperatures. For the cooling of gases, they found
the following equation for the reference temperature:

\[ T = 0.4 \left( T_w - T_b \right) + T_b \]  

(3-10)

where

- \( T_w \) = wall temperature
- \( T_b \) = bulk gas temperature

Most frequently however, the constant 0.4 in the above equation is
changed to 0.5 to get what is called the film temperature.
There is an obvious difference between the two methods of correction that is not well explained in the literature. In the absence of a definitive convention, the film temperature is used to evaluate the gas properties as this appears to be the most common practice.

3.2.4 Surface Roughness

If a surface is sufficiently rough, an increase in the heat transfer coefficient relative to that for a smooth surface can result. This increase is a function of both the surface roughness and the flow Reynolds number. In general, for a given surface roughness, there is a minimum Reynolds number before an increase in the heat transfer coefficient occurs. Heat transfer correlations for heat transfer enhancement due to surface roughness are scarce. One common approach to account for this effect is to use the Colburn analogy [12] to relate the convective heat transfer coefficient to the fluid friction by the following equation:

$$St \cdot Pr^{2/3} = f/2$$

Where $St$ is the Stanton number defined as $h/(\rho C_p V)$, $Pr$ is the Prandtl number, and $f$ is the flow friction factor. The increase in heat transfer is thus related to the increase in the friction factor due to surface roughness. Reference to a Moody chart (see for example ref. 19) indicates that for the peak Reynolds numbers expected in the range of 300,000 to 400,000, the line for a relative roughness of 0.001 (based on a roughness of $\epsilon=0.00015$ ft) has just started to deviate from the friction factor line for a smooth pipe. This indicates that there is some heat transfer enhancement due to surface roughness at the peak exhaust flows that occur during the blow down phase but that there is little increase in the heat transfer during the rest of the exhaust
process. Application of a correction factor for surface roughness is complicated by the need for a correlation to find the increase in friction factor as a function of surface roughness and Reynolds number and data for the appropriated values to be used for the surface roughness of the inside of the exhaust manifold.

The correction used by Caton [7] for surface roughness was based on the results of Nunner. The correction factor in this case is also based on the increase in the flow friction factor relative to the friction factor for smooth pipe and is based on data up to flow Reynolds numbers of 80,000. It was not felt that this correlation would be appropriate for the present study.

Due to the marginal effect on convective heat transfer in the present application and the difficulty in applying an accurate correlation, the heat transfer enhancement due to surface roughness was not included in this study. It should be noted that the heat transfer calculations that follow underestimate the heat transfer due to the omission of the surface roughness effects.

3.2.5 Viscous Heating

Viscous heating of the flow can result for flows of sufficiently high Mach number due to the shear forces between the fluid and the wall and the work done as the fluid moves against these forces. In order for viscous heating to be significant, the Mach number of the flow has to meet the following requirement:

\[ M^2 \geq \frac{1}{\gamma-1} = 2.5 \]  \hspace{1cm} (3-11)

where

- \( M \) = Mach number
- \( \gamma \) = ratio of specific heats
The maximum velocity in the exhaust manifold occurs during blow-down in the port. The engine simulation program for a representative engine gave a maximum flow rate in the port during this period of \( M = 0.5 \). On the basis of this velocity it is seen that viscous heating effects can be neglected in determining the exhaust manifold heat transfer.

3.2.6 Summary

There are only two correction factors that are significant and need to be considered with respect to the heat transfer coefficient in the exhaust manifold. They are the enhancement of heat transfer due to the short length of pipe under consideration (entrance effects) and that due to pipe bends. Combining the formula for fully developed flow for a straight pipe with these correction factors gives the correlation used for the heat transfer coefficient for sections of the pipe downstream of the port section and for the exhaust port during the periods when the exhaust valve is fully open or closed:

\[
Nu = 0.023 C_{EE} C_{BP} Re^{0.8} Pr^{0.3} \quad (3-12)
\]

3.3 Steady-State Heat Transfer Through a Composite Wall

In all of the heat transfer calculations for the exhaust manifold, the heat transfer from the gas to the walls is considered separately from the heat transfer from the walls to the environment. It is expected that the simulation would normally would be run with the inside wall temperatures calculated based on the operating conditions of the engine. It will also accept specified inside wall temperatures.
For the predicted wall temperatures, the program must balance the two heat transfer rates averaged over a complete engine cycle. During the first cycles of a simulation, the program initially calculates the steady-state heat transfer through the manifold walls. The heat transfer through the manifold wall is considered to be one-dimensional. The rate of heat transfer through the wall is found by:

\[ Q = A \cdot U \cdot (T_w - T_c) \]  

where

- \( Q \) = heat transfer rate
- \( A \) = inside surface area
- \( U \) = an overall heat transfer coefficient for the wall from the inside surface to the outside
- \( T_w \) = the inside wall surface temperature
- \( T_c \) = the outside (cold) surface temperature or cooling medium temperature.

Since low heat rejection engines can be expected to have manifold walls composed of layers of materials, the program is set up to accept up to accept three user defined layers each of a specified thickness and thermal conductivity. The overall heat transfer coefficient, \( U \), for a cylindrical composite wall is found from [19]:

\[
(AU)^{-1} = \frac{\ln (R_2 / R_1)}{2 \pi L k_a} + \frac{\ln (R_3 / R_2)}{2 \pi L k_b} + \ldots + \frac{1}{2 \pi L R_o h_o}
\]

where

- \( R_1 \) = the inside radius of the inner layer of material
- \( R_2 \) = the outside radius of the inner layer of material
\( R_3 \) = the outside radius of the second layer of material
\( k_a \) = the thermal conductivity of the inside layer of material
\( k_b \) = the thermal conductivity of the second layer of material
\( R_o \) = the outside radius of the wall
\( h_o \) = the outside heat transfer coefficient

The outside boundary condition must be specified by the user. It can be specified as either the outside surface temperature of the manifold wall or as an ambient temperature and heat transfer coefficient. In the case of specified outside wall temperatures, the last term in Equation 3-14 is dropped.

3.4 Determination of the Inside Wall Temperature

The inside wall surface temperature is calculated for each section of the exhaust manifold individually. Initially, whether the simulation is being run with steady-state wall temperatures or as a transient wall temperature calculation, the wall temperature is held constant during a complete cycle iteration. Since the correct temperature is not known at the start of the simulation, initial guesses are used for the inside wall temperatures. Based on the results from each cycle, the wall temperature is updated until convergence is achieved. Then, if transient wall temperature calculations are desired, the program continues additional cycle iterations and calculates the transient wall temperatures until convergence is again reached. During the transient calculations, the constant, steady-state, wall temperatures are added to
the perturbed solution to find the temperature distribution in the walls as a function of time. Since the transient wall temperatures only penetrate a very short distance into the wall [11], only a few cycles are required for the temperature profile to stabilize. The program is set up to run a minimum of two cycles for the transient temperature calculations to allow the temperature profiles in the wall to become established.

3.4.1 Steady-State Wall Temperature Determination

The method used for the wall temperature calculations was developed by Assanis [11] for use in the same program. The instantaneous heat transfer rate to the walls is calculated from:

\[ Q_w(t) = h(t) (T_g(t) - T_w) \]  \hspace{1cm} (3-15)

where

- \( h(t) \) = instantaneous heat transfer coefficient calculated as described in Sections 3.1 and 3.2
- \( T_w \) = mean inside wall temperature (constant over the cycle)
- \( T_g \) = the instantaneous gas temperature for the section

Based on the results from the previous cycle, an updated wall temperature can be found using the following equation:

\[ \dot{T}_w = \frac{h T_g + U T_c}{\dot{h} + U} \]  \hspace{1cm} (3-16)

where the prime denotes the updated wall temperature to be used in the next cycle iteration. In order to carry out this calculation both the
heat transfer coefficient and the product of the heat transfer coefficient and the gas temperature are integrated out over the complete cycle to find their time average values.

3.4.2 Transient Temperature Calculations

Transient wall temperature calculations are carried out by applying the Fourier equation in one dimension:

\[
\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2}
\]  

(3-17)

This equation is applied using a finite difference analysis that calculates the wall temperature as a function of time throughout the cycle based on the heat transfer from the gas to the wall at that instant. As discussed above, the transient temperature calculations are carried out after the steady-state solution has been found and the results are superimposed to find the temperature profile. The development of the finite difference technique may be found in References 11 and 22.

There is a slight error introduced in the finite difference scheme applied to the exhaust manifold because the scheme is based on plane surface heat transfer while the exhaust manifold is cylindrical. However, this error is negligible because the penetration depth of the transient temperature variations is small with respect to the inside radius of the exhaust manifold.
3.5 Turbine Connecting Pipe Heat Transfer

The heat transfer for the connecting pipe between the turbocharger and the power turbines is treated in the same way as for the exhaust manifold. The heat transfer coefficient is based on turbulent pipe flow using equation (3-12). The correction factor for the entrance effects is calculated based on a new entrance to the pipe at the turbine exhaust. The wall thermal conductivity and the inside wall surface temperature are calculated in the same manner described in Sections 3.3 and 3.4.
CHAPTER 4.
PRESSURE LOSSES

The pressure drop for the exhaust manifold of a 6 cylinder diesel engine was found by Primus [6] to be a factor of 10 to 15 times higher than the pressure drop calculated based on the friction factor for turbulent pipe flow. This increase in the flow losses is apparently due to the complex shape of the manifold and the interaction of the flow with the open passageways from other cylinders. For this reason, a specified flow loss factor is used for the calculation of the exhaust manifold pressure drop. Thus, the pressure drop at any instant of time is found from:

\[ p = K \left( \rho V^2 / 2 \right) \]  

(4-1)

where

- \( p \) = pressure drop in Pascals
- \( V \) = instantaneous velocity in the manifold
- \( \rho \) = manifold density

The velocity of the flow used in the pressure drop calculation for the exhaust manifold is based on the instantaneous mass flow through the turbine and the cross sectional area of the first runner. Similarly, the velocity of the flow for the connecting pipe between the turbines is based on the average turbine mass flow and the cross sectional area of the turbine connecting pipe.

Primus found that suitable values for \( K \) for the exhaust manifold were between 2 and 3.5. However, in the present study loss coefficients in this range gave an excessive pressure drop through the exhaust
manifold. Values of K between 0.5 and 1.0 gave more reasonable results in the current investigation.

There are two reasons for the difference between the two flow loss factors. The first is the fact that in the present study the turbine inlet flow was used to determine the flow velocity. The turbine inlet flow is in general higher than the mass flow through a runner and a pressure drop calculation based on this value will yield a higher result. The second is the sensitivity of the pressure drop to the manifold diameter used for the calculation. For a constant mass flow, the pressure drop varies with the diameter of the manifold raised to the fourth power. A slight difference in the diameter used for the representative section when calculating the pressure drop could easily explain the difference in flow loss factor between the two studies. A flow loss factor of 0.5 was used for the exhaust manifold in all of the calculations presented in this work.
CHAPTER 5.
THERMODYNAMIC EQUATIONS

5.1 General Equations

The three main control volumes of interest in the program are the intake manifold, the master cylinder, and the exhaust manifold. For each of these components the program determines the Thermodynamic State of the gases by integrating the derivatives with respect to time of the temperature, mass, pressure, and fuel fraction. The time derivatives for these properties are determined from the following equations based on the conservation of mass, conservation of fuel mass, the ideal gas law, and the First Law of Thermodynamics [11,22]:

Change in Mass with respect to time:

\[ \dot{m} = \sum_{j} \dot{m}_j \quad (5-1) \]

Change in Fuel Fraction with respect to time:

\[ \dot{F} = \sum_{j} \dot{m}_j \phi_j - mF \quad (5-2) \]

Change in Temperature with respect to time:

\[ \dot{T} = \frac{B}{A} \left[ \frac{m}{m} (1 - \frac{h}{B}) - \frac{V}{V} - \frac{C}{B} \phi + \frac{1}{Bm} \left( \sum_{j} \dot{m}_j h_j - Q_w \right) \right] \quad (5-3) \]

where

\[ A = c_p + \frac{(\partial p/\partial T)}{(\partial p/\partial \rho)} (\frac{1}{\rho} - c_T) \]

\[ B = \frac{1}{(\partial p/\partial \rho)} (1 - \rho c_T) \]

\[ C = c_\phi + \frac{(\partial \phi/\partial \phi)}{(\partial p/\partial \rho)} (\frac{1}{\rho} - c_T) \]
\[ \frac{\phi}{(F/A)_{\text{stoic}}} = \frac{E}{(1-F)^2} \]

Change in Pressure with respect to time:

\[ \dot{p} = \frac{-\partial \dot{p}}{\partial \phi} \left( - \frac{V}{V} - \frac{1}{\rho} \frac{\partial \phi}{\partial T} - \frac{1}{\rho} \frac{\partial \phi}{\partial \phi} \phi + \frac{V}{m} \right) \]

(5-4)

The exhaust manifold is further broken down into smaller sub-control volumes that are contained within the manifold control volume (see Chapter 2). For these sub-control volumes, the pressure derivative is that which is determined for the average exhaust manifold based on equation 5-4. With the pressure derivative an independent variable determined for the average manifold, the mass derivatives for the sub-control volumes become dependent variables. Rearranging equation (5-4) gives:

\[ m_1 = \frac{V_1}{m} \left[ \frac{\partial \rho}{\partial p} \dot{p} + \frac{\partial \rho}{\partial T} \dot{T}_1 + \frac{\partial \rho}{\partial \phi} \phi \right] \]

(5-5)

where the subscript \( i \) refers to the different sub-control volumes of the exhaust manifold.

By applying equations (5-1),(5-2),(5-3), and (5-5) to each separate sub-control volume of the manifold, the properties of each section can be determined.

The mass flow between sections of the exhaust manifold is determined by the mass flow rate through the exhaust valve and the rate of mass storage (i.e. retention) within any sections upstream of the section in question due to changes in the properties of those sections. Thus, the
mass flow out of the port section and into the runner section is found by subtracting the change in mass in the port as found by equation (5-5) from the mass flow through the exhaust valve.

5.2 Solution of the Equations

The solution of the simultaneous equations that result for the exhaust manifold is obtained through an iterative procedure. A Gauss-Seidel scheme is used that repeats the calculation of the derivatives for the exhaust manifold until convergence is achieved [29]. For the initial guess for the unknown time derivatives, the values from the previous call to the subroutine is used. The convergence of the new values for the derivatives is checked using Euclidean norms via the following equation:

\[
\frac{||u^{(s+1)} - u^s||_2}{||u^{(s+1)}||_2} < \epsilon \quad (5-6)
\]

where

- \( u^{(s+1)} \) = the new value of manifold derivatives
- \( u^{(s)} \) = the old value of the manifold derivatives
- \( \epsilon \) = the error tolerance

The Euclidean norm is defined by:

\[
||u^{(s)}||_2 = \left( \sum_{i=1}^{n} (u^{(s)}_i)^2 \right)^{1/2} \quad (5-7)
\]

The proper error tolerance, \( \epsilon \), was found by doing a series of runs with different values of \( \epsilon \). For large values of \( \epsilon \) the simulation ran longer because extra cycles were required for the program to converge. With small values of \( \epsilon \) run times increased because extra loops were made through the exhaust manifold iteration. A minimum run time was observed
when the trade-off between these two effects was balanced. The optimal value for \( \epsilon \) was determined in this manner to be \( 5 \times 10^{-5} \).

In this application the iterative procedure has certain advantages over direct methods (e.g. Gauss elimination) for the solution of the set of simultaneous equations for the exhaust manifold. First, it is more flexible. The program accepts the total number of control volumes to be used for the exhaust manifold as an input parameter. Given the complexity of the equations for the exhaust manifold, building in the same flexibility for a direct solution would have been considerably more difficult. Second, the resulting program is easier to follow for someone unfamiliar with the code. With the iterative procedure, it is not necessary to set up the actual matrix. The equations appear in the program essentially as they are written above. Finally, the computation time for the iterative solution is competitive with direct solution procedures. Since the program progresses in small steps, the new values for the derivatives are not changed much from the old values and convergence is reached rapidly. On the average, the loop for the exhaust manifold derivatives is passed through 1.5 times per call to the subroutine where they are calculated. The average number of passes through the loop is less than two because the predictor-corrector scheme of the integration routine often results in the exhaust manifold derivatives being calculated twice with only slightly different sets of conditions. For these calls to the subroutine for the exhaust manifold derivatives, the change in the values for the derivatives is not large enough to exceed the convergence tolerance and the loop is not repeated.
5.3 Time Average vs. Mass Average Temperatures

During the exhaust process, both the mass flow and the gas temperature of the exhaust gases vary substantially. In addition, following the exhaust process, there is a relatively long period of time when the gases are stagnant in sections of the manifold. Consequently, it becomes important to define what is the appropriate mean gas temperature.

The time average temperature is the simple mean gas temperature over a complete engine cycle. It is found by:

$$\overline{T}_t = \frac{\int_0^{720} T(\theta) \, d\theta}{720}$$  \hspace{1cm} (5-8)

where, $\overline{T}_t$ = the time mean gas temperature

$T(\theta)$ = the instantaneous gas temperature

The mass average temperature is defined by the average specific enthalpy of the gas that flows through a control volume. It can be found from:

$$\overline{T}_m = \frac{\int_0^{720} C_p \dot{m}(\theta) T(\theta) \, d\theta}{\int_0^{720} C_p \dot{m}(\theta) \, d\theta}$$  \hspace{1cm} (5-9)

where $\overline{T}_m$ = the mass average temperature

$\dot{m}(\theta)$ = the instantaneous mass flow through a control volume

$C_p$ = the specific heat of the gas under constant pressure
If $C_p$ is taken as constant over the temperature range of the exhaust gases, the above equation simplifies to:

$$
\bar{T}_m = \frac{\int_0^{720} \dot{m}(\theta) T(\theta) \, d\theta}{\int_0^{720} \dot{m}(\theta) \, d\theta}
$$

(5-10)

Both the time average and the mass average temperatures are calculated by the code. It will be seen that both of these mean temperatures are important.
CHAPTER 6.
MODEL APPLICATION AND BEHAVIOR

The primary engine of interest for use of the exhaust manifold model developed in this study is a turbocompounded diesel engine. Turbocompounded diesel engines, although still largely experimental, show the best possibility of benefiting from decreased heat transfer in the engine or exhaust manifold. In order to exercise the model and quantify the exhaust manifold losses in a real engine, the program was run for a cooled Cummins 6 cylinder NH engine in a turbocompounded configuration. This engine has been the subject of numerous other studies [11,23,24,25,26] and Cummins has made experimental test data available for calibration of the engine performance. Table 6.1 gives the specifications of the engine studied.

The program was run initially to match the program output with the Cummins data. This calibration was necessary because many of the empirical correlations require constants that may vary with engine design. Although the program had been calibrated for the reference engine by Assanis [11], the addition of the extra heat losses in the exhaust manifold required some recalibration of the engine in order to best match the experimental data. Following the calibration, parametric studies were used to investigate the effects of exhaust manifold insulation on engine performance for both normally cooled engines and low heat rejection engines. This chapter discusses the initial engine calibration and presents the general behavior of the model. The results of the parametric studies with insulation are discussed in Chapter 8.
6.1 Manifold Configuration

The exhaust manifold configuration for this engine was chosen to be ports, runners and a plenum. The port represented the portion of the exhaust from the exhaust valves to where the gases exit from the head. The runner represented the portion of the exhaust manifold downstream of the port where the gases from one cylinder flowed without mixing with the exhaust from other cylinders. The plenum represented the portion of the exhaust manifold where the exhaust gases from all of the cylinders mixed before entering the turbine. The layout for this system is shown in Figure 2.1.

6.2 Initial Calibration of the Engine

Table 6.2 shows the comparison of the simulation output versus the engine data provided by Cummins at 1600 and 1900 RPM at peak torque. 1900 RPM is the highest speed for which data were available. 1600 RPM is near mid-speed. From the table it can be seen that the agreement between the predicted and measured brake thermal efficiencies is within 2.5% with the predicted values higher than the measured. The predicted air flow was about 1% higher for the medium speed case and about 1.5% lower for the high speed case. The lower air flow in the high speed case was due to a 1.9% error in the intake manifold temperature as the intake manifold pressure was actually slightly higher for the simulation.

Table 6.3 shows the comparison for two lower loads at 1600 RPM. The loads correspond to approximately 50% and 25% of the peak torque fueling rate at that speed. For the lower loads, the errors between the simulation and the measured values increase. The engine performance is
predicted to be about 25% lower than the measured value for the lowest load case. Certainly, it is more difficult to accurately predict engine performance at lower loads. In general, at lower engine loads the predictions are more sensitive to the assumptions made and the models used for heat transfer and friction. As the losses increase in proportion to the brake output from the engine, slight errors in the correlations used for these losses are amplified and result in a larger percentage error in the performance predictions.

Also, the constants used for the combustion correlations probably need to be recalibrated for the lighter load conditions. For example, the burn duration is a specified input and for the tabulated runs was not readjusted. For lower loads the quantity of fuel injected decreases, the length of time necessary to inject the fuel decreases and it would be expected that the burn duration should also become shorter. However, to properly recalibrate the engine for the lower loads would require cylinder pressure profiles or combustion heat release rates. One run was done with the combustion duration specified as 80 crank angle degrees, a decrease of 36% from the value used for the run presented in Table 6.3. For the decreased burn duration, the reciprocator power output increased 9.4%. It is clear that this parameter needs to be modelled as a function of speed and load.

Another reason for the discrepancy between the predicted engine performance and the measured values may be due to the turbocharger maps used. For the 25% load case the predicted turbine efficiency is 20% lower than the measured values. This error propagates through the engine lowering the air flow and thus raising temperatures. One possible explanation for the error in the turbine efficiency is
discussed below. The accuracy of the performance predictions for the 50% load case fell between those for the low load case and the peak torque case. For the half load case, the engine performance was within 10% of the measured values. The air flow, manifold pressures, and manifold temperatures for this run showed similar agreement with the experimental results as the overall performance.

The exhaust manifold temperatures predicted with the exhaust manifold model developed in this study were lower than the measured values for all of the calibration cases. The disagreement in the port temperatures ranged from 50 to 75K. For the turbine inlet temperatures the discrepancy ranged from 10 to 30K. The error in the exhaust port temperatures is not surprising in light of the fact that the experimental data indicate a lower port temperature than turbine inlet temperature (see for example the measured gas temperatures for the exhaust port and turbine inlet on Tables 6.2 and 6.3). The problem obtaining an accurate measurement for the port gas temperature in an engine is not new [27] and appears to be is due to the highly unsteady nature of the gas temperature and mass flow through the port. These oscillations result in erroneous indications for port gas temperatures and the experimental results for the port have to be viewed with a degree of skepticism. The flow at the turbine inlet is more steady and as a result, the measurement error would not be expected to be as great as for the port. The reason for the disagreement at the turbine inlet is unclear and could be an error in either the experimental data or the predicted heat loss in the engine or in the manifold.

The calibration of the engine with the exhaust manifold heat transfer calculated in the present study predicted a lower the heat
transfer in the cylinder. The previous calibration of the simulation program done by Assanis [11] predicted exhaust manifold heat losses approximately one-third those predicted with this refined exhaust heat transfer model. In order to achieve the best match with the experimental data it was necessary to compensate for the increased heat transfer in the exhaust manifold by decreasing the cylinder heat transfer. As seen on Table 6.2 the volumetric efficiency and the brake horsepower with the adjusted cylinder heat transfer agree well with the experimental data. This agreement is an indication that the calibration is reasonable. The lowering of the cylinder heat transfer was achieved by raising the piston and head temperatures and decreasing the convective heat transfer constant coefficient by 22%. The total cylinder heat transfer decreased from about 13-14% at peak torque to 10-11%.

The calibration presented in Tables 6.2 and 6.3 was done with the turbine efficiency for the turbocharger multiplied by a factor of 1.08. Without the multiplication factor the manifold pressures and the engine air flow were lower than the experimental values by about 6-8%. It is interesting to note that under the low air flow condition, the turbine inlet temperature matched the experimental data within 10K for all cases. However, it was felt that it was unlikely that the experimental data for the air flow and the manifold pressures would all be in error and the adjustment was made to bring the air flow up. The adjustment to the turbine efficiency is not unjustified. It is possible that the maps used in the simulation program were not matched to the turbomachinery used for the experiment. The experimental data for the turbocharger combined efficiency and temperature drop across the turbocharger (for a
similar pressure ratio) both indicated a higher efficiency than was found by using the unadjusted turbine maps. All of the parametric studies for this engine were done with the multiplication factor of 1.08 for the turbine efficiency. The resulting turbine efficiency with the engine at peak torque was about 80% for the adjusted maps. The combined turbocharger efficiency with the correction factor was within one percentage point of the measured values for all cases except the medium speed low load. As discussed above, the turbine efficiency for the lowest load case studied was 20% lower than the measured efficiency with the correction factor used. Without the correction factor, the error would have been greater.

6.3 Boundary Conditions for the Exhaust Manifold

For the outside boundary conditions of the manifold with normal cooling, heat transfer coefficients were chosen from the literature and operating ambient temperatures were assumed. The port section is contained inside the head and surrounded by cooling water. A water temperature of 365K was used with a heat transfer coefficient of 6540 W/m²/K [2]. For the runner, plenum, and turbine connecting pipe, an ambient temperature of 322K was used with a heat transfer coefficient of 56.8 W/m²/K based on Robertson's work[4].

To test whether or not Robertson's value for the heat transfer coefficient is reasonable, empirical heat transfer correlations were applied to calculate the air velocity that would correspond to the heat transfer coefficient used. The heat transfer coefficient for flow around a submerged cylindrical body is found from [19]:
Using equation 6-1 and solving for the velocity for the heat transfer coefficient of 56.8 W/m\(^2\)/K yields a cooling air speed of 11 m/sec.

The exhaust manifold could also be thought of as a horizontal cylinder. In this case, flow over a flat plate would be a more accurate representation of the cooling air flow. The heat transfer coefficient for flow over a flat plate is found from [19]:

\[
\text{Nu} = 0.037 \text{Re}^{0.8} \text{Pr}^{0.3} \tag{6-2}
\]

Again by solving Equation 6-2 for the velocity, one finds a cooling air velocity of 18 m/sec. Both of these values are of comparable magnitude but they seem excessive for a laboratory test cell. However, when a simulation was run with the heat transfer coefficient for the outside of the runners and plenum set equal to 25 W/m\(^2\)/K, the resulting wall temperatures were above 800K. This temperature is high enough to cause the manifold to glow noticeably. Since this is not the case under normal test conditions, the reduced heat transfer coefficient is evidently too low. In the absence of more data, Robertson's value for the heat transfer coefficient for the air cooled sections of the manifold was used.

The uncertainty of the boundary conditions for the exhaust both inside and outside of the head is not a source of significant error. Because of the very high heat transfer coefficient of the cooling water
surrounding the port, most of the resistance to heat transfer is between the exhaust gases and the inside surface of the port. Thus, the port heat transfer is not sensitive to the outside boundary conditions in the range expected for the outside heat transfer coefficient. For the runner and plenum, the opposite is true. The thermal resistance is larger on the outside of the manifold where it is cooled by quiescent air than on the inside where exhaust gases flow with an average velocity on the order of 100 m/sec. For these sections the heat transfer is strongly dependent on the outside boundary conditions chosen. However, it turns out that the heat transfer coefficient to the ambient is so low that the total amount of heat lost from the runners and plenum is not a major portion of the heat loss from the manifold as a whole. For the cases with manifold insulation, the decreased thermal conductivity of the manifold walls provides an additional resistance to heat transfer and the sensitivity of the exhaust manifold heat transfer to the choice of boundary conditions is further decreased.

6.4 Behavior of a Representative Run

After having set up the model and calibrating the engine against experimental data, it is useful to observe how the exhaust manifold model works for the different sections of the manifold. In order to give a representation of the how the exhaust manifold variables behave over an engine cycle, a simulation was done at 1600 RPM and peak engine torque. Normal heat transfer was used. The results of this simulation were plotted out for one engine cycle (720 degrees crank angle) and demonstrate typical cycle variations.
6.4.1 Manifold Pressure and Pressure Drop: The exhaust manifold pressure and the pressure drop for the flow through the exhaust manifold are shown in Figure 6.1. The pressure fluctuations seen in this figure are applied to each sub-control volume of the exhaust manifold.

6.4.2 Mass Flow Along Exhaust Manifold: Figure 6.2 shows the exhaust mass flow between control volumes. The top plot is for the flow through the exhaust valve and into the port. This flow follows the classic exhaust pattern with two peaks. The first peak occurs during the blow-down period when the flow is driven by the high pressure drop across the valve resulting from the cylinder pressure being relatively higher than the exhaust manifold pressure during this phase. The second peak occurs during the exhaust stroke when the flow is driven by the piston motion in the cylinder and there is little pressure drop across the exhaust valve. During the period that the valve is closed, the mass flow is equal to zero.

The second plot is for the flow between the port and the runner. During the valve open period, the flow between the port and the runner is very close to the flow through the valve. The difference between these two flows is caused by the changing pressure and temperature in the port control volume which affects the mass balance (i.e. the mass retained) for the port. During the period when the valve is closed, some flow between the port and the runner is apparent. This is largely due to the pressure oscillations in the exhaust manifold. When the exhaust manifold pressure increases, mass must flow back into the port from the runner because of the increase in density in the port. When the exhaust manifold pressure decreases, the opposite occurs and mass flows out of the port even though the exhaust valve may be closed.
This flow is a result of the uniform pressure assumption for the overall manifold in this model. In an engine set up for pulse turbocharging the effect of the pressure oscillations may not be as large as the model suggests.

The third plot is for the mass flow between the runner and the plenum. During the period when the valve is closed, the flow between the runner and the plenum oscillates as it did for the flow between the port and the runner but with greater amplitude. This difference is due to the larger volume that must be filled and emptied as the exhaust manifold pressure oscillates. The period of these oscillations corresponds to the period between exhaust processes for the engine.

In all three plots a dip can be seen in the mass flow during the exhaust stroke. This drop in mass flow occurs when the next cylinder's exhaust valve opens and the exhaust manifold pressure increases as a result. Again, the drop in mass flow during the exhaust stroke when the next cylinder exhaust valve opens is due to the uniform pressure assumption for the manifold and would not necessarily be seen in an engine set up for pulse turbocharging.

6.4.3 Exhaust Manifold Gas Temperatures: Figure 6.3 shows the gas temperature of the cylinder, port, and runner during an engine cycle. It can be seen that the port oscillations are close to 200K peak to peak with the engine speed of 1600 RPM. The magnitude of the oscillations varies inversely with the engine speed decreasing to 150K at 1900 RPM. The magnitude of the oscillations also decreases with decreasing engine load. This figure shows that the dominant factor in the exhaust temperature oscillations is not due to exhaust heat transfer but rather closely follows the temperature of the gases flowing into the port from
the cylinder. The temperature of the cylinder gases is varying during the exhaust process due to the combined effects of the pressure change in the cylinder, the heat transfer in the cylinder, and the work exchange between the piston and the cylinder gases. Thus, the gas temperature variations in the exhaust manifold are largely governed by the thermodynamics inside the cylinder and to a lesser degree by the heat transfer and mixing in the manifold.

During the period when the exhaust valve is closed, the exhaust manifold gas temperatures are relatively low compared with the temperatures during the period with the exhaust valve open and they oscillate in a near sinusoidal fashion. The low temperature is a result of the temperature of the gases that flow into the manifold just before the exhaust valve closes. Since the exhaust valve is closed during approximately two-thirds of each engine cycle, the low temperature of the exhaust gases during this period has the effect of lowering the indicated temperature reading of a thermocouple placed in the exhaust manifold even though the mass flow during this period is nil. This phenomenon may explain why experimental port temperature readings tend to be lower than gas temperature readings downstream of the port. The magnitude of the oscillations while the exhaust valve is closed corresponds to the temperature changes due to isentropic compression and expansion of the gases.

6.4.4 Plenum and Averaged Manifold Gas Temperature: Figure 6.4 shows the plenum temperature and the averaged manifold gas temperatures during a complete engine cycle. For both volumes, the period of the temperature oscillations is the time between exhaust events between cylinders. The figure shows that the plenum temperature peak to peak
oscillation is larger in magnitude than the averaged manifold oscillation. The increase is due to the smaller volume of the plenum and hence less dilution with gases in the other sections of the manifold. The plenum temperature is a more accurate representation of the temperature oscillations that would occur at the turbine inlet in a real engine. The averaged manifold gas temperature represents the temperature that would be predicted for a single volume filling and emptying model for the exhaust manifold.

6.4.5 Manifold Heat Transfer Coefficients: Figure 6.5 shows the heat transfer coefficients predicted in the three sections of the exhaust manifold. The top plot is of the port heat transfer coefficient. On the plot the three phases of the valve open period are delineated. The correlation for the first phase results in the heat transfer coefficient rising very rapidly when the valve first comes open and then remaining relatively constant until the correlation for turbulent pipe flow (Eq. 3-12) is applied. The correlation applied during the valve closing phase (Eq. 3-3) results in a sharp decrease in the heat transfer coefficient which may be an indication that the correlation for this phase is not particularly well suited for the engine of this study. Since the port heat transfer during the valve closing phase is not a significant portion of the total port heat transfer, any error introduced by the correlation used is negligible.

The middle plot is for the runner heat transfer coefficient. The turbulent pipe flow correlation is applied during the entire engine cycle for this section of the exhaust manifold and the heat transfer coefficient closely parallels the absolute value of the mass flow.
The bottom plot is for the plenum. Although not shown on the figure for the mass flows in the exhaust manifold, the mass flow through the plenum is the sum of the mass flows from all of the cylinders in the engine (the mass flow from the master runner is added to the mass flow from the runners from the other cylinders in the engine). Hence, the heat transfer coefficient is relatively constant with peaks as each cylinder goes through blow-down.

6.4.6 Heat Transfer Rates: Figure 6.6 shows the heat transfer rates for each section of the exhaust during an engine cycle. Again, the three phases of the exhaust port heat transfer can be clearly seen. The plenum heat transfer is negative for short periods when the gas temperature goes below the wall temperature.

6.5 Problems With the Turbomachinery Maps

As part of the debugging process for the program, a problem occurred at low engine speeds with the compressor going off of the map along the surge line. To alleviate this problem, the map was extended using a linear extrapolation from the previous two points on the map and a flag was added to print a warning in the program output when the compressor went into the region of the extended map. For the cases run with normal heat transfer, the warning appeared for engine speeds of 1500 RPM and below. For this reason, few calculations were done at the lower engine speeds as their validity would be in doubt. In some cases, the results from a low speed run have been included for reference or to show a continuing trend.

For the studies done with reduced heat transfer, the compressor went beyond the normal map range with an engine speed of 1600 RPM. For this
reason the parametric studies for the insulated cases were conducted at an engine speed of 1900 RPM.

6.6 Time Averaged vs. Mass Averaged Temperatures

The averaging of the gas temperature over an engine cycle is done on both a mass basis and a time basis (see Chapter 5). Figure 6.7 gives an indication of how the two temperature indications vary along the length of the exhaust manifold for a run at 1600 RPM and peak torque. The temperatures are plotted with the abscissa normalized with respect to heat transfer surface area. The mass average temperature shows a steady decrease in temperature along the manifold starting with the average exhaust temperature through the exhaust valve. The time averaged temperature is considerably lower than the mass averaged temperature in the port but increases steadily along the length of the manifold. At the turbine inlet, the two temperatures have converged to agree within one degree. Table 6.4 gives the time average and the mass average temperature results for a range of operating conditions. For the high load cases, the difference between the mass average and the time average temperature increases with decreasing speed. The difference in the port is 50K at 1900 RPM and increases to 91K at 1400 RPM. For the lighter load cases, the difference between the mass average temperature and the time average temperature decreases.

The reason for the difference between the two averages for the exhaust temperatures can be explained by the temperature profile of the exhaust gases over an engine cycle (see Subsections 6.4.3 & 6.4.4). The mass averaged temperature reflects those periods with high mass flow and high temperature. The time average temperature ignores the mass flow
and gives equal weight to the gas temperature during the relatively long period when the exhaust valve is closed. In the plenum, the exhaust gases from all of the cylinders are mixed and the gas is much more uniform in temperature and flow rate. With the uniform flow and temperature, the difference between the mass average temperature and the time average temperature disappears.

The discrepancy between the mass average and time average gas temperatures may help explain why the port temperature measurements in real engines are typically lower than temperatures further downstream. The thermocouple measuring the port temperature is surrounded by relatively cool exhaust gases during two thirds of the cycle. During this time the thermocouple continues to exchange heat with the gas around it even though the mass flow is zero on the average. This cool down period results in an artificially low temperature reading for the port and a temperature reading that is somewhere between the time average and the mass average temperatures.

6.7 Distribution of heat losses in the exhaust

The results shown on Tables 6.2 and 6.3 indicate that the exhaust manifold heat losses are approximately 5 to 6% of the fuel energy input to the engine. The distribution of these losses is shown on Figure 6.8. It is interesting to note that although the port surface area is only 37% of the total surface area of the manifold, it accounts for 76% of the heat loss. This distribution is due to the higher heat transfer coefficients both inside and outside the port. The outside heat transfer coefficient is more than an order of magnitude greater than that for the runners and plenum and results in the wall temperature for
the port of about 400K. The runners and the plenum have inside wall temperatures between 700K and 800K. This difference translates into an average temperature drop between the gas and the inside wall of 370K for the port compared to 70K for the runner and 30K for the plenum. The heat transfer in the port is also enhanced by a larger heat transfer coefficient than that for the runner. This comparison is based on engine operating conditions of 1600 RPM and peak torque and normal cooling. It is representative of the results for other cooled engine operating conditions.

6.8 Distribution of energy in the engine

The energy balance for the engine system from the intake valve to the turbine inlet is shown on Figure 6.9 for an engine speed of 1900 RPM and full load.
CHAPTER 7.
8 CYLINDER TURBOCHARGED ENGINE EXPERIMENT

Following some of the initial presentations of results for the current study, Cummins Engine Co. expressed an interest in validating the exhaust manifold model by doing an experiment on a specially instrumented test engine. The experimental engine was an 8 cylinder turbocharged engine which was instrumented with thermocouples along one exhaust gas flow path. The original intent of the experiment had been to measure transient gas temperatures in the exhaust to be able to confirm the peak to peak gas temperature variations predicted by the simulation. Unfortunately, due to instrumentation problems only slow time response temperature measurements were obtained. Nonetheless, the experiment was useful in providing a more complete basis for comparison of the results predicted by the simulation with actual test data. This chapter discusses the experimental set-up and compares the test results with the simulation output.

7.1 Description

The test engine was a 14.8 liter (903 in$^3$) displacement turbocharged V8 engine. The engine specifications are given on Table 7.1. The engine has a single intake manifold located between the two banks of cylinders and 4 exhaust runners along the outside of the engine flowing towards the rear where the turbocharger is located. A schematic of the engine layout is shown on Figure 7.1. Each exhaust runner receives the exhaust flow from two cylinders and remains separate from the runners from the other cylinders up to a pulse converter attached to the inlet
of the turbocharger. The runner for the numbers 2 and 4 cylinders was intrumented with 4 thermocouples along its length. The first thermocouple was located just outside the #4 cylinder port. The second thermocouple was located downstream of the junction where the exhaust flow from the two cylinders came together. The third and fourth thermocouples were located at the pulse converter inlet and outlet respectively. The approximate locations of these thermocouples is also shown on Figure 7.1.

The turbocharger maps for the experimental engine were not available at the time the present study was done. As a substitute, the turbocharger maps from the 6 cylinder engine were adjusted to give consistent operating conditions compared with the actual test data. The mass flow was adjusted for both the compressor and the turbine and the turbine efficiency was also adjusted to get a good fit with the measured turbocharger performance. The effective valve areas for the 8 cylinder engine and the exhaust manifold geometry were supplied by Cummins. The port geometry was found from measurements taken on a head from the V-903 test engine in the Sloan Automotive Laboratory.

7.2 Description of simulation model

The exhaust manifold layout for the experimental engine required a different configuration of the exhaust manifold modell than that presented in Chapter 6. In this case, the exhaust flow path was divided into a port, a first runner, a second runner and a plenum. The second runner received the flow from the master cylinder and also the flow from a second cylinder with a phase shift of 270 degrees. The phase shift was determined from the firing order of the engine. The other three
components of the exhaust were essentially the same as for the 6
cylinder engine discussed in Chapter 6. The only other exception was
that the plenum received the exhaust flow from the master second runner
and from three others like it with the appropriate phase shifts applied.
The heat transfer correlations were not changed. The layout of this
model is shown on Figure 7.2.

7.3 Calibration of the engine simulation

The simulation calibration was done with particular attention paid
to matching predicted and experimental values for the intake and exhaust
manifold pressures, the engine air flow, and the engine power output.
The cylinder heat transfer was adjusted to match the volumetric
efficiency with respect to the intake manifold, typical cylinder wall
temperatures measured on a similar V-903 engine [28], and the gas
temperature at the turbine inlet. Table 7.2 shows a comparison of the
simulation output and the experimental test data.

The agreement between the simulation and the experimental data was
very good for the two speeds tested. Unfortunately, the only other
speed for which a complete set of data was available was 1500 RPM.
Engine operation at this speed could not be simulated because it fell
outside the region covered by the turbocharger maps. To some extent the
matching between the simulation output and the experimental values was
aided by the lack of information about the particular engine being
tested. Because of the sparsity of information, the turbine maps and
the combustion constants had to be adjusted until good agreement was
achieved. In order to do a proper calibration and validation of the
simulation, the correct turbocharger maps and transient cylinder
pressure traces would be required. However, for the purposes of the present study, the main concern was to have the proper mass flow through the exhaust manifold at approximately the correct temperature compared with the experimental data. This goal was achieved with the engine simulation. The air flow through the engine was within 0.5% for the two cases run and the turbine inlet temperature was within 10K for the 2600 RPM run and within 3K for the 2200 RPM case.

7.4 Manifold temperature predictions vs. measurements

With the simulation calibrated to give the same conditions in the exhaust manifold as for the experimental engine, a comparison can be made between the predicted gas temperatures along the length of the manifold against the measured values. Such a comparison is useful for observing the differences between the time average temperature and the mass average temperature and seeing where the experimentally measured temperatures fall with respect to the two predicted values. Surely, the mass averaged temperature is the more correct indication of the energy content of the gas but it was seen in Chapter 6 that at least the port thermocouple readings were not mass average values.

One of the difficulties with comparing the simulation results with the engine test data is the cylinder to cylinder variation. This inconsistency is seen in the port gas temperatures which were recorded for all 8 cylinders. Due to cylinder to cylinder variations (for example in air flow or fuel flow to individual cylinders or cylinder heat transfer) and/or due to experimental error, the port temperature measurements differ by as much as 30K from the mean port temperature. The #4 cylinder was 9K above the average for the 2600 RPM run and 19K
above the average for the 2200 RPM run. The #2 cylinder was 6K and 10K above the average for the same two runs. Figure 7.3 has two bar charts showing the port gas temperature indication with respect to the average temperature for each of the cylinders for the two engine speeds studied. The exhaust gas temperature data presented on Table 7.2 is based on the measurements taken in the instrumented section of the exhaust.

Figures 7.4 and 7.5 are plots of the predicted mass average temperature, the predicted time average temperature and the experimental gas temperature along the length of the exhaust manifold. The first figure is for the 2600 RPM case and the second is for the 2200 RPM case. The points have been located along the abscissa based on the heat transfer surface area upstream of the location of the temperature indication normalized with respect to the total surface area of the manifold. Note that the thermocouple #2 was somewhat downstream of the division between the first and second runners because of the location of the thermocouple on the test engine (see Figure 7.1). The same trend can be seen on both plots. For the port, the measured temperature fell approximately half way between the time averaged and the mass averaged temperatures of the simulation. The error between the measured port temperature and the mass averaged temperature was 58K for the 2600 RPM run and 40K for the 2200 RPM run. For thermocouple #2 the measured temperature increased and was about 20K above the mass averaged. From the second thermocouple downstream to the turbine inlet the measured gas temperature decreased steadily. The rate of decrease for the measured temperature was slightly greater than that for the mass average predicted temperatures and the two temperatures agreed at the turbine inlet. Based on the slope of the measured temperatures for the last
three thermocouples, it is feasible that the exhaust manifold heat transfer was underestimated in this portion of the manifold even though a relatively high heat transfer coefficient was used for the outside boundary condition (see Chapter 6). It is apparent from Figures 7.4 and 7.5 that the port thermocouple reading gave considerable error when compared to the mass averaged temperature. However, further downstream after the junction for the exhaust from another cylinder, the thermocouple reading agreed well with the mass averaged temperature. Unfortunately, the manifold lost heat in the length upstream of these thermocouples and the resulting temperature drop would have to be taken into account when estimating engine exhaust temperatures. Based on the simulation, the heat loss can cause a temperature drop of up to 50K before the first thermocouple whose reading agrees with the mass averaged temperature in the exhaust manifold.

Figure 7.6 is a reproduction of the time averaged temperature traces taken by Cummins during the experiment with the engine running at 2600 RPM and rated load. The lines traced are only the deviations from the time average temperature and do not reflect the relative average gas temperatures. A marked difference is seen between the temperature fluctuations at the port (Thermocouple #1) and those for the three thermocouples downstream. The port peak to peak gas temperature fluctuations over the engine cycle were 20K (36F) whereas the other three thermocouples had variations of less than 5K. It is clear from a comparison with the magnitude of the predicted gas temperature variations presented in Chapter 6 that there was considerable damping in the time response of the thermocouples. Although they were made of 0.030" diameter wire, the time response of the thermocouples was too
slow to accurately measure cyclic gas temperature variations. However, taking into account that the thermocouples all had approximately the same time response, the port thermocouple evidently was exposed to far greater fluctuations in flow conditions. This is supported by the four times greater magnitude in the port thermocouple temperature variation over the cycle. This suggests that the high variation in the port flow conditions is directly related to the port thermocouple average temperature reading and accounts for the inaccuracy in port temperature measurements using slow time response measuring devices. In contrast to the inaccuracy in the port temperature measurement, the thermocouples located downstream are in zones where the flow is more steady and the temperature measurements agree better with the predicted mass average temperatures as a result. The relatively low magnitude of the thermocouple temperature fluctuations in the other three locations downstream of the port also indicates that the exhaust flow pulses may be spread out more quickly than the simulation would suggest.

7.5 Predicted thermocouple temperatures

It was seen in the previous section that the thermocouple measurements in an operating engine agree with neither the time average nor the mass average temperatures. This presents a problem in comparing the predicted results with the experimental results that prohibits confirmation of the accuracy or inaccuracy of the predicted gas temperatures. It also leads to speculation as to the accuracy of the thermocouple measurements themselves. To investigate this question a study was done to see if the thermocouple readings could be predicted based on the gas flow conditions over an engine cycle predicted by the
simulation program. A similar study was done by Caton [27] for a spark ignition engine and this study was based on his work.

The predicted thermocouple temperature is found by integrating in time the change in thermocouple temperature due to the flow of heat to and from the thermocouple body. The relationship between the change in energy in the thermocouple and the change in temperature assuming a uniform temperature distribution within the thermocouple is defined by the following:

$$\dot{Q} = \rho V C \frac{dT_{TC}}{dt}$$

(7-1)

Where,

- $Q$ = The change in energy with respect to time of the thermocouple
- $\rho$ = the thermocouple density
- $V$ = the thermocouple volume
- $T_{TC}$ = the thermocouple temperature

The heat balance on the thermocouple is found from the sum of the heat transfer due to convection between the thermocouple and the gas and due to radiation between the thermocouple and the manifold wall. The radiation between the thermocouple and the gas is neglected because of the low emissivity of the gas in the exhaust and the low temperature drop between the gas and the thermocouples. The convection heat transfer for a submerged sphere is found from:

$$\dot{Q} = h A (T_G - T_{TC})$$

(7-2)

where the heat transfer coefficient, $h$, is found from:

$$h = \frac{k}{D} \left( 2 + (0.4 \text{ Re}^{0.5} + 0.06 \text{ Re}^{0.66}) \text{ Pr}^{0.3} \right)$$

(7-3)
The radiation heat transfer from the thermocouple to the manifold wall is found from the Stefan-Boltzman law:

\[ Q = \varepsilon \sigma A (T_{TC}^4 - T_W^4) \]  

(7-4)

Combining equations 7-1 through 7-4 and converting from real time to crank angle gives an overall equation for the change in thermocouple temperature with respect to crankangle.

\[ \frac{dT_{TC}}{d\theta} = \frac{1}{\rho C} \frac{A}{V} \left[ h(T_g - T_{TC}) - \varepsilon \sigma (T_{TC}^4 - T_W^4) \right] \frac{1}{d\theta / dt} \]  

(7-5)

A new subroutine was written to apply equation 7-5 at the four locations where thermocouples were located on the experimental engine. The heat transfer was calculated on a transient basis over the engine cycle for each of the four thermocouples. The heat transfer rates were determined based on the instantaneous values for the mass flow and gas temperature for the control volumes immediately upstream of the simulated thermocouples. In order to better simulate the location of thermocouple #2 which was located approximately halfway between the junction of the runners from the #4 and #2 cylinders and the pulse converter, the second runner described in section 7.2 was divided into two to make a total of 5 control volumes for the exhaust manifold.

The following values were used for the constants in equation 7-5:

\[ \rho = 8.68 \times 10^3 \text{ kg/m}^3 \]
\[ C = 481 \text{ J/kg/K} \]
\[ \varepsilon = 0.85 \]

The surface area to volume ratio was based on a sphere of radius 0.76 mm (0.030 in).
The thermocouple was actually fabricated from 0.76 mm wire fused together to form a junction. The shape of the junction could not properly be called either spherical or cylindrical but simulations using a cylindrical geometry and heat transfer correlation gave results very close to those using spherical geometry. Only the results of the spherical geometry simulations are presented here.

Figures 7.7 and 7.8 are reproductions of 7.4 and 7.5 with the predicted thermocouple temperatures added. On both figures 7.7 and 7.8 it can be seen that the predicted port thermocouple temperatures are very close to the measured values. For the high speed case, the agreement was within 7K and for the medium speed case, the agreement was within 13K.

The agreement for the predicted thermocouples further downstream was not as good. The predicted thermocouple temperatures did not increase as much as the measured temperatures downstream of the addition of the flow from another cylinder. For these thermocouples, the error between the predicted and measured thermocouple temperatures was in the range of 30 to 50K. At the plenum and turbine inlet, the flow from all of the cylinders has converged and the model agrees well with the actual engine. At this point the mass average, time average, measured, and predicted thermocouple temperatures all agree within a few degrees K.

To determine the importance of radiation on the thermocouple temperature, the simulation was run at 2600 RPM with the thermocouple radiation heat transfer omitted. The results of this run have also been included on figure 7.7. With the radiation heat transfer removed, the predicted port thermocouple temperature increased 11K.

Figure 7.9 shows the transient thermocouple temperature predicted
for the port, the second runner and the turbine inlet thermocouples. The scale on this plot is in Fahrenheit degrees to match that of figure 7.6. The time response for the simulated thermocouples was very slow. Although gas temperature variations of 150K were predicted for the port gas temperature in the simulation, the thermocouple temperature variation was only about 3.5K over the complete engine cycle.

When compared to the transient measurements made by Cummins (Fig. 7.6), the lower magnitude of the simulated thermocouple temperature variation over an engine cycle would indicate that either the thermocouple surface area to volume ratio or the heat transfer coefficient between the gas and the thermocouples was low for the simulation. However, the simulation did confirm that the time response of the experimental thermocouples was very slow with respect to the transients in the exhaust manifold. For this reason the experimental measurements were considered to be steady-state temperatures only. The general shape of the of the temperature traces over the engine cycle is similar for both the predicted and the measured thermocouple temperatures.
CHAPTER 8.
PARAMETRIC STUDIES

With the model verified and calibrated, it is ready to be used for parametric studies to investigate what advantages, if any, may be achieved by insulating the exhaust manifold of an internal combustion engine. The results of the calibration runs indicate that approximately 5% of the energy supplied to the engine is lost due to heat transfer in the exhaust system. This loss represents a significant amount of heat—roughly 12% of the work output of the engine. If a portion of the energy lost in the manifold could be recovered, it would result in an improvement in the efficiency of the engine.

To study this question, a number of simulations were done with different levels of exhaust manifold insulation. The first investigation was about how the reciprocator exhaust gas temperature and the turbine inlet gas temperature are affected by exhaust manifold insulation. Then, one set of simulations was done with normal insulation in the cylinder and another set with the head and piston insulated to cut heat losses in the cylinder by approximately 50%. These were done to study the effect of exhaust manifold insulation on overall engine performance. Investigations of this type are useful in determining if there are worthwhile gains to be achieved with exhaust manifold insulation or if as was found for the case of cylinder insulation [11], that the benefits of insulating the exhaust are marginal.

8.1 Basic Assumptions

For the purposes of studying the effects of exhaust manifold insulation, the assumptions used to hold various engine parameters
constant were essentially the same as those used by Assanis [11] in his studies of the effects of engine cylinder insulation. They were the following:

1. The engine geometry and calibration constants were not changed.
2. The start of injection (346 degrees BTDC) and the amount of fuel injected were fixed. Although the combustion calibration constants were not varied, the ignition delay and the heat release profile may have changed as engine operating conditions (cylinder temperature, pressure, and equivalence ratio) change. Implicit in this assumption is that no limit was placed on the peak cylinder pressure.
3. The turbomachinery maps were not changed. All runs were done with the turbine efficiency from the original maps multiplied by a factor of 1.08 for the reasons discussed in Chapter 6. With the exception of this correction, the turbomachinery efficiency was determined from the maps based on the operating conditions for the case being simulated.
4. Friction losses were based on reciprocator speed only.
5. Mass flow rates through the system components were allowed to readjust for the case being studied. The new flow rates were determined by the matching between the reciprocator and the turbomachinery based on the operating conditions of each.

8.2 Effect of Decreased Manifold Heat Transfer on Gas Temperatures

It was found in the initial calibration of the engine that the temperature drop between the exhaust valve and the turbine inlet was about 60K at peak torque and remains more or less constant with engine speed. The temperature drop decreases slightly with load down to about
40K at the 25% load case presented (see Tables 6.2 and 6.3). The net result of insulating the exhaust manifold would be to decrease the temperature drop of the gases as they flow through the manifold.

Figure 8.1 shows how the reciprocator exhaust gas temperature and the turbine inlet gas temperature are affected by manifold insulation. The four cases presented are normal heat transfer, exhaust manifold insulation to decrease the heat losses by 33% and 66%, and zero heat transfer in the exhaust. As would be expected, increasing the level of exhaust manifold insulation causes the turbine inlet temperature to increase. It is interesting to note what happens to the reciprocator exhaust gas temperature when the manifold is insulated. The gas temperature leaving the reciprocator decreases as the manifold heat transfer is decreased. The exhaust gases are cooler because of the higher air flow that results when the enthalpy of the gases at the turbine inlet is increased. For the same fuel flow, the higher air flow translates into a lower equivalence ratio and the gas temperatures decrease as a result.

8.3 Effect of Exhaust Manifold Insulation on Engine Performance

The performance of the engine was studied for three levels of exhaust manifold insulation. The first was the base case with no insulation. The second case was the manifold insulated to a level that decreases heat transfer by about 66%. This was done by using a 1.5 mm layer of plasma sprayed zirconia with a thermal conductivity of 0.6 W/m/K and a thermal diffusivity of $0.54 \times 10^{-6}$ m$^2$/s. The ceramic was surrounded by a metal pipe 8 mm thick with a thermal conductivity of 54.4 W/m/K. For the insulated case, the boundary conditions for all
sections of the manifold were the same as those used for the runner and the plenum for the base case. This effectively corresponds to removing the water cooling around the port and replacing it with air cooling. The boundary conditions applied were an ambient temperature of 322K and an ambient heat transfer coefficient of 56.8 W/m²/K. The third case studied was the limiting case of an adiabatic manifold. The heat transfer for the manifold was set identically equal to zero for this case. The adiabatic case was actually done by using the option in the program to treat the exhaust as a simple plenum since the exhaust manifold heat transfer was no longer being calculated. The constant for the heat transfer correlation was set to zero and the pressure drop factor was corrected based on the new diameter to give the same pressure drop through the manifold. The geometry of the plenum was fixed to assure that the volume matched that of the manifold modeled with the sub-control volumes.

All three levels of exhaust manifold heat transfer were run with normal cylinder heat transfer and again with the cylinder partially insulated with a 5 mm layer of a ceramic material, K=0.6 W/m²/K, added to the head and piston. This level of insulation was sufficient to lower the cylinder heat transfer by about 50%. It should perhaps be noted that with present day technology, it is not possible to apply plasma sprayed zirconium with the thicknesses simulated. However, the purpose of the present study was only to study the effects of decreased heat transfer and demonstrate how the program could be used for arbitrary materials. Thus, no implication of the practicality of the materials simulated is intended here.
For the sake of accuracy, all of the calculations presented for the insulated cases were done using transient wall temperature calculations. However, as will be seen below, the difference between the transient wall temperature performance predictions and those with steady-state wall temperatures is very small.

8.3.1 Normal cylinder heat transfer: Table 8.1 gives a summary of the results for the three levels of exhaust manifold insulation with normal heat transfer in the cylinder. The net effect of the insulation was to raise the overall system efficiency from 43.2% to 43.6% for the partially insulated exhaust and to 43.9% for the adiabatic case. The reciprocator performance remained almost unchanged going from 298.3 kW (400 hp) to 299 kW (401 hp) for the insulated cases. Most of the performance improvement was due to gains from the power turbine performance. The power turbine output increased steadily with exhaust manifold insulation. The power turbine output for the uninsulated case was 32.8 kW (44 hp) and increased to 35.8 kW (48 hp) for the partially insulated exhaust and an output of 38.0 kW (51 hp) was predicted for the adiabatic exhaust. This corresponds to a 16% improvement in the power turbine output with the exhaust heat losses removed.

It is somewhat surprising that the reciprocator performance did not improve with the exhaust manifold insulation. Contrary to insulating the cylinder which results in a decrease in air flow, exhaust manifold insulation increases the air flow through the engine. For the adiabatic case the increase was about 5%. An increase in air flow alone in an internal combustion engine would be expected to increase the thermal efficiency [30]. However, the improved air flow came at the expense of higher exhaust manifold pressures and an increase in pumping
losses. The increase in gross mean effective pressure due to the improved air flow was exactly counteracted by the increase in pumping losses.

One beneficial effect on the reciprocator due to insulating the exhaust was the decrease in peak cylinder temperature. Largely due to the increase in air flow, the average cylinder temperature predicted by the simulation decreased by 30K for the partially insulated case and by 40K for the adiabatic case. Although the one zone model used in the program is not applicable to predict actual cylinder temperatures, it can be useful for showing trends and a corresponding decrease in combustion temperatures can be expected in an actual engine.

It should be noted that no effort was made to optimize the engine for operation with the insulated manifold. Neither the turbocharger maps nor the combustion parameters were changed from the original calibration of the engine. It is feasible that the increase in pumping losses could be eliminated by increasing the aspect ratio of the turbocharger turbine while still retaining the benefit of increased air flow through the reciprocator. However, it was not the purpose of the present study to attempt to optimize the engine operation. Optimization of the engine under the new operating conditions would require a considerable investment in time because of the complexity of the system and the number of parameters that need to be adjusted to give the true optimal engine performance.

8.3.2 Partially insulated cylinder: With the cylinder partially insulated the performance gains with exhaust manifold insulation are very similar to those for the uninsulated cylinder case. In this set of runs, the base case overall brake thermal efficiency was 44.8% and
increased to 45.3% for the partially insulated exhaust and 45.5% for the adiabatic exhaust. Again, this represents a potential performance gain of 0.7%. As with the uninsulated cylinder case, most of the performance gain was due to the increase in power output of the power turbine. Table 8.2 gives a summary of the results from this set of runs.

8.4 Effect of Transient Wall Temperatures on Predicted Performance

The simulation code has been set up to do either a steady-state wall temperature calculation or a combined steady-state/transient wall temperature calculation. Assanis [11] found in his study of cylinder heat transfer that the transient variations in wall temperature had no effect on engine performance for the uninsulated case. Whereas for the insulated cylinder, the difference between the steady-state wall temperature performance and the transient wall temperature performance was 0.4% in the overall engine performance. A similar comparison was done in this study to determine the approximate magnitude of the exhaust manifold wall temperature swings and their effect on the engine performance predictions.

The cases studied were the same as used for the initial engine calibration and the parametric studies discussed above. For the uninsulated cylinder and uninsulated exhaust, the piston and head temperature swings were about 14K peak to peak for the piston and cylinder head. This result agrees with Assanis'. For the port, the temperature swings were about 4K peak to peak. For both the runner and the plenum the peak to peak temperature variation was less than 1K. Because the wall temperature variations during an engine cycle were very small compared to the temperature differential between gas and the
wall, the engine performance was unchanged by the inclusion of the transient wall temperature calculation.

Table 8.3 presents a comparison of the results between the engine performance using a steady-state wall temperature and that using a transient wall temperature calculation. The two cases listed are for the exhaust manifold partially insulated with the cylinder uninsulated and with both the cylinder and the exhaust manifold partially insulated. Both sets were run at 1900 RPM and peak torque. For the uninsulated cylinder, a very slight increase in air flow occurred when transient wall temperatures were used, however the effect was not enough to change the overall engine performance. For the insulated cylinder, the effect of using transient wall temperatures for the simulation resulted in an increase in overall engine efficiency from 45.2% to 45.3%. This difference is insignificant.

For the insulated exhaust manifold the peak to peak temperature swings for the transient wall temperature calculation were modest. For the port the difference between the maximum and minimum wall temperatures over an engine cycle was 22K. For the runner it was 15K and for the plenum it was 4K. The plenum sees continuous mass flow so that the heat transfer in this control volume is much more steady. Figure 8.2 shows the port gas temperature and the port wall temperature over a complete engine cycle at 1900 RPM. From the figure, it can be seen why the transient wall temperature calculation has such little effect on the manifold heat transfer. The temperature drop between the port gas and the wall during blow-down when most of the heat transfer in the port takes place, was about 170K. The increase in wall temperature of 22K represented a decrease of less than 13% in the temperature.
differential driving the heat transfer. Since only about 2% of the engine energy input is lost due to heat transfer in the insulated exhaust, the change in heat transfer for the transient wall calculation was only about 0.25% of the fuel energy. Thus, transient wall temperature calculations for the exhaust manifold do not provide a significant change in the performance predictions from the simulation code. The insignificance of the transient wall temperature calculations on the engine performance gives further demonstration that the plane geometry finite difference scheme used for the calculation is justified.
CHAPTER 9.
CONCLUSIONS

This thesis presents the results of development of a model for the flow process in the exhaust manifold of an internal combustion engine for use in a computer simulation code. The main concentration was on the heat losses from the gases to the manifold wall as they flow through the exhaust manifold. The model developed would be useful for the computation of thermal stresses that result in the exhaust manifold walls due to transient wall temperatures. The conclusions for the two particular engines studied are as follows:

1. The empirical correlations for the heat transfer in the exhaust manifold of an internal combustion engine result in a predicted heat loss of 4 to 5% of the engine energy input. If the predictions are correct, it would indicate that cylinder heat transfer has been overestimated by past researchers of the engine studied.

2. The predicted potential benefit from insulating the exhaust manifold of a turbocompounded engine is up to 0.7 percentage points gain in engine efficiency. Almost all of the gain came from an increase in the power turbine output.

3. Insulation in the exhaust manifold can result in lower cylinder temperatures by up to 30K for a constant fuel rate.

4. A substantial difference was observed between the time average and the mass average gas temperature predictions in the exhaust ports and runners. The temperature disagreement decreases as a function of distance along the exhaust flow path and may help explain some of
the problems of measuring exhaust port temperatures in operating internal combustion engines.

5. Comparison of test results from specially instrumented turbocharged engine showed that the measured gas exhaust gas temperatures increased downstream of the junction combining the exhaust flow from more than one cylinder and agreed with the mass average temperature predicted by the simulation.

6. Thermocouple temperatures were predicted using the simulation code. The predicted temperatures in the port agreed well with the measured temperatures. The agreement between the predicted thermocouple temperatures downstream of the port and the measured values was not as good.

Other areas be investigated are the benefits of reduced cooling load due to exhaust manifold insulation and optimization of engine performance following reductions in heat transfer. The current study did not attempt to optimize injection timing, fuel burn rate, or the turbomachinery with reductions in heat transfer.
REFERENCES


TABLE 6.1

CUMMINS 6 CYLINDER NH
ENGINE SPECIFICATIONS

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
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</thead>
<tbody>
<tr>
<td>Bore</td>
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</tr>
<tr>
<td>Stroke</td>
<td>15.2 cm (6.0 in)</td>
</tr>
<tr>
<td>Connecting Rod Length</td>
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</tr>
<tr>
<td>Compression Ratio</td>
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</tr>
<tr>
<td>Engine Displacement</td>
<td>14.0 l</td>
</tr>
<tr>
<td>Intake Manifold Volume</td>
<td>5.5 l</td>
</tr>
<tr>
<td>Exhaust Manifold Volume</td>
<td>5.0 l</td>
</tr>
<tr>
<td>Injection Timing</td>
<td>14 deg BTDC</td>
</tr>
<tr>
<td>Valve Timings:</td>
<td></td>
</tr>
<tr>
<td>Intake valve opens</td>
<td>11 deg BTDC</td>
</tr>
<tr>
<td>Intake valve closes</td>
<td>32 deg ABDC</td>
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<tr>
<td>Exhaust valve opens</td>
<td>35 deg BBDC</td>
</tr>
<tr>
<td>Exhaust valve closes</td>
<td>16 deg ATDC</td>
</tr>
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</table>
### TABLE 6.2
INITIAL CALIBRATION OF CUMMINS 6 CYLINDER TURBO-COMPOUNDED ENGINE
PEAK TORQUE - 1900 & 1600 RPM

<table>
<thead>
<tr>
<th>Engine Speed</th>
<th>1900 RPM</th>
<th>1600 RPM</th>
</tr>
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<tbody>
<tr>
<td>Fuel Rate</td>
<td>64.5 kg/hr</td>
<td>56.7 kg/hr</td>
</tr>
</tbody>
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<table>
<thead>
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<th>TEST BLOCK(^1)</th>
<th>SIM.</th>
<th>TEST BLOCK(^1)</th>
<th>SIM.</th>
</tr>
</thead>
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<td><strong>Engine Breathing</strong></td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Intake Manifold Temp. (K)</td>
<td>318.</td>
<td>324.</td>
<td>316.</td>
<td>317.</td>
</tr>
<tr>
<td>Intake Manifold Press (atm)</td>
<td>2.44</td>
<td>2.47</td>
<td>2.39</td>
<td>2.46</td>
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<tr>
<td>Intake Air Flow (kg/s)</td>
<td>0.546</td>
<td>0.538</td>
<td>0.458</td>
<td>0.463</td>
</tr>
<tr>
<td>Int. Volumetric Eff. (%)</td>
<td>90.6</td>
<td>91.0</td>
<td>91.8</td>
<td>91.8</td>
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<tr>
<td><strong>Exhaust Manifold Heat Transfer</strong></td>
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<td></td>
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</tr>
<tr>
<td>Mean Exhaust Temp. (K)</td>
<td>830.</td>
<td>896.</td>
<td>803.</td>
<td>879.</td>
</tr>
<tr>
<td>Turbine Inlet Temp.(K)</td>
<td>864.</td>
<td>834.</td>
<td>841.</td>
<td>816.</td>
</tr>
<tr>
<td>Temperature Drop (K)</td>
<td>-34.</td>
<td>62.</td>
<td>-38.</td>
<td>63.</td>
</tr>
<tr>
<td>Port Wall Temp.(^2) (K)</td>
<td>n/a</td>
<td>406.</td>
<td>n/a</td>
<td>401.</td>
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<td>Runner Wall Temp.(^2) (K)</td>
<td>n/a</td>
<td>748.</td>
<td>n/a</td>
<td>726.</td>
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<td>Heat Transfer to Walls(^2) (% of heat input)</td>
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<td>5.0</td>
<td>n/a</td>
<td>4.9</td>
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<td></td>
<td></td>
</tr>
<tr>
<td><strong>Turbomachinery</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>Boost Pressure Ratio</td>
<td>2.66</td>
<td>2.68</td>
<td>2.54</td>
<td>2.62</td>
</tr>
<tr>
<td>T/C Speed (KRPM)</td>
<td>64.8</td>
<td>65.1</td>
<td>60.5</td>
<td>61.3</td>
</tr>
<tr>
<td>Exhaust Manifold Press (atm)</td>
<td>3.05</td>
<td>3.38</td>
<td>2.58</td>
<td>2.90</td>
</tr>
<tr>
<td>T/C Turbine Pressure Ratio</td>
<td>1.96</td>
<td>2.00</td>
<td>1.84</td>
<td>1.91</td>
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<tr>
<td>Turbine Exh. Temp. (K)</td>
<td>739.</td>
<td>728.</td>
<td>730.</td>
<td>720.</td>
</tr>
<tr>
<td>P/Turbine Exh. Temp.(^2) (K)</td>
<td>n/a</td>
<td>665.</td>
<td>n/a</td>
<td>668.</td>
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<tr>
<td>T/C Combined Eff. (%)</td>
<td>0.62</td>
<td>0.62</td>
<td>0.65</td>
<td>0.65</td>
</tr>
<tr>
<td>Power Turbine Eff.(^2) (%)</td>
<td>n/a</td>
<td>.79</td>
<td>n/a</td>
<td>.78</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td><strong>System</strong></td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heat Transfer to Recip. Walls(^2) (% of heat input)</td>
<td>n/a</td>
<td>10.3</td>
<td>n/a</td>
<td>10.7</td>
</tr>
<tr>
<td>Reciprocator Brake Power (hp)</td>
<td>407.</td>
<td>400.</td>
<td>381.</td>
<td>373.</td>
</tr>
<tr>
<td>P/Turbine Brake Power (hp)</td>
<td>46.</td>
<td>44.</td>
<td>30.</td>
<td>30.</td>
</tr>
<tr>
<td>Reciprocator BSFC (lb/hp/hr)</td>
<td>0.349</td>
<td>0.355</td>
<td>0.327</td>
<td>0.335</td>
</tr>
<tr>
<td>Compounded BSFC (lb/hp/hr)</td>
<td>0.313</td>
<td>0.320</td>
<td>0.303</td>
<td>0.310</td>
</tr>
</tbody>
</table>

\(^1\) Test data furnished by Cummins Engine Co.
\(^2\) n/a refers to data not available for this study
### TABLE 6.3

**INITIAL CALIBRATION OF CUMMINS 6 CYLINDER TURBO-COMPOUNDED ENGINE**

1600 RPM - 50% and 25% Peak Torque Fueling Rate

<table>
<thead>
<tr>
<th>FUEL RATE</th>
<th>29.4 kg/hr</th>
<th>14.0 kg/hr</th>
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<td>TEST</td>
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<tr>
<td></td>
<td>BLOCK¹</td>
<td>BLOCK¹</td>
</tr>
<tr>
<td>Engine Breathing</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Intake Manifold Temp. (K)</td>
<td>308.</td>
<td>311.</td>
</tr>
<tr>
<td>Intake Manifold Press (atm)</td>
<td>1.66</td>
<td>1.59</td>
</tr>
<tr>
<td>Intake Air Flow (kg/s)</td>
<td>0.322</td>
<td>0.306</td>
</tr>
<tr>
<td>Int. Volumetric Eff. (%)</td>
<td>90.7</td>
<td>91.5</td>
</tr>
<tr>
<td>Atm. Volumetric Eff. (%)</td>
<td>150.</td>
<td>146.</td>
</tr>
</tbody>
</table>

| Exhaust Manifold Heat Transfer |            |            |
| Mean Exhaust Temp. (K) | 672. | 745. | 553. | 602. |
| Turbine Inlet Temp. (K) | 694. | 689. | 572. | 562. |
| Temperature Drop (K) | -22. | 56. | -19. | 40. |
| Port Wall Temp. (K)² | n/a | 383. | n/a | 373. |
| Runner Wall Temp. (K)² | n/a | 598. | n/a | 485. |
| Heat Transfer to Walls² | n/a | 5.4 | n/a | 5.6 |
| (% of heat input) |            |            |

| Turbomachinery |            |            |
| Boost Pressure Ratio | 1.75 | 1.69 | 1.31 | 1.23 |
| T/C Speed (KRPM) | 43.4 | 44.6 | 33.7 | 31.1 |
| Exhaust Manifold Press (atm) | 1.90 | 2.00 | 1.58 | 1.57 |
| T/C Turbine Pressure Ratio | 1.58 | 1.53 | 1.35 | 1.33 |
| Turbine Exh. Temp. (K) | 633. | 638. | 542. | 537. |
| P/Turbine Exh. Temp.² (K) | n/a | 608. | n/a | 521. |
| T/C Combined Eff. (%) | 0.59 | 0.58 | 0.51 | 0.41 |
| Power Turbine Eff.² (%) | n/a | 0.65 | n/a | 0.44 |

| System |            |            |
| Heat Transfer to Recip. Walls² | n/a | 13.4 | n/a | 17.2 |
| (% of heat input) |            |            |
| Reciprocator Brake Power (hp) | 187. | 170. | 70. | 54. |
| P/Turbine Brake Power (hp) | 12. | 9. | 6. | 2. |
| Reciprocator BSFC (lb/hp/hr) | 0.346 | 0.381 | 0.442 | 0.568 |
| Compounded BSFC (lb/hp/hr) | 0.325 | 0.361 | 0.407 | 0.548 |

¹ Test data furnished by Cummins Engine Co.
² n/a refers to data not available for this study
TABLE 6.4

TIME AVERAGE VS. MASS AVERAGE GAS TEMPERATURES

<table>
<thead>
<tr>
<th></th>
<th>1900 RPM</th>
<th>1600 RPM</th>
<th>1400 RPM</th>
<th>50% LOAD</th>
<th>25% LOAD</th>
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<tbody>
<tr>
<td><strong>PEAK TORQUE</strong></td>
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<td></td>
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</tr>
<tr>
<td><strong>1600 RPM</strong></td>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>EXHAUST VALVE</strong></td>
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</tr>
<tr>
<td>Mass Average</td>
<td>896.</td>
<td>879.</td>
<td>886.</td>
<td>745.</td>
<td>602.</td>
</tr>
<tr>
<td>Time Average</td>
<td>853.</td>
<td>840.</td>
<td>851.</td>
<td>710.</td>
<td>575.</td>
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<tr>
<td><strong>PORT</strong></td>
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<td></td>
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<tr>
<td>Mass Average</td>
<td>803.</td>
<td>770.</td>
<td>762.</td>
<td>672.</td>
<td>569.</td>
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<tr>
<td>Time Average</td>
<td>843.</td>
<td>831.</td>
<td>842.</td>
<td>696.</td>
<td>562.</td>
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<tr>
<td><strong>RUNNER</strong></td>
<td></td>
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<td></td>
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</tr>
<tr>
<td>Mass Average</td>
<td>815.</td>
<td>794.</td>
<td>796.</td>
<td>675.</td>
<td>559.</td>
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<tr>
<td>Time Average</td>
<td>834.</td>
<td>818.</td>
<td>823.</td>
<td>689.</td>
<td>562.</td>
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<td><strong>PLENUM</strong></td>
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<tr>
<td>Mass Average</td>
<td>834.</td>
<td>817.</td>
<td>823.</td>
<td>691.</td>
<td>562.</td>
</tr>
<tr>
<td>Time Average</td>
<td>834.</td>
<td>818.</td>
<td>823.</td>
<td>689.</td>
<td>562.</td>
</tr>
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</table>
**TABLE 7.1**

**CUMMINS 8 CYLINDER "903" ENGINE SPECIFICATIONS**

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
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<td>Bore</td>
<td>13.97 cm (5.5 in)</td>
</tr>
<tr>
<td>Stroke</td>
<td>12.06 cm (4.75 in)</td>
</tr>
<tr>
<td>Connecting Rod Length</td>
<td>20.81 cm (8.19 in)</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>13.5</td>
</tr>
<tr>
<td>Engine Displacement</td>
<td>14.8 l (902.4 in³)</td>
</tr>
<tr>
<td>Intake Manifold Volume</td>
<td>9.4 l</td>
</tr>
<tr>
<td>Exhaust Manifold Volume</td>
<td>13.0 l</td>
</tr>
<tr>
<td>Injection Timing</td>
<td>342 BTDC</td>
</tr>
<tr>
<td>Valve Timings:</td>
<td></td>
</tr>
<tr>
<td>Intake valve opens</td>
<td>10 deg BTDC</td>
</tr>
<tr>
<td>Intake valve closes</td>
<td>40 deg ABDC</td>
</tr>
<tr>
<td>Exhaust valve opens</td>
<td>44 deg BBDC</td>
</tr>
<tr>
<td>Exhaust valve closes</td>
<td>11 deg ATDC</td>
</tr>
<tr>
<td>Firing order</td>
<td>1-5-4-8-6-3-7-2</td>
</tr>
</tbody>
</table>
### Table 7.2

**CALIBRATION OF CUMMINS 8 CYLINDER TURBO-CHARGED ENGINE**

**PEAK TORQUE - 2200 & 2600 RPM**

<table>
<thead>
<tr>
<th>Engine Speed</th>
<th>2200 RPM</th>
<th>2600 RPM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel Rate</td>
<td>76.5 kg/hr</td>
<td>90.9 kg/hr</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Engine Breathing</th>
<th>TEST BLOCK</th>
<th>SIM.</th>
<th>TEST BLOCK</th>
<th>SIM.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake Manifold Temp. (K)</td>
<td>333.</td>
<td>330.</td>
<td>333.</td>
<td>335.</td>
</tr>
<tr>
<td>Intake Manifold Press (atm)</td>
<td>1.94</td>
<td>1.96</td>
<td>2.38</td>
<td>2.40</td>
</tr>
<tr>
<td>Intake Air Flow (kg/s)</td>
<td>0.523</td>
<td>0.522</td>
<td>0.752</td>
<td>0.748</td>
</tr>
<tr>
<td>Int. Volumetric Eff. (%)</td>
<td>94.</td>
<td>93.8</td>
<td>93.</td>
<td>92.5</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Exhaust Manifold Heat Transfer</th>
<th>TEST SIM.</th>
<th>TEST SIM.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mean Exhaust Temp. (K)</td>
<td>n/a</td>
<td>984.</td>
</tr>
<tr>
<td>Thermocouple #1 - Port (K)</td>
<td>880.</td>
<td>938/829</td>
</tr>
<tr>
<td>T/C #2 - First Runner (K)</td>
<td>936.</td>
<td>922/843</td>
</tr>
<tr>
<td>T/C #3 - Second Runner (K)</td>
<td>917.</td>
<td>907/875</td>
</tr>
<tr>
<td>T/C #4 - Turbine Inlet (K)</td>
<td>899.</td>
<td>899/902</td>
</tr>
<tr>
<td>Temperature Drop (K)</td>
<td>n/a</td>
<td>85.</td>
</tr>
<tr>
<td>Port Wall Temp. (K)</td>
<td>n/a</td>
<td>394.</td>
</tr>
<tr>
<td>Runner Wall Temp. (K)</td>
<td>n/a</td>
<td>772.</td>
</tr>
<tr>
<td>Heat Transfer to Walls (%)</td>
<td>n/a</td>
<td>5.8</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Turbomachinery</th>
<th>TEST SIM.</th>
<th>TEST SIM.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boost Pressure Ratio</td>
<td>2.00</td>
<td>2.02</td>
</tr>
<tr>
<td>T/C Speed (KRPM)</td>
<td>48.4</td>
<td>51.0</td>
</tr>
<tr>
<td>T/C Turbine Pressure Ratio</td>
<td>1.42</td>
<td>1.46</td>
</tr>
<tr>
<td>Turbine Exh. Temp. (K)</td>
<td>822.</td>
<td>837.</td>
</tr>
<tr>
<td>T/C Combined Eff. (%)</td>
<td>n/a</td>
<td>.69</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>System</th>
<th>TEST SIM.</th>
<th>TEST SIM.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Transfer to Recip. Walls (%)</td>
<td>n/a</td>
<td>9.9</td>
</tr>
<tr>
<td>Reciprocator Brake Power (hp)</td>
<td>507.</td>
<td>506.</td>
</tr>
<tr>
<td>BSFC (lb/hp/hr)</td>
<td>0.332</td>
<td>0.334</td>
</tr>
</tbody>
</table>

---

1. Test data furnished by Cummins Engine Co.
2. n/a refers to data not available for this study
3. Simulation predicted temperatures are given as mass averaged/time averaged
### TABLE 8.1

**1900 RPM PEAK TORQUE**

NORMAL CYLINDER PEAK TORQUE TRANSFER
THREE LEVELS OF EXHAUST HEAT TRANSFER

<table>
<thead>
<tr>
<th></th>
<th>NORMAL HEAT TRANS</th>
<th>PARTIAL INSULATION</th>
<th>ADIABATIC</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Reciprocator</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heat Transfer to Walls (% of heat input)</td>
<td>10.3</td>
<td>10.2</td>
<td>10.3</td>
</tr>
<tr>
<td>Mean Exhaust Temp. (K)</td>
<td>896.</td>
<td>879.</td>
<td>871.</td>
</tr>
<tr>
<td>Intake Air Flow (kg/s)</td>
<td>0.538</td>
<td>0.558</td>
<td>0.568</td>
</tr>
<tr>
<td>Intake Volumetric Eff. (%)</td>
<td>91.0</td>
<td>91.1</td>
<td>91.1</td>
</tr>
<tr>
<td>Atm. Volumetric Eff. (%)</td>
<td>213.</td>
<td>221.</td>
<td>225.</td>
</tr>
<tr>
<td>Peak Cylinder Pressure (atm)</td>
<td>124.</td>
<td>128.</td>
<td>131.</td>
</tr>
<tr>
<td>Peak Cylinder Temperature (K)</td>
<td>1660</td>
<td>1630</td>
<td>1620</td>
</tr>
<tr>
<td>Pumping MEP (atm)</td>
<td>-1.35</td>
<td>-1.39</td>
<td>-1.42</td>
</tr>
<tr>
<td>Gross MEP (atm)</td>
<td>16.2</td>
<td>16.3</td>
<td>16.3</td>
</tr>
<tr>
<td>Brake MEP (atm)</td>
<td>13.3</td>
<td>13.3</td>
<td>13.3</td>
</tr>
<tr>
<td>Brake Thermal Eff. (%)</td>
<td>38.9</td>
<td>39.0</td>
<td>39.0</td>
</tr>
<tr>
<td><strong>Exhaust Manifold</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heat Transfer to Walls (% of heat input)</td>
<td>5.0</td>
<td>1.7</td>
<td>0.0</td>
</tr>
<tr>
<td>Temperature Drop (K)</td>
<td>62.</td>
<td>20.</td>
<td>0.0</td>
</tr>
<tr>
<td>Port Wall Temp. (K)</td>
<td>406.</td>
<td>791.</td>
<td>--</td>
</tr>
<tr>
<td>Runner Wall Temp. (K)</td>
<td>748.</td>
<td>780.</td>
<td>--</td>
</tr>
<tr>
<td><strong>Turbomachinery</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>T/C Turbine Pressure Ratio</td>
<td>2.00</td>
<td>2.04</td>
<td>2.06</td>
</tr>
<tr>
<td>Boost Pressure Ratio</td>
<td>2.68</td>
<td>2.78</td>
<td>2.84</td>
</tr>
<tr>
<td>T/C Speed (K RPM)</td>
<td>65.1</td>
<td>67.0</td>
<td>68.2</td>
</tr>
<tr>
<td>Turbine Inlet Temp. (K)</td>
<td>834.</td>
<td>858.</td>
<td>868.</td>
</tr>
<tr>
<td>Turbine Exh. Temp. (K)</td>
<td>728.</td>
<td>746.</td>
<td>753.</td>
</tr>
<tr>
<td>T/C Combined Eff. (%)</td>
<td>0.62</td>
<td>0.62</td>
<td>0.62</td>
</tr>
<tr>
<td>Power Turbine Eff. (%)</td>
<td>0.79</td>
<td>0.79</td>
<td>0.79</td>
</tr>
<tr>
<td><strong>System</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Diesel Brake Power (hp)</td>
<td>400.</td>
<td>401.</td>
<td>401.</td>
</tr>
<tr>
<td>P/Turbine Brake Power (hp)</td>
<td>44.</td>
<td>48.</td>
<td>51.</td>
</tr>
<tr>
<td>Overall Brake Eff. (%)</td>
<td>43.2</td>
<td>43.6</td>
<td>43.9</td>
</tr>
</tbody>
</table>

---

1. 1900 rpm, 64.5 kg/hr Fuel Rate
2. 1.5 mm of Ceramic Material K=0.6 W/m/K
TABLE 8.2

1900 RPM PEAK TORQUE
INSULATED CYLINDER
THREE LEVELS OF EXHAUST HEAT TRANSFER

<table>
<thead>
<tr>
<th></th>
<th>NORMAL HEAT TRANS</th>
<th>PARTIAL INSULATION</th>
<th>ADIABATIC</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Reciprocator</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heat Transfer to Walls</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(% of heat input)</td>
<td>4.9</td>
<td>4.8</td>
<td>4.8</td>
</tr>
<tr>
<td>Mean Exhaust Temp. (K)</td>
<td>937.</td>
<td>918.</td>
<td>911.</td>
</tr>
<tr>
<td>Intake Air Flow (kg/s)</td>
<td>0.548</td>
<td>0.569</td>
<td>0.579</td>
</tr>
<tr>
<td>Int. Volumetric Eff. (%)</td>
<td>86.0</td>
<td>86.2</td>
<td>86.3</td>
</tr>
<tr>
<td>Peak Cylinder Pressure (atm)</td>
<td>131.</td>
<td>136.</td>
<td>138.</td>
</tr>
<tr>
<td>Peak Temperature (K)</td>
<td>1730</td>
<td>1710</td>
<td>1700</td>
</tr>
<tr>
<td>Pumping MEP (atm)</td>
<td>-1.24</td>
<td>-1.28</td>
<td>-1.32</td>
</tr>
<tr>
<td>Gross MEP (atm)</td>
<td>16.5</td>
<td>16.6</td>
<td>16.7</td>
</tr>
<tr>
<td>Brake MEP (atm)</td>
<td>13.7</td>
<td>13.7</td>
<td>13.7</td>
</tr>
<tr>
<td>Brake Thermal Eff. (%)</td>
<td>40.1</td>
<td>40.2</td>
<td>40.2</td>
</tr>
<tr>
<td><strong>Exhaust Manifold</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Heat Transfer to Walls</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(% of heat input)</td>
<td>5.4</td>
<td>1.8</td>
<td>0.0</td>
</tr>
<tr>
<td>Temperature Drop (K)</td>
<td>62.</td>
<td>19.</td>
<td>0.0</td>
</tr>
<tr>
<td>Port Wall Temp. (K)</td>
<td>412.</td>
<td>830.</td>
<td>--</td>
</tr>
<tr>
<td>Runner Wall Temp. (K)</td>
<td>786.</td>
<td>820.</td>
<td>--</td>
</tr>
<tr>
<td><strong>Turbomachinery</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>T/C Turbine Pressure Ratio</td>
<td>2.03</td>
<td>2.07</td>
<td>2.09</td>
</tr>
<tr>
<td>Boost Pressure Ratio</td>
<td>2.90</td>
<td>3.01</td>
<td>3.07</td>
</tr>
<tr>
<td>T/C Speed (KRPM)</td>
<td>67.3</td>
<td>69.3</td>
<td>70.5</td>
</tr>
<tr>
<td>Turbine Inlet Temp. (K)</td>
<td>874.</td>
<td>897.</td>
<td>909.</td>
</tr>
<tr>
<td>Turbine Exh. Temp. (K)</td>
<td>760.</td>
<td>778.</td>
<td>786.</td>
</tr>
<tr>
<td>P/Turbine Exh. Temp. (K)</td>
<td>693.</td>
<td>708.</td>
<td>720.</td>
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<tr>
<td>T/C Combined Eff. (%)</td>
<td>0.63</td>
<td>0.63</td>
<td>0.63</td>
</tr>
<tr>
<td>Power Turbine Eff. (%)</td>
<td>0.79</td>
<td>0.78</td>
<td>0.78</td>
</tr>
<tr>
<td><strong>System</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Diesel Brake Power (hp)</td>
<td>412.5</td>
<td>414.</td>
<td>414.</td>
</tr>
<tr>
<td>P/Turbine Brake Power (hp)</td>
<td>48.0</td>
<td>52.</td>
<td>55.</td>
</tr>
<tr>
<td>Overall Brake Eff. (%)</td>
<td>44.8</td>
<td>45.3</td>
<td>45.5</td>
</tr>
</tbody>
</table>

1 1900 RPM, 64.5 kg/hr Fuel Rate
2 5 mm of Ceramic Material on Piston and Head, K=0.6 W/m/K, approximately 50% decrease in cylinder heat transfer
3 1.5 mm of Ceramic Material, K=0.6 W/m/K
<table>
<thead>
<tr>
<th>STEADY-STATE vs TRANSIENT WALL TEMPERATURE</th>
<th>ENGINE PERFORMANCE PREDICTIONS</th>
</tr>
</thead>
<tbody>
<tr>
<td>1900 RPM - PEAK TORQUE</td>
<td>1900 rpm, 64.5 kg/hr Fuel Rate</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>NORMAL CYLINDER H.T.</th>
<th>PARTIALLY INSUL EXH$^2$</th>
<th>PARTIALLY INSULATED CYLINDER$^3$ AND EXHAUST$^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>STEADY-STATE</td>
<td>TRANSIENT</td>
<td>STEADY-STATE</td>
</tr>
<tr>
<td>Reciprocator Heat Transfer to Walls (% of heat input)</td>
<td>10.3</td>
<td>10.2</td>
<td>4.8</td>
</tr>
<tr>
<td>Mean Exhaust Temp. (K)</td>
<td>880.</td>
<td>879.</td>
<td>922.</td>
</tr>
<tr>
<td>Intake Air Flow (kg/s)</td>
<td>0.556</td>
<td>0.558</td>
<td>0.566</td>
</tr>
<tr>
<td>Intake Volumetric Eff. (%)</td>
<td>91.0</td>
<td>91.1</td>
<td>85.6</td>
</tr>
<tr>
<td>Atm. Volumetric Eff. (%)</td>
<td>220.</td>
<td>221.</td>
<td>224.</td>
</tr>
<tr>
<td>Pumping MEP (atm)</td>
<td>-1.39</td>
<td>-1.39</td>
<td>-1.26</td>
</tr>
<tr>
<td>Gross MEP (atm)</td>
<td>16.3</td>
<td>16.3</td>
<td>16.6</td>
</tr>
<tr>
<td>Brake MEP (atm)</td>
<td>13.3</td>
<td>13.3</td>
<td>13.7</td>
</tr>
<tr>
<td>Brake Thermal Eff. (%)</td>
<td>38.9</td>
<td>39.0</td>
<td>40.1</td>
</tr>
<tr>
<td>Peak to Peak Temp. swing (K)</td>
<td>0.</td>
<td>14.</td>
<td>0.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Exhaust Manifold</th>
<th>Piston and cylinder head</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Transfer to Walls (% of heat input)</td>
<td>1.8</td>
</tr>
<tr>
<td>Temperature Drop (K)</td>
<td>23.</td>
</tr>
<tr>
<td>Port Wall Temp. (K)</td>
<td>789.</td>
</tr>
<tr>
<td>Runner Wall Temp. (K)</td>
<td>778.</td>
</tr>
<tr>
<td>Peak to Peak Port Temp. swing(K)</td>
<td>0.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Turbomachinery T/C Turbine Pressure Ratio</th>
<th>2.04</th>
<th>2.04</th>
<th>2.06</th>
<th>2.07</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boost Pressure Ratio</td>
<td>2.78</td>
<td>2.78</td>
<td>3.01</td>
<td>3.01</td>
</tr>
<tr>
<td>T/C Speed (KRPM)</td>
<td>67.0</td>
<td>67.0</td>
<td>69.2</td>
<td>69.3</td>
</tr>
<tr>
<td>Turbine Inlet Temp. (K)</td>
<td>857.</td>
<td>858.</td>
<td>902.</td>
<td>897.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>System</th>
<th>401.</th>
<th>401.</th>
<th>413.</th>
<th>414.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel Brake Power (hp)</td>
<td>48.</td>
<td>48.</td>
<td>52.</td>
<td>52.</td>
</tr>
<tr>
<td>P/Turbine Brake Power (hp)</td>
<td>43.6</td>
<td>43.6</td>
<td>45.2</td>
<td>45.3</td>
</tr>
<tr>
<td>Overall Brake Eff. (%)</td>
<td>43.6</td>
<td>43.6</td>
<td>45.2</td>
<td>45.3</td>
</tr>
</tbody>
</table>

$^1$1900 rpm, 64.5 kg/hr Fuel Rate
$^2$1.5 mm of Ceramic Material K=0.6 W/m/K, α=5.45x10$^{-7}$ m$^2$/s
$^3$5 mm of Ceramic Material K=0.6 W/m/K, α=5.45x10$^{-7}$ m$^2$/s
Figure 1.1  Schematic of turbocompounded engine.
Simplest form of exhaust manifold model.

Figure 2.1
Figure 2.2  Typical head arrangement with two exhaust valves per cylinder.
Figure 3.1 Entrance effects on heat transfer in turbulent pipe flow.

Boelter, Young, and Iversen [13]
Figure 6.1  Plot of manifold pressure and pressure drop over an engine cycle. 1600 RPM, 56.7 kg/hr.
Figure 6.2  Plot of mass flow between exhaust manifold control volumes over an engine cycle. 1600 RPM, 56.7 kg/hr.
Figure 6.3  Plot of exhaust manifold control volume gas temperatures over an engine cycle. 1600 RPM, 56.7 kg/hr.
Figure 6.4  Plot of plenum and average manifold gas temperatures over an engine cycle.  1600 RPM, 56.7 kg/hr.
Figure 6.5  Plot of manifold heat transfer coefficients over an engine cycle. 1600 RPM, 56.7 kg/hr.
Figure 6.6  Plot of manifold heat transfer rates over an engine cycle.
1600 RPM, 56.7 kg/hr.
Figure 6.7  Mass average and time average gas temperatures along the exhaust flow path in the manifold. 1600 RPM, 56.7 kg/hr.
Figure 6.8  Distribution of heat losses in the exhaust manifold.
1900 RPM, 64.5 kg/hr.
Figure 6.9  Distribution of energy in a turbocompounded engine from the intake valve to the turbine inlet.
1900 RPM, 64.5 kg/hr.
Figure 7.1 Schematic of Cummins 8 cylinder turbocharged 903 engine.
Figure 7.2 Model configuration used for 8 cylinder engine simulation.
Figure 7.3 Measured port gas temperatures for individual cylinders.
Figure 7.4 Predicted mass average and time average gas temperatures and measured temperatures along the length of the manifold. 2600 RPM, 90.9 kg/hr fuel rate.
Figure 7.5  Predicted mass average and time average gas temperatures and measured temperatures along the length of the manifold 2200 RPM, 76.5 kg/hr fuel rate.
CYL 2 & 4 EXHAUST BRANCH TEMPERATURES

V903-600 @ 2600, HT5C TURBO
AC COUPLED AC SIGNAL

.030" WIRE DIAM THERMOCOUPLE

1. #4 EXHAUST PORT
2. DOWNSTRM OF #2 PORT
3. BEFORE PULSE CONVERTER
4. TURBINE INLET (CASING)

Figure 7.6 Reproduction of Cummins transient temperature measurements in the exhaust manifold at 2600 RPM and rated load.
Figure 7.7  Thermocouple predicted temperatures along the length of the manifold.  2600 RPM, 90.9 kg/hr fuel rate.
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A COMPUTER MODEL FOR THE HEAT LOSSES
IN THE EXHAUST MANIFOLD OF AN
INTERNAL COMBUSTION ENGINE

by

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B.Sc. Mechanical Engineering
Cornell University
(1974)

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Massachusetts Institute of Technology
JUL 28 1986

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APPENDIX A

PROGRAMING INFORMATION
SUBROUTINE HIERARCHY

FLOW CHART SHOWING ORDER OF CALLS TO SUBROUTINES DURING SIMULATION.

- PROGRAM RUN SET-UP ---

- START OF ITERATION ---

- INTAKE PROCESS: IVO - IVC ---

- COMPRESSION PROCESS ***: IVC - TINJ ---

- IGNITION DELAY PERIOD: TINJ - TIGN ---
SUBROUTINE HIERARCHY (CONTINUED)

--- COMBUSTION, EXPANSION PROCESS: TIGN - EVO ---

SIM → STEPCA → ODERT → DERT1 → CMBSTN → CSAVDV
    → VLVSUM (I.V.) → STEP1 → GCMF → FLAME → INTRP → DIFEQ
    → VLVSUM (E.V.) → EXSUM * → RESULT → DATA → PFDIF **

--- EXHAUST PROCESS: EVO - IVO ---

→ STEPCA → ODERT → DERT1 → EXAUST → VACDEX
    → VLVSUM (I.V.) → STEP1 → GCMF → MFLRT → CSAVDV → DIFEQ
    → VLVSUM (E.V.) → EXSUM * → RESULT → DATA → PFDIF **

→ QEND

--- END OF ITERATION ---

→ EXWRT

OTHER GENERAL PURPOSE SUBROUTINES CALLED FROM VARIOUS LEVELS:

- THERMO
- HPROP
- CPROP
- TRANSP
- DELH
- ITRATE

FUNCTIONS:

- EQR
- FAHR

- ONLY CALLED FOR EXHAUST MANIFOLD SUB-MODEL
- ** ONLY CALLED FOR TRANSIENT WALL TEMPERATURE CALCULATION
- *** IF THE IGNITION DELAY PERIOD IS SPECIFIED, THE COMPRESSION PROCESS IS FROM IVC TO TIGN AND THE LOOP FOR THE IGNITION DELAY PERIOD IS BYPASSED
SCHEMATIC OF VARIABLES FOR EXHAUST MANIFOLD FOR 6 CYLINDER CONFIGURATION WITH PORT, RUNNER, AND PLENUM

VALUES OF CONFIGURATION INPUT PARAMETERS:

NUMEX = 3
NPIPE = 1, 1, 6
NSHFT = 0, 0, 120

DEFINITION OF VARIABLE NAMES:

EMFLO(I) = MASS FLOW BETWEEN CONTROL VOLUMES
EHFLO(I) = SPECIFIC ENTHALPY OF FLOW BETWEEN C.V.s
EFFLO(I) = FUEL FRACTION OF FLOW BETWEEN C.V.s

MASSEX(J) = STORAGE ARRAY FOR EXHAUST VALVE MASS FLOW
HCYLEX(J) = STORAGE ARRAY FOR EXH. VALVE FLOW SPEC ENTHALPY
FCYLEX(J) = STORAGE ARRAY FOR EXH. VALVE FLOW FUEL FRACTION

ENGM(2) = TOTAL MASS FLOW THROUGH EXHAUST VALVES
HMDOT(2) = TOTAL ENTHALPY FLOW THROUGH EXHAUST VALVES
FMDOT(2) = TOTAL FUEL FRACTION FLOW THROUGH EXHAUST VALVES

XMASS(1,J) = STORAGE ARRAY FOR MASS FLOW BETWEEN C.V.s
XH(1,J) = STORAGE ARRAY FOR SPECIFIC ENTHALPY OF FLOW BETWEEN C.V.s
XF(1,J) = STORAGE ARRAY FOR FUEL FRACTION OF FLOW BETWEEN C.V.s

ENGMX(4) = TOTAL MASS FLOW FROM OTHER 5 CYLINDERS
HMDOTX(4) = TOTAL ENTHALPY FLOW FROM OTHER 5 RUNNERS
FMDOTX(4) = TOTAL FUEL FRACTION FLOW FROM OTHER 5 RUNNERS

Q(I) = HEAT TRANSFER FROM GAS TO WALL FOR EACH CONTROL VOLUME

TRBM = FLOW TO TURBINE
SCHEMATIC OF VARIABLES FOR EXHAUST MANIFOLD FOR 8 CYLINDER CONFIGURATION WITH PORT, FIRST AND SECOND RUNNERS, AND PLENUM

VALUES OF CONFIGURATION INPUT PARAMETERS:
NUMEX = 4
NPIPE = 1, 1, 2, 4
NSHFT = 0, 0, 270, 180

DEFINITION OF VARIABLE NAMES:
EMFLO(I) = MASS FLOW BETWEEN CONTROL VOLUMES
EHFLO(I) = SPECIFIC ENTHALPY OF FLOW BETWEEN C.V.s
EFFLO(I) = FUEL FRACTION OF FLOW BETWEEN C.V.s
MASSEX(J) = STORAGE ARRAY FOR EXHAUST VALVE MASS FLOW
HCYLEX(J) = STORAGE ARRAY FOR EXH. VALVE FLOW SPEC ENTHALPY
FCYLEX(J) = STORAGE ARRAY FOR EXH. VALVE FLOW FUEL FRACTION
ENGM(2) = TOTAL MASS FLOW THROUGH EXHAUST VALVES
HMDOT(2) = TOTAL ENTHALPY FLOW THROUGH EXHAUST VALVES
FMDOT(2) = TOTAL FUEL FRACTION FLOW THROUGH EXHAUST VALVES
XMASS(K,J) = STORAGE ARRAY FOR MASS FLOW BETWEEN C.V.s
XH(K,J) = STORAGE ARRAY FOR SPECIFIC ENTHALPY OF FLOW BETWEEN C.V.s
XF(K,J) = STORAGE ARRAY FOR FUEL FRACTION OF FLOW BETWEEN C.V.s
ENGMX(4) = TOTAL MASS FLOW FROM OTHER 5 CYLINDERS
HMDOTX(4) = TOTAL ENTHALPY FLOW FROM OTHER 5 RUNNERS
FMDOTX(4) = TOTAL FUEL FRACTION FLOW FROM OTHER 5 RUNNERS
Q(I) = HEAT TRANSFER FROM GAS TO WALL FOR EACH CONTROL VOLUME
TRBM = FLOW TO TURBINE
APPENDIX B

SUMMARY OF COMPUTER CODE CHANGES
SUMMARY OF COMPUTER CODE CHANGES
FOR THE PRESENT STUDY

General:

The program was rearranged to move the call statements for the subroutines that were called for each time step from the main program to a single subroutine. This change simplified the main program and facilitated making changes that would otherwise have been done in each of the 5 process loops in the main program.

Subroutine ENGPAR was eliminated and provisions were made to include all of the engine geometry parameters in the input list. Formerly the stroke, control rod length and the cylinder heat transfer surface areas were calculated in ENGPAR as functions of the bore.

The input of the valve profiles was changed to accept a flexible number of coordinates in the input files. The reading of the input valve and the exhaust valve areas continues until the end of the respective file is reached and the number of points input is adjusted accordingly. Currently, the program is set up to accept up to 120 points to define each valve opening profile. A flag was included to warn the user if this value is exceeded.

The storage of the mass flow, specific enthalpy, and fuel fraction of the flow with respect to the master cylinder was changed to be referenced to the number of crank angle degrees after the intake valve or the exhaust valve opens. This change allows the program to accept variable valve timings without recompiling the program.

The number of equations to be integrated is defined as a parameter at the beginning of the main program. This allows the number of equations being integrated to be changed in one place with the necessary information passed to the various subroutines. Formerly, a change in the total number of variables in the integration required dimension statements changes in numerous subroutines.
The program was modified to only accept time steps of one crank angle at the main program level. Provisions were made to specify a longer interval for the program output.

Flags were included in the turbomachinery subroutines ICMAP and IPTMAP to warn the user when the program solution was going off of the turbomachinery maps input.

Changes to Specific Subroutines:

DIFEQ: DIFEQ was modified to include the calculation for the exhaust manifold sub-control volume derivatives for pressure, temperature, and fuel fraction. It was also changed to include the extra variables for the cycle averages for the exhaust manifold control volumes.

QDP: QDP was split into two subroutines. QMAN calculates the manifold heat transfer for the intake manifold and in the case of the single plenum for the exhaust manifold and the turbine connecting pipe. DPMAN calculates the pressure drop in a manifold section.

INTAKE, EXAUST: These subroutines were modified to calculate the jet Reynolds number for the opening and closing phases of the exhaust port heat transfer.

ODERT, DERT1, STEP1: The integration subroutines were modified to pass the number of equations (NEQN) down to the process subroutines in the call statements to allow flexible dimensioning of the state variable arrays.

DERT1: DERT1 was also modified to eliminate the call to subroutine ERRCHK. This was done to eliminate compatibility problems with an IBM computer. Subroutine ERRCHK is no longer used.

VACDEX: The subroutine for the interpolation of the exhaust valve effective flow area table was modified to assign the proper value for the phase of the valve opening for the port heat transfer correlation.
New Subroutines for the Exhaust Manifold Model:

MANPAR: This subroutine was written to read in the exhaust manifold input data and to reduce this data down to constants used in running the program. The data is also transferred into global arrays for the transient wall temperature calculations.

EXWRITE: EXWRITE was written to output the exhaust manifold input data and the results for the exhaust manifold at the end of a simulation.

HTRATE: This subroutine was written to calculate the manifold heat transfer rate at each time step. It supercedes QMAN when the exhaust manifold sub-control volume option is in effect.

EXSUM: This subroutine is the counterpart to VLVSUM but for the exhaust manifold control volumes. It records the mass flow, specific enthalpy, and the fuel fraction of the flow between the master exhaust manifold control volumes and retrieves this information to determine what the flows are from the phantom control volumes at each time step. Unlike VLVSUM it only includes the flow from the other cylinders of the engine in the flow values that it returns excluding the master cylinder.
APPENDIX C

LISTING OF PROGRAM
A COMPUTER SIMULATION OF THE TURBO-CHARGED
TURBO-COMPOUNDED DIESEL ENGINE SYSTEM:
A PERFORMANCE PREDICTIVE MODEL

BY

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VERSION 4.0
JANUARY 1986

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CAMBRIDGE, MASS 02139
PURPOSE

THIS PROGRAM IS A ZERO-DIMENSIONAL SIMULATION OF THE
OPERATING CYCLE OF A TURBOCHARGED TURBOCOMPONDED DIESEL.
THE PROGRAM CALCULATES THE PERFORMANCE OF THE RECIPROCATOR
AND THE OTHER SYSTEM COMPONENTS, AS A FUNCTION OF OPERATING
CONDITIONS AND ENGINE DESIGN.
THE PROGRAM CAN BE USED IN THE FOLLOWING OPERATING MODES:
I) TURBOCHARGED TURBOCOMPONDED OR TURBOCHARGED ONLY.
II) SPECIFIED COMPONENT WALL SURFACE TEMPERATURES, OR
PREDICTED WALL TEMPERATURES FOR A GIVEN WALL STRUCTURE.
III) STEADY-STATE OR STEADY-STATE PLUS CYCLIC-TRANSIENT WALL
CONDUCTION MODELS CAN BE USED.
IV) RADIATION CAN BE MODELED BASED ON FLAME TEMPERATURE, OR
ON BULK GAS TEMPERATURE (ANNAND'S MODEL)
V) SPECIFIED IGNITION DELAY OR PREDICTED IGNITION DELAY.
VI) EXHAUST SUB-CONTROL VOLUME OR SINGLE FILLING AND EMPTYING
PLENUM MODEL.

DESCRIPTION OF PARAMETERS:
PARAMETER INPUT OUTPUT DESCRIPTION
I OPERATING PARAMETERS
ERPM YES NO ENGINE RPM
INJ YES NO INJECTION TIMING (DEG)
TIGN YES NO IGNITION TIMING (SPDEL=.FALSE. ONLY)
FMIN YES NO FUEL MASS INJECTED PER CYL PER CYCLE (G)
FR NO YES FUELING RATE (LB/MIN)
PATM YES NO ATMOSPHERIC PRESSURE (ATM)
TATM YES NO ATMOSPHERIC TEMPERATURE (K)

II ENGINE DESIGN PARAMETERS
ICYL YES NO # OF ENGINE CYLINDERS
BORE YES NO ENGINE BORE (CM)
STROKE YES NO ENGINE STROKE (CM)
CONRL YES NO CONNECTING ROD LENGTH (CM)
AHEAD YES NO HEAD SURFACE AREA (CM^2)
-138-

121 C APSTON YES NO PISTON SURF AREA (CM^2)
122 C CLVTD C YES NO CYLINDER VOLUME AT TDC (CM^2)
123 C DVOLUM NO YES TOTAL SWPT VOLUME OF A CYLINDER (M^3)
124 C DVOL NO YES TOTAL SWPT VOLUME OF ENGINE (M^3)
125 C TIVO YES NO INTAKE VALVE OPENS (DEG)
126 C TVC YES NO INTAKE VALVE CLOSES (DEG)
127 C TEVO YES NO EXHAUST VALVE OPENS (DEG)
128 C TEVC YES NO EXHAUST VALVE CLOSES (DEG)
129 C FUELT P YES NO FUEL TYPE - SEE SBRTN FUELDT FOR SPECS.
130 C CFR2 YES NO COEFF FOR LINEAR TERM IN MECH FRIC CALC
131 C CFR3 YES NO COEFF FOR QUADRATIC TERM IN MECH FRIC CALC
132 C CFCTR YES NO DIFFN BURNING CONSTANT - SEE SBTN CMSTN
133 C TIVM YES NO NOMINAL BURNING DURATION (CA) - SEE CMSTN

3 LOGICAL SWITCHES FOR SIMULATION OPTIONS SPECIFICATION

136 C POWER YES NO TRUE IF ENGINE IS TURBOCOMPOUNDED
138 C SPTEMP YES NO TRUE FOR SPECIFIED WALL TEMPERATURES
139 C TRANS YES NO TRUE FOR TRANSIENT WALL TEMPERATURE CALC.
140 C SPTEMP MUST BE FALSE FOR TRANSIENT CALC.
141 C ANNAND YES NO TRUE FOR ANNAND APPROXIMATION FOR CYLINDER
142 C RADIATION HEAT TRANSFER. FALSE FOR RADIATION
143 C HEAT TRANSFER BASED ON ADIABATIC FLAME TEM.
144 C SPDEL YES NO TRUE FOR SPECIFIED IGNITION DELAY, FALSE FOR
145 C CALCULATED IGNITION DELAY PERIOD.
146 C EXSUB YES NO TRUE FOR EXHAUST MANIFOLD SUB-CONTROL VOLUME
147 C MODEL. FALSE FOR SINGLE PLENUM.

4 MANIFOLD DESIGN PARAMETERS (FOR SINGLE EXHAUST MANIFOLD VOLUME)

150 C (J=1 FOR INTAKE, J=2 FOR EXHAUST, J=3 FOR CONNECTING PIPE)
152 C
153 C ELNG(J) YES NO LENGTH (M)
154 C EDIAM(J) YES YES EFFECTIVE DIAMETER (M)
155 C EAREA(J) NO YES INTERNAL SURFACE AREA OF MANIFOLD
156 C (M^2)
157 C ECROSS(J) NO YES CROSS-SECTIONAL AREA OF MANIFOLD
158 C (M^2)
159 C EVOLUME(J) NO YES VOLUME (M^3)
160 C

5 INTERCOOLER DESIGN

161 C
163 C HI(1) NO YES HEAT EXCHANGER EFFECTIVENESS
164 C HI(2) YES NO COOLANT INLET TEMPERATURE (K)
165 C HI(3) NO YES AIR TEMPERATURE AT
166 C INTERCOOLER OUTLET (K)
167 C HI(4) NO YES SPECIFIC ENTHALPY OF AIR
168 C AT T = HI(3) (J/kg/K)
169 C HI(5) YES NO HEAT TRANSFER COEFFICIENT x AREA (W/K)
170 C

6 TURBOMACHINERY DATA

172 C
174 C B(1) YES NO ROTATIONAL INERTIA OF TURBOCHARGER
175 C ROTOR
176 C B(2) YES NO ROTATIONAL DAMPING OF TURBOCHARGER
177 C ROTOR
178 C PTTEF YES NO POWER TURBINE TRANSMISSION EFFICIENCY
179 C CSC YES NO COMPRESSOR EFFICIENCY SCALE FACTOR
180 C TSC YES NO TURBINE EFFICIENCY SCALE FACTOR
7 SYSTEM PRESSURE LOSSES (PASCALS) & LOSS FACTORS

**SPECIFIED PRESSURE DROP:**
- **DP(1)** YES NO COMRESSOR EXIT - INTERCOOLER INLET
- **DP(2)** NO NO INTERCOOLER INLET - INTAKE MANIFOLD
- **DP(3)** NO NO EXHAUST MANIFOLD - TURBINE INLET
- **DP(4)** NO NO TURBINE EXIT - POWER TURBINE INLET
- **DP(5)** YES NO POWER TURBINE EXIT - ATMOSPHERIC

**LOSS FACTORS:**
- **EMKT(1)** YES NO INTERCOOLER INLET - INTAKE MANIFOLD
- **EMKT(2)** YES NO EXHAUST MANIFOLD - TURBINE INLET
- **EMKT(3)** YES NO TURBINE EXIT - POWER TURBINE INLET
- SEE SRBN DP MAN FOR USAGE

8 TURBULENCE SUB-MODEL CONSTANTS

- **CBETA** YES NO TURBULENT DISSIPATION CONSTANT

9 HEAT TRANSFER

**CONSTANTS:** 
- **NU** = **CONS’T (REYNOLDS NO.)** EXP’NT
- **CONHT** YES NO CONS’T (FOR CYLINDER)
- **ECONHT(1)** YES NO CONS’T FOR INTAKE MANIFOLD
- **ECONHT(2)** YES NO CONS’T FOR EXHAUST MANIFOLD
- **ECONHT(3)** YES NO CONS’T FOR TURBINE CONNECTING PIPE
- **EXPHT** YES NO EXP’NT
- **CRAD** YES NO RADIATION CONS’T (ANNAND=.TRUE.)

**TEMPERATURES:**
- **TPSTON** YES YES PISTON WALL TEMPERATURE (K) (INITIAL)
- **THEAD** YES YES HEAD WALL TEMPERATURE (K) (INITIAL)
- **TCW** YES YES CYLINDER LINER WALL TEMP (K) (INITIAL)
- **ETWALL(I)** YES YES MANIFOLD WALL TEMPS (K) (EXSUB=.FALSE.)

10 FUEL AND AIR SPECIFICATIONS

- **CX** YES NO NUMBER OF CARBON ATOMS IN THE FUEL (8.0 FOR C8H18)
- **DEL** YES NO MOLAR C:H RATIO OF THE FUEL
- **PSI** YES NO MOLAR N:O RATIO OF AIR
- **QLOWER** YES NO LOWER HEATING VALUE OF FUEL (MJ/KG)

11 INITIAL GUESSES AT THE START OF INTAKE PROCESS

- **PSTART** YES NO INITIAL PRESSURE IN CYLINDER (ATM)
- **TSTART** YES NO INITIAL TEMPERATURE IN CYLINDER (K)
### 12 Output Time Increments

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<thead>
<tr>
<th>Variable</th>
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<tr>
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<thead>
<tr>
<th>Variable</th>
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### Definition of System State Variables

(All are double precision)

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<th>Variable</th>
<th>Description</th>
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<tbody>
<tr>
<td>DT</td>
<td>Time (deg)</td>
</tr>
<tr>
<td>DY(1)</td>
<td>Mass induced into chamber through intake valve (kg)</td>
</tr>
<tr>
<td>DY(2)</td>
<td>Mass exhausted from chamber through exhaust valve (kg)</td>
</tr>
<tr>
<td>DY(3)</td>
<td></td>
</tr>
<tr>
<td>DY(4)</td>
<td>Fraction of fuel mass burned</td>
</tr>
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<tr>
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<tr>
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DY(54) TIME-AVERAGED EFFECTIVE LINEARIZED HEAT TRANSFER COEFF. FROM GAS TO PISTON
DY(55) TIME-AVERAGED PRODUCT OF GAS/PISTON HEAT TRANSFER COEFF. TIMES GAS TEMPERATURE
DY(56) TIME-AVERAGED EFFECTIVE LINEARIZED HEAT TRANSFER COEFF. FROM GAS TO CYL. HEAD
DY(57) TIME-AVERAGED PRODUCT OF GAS/CYL. HEAD HEAT TRANSFER COEFF. TIMES GAS TEMPERATURE
DY(58) TIME-AVERAGED EFFECTIVE LINEARIZED HEAT TRANSFER COEFF. FROM GAS TO CYL. LINER
DY(59) TIME-AVERAGED PRODUCT OF GAS/CYL. LINER HEAT TRANSFER COEFF. TIMES GAS TEMPERATURE
DY(60) TIME-AVERAGED EFFECTIVE LINEARIZED HEAT TRANSFER COEFF. FROM GAS TO CYL. LINER EXP.
DY(61) - SPARE
DY(62) - SPARE

C EXHAUST MANIFOLD SUB-MODEL VARIABLES:
DY(63) EXHAUST PORT MASS (KG)
DY(64) EXHAUST PORT TEMPERATURE (K)
DY(65) EXHAUST PORT FUEL FRACTION
DY(66) TIME AVERAGED PORT TEMPERATURE
DY(67) MASS AVERAGED PORT TEMPERATURE
DY(68) INTEGRATED HEAT TRANSFER FOR PORT
DY(69) TIME AVERAGED HEAT TRANS COEF FOR PORT
DY(70) TIME AVERAGED PRODUCT OF PORT HEAT TRANS COEF. TIMES GAS TEMPERATURE

ADD INTEGER MULTIPLES OF 8 TO GET INDICES FOR VARIABLES FOR SECTIONS DOWNSTREAM OF THE PORT.

SYSTEM TEMPERATURE DEFINITIONS
RTEMP(1) = COMPRESSOR INLET TEMPERATURE (INPUT)
RTEMP(2) = COMPRESSOR DISCHARGE TEMPERATURE (OUTPUT)
RTEMP(3) = TURBINE OUTLET TEMPERATURE (OUTPUT)
RTEMP(4) = POWER TURBINE INLET TEMPERATURE (OUTPUT)
RTEMP(5) = POWER TURBINE OUTLET TEMPERATURE (OUTPUT)

SYSTEM PRESSURE DEFINITIONS
PINLET = COMPRESSOR INLET PRESSURE (INPUT)
PRSS(1) = COMPRESSOR DISCHARGE PRESSURE (OUTPUT)
PRSS(2) = TURBINE INLET PRESSURE (OUTPUT)
PRSS(3) = TURBINE OUTLET PRESSURE (OUTPUT)
PRSS(4) = POWER TURBINE INLET PRESSURE (OUTPUT)
PRSS(5) = POWER TURBINE OUTLET PRESSURE (INPUT)

SYSTEM MASS FLOW RATE DEFINITIONS
RMSS(1) = AVERAGE ENGINE INTAKE MASS FLOW
RMSS(2) = NOT USED
RMSS(3) = CORRECTED COMPRESSOR MASS FLOW (LB/MIN)
RMSS(4) = CORRECTED TURBINE MASS FLOW (LB/MIN)
RMSS(5) = CORRECTED POWER TURBINE MASS FLOW (LB/MIN)

SYSTEM SPECIFIC ENTHALPY DEFINITIONS
H(1) = SPECIFIC ENTHALPY AT ATMOSPHERIC CONDITIONS
H(2) = SPECIFIC ENTHALPY OF ENGINE EXHAUST
H(3) = SPECIFIC ENTHALPY CHANGE ACROSS COMPRESSOR
H(4) = SPECIFIC ENTHALPY CHANGE ACROSS TURBINE
H(5) = SPECIFIC ENTHALPY CHANGE ACROSS POWER TURBINE

CONVERSION AND CORRECTION FACTORS

RCORR(1) = POWER TURBINE GEAR RATIO, (TURBINE RPM/ RCP RPM) (INPUT)
RCORR(2) = MASS FLOW CONVERSION FACTOR FROM LB/MIN TO KG/DEG
1 LB/MIN = 0.45359/60 KG/SEC = .00756 * ESPD KG/DEG
1/RCORR(2) CONVERTS FORM KG/DEG TO LB/MIN
RCORR(3) = TEMPERATURE CORRECTION FACTOR FOR COMPRESSOR MAP
RCORR(4) = TEMPERATURE CORRECTION FACTOR FOR TURBINE MAP
RCORR(5) = TEMPERATURE CORRECTION FACTOR FOR POWER TURBINE MAP

DISCRIPTION OF UNIT NUMBERS FOR INPUT AND OUTPUT FILES

UNIT-3, FILE = COMAP.DAT: COMPRESSOR MAP INPUT FILE
UNIT-4, FILE = TMAP.DAT : TURBINE MAP INPUT FILE
UNIT-5, FILE = PTMAP.DAT: POWER TURBINE MAP INPUT FILE

UNIT-6, FILE = OUT1.DAT OR OUT2.DAT : MAIN OUTPUT FILES
UNIT-7, FILE = TT: TERMINAL OUTPUT
UNIT-8, FILE = INPUT.DAT : MAIN INPUT FILE (SEE NAMELIST INPUT)
UNIT-9, FILE = EXHMAN.DAT: EXHAUST MANIFOLD INPUT
     (SEE MANPAR.FOR)
UNIT-10, FILE = CHEAT.DAT : CYLINDER LINER INPUT FILE
     (SEE CYLPAR.FOR)
UNIT-11, FILE = PHEAT.DAT : PISTON AND HEAD INPUT FILE
     (SEE PARFIN.FOR)

UNIT-12, FILE = FLOW.DAT : TURBULENT FLOW OUTPUT FILE
UNIT-13, FILE = HEAT.DAT : CYLINDER HEAT TRANSFER OUTPUT FILE
UNIT-14, FILE = RAD.DAT : CYLINDER RADIATION HEAT TRANS OUTPUT
UNIT-15, FILE = RMAS.DAT : CYCLE BY CYCLE OUTPUT OF MASS FLOWS

UNIT-36, FILE = FINP.DAT : PISTON TEMPERATURE OUTPUT
     (TRANSIENT CALCULATION ONLY)
UNIT-37, FILE = FINH.DAT : HEAD TEMP. OUTPUT (TRANS CALC ONLY)
UNIT-38, FILE = PORT.DAT : PORT TEMP. OUTPUT
     " " "
UNIT-39, FILE = RUNNER.DAT: RUNNER TEMP. OUTPUT
     " " "
UNIT-40, FILE = PLENUM.DAT: PLENUM TEMP. OUTPUT
     " " "

UNIT-75, FILE = TABLIN.DAT: INTAKE VALVE FLOW AREA TABLE INPUT
UNIT-76, FILE = TABLEX.DAT: EXHAUST VALVE FLOW AREA TABLE INPUT

 THESE FILES ARE CURRENTLY OPENED AS NECESSARY IN THE MAIN
PROGRAM AND IN SBRTN FILEDEF FOR A VAX-750

REMARKS

ALL CRANK ANGLE DATA ARE REFERENCED TO 0.0 DEG ● TDC
OF THE INTAKE STROKE (START OF THE INTAKE STROKE)
SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED:

1 WORKING SUBROUTINES

- STEPCA
- INTAKE
- CMPRES
- CMBSRN
- EXAUST
- GCMP
- GIDEL

2 SPECIAL UTILITY SUBROUTINES

- VACDIN
- VACDEX
- CSAVDV

3 GENERAL UTILITY SUBROUTINES

- PARFIN
- DIFEQ
- DELH
- TRANSP
- QMAN
- ODERT
- ERRCHK
- DATA
- CYLPAR
- PFDIF
- ICMAP
- THERMO
- MFLRT
- DPMAN
- DERTI
- INTRP
- EXWRIT
- FLAME
- IPTMAP
- CPROP
- QEND
- EXSUM
- ITRATE
- RESULT
- FAHR

Written by D. N. Assanis et al.
Edited by D. N. Assanis & R. M. Frank

Logical SPDEL, POWER, SPTEMP, ANNAND, TRANS, EXSUB

Title: 80

Integer SIZC, SIZT, SIZPT, SIZ1, SIZ2, SIZ3, FUELTP

Note that the value of parameter NEQN must be greater than or equal to 60+8*(NUMEXT). NUMEXT is defined in SBRTN MANPAR

Parameter (NEQN=102)

Note that the dimension of array WORK is based on NEQN, currently work is dimensioned for NEQN less than or equal to 110. For larger values of NEQN work should be redimensioned in subroutines STEPCA, and ODERT. See ODERT for required dimension for work.

Dimension WORK(2410), IWORK(5)

Real*8 DT, DY(NEQN), TOUT, RELERR, ABSERR, WORK, REROOT, AEROOT

Real MW, MMM, MSTART, MACRSC, MASS, MKESTA,
& MAXERR, MFINAL, MASSIN, MASEX, MASSX3, KIL, MIL

Integer IROW, EROW

Dimension TABLIN(120,2), TABLEX(120,2)

Parameter (PI=3.1415927, 10^0, 11^1, 12^2)

Parameter (SIZ=6, SIZT=6, SIZPT=6, SIZ1=8, SIZ2=8, SIZ3=11)

Parameter (CEN = 1.62, KIL = 1.58, MIL = 1.56, ERG = 1.57,
& ATPA = 1.913458E5, HGPA = 0.09534E-3, PSIPA = 6.84584E3,
& CSFC = 1.844E-3, HPKW = 0.7457)

Parameter (NCV=6)

Dimension Y0(NEQN-20), Y(NEQN).
& MASSIN(270), MASSEX(270),
& FCYLIN(270), FCYLEX(270),
& HCYLIN(270), HCYLEX(270)
C DIMENSION EVOLME(3), ELNG(3)
C
DIMENSION NCLA(6)
DIMENSION QAVT(2)
DIMENSION PTW(10,3,51)
C DIMENSION TNEW(10)
C
COMMON/ITERAS/ ITERAS, ISTEDY
COMMON/PRINT/ TPRINT, TSCREEN
COMMON/SPTEMP/ SPTEMP
COMMON/ANNAND/ ANNAND
COMMON/EXSUB/ EXSUB
COMMON/POWER/POWER
COMMON/TABSIZ/ IRON, EROW
COMMON/TABLIN/ TABLIN
COMMON/TABLEX/ TABLEX
COMMON/ICYL/ ICYL
COMMON/I/ PINLET
COMMON/EPARAM/ BORE, STROKE, CONRL, CYLCA, CSATDC, CMTIO, CLVTDC
COMMON/TCPAR/ B(2)
COMMON/VLYOPN/ IINOPN, IEXOPN
COMMON/TIMES/ TIVO, TEC, TIVC, TINJ, TIGN, TEVO
COMMON/ITRML/ MAXTRY, MAXERR
COMMON/INTRR/ RELERR, ABSERR, REROOT, AEROOT
COMMON/ED&K/ ENKT(3)
C
COMMON/RADN/ TCHTR, TRHTR, THR, CHTRAP, CHTRAH, CHTRAW,
& RHTRAP, RHTRAH, RHTRAW
COMMON/TOPO/ TOAIR, POAIR, GAMAIR
COMMON/FLAME/ TAIR, PAIR, TFLAME, TRAD, EMIS
COMMON/NCLA/ NCLA, NCC
COMMON/CYLP/ CDIAM(6), CTHIK(6,3), CCOND(6,3), CHCOOL(6),
& CTCOOL(6), CUOVE(6)
COMMON/NPLA/ NPLA(18), NPC
COMMON/PARF/ PDELX(1.3), PCNUM(1.3), INNODE(10,3), PCOND(10,3),
& PHEFF(19,3), PDIFU(16,3), PHCOOL(10), PTCOOL(10),
& PUOVE(10), PSURFA(10)
COMMON/PFD/ PTW
COMMON/HTCOTG/ HTC(10), TGAS(10)
COMMON/QSOL/ QWALL(10), QAVE(10), QSS(10)
COMMON/TEMPS/ NTEMP, TMALL(10), TSS(10), TSS2(10), TCW, TCW2
COMMON/PERFCT/ PERI, FACT, DELT
COMMON/HTRC/ CONHT, EXPH
COMMON/CRAD/ CRAD
COMMON/Areas/ AHEAD, APSTON
COMMON/TURB/ CBETA, MACRSC, UPRIME, VMKE, VPISO
COMMON/HEATS/ CVHTRN, HTRCOE, HTPAPI, HTPAHD, HTPACW, HTRAPI,
& HTRAWD, HTRACW, HTRAN
C
COMMON/SUMIT/ ENGM(2), FMDOT(2), HMDOT(2)
COMMON/EHF/ENT(8), HAL(8), FAL(8), TFLAG, TAL(8)
COMMON/BURN/ FRPM
COMMON/D/ ERPM
COMMON/DTDTH/ ESPD
COMMON/FUEL/ FUELTP, CX, HY, DEL, PSI, AFRAST, QLOWER, HFORM
COMMON/ARRAY/ MASSIN, MASSEX, FCYLIN, FCYLEX, HCYLIN, HCYLEX
COMMON/MSTA/ MSTART
COMMON/MA/ YPM(2), YPH(2), YPF(2)
COMMON/RHAAS/ RHO, MASS, VOLUME, HH, GAMMA
COMMON/VALVE/ VIV, VEV
COMMON/B / CPM(2), HM(2), MMH(2), GM(2), RHOM(2)
COMMON/X / RTMP(5), H(5), RMASS(5), ROCR(5)
COMMON/NEW/ ASP(3), PR(3), PRSS(5), DP(5), HI(5)
& TMAP(2), PTMAP(2), CMAP(2)
& CM(SIZC,SIZ1,3), TM(SIZT,SIZ2,3), PM(SIZPT,SIZ3,3)
& CRPM(SIZC), TRPM(SIZT), PTRPM(SIZPT), PSTD(3), TSTD(3)
COMMON/PROFILE/ DTBN, ALPHA, CSP1, CSP2, CSD1, CSD2
COMMON/FRATE/
COMMON/QPI/ EDIAM(3), EAREA(3), ECROSS(3), ETWALL(3), ECONHT(3)
COMMON/OQP/ EPRF(3), EVBLK(3), EREF(3), ENUF(3), ENHOE(3)
& EQDOT(3)
COMMON/FLOPRO/ EMFLO(NCV+1), EHFL0(NCV+1), EFFLO(NCV+1)
COMMON/B/ ERHO(NCV), DYNVIS(NCV), EH(5), EFFR(NCV)
COMMON/ARRAYX/ XM(2,720), XR(2,720), XH(2,720)
COMMON/OQ2/ QLOSS(2,720)
COMMON/SUMITX/ ENGMX(NCV), FMDOTX(NCV), HMDOTX(NCV), QDOTX
COMMON/EXNUM/ NUMEX, NUMEXT
COMMON/EXAR/ NPIPE(NCV), NSHFT(NCV), NUMSEG(NCV)
COMMON/EXGEOM/ EXLNG(NCV), EXDIAM(NCV), EXAREA(NCV)
& EXVOL(NCV), EXXAR(NCV)
COMMON/EXAR/CBP(NCV), RLANG(NCV), ENCEE(NCV)
COMMON/PHASE/ EPHAO, EPHAC, IPHASE, REJ
COMMON/EXTINI/ EXW(NCV)
COMMON/INFO/ IDIFCT, ILOOPC
COMMON/EXHTEM/ AVREXT
NAMELIST/INPUT/ TITLE, POWER, SPTEMP, TRANS,
& ANNAND, SPDEL, ERPM, EXSUB,
& FUELTP, FMIN, TMU, TIGN, TIVO, TIVC, TEVO, TEVC, ICYL,
& BORE, STROKE, COMRL, AHEAD, APSTON,
& CLVTDG, ELONG, EDIAM, HI, B, PTTEF, EDMK, DP, ROCR(5), CFR2,
& CFR3, TAMT, PATM, PINLET, CFACTR, DTBRN, TPSTON, THEAD, TCW,
& ETWALL, EXPHT, COMHT, CRAD, CBETA, ECONHT, PSTART, TSTART,
& PHISTA, MKESTA, TKESTA, YO, PRSS, RTMP, TPRINT, TSCREN,
& CIINTG, CBINTG, CCINTG, CEINTG, AREROT, MAXITS,
& MINITS, DELQ1, DELQ2, DELM1, DELM2, CSC, TSC, PTSC,
& CMSC, TMSC, PTMSC
EXTERNAL INTAKE, CMPRES, CMBSTN, EXAUST, GCMP, GIDEL
C STANDARD DATA SET — OVERRIDE BY USING NAMELIST INPUT
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POWER = .TRUE.
SPDEL = .TRUE.
TRANS = .FALSE.
EXSUB = .TRUE.
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ECONHT(1) =  .035
ECONHT(2) =  .035
ECONHT(3) =  .035
EXPH1 = .8
CRAD  =  2.

C

DP(1)  =  2000.
DP(2)  =  1000.
DP(3)  =  1000.
DP(4)  =  1000.
DP(5)  =  2000.

C

EMKT(1)  =  1.0
EMKT(2)  =  1.0
EMKT(3)  =  1.0

C

PRSS(4)  =  1.3E5
PRSS(5)  =  1.0E5

C

PSTART  =  3.6
TSTART  =  800.
PHISTA  =  0.47
KKESTA  =  2.5E-3
TKESTA  =  1.1E-3
Y0(2)  =  320.
Y0(3)  =  240.E3
Y0(4)  =  0.0
Y0(5)  =  800.
Y0(6)  =  330.E3
Y0(7)  =  0.0336
Y0(8)  =  64.

C

TPRINT  =  8.0
TSCREEN  =  100.0

C

AREROT  =  1.E-4
CIINTG  =  1.E-4
CCINTG  =  1.E-4
CBINTG  =  1.E-5
CEINTG  =  1.E-4
MAXERR  =  1.E-4
MAXTRY  =  100
MAXITS  =  20
MINITS  =  1
DELU1  =  0.20
DELU2  =  0.20
DELM1  =  0.005
DELM2  =  0.002

C

ISTEDY  =  21

C

OPEN INPUT AND OUTPUT FILES:
NOTE THAT FILES ARE ALSO OPENED AT THE END OF MAIN PROGRAM AND
CLOSED IN MAIN PROGRAM, AND IN SBTNS CYLPR, PARFIN, & MANPAR
CALL FILEDEF
READ NAMELIST INPUT
READ (8, INPUT)

CCOUT
WRITE(16, *) TITLE
WRITE(16, *) 'ENGINE SPEED = ', ERPM
WRITE(16, *) 'FUEL INJECTED/CYL/CYCLE = ', FMIN
CLOSE(UNIT=16)

READ (8, INPUT)
CCOUT
READ IN COMPRESSOR, TURBINE AND POWER TURBINE MAPS:
CSC, TSC, AND PTSC SCALE EFFICIENCIES
CMSC, TMSC, AND PTMSC SCALE MASS FLOWS

COMPRESSOR:
READ (3, *) PSTD(1), TSTD(1)
DO 2 I = 1, SIZC
   READ (3, *) CRPM(I)
   DO 1 J = 1, SIZ1
      READ (3, *) (CM(I, J, K), K = 1, 3)
      CM(I, J, 2) = CM(I, J, 2) * CSC
      CM(I, J, 1) = CM(I, J, 1) * CMSC
   1 CONTINUE
2 CONTINUE

TURBINE:
READ (4, *) PSTD(2), TSTD(2)
DO 4 I = 1, SIZT
   READ (4, *) TRPM(I)
   DO 3 J = 1, SIZ2
      READ (4, *) (TM(I, J, K), K = 1, 3)
      TM(I, J, 2) = TM(I, J, 2) * TSC
      TM(I, J, 1) = TM(I, J, 1) * TMSC
   3 CONTINUE
4 CONTINUE

POWER TURBINE:
IF (.NOT. POWER) GOTO 7
READ (5, *) PSTD(3), TSTD(3)
DO 6 I = 1, SIZPT
   READ (5, *) PTRPM(I)
   DO 5 J = 1, SIZ3
      READ (5, *) (PTM(I, J, K), K = 1, 3)
      PTM(I, J, 2) = PTM(I, J, 2) * PTSC
      PTM(I, J, 1) = PTM(I, J, 1) * PTMSC
   5 CONTINUE
6 CONTINUE

READ IN INTAKE VALVE FLOW AREAS:
7 CONTINUE
IROW = 0
DO 77 I = 1, 120
   READ(75, *, END=772) (TABLIN(I, J), J=1, 2)
   CONVERT AREAS FROM SQ. FEET INTO SQ. METERS
   TABLIN(I, 2) = TABLIN(I, 2) * 0.092903
   IROW = IROW + 1
771 CONTINUE
WRITE (8, *) 'INTAKE VALVE DATA LIST TOO LONG, SHORTEN LIST OR',
& 'REDIMENSION ARRAY TABLIN IN MAIN PROGRAM AND VACDIN'

C
C READ IN EXHAUST VALVE FLOW AREAS:
    772 EROW = 0
    DO 774 I = 1, 120
         READ(76.*, END=774) (TABLEX(I,J), J=1,2)
C CONVERT AREAS FROM SQ.FEET INTO SQ.METERS
    TABLEX(I,2) = TABLEX(I,2) * 0.092903
    EROW = EROW + 1
C 773 CONTINUE
    WRITE (6,*) 'EXHAUST VALVE DATA LIST TOO LONG, SHORTEN LIST OR',
    & 'REDIMENSION ARRAY TABLEX IN MAIN PROGRAM AND VACDEX'
C
C 774 CLOSE (UNIT=3)
    CLOSE (UNIT=4)
    CLOSE (UNIT=5)
    CLOSE (UNIT=6)
    CLOSE (UNIT=75)
    CLOSE (UNIT=76)
C
C CALL FUELD
    FSTART = PHISTA / (PHISTA + AFRAST)
C
C CONVERT INPUT PARAMETERS TO SI UNITS:
    BORE = BORE / 100.
    STROKE = STROKE / 100.
    CONRL = CONRL / 100.
    APSTON = APSTON * 1.E-4
    AHEAD = AHEAD * 1.E-4
C
C PSTART = PSTART * ATPA
    PATM = PATM * ATPA
    PINLET = PINLET * ATPA
    CLVTDC = CLVTDC * 1.E-6
    FMIN = FMIN * 1.E-3
C
C INITIALIZE CHAMBER GEOMETRY DATA AND CALCULATE BASIC
C GEOMETRIC ENGINE PARAMETERS AND CONSTANTS:
    IF (AHEAD .EQ. 0.) AHEAD = 1.25 * PI * BORE * BORE / 4
    IF (APSTON .EQ. 0.) APSTON = PI * BORE * BORE / 4
C
C DVOLUM = PI * BORE * BORE * STROKE/4.
    DVOL = FLOAT(ICYL) * DVOLUM
    CMRTIO = (CLVTDC + DVOLUM)/CLVTDC
    CYLCA = .7853982 * BORE * BORE
    ESPD = 1./((.6 * ERPM)
C
C DELT = ESPD
    PERI = 720. * DELT
C
C READ IN CYLINDER HEAT TRANSFER DATA:
    IF (SPTEMP) GOTO 10
C
C CALL PARFIN
    CALL CYLPAR
C
C 10 TWALL(1) = TPSTON
    TWALL(2) = THEAD
    TSS(1) = TPSTON
    TSS(2) = THEAD
    TCWIN = TCW
C
C PERFORM GEOMETRIC CALCULATIONS FOR INTAKE AND EXHAUST
C MANIFOLDS AND TURBINE CONNECTING PIPE:
NMAN = 3
IF (EXSUB) NMAN = 1
C
EDIAM(2) & (3) AND ELNG(2) & (3) IGNORED FOR EXHAUST SUB-MODEL
C
DO 8 J = 1, NMAN
EAREA(J) = PI * EDIAM(J) * ELNG(J)
ECROSS(J) = PI * EDIAM(J) * EDIAM(J) / 4.
EVOLME(J) = ECROSS(J) * ELNG(J)
8 CONTINUE
C
READ IN AND INITIALIZE EXHAUST MANIFOLD GEOMETRY AND
C PROPERTIES:
NTEMP = 2
NUMEX = 0
IF (EXSUB) THEN
CALL MANPAR
NTEMP = NTEMP + NUMEX
ENDIF
C
C APPROXIMATE AVERAGE ENGINE INTAKE MASS FLOW (LB/MIN):
CALL THERMO(YB(2),YB(3),YB(4),HMM(1),CPM(1),G3,G4,RHOM(1),G5,
& G6,G7,GM(1),MMM(1),G10,G11,G12)
RCORR(2) = .0075667 * ESPD
RMASS(1) = 0.9 * DVOL * RHOM(1) /720. /RCORR(2)
C
C ASSIGN INITIAL VALUES TO ENTHALPIES, TEMPERATURES,
C MASS FLOW RATES AND EQUIVALENCE RATIOS:
C
FUELRT = FMIN * FLOAT(ICYL) / 720. /RCORR(2)
CALL THERMO ( Y0(2), Y0(3), 0., HGIN, XXB, XCC, XXD,
& XFE, XXF, XXG, XXH, XXI, XXJ, XXX, XXL, XXM)
CALL THERMO ( Y0(6), Y0(7), FSTART, HGEX, XXB, XCC, XXD,
& XGE, XXXF, XXG, XXH, XXI, XXJ, XXX, XXL, XXM)
C
ITIVO = NINT(TIVO)
ITIVC = NINT(TIVC)
ITEVO = NINT(TEVO)
ITEVC = NINT(TEVC)
INOPN = ITIVC - ITIVO
DO 776 I = 1, INOPN
AAA = FLOAT(I) / INOPN * PI
MASSIN(I) = 360. /ICYL / INOPN * PI * RMASS(1)
& * RCORR(2) * SIN(AAA)
FCYLIN(I) = 0.
HCYLIN(I) = HGIN
776 CONTINUE
C
IEXOPN = ITEVC - ITEVO
DO 777 I = 1, IEXOPN
BBB = FLOAT(I) / IEXOPN * PI
EMASS = 360. /ICYL / IEXOPN * PI * (RMASS(1)+FR)
& * RCORR(2) * SIN(BBB)
MASSEX(I) = EMASS
FCYLEX(I) = FSTART
HCYLEX(I) = HGEX
777 CONTINUE
C
TE = Y0(6)
Y0(8) = FSTART
CALL THERMO (RTEMP(1), Y0(4), H(1), X1, X2, X3,
& X4, X5, X6, X7, X8, X9, X10, X11, X12)
CALL THERMO (TE, Y0(7), Y0(8), H(2), V1, V2, V3,
& V4, V5, V6, V7, V8, V9, V10, V11, V12)

CALL THERMO (TE, Y0(7), Y0(8), H(2), V1, V2, V3,
& V4, V5, V6, V7, V8, V9, V10, V11, V12)

C INITIALIZE SOME SYSTEM TEMPERATURES:
RTEMP(3) = 0.85 * Y0(8)
RTEMP(2) = Y0(2)
RTEMP(4) = RTEMP(3)
RTEMP(5) = RTEMP(4)

C INITIALIZE VARIOUS SYSTEM Pressures:
PRSS(1) = Y0(3) + DP(1) + DP(2)
PRSS(2) = Y0(7) - DP(3)
PRSS(5) = PATM + DP(5)

C FIND COMPRESSOR, TURBINE AND POWER TURBINE MAP
C TEMPERATURE CORRECTION FACTORS:
RCORR(3) = SQRT(TSTD(1)/RTEMP(1))
RCORR(4) = SQRT(TSTD(2)/Y0(6))
IF (POWER) RCORR(5) = SQRT(TSTD(3)/Y0(6))

C CALCULATE MANIFOLD MASSES FROM OTHER INITIAL STATES,
GIVEN VOLUME AND IDEAL GAS LAW, FOR EACH MANIFOLD:
CALL THERMO(Y6(6), Y0(7), Y0(8), H0M(2), CPM(2), Q3, Q4, R0M(2), Q5,
& Q6, Q7, GM(2), MM(2), Q10, Q11, Q12)
Y0(1) = R0M(1) * EVOLME(1)
Y0(5) = R0M(2) * EVOLME(2)

IF (.NOT.EXSUB) GOTO 780
TEMP = Y0(8)
FR = Y0(8)
VOLSUM = 0.
DO 779 I = 1, NUMEX
IMASS = 43 + (I-1)*8
Y0(IMASS) = R0M(2) * EXVOL(I)
Y0(IMASS+1) = TEMP
Y0(IMASS+2) = FR
VOLSUM = VOLSUM + NUMSEG(I)*EXVOL(I)
779 CONTINUE

C SUBSTITUTE CROSS SECTIONAL AREAS FOR PRESSURE DROP CALCULATIONS
ECROSS(2) = EXXAR(2)
IF (POWER) ECROSS(3) = EXXAR(NUMEXT)
ERHO(2) = R0M(2)
ERHO(4) = R0M(2)

C~~~~~~~~~~~~~~~~~~~~~~~~~~~~~
C START OF CURRENT CYCLE ITERATION
C~~~~~~~~~~~~~~~~~~~~~~~~~~~~~

DO 780 ITERAS = 1, MAXITS
WRITE (7,880) ITERAS, MAXITS

C CALCULATE MASS IN CYLINDER:
CALL THERMO ( TSTART, PSTART, FSTART, HSTART, XXB, XXC, XXD, 
  & RHO, XXE, XXF, XXG, XXX, XXI, XXJ,  
  XXK, XXL) 
CALL CSAVDV (TIVO, XXA, VOLUME, XXB) 
MSTART = RHO * VOLUME

PREPARE OUTPUT FILES FOR OUTPUT FROM THIS ITERATION:
REWIND 6
REWIND 12
REWIND 13
REWIND 14
REWIND 30
REWIND 37
REWIND 38
REWIND 39
REWIND 40

WRITE MAIN HEADINGS AND ECHO INPUT PARAMETERS:
WRITE (6,222)
IF (.NOT. POWER) WRITE (6,2899)
IF (POWER) WRITE (6,2900)
WRITE (6,222)
WRITE (6,2901)
WRITE (6,222)
WRITE (6,2914)
WRITE (6,*) TITLE
IF (SPDEL) WRITE (6,2905)
IF (.NOT.SPDEL) WRITE (6,2906)
IF (SPTEMP) WRITE (6,2907)
IF (.NOT.SPTEMP .AND. ITERAS .LT. ISTEDY) WRITE (6,2908)
IF (.NOT.SPTEMP .AND. ITERAS .GT. ISTEDY) WRITE (6,2909)
IF (ANNAND) WRITE (6,2910)
IF (.NOT. ANNAND) WRITE (6,2911)
IF (EXSUB) WRITE (6,2931)
IF (.NOT. EXSUB) WRITE (6,2932)
WRITE (6,222)
WRITE (6,2902)
IF (FUELTP .EQ. 1) WRITE (6,2903)
IF (FUELTP .EQ. 2) WRITE (6,2904)
WRITE (6,2913) ERPM, TINJ, FMIN*KIL, FUELRT, PINLET/ATPA,
& RTEMP(1), PATM/ATPA, TATM
WRITE (6,222)

WRITE (6,111)
WRITE (6,*) 
WRITE (6,*) ' >>>>> ENGINE DESIGN PARAMETERS'
WRITE (6,*) 
WRITE (6,2919) ICYL, BORE*CEN, STROKE*CEN, CONRL*CEN, CMRTIO,
& DVOLUM*MIL, CLVTD*CEN, DVOL*KIL, CFR2, CFR3,
& TIVO, TIVC, TEVO, TEVC
WRITE (6,222)

IF (EXSUB) GOTO 781
WRITE (6,*) ' >>>>> MANIFOLD DIMENSIONS INTAKE',
& ' EXHAUST'
WRITE (6,*) 
WRITE (6,*) 'LENGTH (M) ', ELNG(1), ELNG(2)
WRITE (6,*) 'DIAMETER (M) ', EDIAM(1), EDIAM(2)
WRITE (6,*) 
WRITE (6,*) 'CROSS-SECTIONAL AREA (M**2) ', ECROSS(1),
& ECROSS(2)
WRITE (6,*) 'INTERNAL SURFACE AREA (M**2)', EAREA(1).
& EAREA(2)
WRITE (6,*) ',
WRITE (6,*) 'VOLUME (LT)', EVOLME(1)*1000.,
& EVOLME(2)*1000.
WRITE (6,*) '

C
WRITE (6,222) '>>> TURBINE CONNECTING PIPE DIMENSIONS'
WRITE (6,*) '
WRITE (6,*) 'LENGTH (M)', ELNG(3)
WRITE (6,*) 'DIAMETER (M)', EDIAM(3)
WRITE (6,*) '
WRITE (6,*) 'CROSS-SECTIONAL AREA (M**2)', ECROSS(3)
WRITE (6,*) 'INTERNAL SURFACE AREA (M**2)', EAREA(3)
WRITE (6,*) '
WRITE (6,*) 'VOLUME (LT)', EVOLME(3)*1000.
GOTO 790

C
CONTINUE
WRITE (6,*) '>>> INTE MANIFOLD DIMENSIONS'
WRITE (6,*) '
WRITE (6,*) 'LENGTH (M)', ELNG(1)
WRITE (6,*) 'DIAMETER (M)', EDIAM(1)
WRITE (6,*) '
WRITE (6,*) 'CROSS-SECTIONAL AREA (M**2)', ECROSS(1)
WRITE (6,*) 'INTERNAL SURFACE AREA (M**2)', EAREA(1)
WRITE (6,*) '
WRITE (6,*) 'VOLUME (LT)', EVOLME(1)*1000.
WRITE (6,*) '
WRITE (6,*) '
CALL EXWRIT(NEQN, DY, 1)

C
WRITE (6,222) '>>> TURBOMACHINERY DATA'
WRITE (6,*) '
WRITE (6,*) 'T/C INERTIA (KG-M**2)', B(1)
WRITE (6,*) 'T/C DAMPING (KG-M**2/S)', B(2)
WRITE (6,*) '
WRITE (6,*) 'P:TURBINE TRANSMISSION EFFIC.', PTTEF
WRITE (6,*) 'P:TURBINE GEAR RATIO', RCORR(1)
WRITE (6,*) '
WRITE (6,*) 'COMPRESSOR EFF. SCALE FACTOR - CSC ', CSC,
& COMP MASS FLOW SCALE FACTOR - CMSC ', CMSC
WRITE (6,*) '
WRITE (6,*) 'TURBINE EFF. SCALE FACTOR - TSC ', TSC,
& TURB MASS FLOW SCALE FACTOR - TMSC ', TMSC
WRITE (6,*) '
WRITE (6,*) 'POWER TURB EFF. SCALE FACTOR- PTSC ', PTSC,
& PWR TURB MASS FLOW SCL FACTOR- PTMSC', PTMSC
WRITE (6,*) '
WRITE (6,222) '>>> SYSTEM PRESSURE LOSSES'
WRITE (6,*) '
WRITE (6,*) 'SPECIFIED PRESSURE DROPS (PA):'
WRITE (6,*) 'COMPRESSOR EXIT - INTERCOOLER INLET - DP(1)', DP(1)
WRITE (6,*) 'POWER TURBINE EXIT - ATMOSPHERIC - DP(5)', DP(5)
WRITE (6,*) '

C

WRITE (6,*) 'SPECIFIED LOSS FACTORS:
WRITE (6,*) 'INTERCOOLER INLET - INTAKE - EMKT(1)
& EMKT(1)
WRITE (6,*) 'EXHAUST MANIFOLD - TURBINE INLET - EMKT(2)
& EMKT(2)
IF (POWER) WRITE (6,*)
& 'TURBINE CONNECTING PIPE EMKT(3)
WRITE (6,222)
C
WRITE (6,*) '>>> HEAT TRANSFER AND TURBULENCE PARAMETERS'
WRITE (6,8861) DO 18 I = 1, NPC
IF (I.EQ.1) WRITE (6,8862)
IF (I.EQ.2) WRITE (6,8863)
WRITE (6,8865) (ILAYER, ILAYER=1, NPLA(I))
WRITE (6,8867) (PTHIK(I,J), J=1, NPLA(I))
WRITE (6,8868) (PCOND(I,J), J=1, NPLA(I))
WRITE (6,8869) (PDIFU(I,J), J=1, NPLA(I))
WRITE (6,8870) (INNODE(I,J), J=1, NPLA(I))
WRITE (6,8871) INT(FACT * INNODE(I,1))
WRITE (6,8872) (PCNUM(I,J), J=1, NPLA(I))
WRITE (6,8880) (PTCOOL(I), PUOVE(I))
WRITE (6,8881) (PTCOOL(I), PUOVE(I))
WRITE (6,8882) (PTCOOL(I), PUOVE(I))
C
DO 19 I = 1, NCC
WRITE (6,8864)
WRITE (6,8865) (ILAYER, ILAYER=1, NCLA(I))
WRITE (6,8866) CDIAM(I)
WRITE (6,8867) (CTHIK(I,J), J=1, NCLA(I))
WRITE (6,8868) (CCOND(I,J), J=1, NCLA(I))
WRITE (6,8880) (CTCOOL(I), CUOVE(I))
WRITE (6,8881) (CTCOOL(I), CUOVE(I))
C
WRITE (6,*)
C
WRITE (6,222)
C
WRITE (6,1111)
C
WRITE (6,8881)
DO 18 I = 1, NPC
WRITE (6,1111)
WRITE (6,1112)
WRITE (6,1113)
WRITE (6,1114)
WRITE (6,1115)
WRITE (6,1116)
WRITE (6,1117)
WRITE (6,1118)
WRITE (6,1119)
WRITE (6,1120)
WRITE (6,1121)
WRITE (6,1122)
WRITE (6,1123)
WRITE (6,1124)
WRITE (6,1125)
WRITE (6,1126)
WRITE (6,1127)
WRITE (6,1128)
WRITE (6,1129)
WRITE (6,1130)
WRITE (6,1131)
WRITE (6,1132)
WRITE (6,1133)
WRITE (6,1134)
WRITE (6,1135)
WRITE (6,1136)
WRITE (6,1137)
WRITE (6,1138)
WRITE (6,1139)
WRITE (6,1140)
WRITE (6,1141)
WRITE (6,1142)
WRITE (6,1143)
WRITE (6,1144)
WRITE (6,1145)
WRITE (6,1146)
WRITE (6,1147)
WRITE (6,1148)
WRITE (6,1149)
WRITE (6,1150)
WRITE (6,1151)
WRITE (6,1152)
WRITE (6,1153)
WRITE (6,1154)
WRITE (6,1155)
WRITE (6,1156)
WRITE (6,1157)
WRITE (6,1158)
WRITE (6,1159)
WRITE (6,1160)
WRITE (6,1161)
WRITE (6,1162)
WRITE (6,1163)
WRITE (6,1164)
WRITE (6,1165)
WRITE (6,1166)
WRITE (6,1167)
WRITE (6,1168)
WRITE (6,1169)
WRITE (6,1170)
WRITE (6,1171)
WRITE (6,1172)
WRITE (6,1173)
WRITE (6,1174)
WRITE (6,1175)
WRITE (6,1176)
WRITE (6,1177)
WRITE (6,1178)
WRITE (6,1179)
WRITE (6,1180)
WRITE (6,1181)
WRITE (6,1182)
WRITE (6,1183)
WRITE (6,1184)
WRITE (6,1185)
WRITE (6,1186)
WRITE (6,1187)
WRITE (6,1188)
WRITE (6,1189)
WRITE (6,1190)
WRITE (6,1191)
WRITE (6,1192)
WRITE (6,1193)
WRITE (6,1194)
WRITE (6,1195)
WRITE (6,1196)
WRITE (6,1197)
WRITE (6,1198)
WRITE (6,1199)
WRITE (6,1200)
WRITE (6,2921) MAXITS, ITERAS, TPRINT, TSCREN,
& CINTG, CCINTG, CBINTG, CEINTG, AREROT
WRITE (6,2922) DELM1, DELM2, DELQ1, DELQ2
WRITE (6,222)
WRITE (6,111)
WRITE (6,2998)
WRITE (6,222)
WRITE (6,2999)
WRITE (6,222)
WRITE (6,3110)
WRITE (6,3596)
WRITE (6,222)
WRITE (6,2998)
WRITE (6,222)
WRITE (6,3110)
WRITE (6,3596)
WRITE (6,222)
WRITE (6,2998)
WRITE (12,111)
WRITE (12,6111)
WRITE (12,222)
WRITE (12,3110)
WRITE (12,6594)
WRITE (12,222)
WRITE (12,111)
WRITE (12,6111)
WRITE (12,222)
WRITE (12,3110)
WRITE (12,6594)
WRITE (12,222)
WRITE (12,111)
WRITE (12,6111)
WRITE (12,222)
WRITE (12,3110)
WRITE (12,6594)
WRITE (12,222)
WRITE (13,111)
WRITE (13,6111)
WRITE (13,222)
WRITE (13,3110)
WRITE (13,7592)
WRITE (13,222)
WRITE (13,111)
WRITE (13,6111)
WRITE (13,222)
WRITE (13,3110)
WRITE (13,7592)
WRITE (13,222)
WRITE (13,111)
WRITE (13,6111)
WRITE (13,222)
WRITE (13,3110)
WRITE (13,7592)
WRITE (13,222)
WRITE (14,111)
WRITE (14,8111)
WRITE (14,3110)
WRITE (14,8592)
WRITE (14,222)
ENDIF
WRITE HEADINGS FOR TRANSIENT WALL TEMPERATURE PROFILES
IF (.NOT. ANNAND) THEN
WRITE (14,111)
WRITE (14,8111)
WRITE (14,8592)
WRITE (14,222)
ENDIF
C
C WRITE HEADINGS FOR TRANSIENT WALL TEMPERATURE PROFILES
IF (TRANS .AND. ITERAS.GT.ISTEDY) THEN
WRITE (36,111)
WRITE (36,9111) TITLE
WRITE (36,222)
C
WRITE (37,111)
WRITE (37,9112) TITLE
WRITE (37,222)
C
IF (EXSUB) THEN
DO 15 I = 38, 40
WRITE (1,111)
WRITE (1,9113) TITLE
WRITE (1,222)
15 CONTINUE
ENDIF
C
C INITIALIZE PARAMETERS FOR CALL TO SUBROUTINE ODERT:
DO 16 I = 1, NEQN
DY(I) = 0.
16 C
C INITIALIZE MASTER CYLINDER VARIABLES:
DY(6) = MKESTA
DY(7) = TKESTA
initialize intake manifold and exhaust manifold variables:

\[ Dy(11) = T_{START} \]
\[ Dy(12) = P_{START} \]
\[ Dy(20) = F_{START} \]

C

DO 161 I = 1, 9

161 Dy(20+I) = Y0(I)

C

initialize exhaust manifold section variables:

IF (.NOT.EXSUB) GOTO 162

DO 163 I = 1, NUMEX

IREF = 43 + (I-1)*8

Dy(20+IREF) = Y0(IREF)

Dy(21+IREF) = Y0(IREF+1)

Dy(22+IREF) = Y0(IREF+2)

163 CONTINUE

C

HEATI = 0.0

WORKI = 0.0

VIV = 0.0

WRITE(6,4216) TIVO,Dy(12)/ATPA, Dy(11), Dy(1)*KIL, Dy(2)*KIL,

& EQR(Dy(2E)), Dy(23)/ATPADy(22), Dy(27)/ATPADy(26), IFLAG

AEROOT = AREROT

REROOT = AREROT

C

START OF INTAKE PROCESS (TIVO - TIVC)

C

CALL VLVSUM (IIN, 1, Dy(24), HM(1), MASSIN, FCYLIN, HCYLIN,

& IINOPN)

CALL VLVSUM (IEX, 2, Dy(28), HM(2), MASSEX, FCYLEX, HCYLEX,

& IEXOPN)

C

NCALL = NINT(TEND - DT)

NCALL: NO. OF TIMES INTEGRATING SUBROUTINE IS CALLED

DO 60 NC = 1, NCALL

TOUT = DT + 1.

CALL STEPCA(INTAKE, NEQN, Dy, DT, TOUT, IFLAG, WORK, IWOR...

C

IF (I .EQ. 2) GO TO 200

HEATI = Dy(8) + Dy(9) + Dy(10)
1321 WORKI = DY(16)
1322 TEND = TIVC
1323 GO TO 30
1324 C
1325#############################################################
1326 C
1327 C END OF INTAKE PROCESS
1328 C
1329#############################################################
1330 C
1331 200 CONTINUE
1332 C
1333 ZMAST = MSTART + DY(1) - DY(2)
1334 C
1335 C CALCULATE TOTAL MASS OF AIR INDUCTED IN THIS CYCLE (AMIN):
1336 C AMIN = DY(1)
1337 C
1338 C CALCULATE VOLUMETRIC EFFICIENCY RELATIVE TO INTAKE MANIFOLD
1339 C CONDITIONS (VOLEFI) AND ATMOSPHERIC CONDITIONS (VOLEFA):
1340 C TIM = DY(22)
1341 C PIM = DY(23)
1342 C VOLEFI = 100. * AMIN / (DVOLUM * RHOM(1) )
1343 C VOLEFA = VOLEFI * (PIM/PATM) * (TATM/TIM)
1344 C WRITE (7,891) VOLEFI
1345 C
1346 C WRITE (6,222)
1347 CNEWPAGE WRITE (6,111)
1348 C
1349 WRITE (6,3111)
1350 WRITE (6,3597)
1351 WRITE (6,222)
1352 C
1353 WRITE (12,222)
1354 WRITE (12,111)
1355 WRITE (12,3111)
1356 WRITE (12,6594)
1357 WRITE (12,222)
1358 C
1359 WRITE (13,222)
1360 WRITE (13,111)
1361 WRITE (13,3111)
1362 WRITE (13,7592)
1363 C
1364 C
1365#############################################################
1366 C
1367 C OPTION 1: START OF COMPRESSION PROCESS (TIVC - TINJ)
1368 C OPTION 2: START OF COMPRESSION & IGNITION DELAY PROCESS (TIVC - TIGN)
1369 C
1370#############################################################
1371 C
1372 IFLAG = 1
1373 ABSERR = CCINTG
1374 RELERR = CCINTG
1375 TEND = TINJ
1376 IF (SPDEL) TEND = TIGN
1377 C
1378 NCALL = NINT(TEND - DT)
1379 C
1380 DO 220 NC = 1, NCALL
TOUT = DT + 1.

CALL STEPCA(CMPRES, NEQN, DY, DT, TOUT, IFLAG, WORK, IWORK, GCMP, 2)

C 220 CONTINUE
C
C IF (SPDEL) GO TO 227
C
C C 251 IFLAG = 1
C
ABSERR = CCINTG
RELERR = CCINTG
TEND = TEVO
NCALL = NINT(TEND - DT)

C DO 225 NC = 1, NCALL
C
TOUT = DT + 1.
C
CALL STEPCA(CMPRES, NEQN, DY, DT, TOUT, IFLAG, WORK, IWORK, GIDEL, 2)
C
IF (IFLAG .EQ. 8) GO TO 226

C 225 CONTINUE
C
C 226 WRITE(7,882) DT, DY(12)/ATPA, IFLAG
C
WRITE(8,4211) DT, DY(12)/ATPA, DY(11), EQR(DY(20)), DY(23)/ATPA,
C
& UPRIME, CVHTRN, MACRSC*CEN
C
WRITE(13,7210) DT, HTRCOE, HTPAPI/KIL, HTPAHD/KIL,
C
& HTPACW/KIL, THTRAN/KIL

C
C 227 HEATC = DY(8) + DY(9) + DY(10) - HEATI
C
WORKC = DY(16) - WORKI
C
C END OF COMPRESSION PROCESS
C
C REINITIALIZE 'ODERT' FOR START OF COMBUSTION:

IFLAG = 1
TIGN = DT
TIDEL = TIGN - TINJ
DELMS = TIDEL * 1000. * ESPD

C
DY(4) = 0
MASS = MSTART + DY(1) - DY(2) + DY(4)
AIRMAS = MASS * (1. - DY(28))
PHIOVE = (FMIN / AIRMAS) * AFRAST

ALPHA = 0.
DELMIN = (0.926 + PHIOVE**0.37)**(1./0.26)
IF (DELMS .GT. DELMIN)
C
ALPHA = 1. - 0.926 * PHIOVE**0.37 / DELMS**0.28

C THE FOLLOWING COMBUSTION CORRELATION CONSTANTS MUST BE
C CALIBRATED AGAINST ENGINE CYLINDER PRESSURE DATA:
\text{CSP1} = 2. + 1.25 \times 10^{-8} \times (\text{DELMS} \times \text{ERPM})^{2.4}

\text{CSP2} = 5000.

\text{CSD1} = 14.2 / \text{PHIOVE}^{0.644}

\text{CSD2} = \text{CFACTR} \times \text{CSD1}^{0.25}

\text{TOAIR} = \text{DY(11)}

\text{POAIR} = \text{DY(12)}

\text{CALL THERMO (TOAIR, POAIR, X1, X2, X3, X4, X5, X6, X7, X8,}
& \text{ GAMAIR, X9, X10, X11, X12)}

\text{WRITE (6,111)}

\text{WRITE (6,3112)}

\text{WRITE (6,3598)}

\text{WRITE (6,222)}

\text{WRITE (12,111)}

\text{WRITE (12,3112)}

\text{WRITE (12,6594)}

\text{WRITE (12,222)}

\text{WRITE (13,111)}

\text{WRITE (13,3112)}

\text{WRITE (13,7592)}

\text{WRITE (13,222)}

\text{DO 350 NC = 1, NCALL}

\text{CALL STEPCA(CMBSTN, NEON, DY, DT, TOUT, IFLAG, WORK, IWORK,}
& \text{ GCMP, 3)}

\text{TOUT = DT + 1.}

\text{350 CONTINUE}

\text{WRITE (6,222)}

\text{WRITE (6,111)}

\text{WRITE (6,3113)}

\text{WRITE (6,3599)}

\text{WRITE (6,222)}

\text{WRITE (12,222)}
WRITE (12,111)
WRITE (12,3113)
WRITE (12,6594)
WRITE (12,222)
WRITE (13,222)
WRITE (13,111)
WRITE (13,3113)
WRITE (13,7592)
WRITE (13,222)

C
VEV = 0.0
C
REINITIALIZE 'ODERT' FOR START OF EXHAUST:
I = 0
IFLAG = 1
ABSERR = CEINTG
RELERR = CEINTG
TEND = 540.
390 I = I + 1
NCALL = NINT(TEND - DT)
C
DO 410 NC = 1, NCALL
   TOUT = DT + 1.
   CALL STEPCA(Exhaust, NEQN, DY, DT, TOUT, IFLAG, WORK, IWORK,
&        GOMP, 4)
410 CONTINUE
C
IF (I .EQ. 2) GO TO 450
C
HEATCE = DY(8) + DY(9) + DY(10) - HEATI
WORKCE = DY(16) - WORKI
TEND = TIVO + 720.
GO TO 390
C
C#fII ######## I####IffII#I|fI|f###I#####I#I#II#IfII##
C
C END OF EXHAUST PROCESS
C
C#
450 CONTINUE
C
HEATE = DY(8) + DY(9) + DY(10) - HEATCE - HEATI
WORKE = DY(16) - WORKCE - WORKI
C
WRITE (6,222)
C WRITE TO RMAS.DAT
WRITE(15,*) 'CYCLE ', ITERAS
WRITE(15,*) 'TOTAL ENG. FLOWS (G/CYCLE)',DY(31)*KILDY(33)*KIL
WRITE(15,*) 'COMP FLOW, TRB FLOW (G/CYCLE)',DY(32)*KIL,DY(34)*KIL
DELMNO = (DY(32) + FMIN*ICYL - DY(34)) / DY(34) * 100.
WRITE(15,*) 'OVERALL NORMALIZED CHG IN MASS IN SYS. (%)=',DELMNO
C
C STORE STATE VARIABLES FOR NEXT ITERATION:
MKESTA = DY(8)
TKESTA = DY(7)
PSTART = DY(12)
TSTART = DY(11)
FSTART = DY(20)

C
DO 468 J = 1, NEQN-20
Y0(J) = DY(J+20)
468 CONTINUE

C
CALCULATE CYCLE-AVERAGE QUANTITIES FOR HEAT TRANSFER
&W NEW WALL TEMPERATURES FOR PREDICTED STEADY-STATE
WALL TEMPERATURE:
CALL QEND(NEQN, DY)
C
CHECK FOR WALL TEMPERATURE CALCULATION:
IF(SPTEMP) GO TO 467
TCW = TCW2
C
CHECK FOR TRANSIENT WALL TEMPERATURE CALCULATION:
IF(ITERAS.GT.ISTEDY) GO TO 469
C
QUASI STEADY-STATE CALCULATION:
UPDATE WALL TEMPERATURE BASED ON RESULTS FROM QEND:
DO 464 I = 1, NUMEXT+2
TWALL(I) = TSS2(I)
TSS(I) = TSS2(I)
464 CONTINUE
C
CHECK CONVERGENCE FOR STEADY-STATE CALCULATION:
IF(ITERAS .LT. MINITS) GOTO 488
IF(ABS((QAVE(1)-QAVT(1))/QAVE(1)) +
& ABS((QAVE(2)-QAVT(2))/QAVE(2)) .LE. DELQ1 .AND.
& ABS((DY(31)-DY(32))/DY(31)) .LE. DELM1 .AND.
& ABS((DY(33)-DY(34))/DY(33)) .LE. DELM1) GO TO 471
C
NO CONVERGENCE - GO ON TO NEXT CYCLE CALCULATION:
GO TO 488
C
HEAT TRANSFER CONVERGENCE CHECK FOR TRANSIENT CASE:
469 IF(ABS((QAVE(1)-QAVT(1))/QAVE(1)) +
& ABS((QAVE(2)-QAVT(2))/QAVE(2)).GT.DELQ2) GOTO 468
C
ASSURE TWO CYCLES OF TRANSIENT CALCULATION:
IF (ITERAS.LE.ISTEDY+1) GOTO 468
C
C
467 IF (ITERAS .GE. MINITS .AND.
& ABS((DY(31)-DY(32))/DY(31)) .LE. DELM2 .AND.
& ABS((DY(33)-DY(34))/DY(33)) .LE. DELM2) GO TO 471
C
CONTINUE TO NEXT CYCLE ITERATION
468 QAVT(1) = QAVE(1)
QAVT(2) = QAVE(2)
470 CONTINUE
C
CONTINUE TO NEXT CYCLE ITERATION
C
END OF CURRENT CYCLE ITERATION
C
CALCULATION RESULTS FOR THIS CYCLE
C
471 H(2) = DY(17) / DY(2)
AVREXH = H(2)
T Guess = 900.
PEM = DY(38)
FUELR = DY(20)
CALL ITRATE ( T Guess, PEM, FUELR, AVREXH, XXA, XXB, XXX,
& XDX, XEX, XXF, XXG, XHX, XXI, XXJ, XXX, XXL)
& AVREX = T Guess
WRITE(7,*), 'TOTAL ENG. FLOWS (G/CYCLE)', DY(31), KIL, DY(33)*KIL
WRITE(7,*), 'COMP FLOW, TRB FLOW (G/CYCLE)', DY(32), KIL, DY(34)*KIL
C
IF (EXSUB) GOTO 479
WRITE(6,111)
WRITE(6,*), '>>> INSTANTANEOUS SYSTEM DATA AFTER', ITERAS,
& 'ENGINE CYCLES'
WRITE(6,*), ''
WRITE(6,222)
WRITE(6,*), '>>> MANIFOLD HEAT TRANSFER DATA INTAKE',
& 'EXHAUST'
WRITE(6,*), ''
WRITE(6,*), 'MANIFOLD WALL TEMPERATURES (K)', ETWALL(1), ETWALL(2)
& ETWALL(3)
WRITE(6,*), ''
WRITE(6,*), 'AVERAGE VELOCITY (M/S)', EVBLK(1), EVBLK(2)
WRITE(6,*), ''
WRITE(6,*), 'REYNOLDS NUMBER (FILM)', EREF(1), EREF(2)
WRITE(6,*), ''
WRITE(6,*), 'PRANDTL NUMBER', EPRF(1), EPRF(2)
WRITE(6,*), ''
WRITE(6,*), 'NUSSLELT NUMBER', ENUF(1), ENUF(2)
WRITE(6,*), ''
WRITE(6,*), 'HEAT TRANSFER COEFF. (W/K/M**2)', EHTCOE(1), EHTCOE(2)
WRITE(6,*), ''
WRITE(6,*), 'HEAT TRANSFER RATE (W)', EQDOT(1), EQDOT(2)
WRITE(6,*), ''
WRITE(6,*), ''
WRITE(6,222)
WRITE(6,*), ''
WRITE(6,*), '>>> TURBINE CONN. PIPE HEAT TRANSFER DATA'
WRITE(6,*), ''
WRITE(6,*), ''
WRITE(6,*), 'PIPE WALL TEMPERATURE (K)', ETWALL(3)
WRITE(6,*), ''
WRITE(6,*), 'AVERAGE VELOCITY (M/S)', EVBLK(3)
WRITE(6,*), ''
WRITE(6,*), 'REYNOLDS NUMBER (FILM)', EREF(3)
WRITE(6,*), ''
WRITE(6,*), 'PRANDTL NUMBER (FILM)', EPRF(3)
WRITE(6,*), ''
WRITE(6,*), 'NUSSLELT NUMBER', ENUF(3)
WRITE(6,*), ''
WRITE(6,*), ''
WRITE(6,*), 'HEAT TRANSFER COEFF. (W/K/M**2)', EHTCOE(3)
WRITE(6,*), ''
WRITE(6,*), ''
WRITE(6,*), ''
WRITE(6,*), ''
WRITE(6,*), ''
WRITE(6,*), ''
WRITE(6,*), ''
WRITE(6,*), ''
WRITE(6,*), ''
WRITE(6,*), ''
WRITE(6,*), ''
WRITE (6,222)

C

479 WRITE (6,111)
WRITE (6,*) '>> TIME-AVERAGED SYSTEM DATA AFTER', ITERAS = 1,
& ' ENGINE CYCLES'
WRITE (6,*)

C

DO 454 J = 1, NEQN
Y(J) = DY(J)

C

454 CONTINUE

C

DEFINE THE FOLLOWING VARIABLES FOR USE IN OUTPUT:
APCD = Y(35) + DP(2) + DP(1)
APTI = Y(36) - DP(3)
APTD = APTI / Y(49)
APPTI = APTD - DP(4)
APPTD = PATM + DP(5)
DELP = Y(35) - Y(36)

WRITE (6,222)
WRITE (6,*) '>>>PRESSURE DATA (ATM) (IN-HG)
WRITE (6,*)
WRITE (6,*) 'COMPRESSOR INLET ',PINLET/ATPA , PINLET+HGPA
WRITE (6,*)
WRITE (6,*) 'COMPRESSOR DISCHARGE',APCD/ATPA , APCD+HGPA
WRITE (6,*)
WRITE (6,*) 'INTAKE MANIFOLD ',Y(35)/ATPA , Y(35)+HGPA
WRITE (6,*)
WRITE (6,*) 'EXHAUST MANIFOLD ',Y(36)/ATPA , Y(36)+HGPA
WRITE (6,*)
WRITE (6,*) 'TURBINE INLET ',APTI/ATPA , APTI+HGPA
WRITE (6,*)
WRITE (6,*) 'TURBINE DISCHARGE ',APTD/ATPA , APTD+HGPA
WRITE (6,*)

IF (.NOT.POWER) GO TO 473

472 WRITE (6,*) 'POWER TURBINE INLET ',APPTI/ATPA , APPTI+HGPA
WRITE (6,*)
WRITE (6,*) 'P. TURBINE DISCHARGE',APPTD/ATPA , APPTD+HGPA
WRITE (6,*)

473 WRITE (6,*) 'DELTA P ENGINE ',DELP/ATPA , DELP+HGPA
WRITE (6,*)
WRITE (6,*)
WRITE (6,222)

C

WRITE (6,*) '>>>TEMPERATURE DATA (K) (F)
WRITE (6,*)
WRITE (6,*)
WRITE (6,*)
WRITE (6,*) 'COMPRESSOR INLET ',RTEMP(1), FAHR(RTEMP(1))
WRITE (6,*)
WRITE (6,*) 'COMPRESSOR DISCHARGE', Y(37), FAHR(Y(37))
WRITE (6,*)
WRITE (6,*) 'INTERCOOLER OUTLET ', Y(38), FAHR(Y(38))
WRITE (6,*)
WRITE (6,*) 'INTAKE MANIFOLD ', Y(39), FAHR(Y(39))
WRITE (6,*)
WRITE (6,*) 'ENGINE EXHAUST ', AVREXT,FAHR(AVREXT)
WRITE (6,*)
WRITE (6,*)
WRITE (6,*) 'TURBINE INLET ', Y(40), FAHR(Y(40))
WRITE (6,*) 'TURBINE EXHAUST ', Y(41), FAHR(Y(41))
WRITE (6,*)
IF (.NOT.POWER) GO TO 475
WRITE (6,*) 'POWER TURBINE INLET ', Y(60), FAHR(Y(60))
WRITE (6,*)
WRITE (6,*) 'P. TURBINE EXHAUST ', Y(42), FAHR(Y(42))
WRITE (6,*)
WRITE (6,*)
475 WRITE (6,*)
WRITE (6,222)

WRITE (6,*) ' >>>>> INTERCOOLER DATA'
WRITE (6,*)
WRITE (6,*) 'INTERCOOLER EFFECTIVENESS ', Y(43)
WRITE (6,*)
WRITE (6,*) 'COOLANT INLET TEMPERATURE ', HI(2)
WRITE (6,*)
WRITE (6,*) 'INTERCOOLER "A*U" (W/K) ', HI(5)
WRITE (6,*)
WRITE (6,111)
IF (.NOT.POWER) GO TO 477
WRITE (6,*) ' >>>> TURBOCHARGER DATA COMPRESSOR TURBINE ',
& ' P. TURBINE'
WRITE (6,*)
WRITE (6,*) 'MAP FLOW (LB/MIN) ', Y(44), Y(45), Y(53)
WRITE (6,*)
WRITE (6,*) 'MAP SPEED (KRPM) ', Y(46), Y(47), Y(48)
WRITE (6,*)
WRITE (6,*) 'PRESSURE RATIOS: ', APCD/PINLET, Y(49), APPD/APPTD
WRITE (6,*)
WRITE (6,*) 'EFFICIENCIES: ', Y(50), Y(51), Y(52)
WRITE (6,*)
GO TO 478

WRITE (6,*) ' >>>> TURBOCHARGER DATA COMPRESSOR TURBINE '
WRITE (6,*)
WRITE (6,*)
WRITE (6,*) 'MAP FLOW (LB/MIN) ', Y(44), Y(45)
WRITE (6,*)
WRITE (6,*) 'MAP SPEED (KRPM) ', Y(46), Y(47)
WRITE (6,*)
WRITE (6,*) 'PRESSURE RATIOS: ', APCD/PINLET, Y(49)
WRITE (6,*)
WRITE (6,*) 'EFFICIENCIES: ', Y(50), Y(51)
WRITE (6,*)
GO TO 478

WRITE (6,222)

WRITE (6,5959) TSS(1), TSS(2), TCW
WRITE (6,222)

OUTPUT EXHAUST MANIFOLD RESULTS:
IF (EXSUB) CALL EXWRIT(NEQN, DY, 2)

CALCULATE MASS REMAINING IN CYLINDER:
CALL THERMO (Y(11), Y(12), Y(20), HFINAL, CSUBP, CSUBT, CSUBF,
& RHO, DRHODT, DRHODP, DRHODF, GAMMA, MW, ADUMY, BDUMY, CDUMY)
CALL CSAVDV (TIVO, ACW, VOLUME, DVT)
MFINAL = RHO * VOLUME

C
CALL CSAVDV (TIVO, ACW, VOLUME, DVT)
MFINAL = RHO * VOLUME

C
CALCULATE NET AND GROSS THERMAL EFFICIENCIES:
THREFN = 100. * DY(16)/(FMIN * QLOWER)
THREFG = 100. * WORKCE/(FMIN * QLOWER)
HEATX = 100. * (DY(8) + DY(9) + DY(10))/(FMIN * QLOWER)
ZPMEP = (WORKI + WORKE)/DVOLUM
ZIMEP = WORKCE / DVOLUM
ZISFC = MIL * 3600. * FMIN/WORKCE

C
CALCULATE MEAN PISTON SPEED IN FPM. THEN, CALCULATE
FRICITION MEP BASED ON MILLINGTON & HARTLES CORRELATION:
VFPFM = 2. * STROKE / 0.3048 * ERPM
CFR1 = CMRTIO - 4.
ZPMEP = (CFR1 + CFR2 * (ERPM/1000.) +
& CFR3 * (VFPFM/1000.)**2 ) * PSIPA
FWORK = ZPMEP / DVOLUM
BWORK = DY(16) - FWORK
THREFB = 100. * BWORK / (FMIN * QLOWER)
ZBMEP = ZIMEP + ZPMEP - ZMEP
ZBSFC = MIL * 3600. * FMIN / BWORK

C
ENERGY BALANCE:
CYHSTA = HSTART * MSTART
CYHIN = (HM(1) * DY(1)) + (FMIN * HFORM)
CYHEX = DY(17)
CYHEAT = HEATI + HEATCE + HEATE
CYWORK = WORKI + WORKCE + WORKE
CYHFIN = HFINAL * MFINAL
DECYCL = CYHFIN + CYHEX + CYHEAT + CYWORK - CYHIN - CYHSTA
DEONHI = 100. * DECYCL /(CYHIN + CYHSTA)
DEONQ = 100. * DECYCL /(FMIN * QLOWER)
WRITE (6,111)
WRITE (6,5910)
WRITE (6,222)
WRITE (6,5920) VOLEFI, VOLEFA, ZPMEP/ATPA, ZPMEP/PSIPA,
& ZIMEP/ATPA, ZIMEP/PSIPA, ZPMEP/ATPA, ZIMEP/PSIPA,
& ZBMEP/ATPA, ZBMEP/PSIPA, ZISFC, ZISFC*CSFC
WRITE (6,5921) ZBSFC, ZBSFC*CSFC,
& THREFG, THREFN, THREFB, HEATX, AVREXT
WRITE (6,222)

C
ENWORK = BWORK * FLOAT(ICYL)
ENHEAT = FLOAT(ICYL) * FMIN * QLOWER
PTWORK = DY(30) + PTTEF
ENBHP = ENWORK/KIL * ERPM/60. / 2.
PTBHP = PTWORK/KIL * ERPM/60. / 2.
TBSFC = MIL* 3600. * FLOAT(ICYL) * FMIN / (ENWORK + PTWORK)
BTHEFF = 100. * (ENWORK + PTWORK) / ENHEAT
WRITE (6,111)
WRITE (6,5922)
WRITE (6,222)
WRITE (6,5923) ENWORK/KIL, PTWORK/KIL, ENHEAT/KIL,
& ENBHP, ENBHP/HPKW, PTBHP, PTBHP/HPKW,
& TBSFC, TBSFC*CSFC, BTHEFF

C
WRITE (6,111)
WRITE (6,5939)
WRITE (6,5946) MSTART*KIL, ZMAST*KIL, AMIN*KIL, FMIN*KIL
WRITE (6,222)
WRITE (6,5960)
WRITE (6,5961) HEAT/KIL, WORK/KIL
WRITE (6,5962) HEAT/C/KIL, WORK/C/KIL
WRITE (6,5963) HEAT/C/KIL, WORK/C/KIL
WRITE (6,5964) HEAT/KIL, WORK/KIL
WRITE (6,5960) CYHST/KIL, CYHIN/KIL, CYHEX/KIL,
& CYHEAT/KIL, CYWORK/KIL, BWORK/KIL, CYHFIN/KIL,
& DECYCL/KIL, DEONHI, DEONQ
WRITE (6,222)
WRITE (6,5989)
WRITE (6,5990) DELMAS - (DY(1) + FMIN - DY(2)) * KIL
DELMAO = DELMAS / DY(2) / 10.
WRITE (6,5997)
WRITE (6,5998) DY(1)*KIL, FMIN*KIL, DY(2)*KIL
WRITE (6,5999) DELMAS, DELMAO
WRITE (6,222)
WRITE (6,5991)
WRITE (6,5992) TIDEL,TIGN,DTBRN,ALPHA,CSP1,CSP2,CSD1,CSD2,CFACTR
WRITE (6,222)
WRITE (6,*)' AVERAGE NUMBER OF CALLS TO DIFEQ/CYCLE THIS RUN: ',
& IDIFCT/ITERAS
CT = FLOAT(ILOOPC)/FLOAT(IDIFCT)
IF (.NOT.EXSUB) WRITE (6,*)' AVERAGE TIMES THRU LOOP FOR EXHAUST',
& ' LOOP PER CALL TO DIFEQ: ', CT
CHECK FOR END OF CURRENT PROGRAM RUN
IF(SPTEMP .OR. .NOT. TRANS) GO TO 616
IF(ITERAS.GT.ISTEDY) GO TO 612
SET-UP FOR TRANSIENT WALL TEMPERATURE CALCULATION:
CLOSE(Unit=6)
OPEN (Unit=6, File = 'OUT2.DAT', Status = 'NEW')
OPEN (Unit=36, File = 'FINP.DAT', Status = 'NEW')
OPEN (Unit=37, File = 'FINH.DAT', Status = 'NEW')
IF (.NOT.EXSUB) GOTO 7811
OPEN (Unit=38, File = 'PORT.DAT', Status = 'NEW')
OPEN (Unit=39, File = 'RUNNER.DAT', Status = 'NEW')
OPEN (Unit=40, File = 'PLENUM.DAT', Status = 'NEW')
7811 ISTEDY = ITERAS
C INITIATE TRANSIENT PROFILES TO ZERO
C AND STEADY-STATE WALL HEAT FLUX TO CURRENT VALUES:
DO 7838 I = 1, NTMP
QSS(I) = QAVE(I)
DO 7838 J = 1, NPLA(I)
  DO 7838 K = 1, INNODE(I,J)
    PTW(I,J,K) = 0.
7838 CONTINUE
C BEGIN ITERATIONS FOR TRANSIENT TEMPERATURE CALCS
GO TO 468
C
880 FORMAT (///,1X,18X,'START OF ITERATION #'+I2,2X,'OF '+I2,
   & ' ALLOWED',///)
882 FORMAT (1H ,2X,'CA = '+F6.2,10X,'P = '+F10.5,28X,'IFG = '+I2)
891 FORMAT (///(' VOLUMETRIC EFFICIENCY = '+1F5.1,' %')///)
111 FORMAT (1X,'----------------------------------------------------------'
   & '----------------------------------------------------------'
   & '----------------------------------------------------------',/
   & '----------------------------------------------------------
   & '----------------------------------------------------------',/
112 FORMAT (1H)
222 FORMAT (1X,'----------------------------------------------------------'
   & '----------------------------------------------------------'
   & '----------------------------------------------------------',/
   & '----------------------------------------------------------
   & '----------------------------------------------------------',/
2899 FORMAT (///,1H,35X,'M.I.T. TURBOCHARGED DIESEL ENGINE',
   & ' CYCLE SIMULATION',///)
2900 FORMAT (///,1H,35X,'M.I.T. TURBOCOMPOUND DIESEL ENGINE',
   & ' CYCLE SIMULATION',///)
2901 FORMAT (/,' >>>>> INPUT DATA',/) 2914 FORMAT (/,' >>>>> OPERATING MODE',/) 2902 FORMAT (/,' >>>>> OPERATING CONDITIONS',/)
2903 FORMAT (/,' FUEL USED IS DIESEL #2') 2904 FORMAT (/,' FUEL USED IS ISO-OCTANE') 2905 FORMAT (/,' SPECIFIED IGNITION DELAY OPTION') 2906 FORMAT (/,' PREDICTED IGNITION DELAY OPTION') 2907 FORMAT (/,' SPECIFIED WALL TEMPERATURE OPTION') 2908 FORMAT (/,' PREDICTED QUASI-STEADY WALL TEMPERATURES') 2909 FORMAT (/,' PREDICTED TRANSIENT WALL TEMPERATURES') 2910 FORMAT (/,' ANNAND RADIATION MODEL') 2911 FORMAT (/,' FLAME RADIATION MODEL') 2931 FORMAT (/,' EXHAUST MANIFOLD SUB-CONTROL VOLUME MODEL') 2932 FORMAT (/,' EXHAUST MANIFOLD SINGLE FILLING & EMPTYING PLENUM') 2913 FORMAT (/,' ENGINE SPEED - ERPM = '+F7.1, ' RPM',
   & ' INJECTION TIMING - TINJ = '+F7.1, ' DEG CA',
   & ' FUEL INJECTED /CYL /CYCLE - FMIN = '+F7.4, ' G',
   & ' TOTAL FUELING RATE = '+F7.4, ' LB/MIN',
   & ' COMPRESSOR INLET PRESSURE -PINLET= '+F10.4, ' ATM',
   & ' COMPRESSOR INLET TEMPERATURE -RTEMP()= '+F8.2, ' K',
   & ' ATMOSPHERIC PRESSURE - PATM = '+F10.4, ' ATM',
   & ' ATMOSPHERIC TEMPERATURE - TATM = '+F8.2, ' K',/)
2919 FORMAT (/,' NUMBER OF CYLINDERS -ICYL = '+I4,/
   & ' CYLINDER BORE = '+F9.3, ' CM',
   & ' CRANKSHAFT STROKE = '+F9.3, ' CM',
   & ' CONNECTING ROD LENGTH - CONRL = '+F9.3, ' CM',/)
1981 & */ ** COMPRESSION RATIO = 'F9.3 /
1982 & */ ** DISPLACED VOLUME = 'F9.3 ' CC',
1983 & */ ** CLEARANCE VOLUME - CLVTDC = 'F9.3 ' CC',
1984 & */ ** ENGINE DISPLACEMENT = 'F9.3 ' LT', /
1985 & */ ** FRICTION CONSTANT 2 - CFR2 = 'F9.3 ,
1986 & */ ** FRICTION CONSTANT 3 - CFR3 = 'F9.3 ,
1987 & */ ** INTAKE VALVE OPENS - TIVO = 'F7.1 , ' DEG CA',
1988 & */ ** INTAKE VALVE CLOSES - TIVC = 'F7.1 , ' DEG CA',
1989 & */ ** EXHAUST VALVE OPENS - TEVO = 'F7.1 , ' DEG CA',
1990 & */ ** EXHAUST VALVE CLOSES - TEVC = 'F7.1 , ' DEG CA')
1991 C
1992 2915 FORMAT('/,' HEAT TRANSFER CONSTANT(ChAMBER) - CONHT = 'F10.4 ,
1993 & */.* HEAT TRANSFER CONSTANT(INT MAN) - ECONHT(1) = 'F10.4 ,
1994 2916 FORMAT('/,' HEAT TRANSFER CONSTANT(CHAMBER) - CONHT = 'F10.4 ,
1995 & */.* HEAT TRANSFER CONSTANT(INT MAN) - ECONHT(1) = 'F10.4 ,
1996 & */.* HEAT TRANSFER CONSTANT(EXH MAN) - ECONHT(2) = 'F10.4 ,
1997 2912 FORMAT('/,' HEAT TRANSFER CONSTANT(C. PIPE) - ECONHT(3) = 'F10.4 ,
1998 2917 FORMAT('/,' HEAT TRANSFER EXPONENT - EXPHT = 'F10.4 ,
1999 & */.* INITIAL PISTON TEMPERATURE - TPSTON = 'F9.2 , ' K ,
2000 & */.* INITIAL CYL HEAD TEMPERATURE - THEAD = 'F9.2 , ' K ,
2001 & */.* INITIAL CYL WALL TEMPERATURE - TCW = 'F9.2 , ' K ,
2002 & */.* INT. MANIFOLD WALL TEMPERATURE - ETWALL(1) = 'F9.2 , ' K ,
2003 & */.* TURBULENT DISSIPATION CONSTANT - CBETA = 'F10.4 ,
2004 2918 FORMAT('/,' HEAT TRANSFER EXPONENT - EXPHT = 'F10.4 ,
2005 & /* INITIAL PISTON TEMPERATURE - TPSTON = 'F9.2 , ' K ,
2006 & /* INITIAL CYL HEAD TEMPERATURE - THEAD = 'F9.2 , ' K ,
2007 & /* INITIAL CYL WALL TEMPERATURE - TCW = 'F9.2 , ' K ,
2008 & /* INT. MANIFOLD WALL TEMPERATURE - ECONHT(1) = 'F9.2 , ' K ,
2009 & */.* EXH. MANIFOLD WALL TEMPERATURE - ECONHT(2) = 'F9.2 , ' K ,
2010 & */.* CONN. PIPE WALL TEMPERATURE - ECONHT(3) = 'F9.2 , ' K ,
2011 & */.* TURBULENT DISSIPATION CONSTANT - CBETA = 'F10.4 ,
2012 C
2013 2930 FORMAT('/,' RADIATION CONSTANT (ANNAND) = 'F10.4 ,
2014 C
2015 2928 FORMAT('/,' >>>>> COMPUTATIONAL PARAMETERS' , /
2016 2921 FORMAT('/,' MAXIMUM # OF ITERATIONS - MAXITS = 'I4 , /
2017 & /* OUTPUT AT ITERATION # = 'I4 , /
2018 & /* TPRINT = 'F9.2 ,
2019 & /* TSSCREEN = 'F9.2 , /
2020 & /* CIINTG = 'F13.6 ,
2021 & /* CIINTG = 'F13.6 ,
2022 & /* CBINTG = 'F13.6 ,
2023 & /* CEINTG = 'F13.6 , /
2024 & /* AREROT = 'F13.6)
2025 2922 FORMAT('/,' DELM1 = 'F13.6 ,
2026 & /* DELM2 = 'F13.6 ,
2027 & /* DELQ1 = 'F13.6 ,
2028 & /* DELQ2 = 'F13.6 , /
2029 C
2030 2998 FORMAT('/,' >>>>> OUTPUT DATA' , /
2031 2999 FORMAT('/,' >>>>> ENGINE CRANK-ANGLE BY CRANK-ANGLE RESULTS' , /
2032 C
2033 3110 FORMAT('/'(1X,' >>>>> START OF INTAKE PROCESS ')/)
2034 C
2035 3111 FORMAT('/'(1X,' >>>>> START OF COMPRESSION PROCESS ')/)
2036 C
2037 3112 FORMAT('/'(1X,' >>>>> START OF COMBUSTION AND EXPANSION PROCESSES
2038 & ')/)
2039 C
2040 3113 FORMAT('/'(1X,' >>>>> START OF EXHAUST PROCESS ')/)
3596 FORMAT ((4X,'CA',8X,'P',8X,'TEMP',8X,'MIN',8X,'MEX',8X, 
  & 'PHI',8X,'PIM',8X,'TIM',8X,'PEM',8X,'TEM',4X, 'IFG', 
  & 5X,'SPEED')/ 
  & (2X,'(DEG)',5X,'(ATM)',7X,'(K)',8X,'(G)',8X,'(G)', 
  & 8X,'(-)',7X,'(ATM)',7X,'(K)',7X,'(ATM)',7X,'(K)', 
  & 4X,'(-)',5X,'(KRPM)')) 
C
3597 FORMAT ((4X,'CA',7X,'P',9X,'TEMP',13X,'XB',15X, 'PHI', 
  & 8X,'PIM',8X,'TIM',8X,'PEM',8X,'TEM',4X,'IFG', 
  & 'SPEED')/ 
  & (2X,'(DEG)',4X,'(ATM)',8X,'(K)',13X,'(-)',14X,'(-)', 
  & 7X,'(ATM)',7X,'(K)',7X,'(ATM)',7X,'(K)', 
  & 4X,'(-)',5X,'(KRPM)')) 
C
3598 FORMAT ((4X,'CA',7X,'P',9X,'TEMP',38X,'THI',6X,'PIM',8X, 
  & 'TIM',8X,'PEM',8X,'TEM',4X,'IFG', 
  & 5X,'SPEED')/ 
  & (2X,'(DEG)',8X,'(ATM)',9X,'(K)',8X,'(G)',8X, 
  & 8X,'(-)',7X,'(ATM)',7X,'(K)',7X,'(ATM)',7X,'(K)', 
  & 4X,'(-)',5X,'(KRPM)')) 
C
3599 FORMAT ((4X,'CA',8X,'P',8X,'TEMP',19X,'MEX',8X,'PHI', 
  & 8X,'PIM',8X,'TIM',8X,'PEM',8X,'TEM',4X,'IFG', 
  & 'SPEED')/ 
  & (2X,'(DEG)',5X,'(ATM)',7X,'(K)',19X,'(G)',8X,'(-)', 
  & 7X,'(ATM)',7X,'(K)',7X,'(ATM)',7X,'(K)', 
  & 4X,'(-)',5X,'(KRPM)')) 
C
C
C
5910 FORMAT ///(3X,>>>>> DIESEL ENGINE PERFORMANCE RESULTS ')/// 
C
5920 FORMAT ///(' VOLUMETRIC EFFICIENCY; (%) ')/ 
  & (' BASED ON: INTAKE / ATM ',2(F9.1))/ 
  & (' PUMPING MEAN EFF. PRESSURE ')/ 
  & (ATM, PSI) : PMEP ',2(F10.2))/ 
  & (' GROSS IND. MEAN EFF. PRESSURE ')/ 
  & (ATM, PSI) : IMEP ',2(F10.2))/ 
  & (' FRICITION MEAN EFF. PRESSURE ')/ 
  & (ATM, PSI) : FMEE ',2(F10.2))/ 
  & (' BRAKE MEAN EFF. PRESSURE ')/ 
  & (ATM, PSI) : BMEE ',2(F10.2))/ 
  & (' GROSS INDICATED S.F.C. ')/ 
  & (G/KW/HR, LB/HP/HR) : ISFC ',2(F10.3))/ 
  & (' BRAKE S.F.C. ')/ 
  & (G/KW/HR, LB/HP/HR) : BSFC ',2(F10.3))/ 
  & (' GROSS INDICATED THERMAL ')/ 
  & (' EFFICIENCY; (%) ',F7.1)/ 
  & (' NET INDICATED THERMAL ')/ 
  & (' EFFICIENCY; (%) ',F7.1)/ 
  & (' CYLINDER BRAKE THERMAL ')/ 
  & (' EFFICIENCY; (%) ',F7.1)/ 
  & (' (CYL. HEAT TRANSFER PER CYCLE)/ ')/ 
  & (' MASS OF FUEL TIMES LHV) : (X) ',F7.1)/ 
  & (' MEAN EXHAUST ')/ 
  & (' TEMPERATURE; (K) ',F7.1)/ 
5922 FORMAT ///(3X,>>>>> TOTAL SYSTEM PERFORMANCE RESULTS ')///
5923 FORMAT (/(' DIESEL WORK '),/)
5924 & (' PER CYCLE; (KJ) ',F10.6)//
5925 & (' POWER TURBINE WORK '),/ ',F10.6)//
5926 & (' PER CYCLE (KJ) ',F10.6)//
5927 & (' TOTAL HEAT INPUT '),/ ',F10.6)//
5928 & (' PER CYCLE (KJ) ',F10.6)//
5929 & (' DIESEL BRAKE POWER '),/ '
5930 & (' (KW, HP) : ENTHP ',2(F10.1))///
5931 & (' POWER TRB. BRAKE POWER '),/ '
5932 & (' (KW, HP) : FTBHP ',2(F10.1))///
5933 & (' OVERALL BRAKE S.F.C. '),/ '
5934 & (' (G/KW/HR, LB/HP/HR) : BSFC ',2(F10.3))///
5935 & (' OVERALL BRAKE THERMAL '),/ '
5936 & (' EFFICIENCY; (%) ',F7.1}//

C

5939 FORMAT (/,' >>>> CYLINDER MASS SUMMARY'/)
5940 FORMAT (/ (' MASS IN CYLINDER AT TIVO = ',F8.5, ' G')//
5941 & (' MASS IN CYLINDER AT TIVC = ',F8.5, ' G')//
5942 & (' MASS OF AIR INDUCED = ',F8.5, ' G')//
5943 & (' MASS OF FUEL INJECTED = ',F8.5, ' G')//

C

5959 FORMAT (/,' >>>> FINAL STEADY-STATE CYLINDER TEMPERATURES'/)
5960 FORMAT (/ (' PISTON TEMPERATURE (K) = ',F4.0, '/
5961 & (' HEAD TEMPERATURE (K) = ',F4.0, '/
5962 & (' LINER TEMPERATURE (K) = ',F4.0, ' )

C

5965 FORMAT (/,' >>>> CYLINDER HEAT & WORK TRANSFERS'/)
5966 FORMAT (/ (' HEAT = ',F10.6, ' KJ', ' (TIVO - 180)')//
5967 & (' WORK = ',F10.6, ' KJ')//

C

5969 FORMAT (/,' >>>> CYLINDER ENTHALPY BALANCE'/)
5970 FORMAT (/ (' INITIAL ENTHALPY /CYL / CYCLE = ',F9.5, ' KJ')//
5971 & (' TOTAL ENTHALPY IN /CYL / CYCLE = ',F9.5, ' KJ')//
5972 & (' TOTAL ENTHALPY OUT/CYL/CYCLE = ',F9.5, ' KJ')//
5973 & (' TOTAL HEAT LOSS / CYL / CYCLE = ',F9.5, ' KJ')//
5974 & (' TOTAL HEAT OUTPUT / CYL / CYCLE = ',F9.5, ' KJ')//
5975 & (' IND. WORK OUTPUT / CYL / CYCLE = ',F9.5, ' KJ')//
5976 & (' BRAKE WORK OUTPUT /CYL/CYCLE = ',F9.5, ' KJ')//
5977 & (' RESIDUAL ENTHALPY /CYL/CYCLE = ',F9.5, ' KJ')//
5978 & (' NET ENERGY GAIN / CYL/CYCLE = ',F9.5, ' KJ')//
5979 & (' ENERGY GAIN)/(H@IVC) = ',F9.5, ' %')//
5980 & (' ENERGY GAIN)/(W/FUEL*LHV) = ',F9.5, ' %')//

C

5991 FORMAT (/,' >>>> COMBUSTION SUMMARY'/)
5992 FORMAT (/ (' IGNITION DELAY PERIOD = ',F8.3, ' DEG CA')//
5993 & (' IGNITION TIMING = ',F8.3, ' DEG CA')//
5994 & (' BURN DURATION = ',F8.3, ' DEG CA')//
5995 & (' WEIGHTING FACTOR = ',F8.3, ' DEG CA')//
5996 & (' PREMIXED CONSTANT 1 = ',F8.3, ' DEG CA')//
5997 & (' PREMIXED CONSTANT 2 = ',F8.3, ' DEG CA')//
5998 & (' DIFFUSION CONSTANT 1 = ',F8.3, ' DEG CA')//
5999 & (' DIFFUSION CONSTANT 2 = ',F8.3, ' DEG CA')//
& //.' CFACTR = ',F8.3,/
C
5993 FORMAT ('/', >>>> OVERALL MASS BALANCE',/)
5994 FORMAT ('/', TOTAL COMPRESSOR MASS FLOW (G/CYCLE) = ',F7.4,
5995 FORMAT ('/', TOTAL FUEL FLOW (G/CYCLE) = ',F7.4,
5996 FORMAT ('/', TOTAL TURBINE MASS FLOW (G/CYCLE) = ',F7.4)
5997 FORMAT ('/', >>>> CYLINDER MASS BALANCE',/)
5998 FORMAT ('/', TOTAL INTAKE VALVE MASS FLOW (G/CYCLE) = ',F7.4,
5999 FORMAT ('/', TOTAL EXHAUST VALVE MASS FLOW (G/CYCLE) = ',F7.4)
6000 FORMAT ('/', >>>> NORMALIZED CHANGE IN MASS (% OF TURBINE MASS',
6001 FORMAT ('/', ',F8.4, %',//' /// THE CHANGE IN MASS','/// AND THE CHANGE IN MASS IN THE ENGINE IF THE','/// PROGRAM HAS NOT CONVERGED EXACTLY')
6111 FORMAT ('///', >>>> TURBULENT FLOW MODEL')
6112 FORMAT ('///', >>>> HEAT TRANSFER DATA')
6113 FORMAT ('///', >>>> RADIATIVE HEAT TRANSFER DATA')
6114 FORMAT ('///', '/// WALL CONDUCTION MODELS')
6115 FORMAT ('///', '/// PISTON WALL STRUCTURE')
6116 FORMAT ('///', '/// CYL. HEAD WALL STRUCTURE')
6117 FORMAT ('///', '/// CYL. LINER WALL STRUCTURE')
6118 FORMAT ('///', '/// LAYER', '3X,3(I4,10X))
6119 FORMAT ('///', '/// INSIDE DIAMETER (M)', 'F10.5)
6120 FORMAT ('///', '/// THICKNESS (M)', '3(F10.5, 5X))
6121 FORMAT ('///', '/// THERMAL CONDUCTIVITY (W/M/K)', '3(F10.3, 5X))
6122 FORMAT ('///', '/// THERMAL DIFFUSIVITY (M2/SEC)', '3(E10.3, 5X))
FORMAT (/, 'NUMBER OF NODES', 3X, 3(I4, 10X))
FORMAT (/, 'NODES WITHIN SKIN DEPTH', 3X, I4)
FORMAT (/, 'COURANT NUMBER', 3(F10.5, 5X))
FORMAT (/, 'OUTSIDE WALL TEMPERATURE (K)', F10.1,)
& /, 'OVERALL CONDUCTIVITY (W/M/K) ', F10.1,/
& /, 'AMBIENT TEMPERATURE (K) ', F10.1,/
& /, 'OUTSIDE HEAT TR. COEFF.(W/M2/K)', F10.1,/
& /, 'OVERALL HEAT TR. COEFF.(W/M2)', F10.1,/
9111 FORMAT (///, 'TRANSIENT TEMPERATURE PROFILES WITHIN PISTON', /
& /, 'UP TO FIRST 15 NODES IN FIRST AND SECOND', /
& ' LAYERS', /, 5X, A)
9112 FORMAT (///, 'TRANSIENT TEMPERATURE PROFILES WITHIN HEAD', /
& /, 'UP TO FIRST 15 NODES IN FIRST AND SECOND', /
& ' LAYERS', /, 5X, A)
9113 FORMAT (///, 'TRANSIENT TEMPERATURE PROFILES IN EXHAUST', /
& /, 'UP TO FIRST 15 NODES IN FIRST AND SECOND', /
& ' LAYERS', /, 5X, A)
612 CONTINUE
CLOSE(UNIT=38)
CLOSE(UNIT=37)
IF (EXSUB) THEN
CLOSE(UNIT=38)
CLOSE(UNIT=39)
CLOSE(UNIT=40)
ENDIF
616 CLOSE(UNIT=6)
STOP
END
SUBROUTINE CMBSTN

PURPOSE
CALCULATES THE TIME RATE OF CHANGE OF PRESSURE, TEMPERATURE, FUEL EQUIVALENCE RATIO, MASS, HEAT TRANSFER, WORK TRANSFER, MEAN KINETIC ENERGY, AND TURBULENT KINETIC ENERGY IN THE MASTER CYLINDER DURING COMBUSTION.

USAGE
CALL CMBSTN (NEQN, DT, DY, DYP)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>NEQN</td>
<td>YES</td>
<td>NO</td>
<td>NUMBER OF EQUATIONS TO BE INTEGRATED</td>
</tr>
<tr>
<td>DT</td>
<td>YES</td>
<td>NO</td>
<td>TIME (DEG)</td>
</tr>
<tr>
<td>DY(1)</td>
<td>YES</td>
<td>NO</td>
<td>MASS INDUCED INTO CHAMBER THROUGH INTAKE VALVE (KG)</td>
</tr>
<tr>
<td>DY(2)</td>
<td>YES</td>
<td>NO</td>
<td>MASS EXHAUSTED FROM CHAMBER THROUGH EXHAUST VALVE (KG)</td>
</tr>
<tr>
<td>DY(4)</td>
<td>YES</td>
<td>NO</td>
<td>FUEL MASS BURNED (-)</td>
</tr>
<tr>
<td>DY(6)</td>
<td>YES</td>
<td>NO</td>
<td>MEAN KINETIC ENERGY IN CHAMBER (J)</td>
</tr>
<tr>
<td>DY(7)</td>
<td>YES</td>
<td>NO</td>
<td>TURBULENT KINETIC ENERGY IN CHAMBER (J)</td>
</tr>
<tr>
<td>DYP(1)</td>
<td>NO</td>
<td>YES</td>
<td>RATE AT WHICH MASS IS INDUCED THROUGH THE INTAKE VALVE (KG/DEG)</td>
</tr>
<tr>
<td>DYP(2)</td>
<td>NO</td>
<td>YES</td>
<td>RATE AT WHICH MASS IS EXHAUSTED THROUGH THE EXHAUST VALVE (KG/DEG)</td>
</tr>
<tr>
<td>DYP(4)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF BURNING OF FUEL MASS FRESH CHARGE (1/DEG)</td>
</tr>
<tr>
<td>DYP(6)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF MEAN KINETIC ENERGY (J/DEG)</td>
</tr>
<tr>
<td>DYP(7)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF TURBULENT KINETIC ENERGY (J/DEG)</td>
</tr>
<tr>
<td>DYP(8)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF HEAT TRANSFER - CYLINDER HEAD (J/DEG)</td>
</tr>
<tr>
<td>DYP(9)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF HEAT TRANSFER - PISTON TOP (J/DEG)</td>
</tr>
<tr>
<td>DYP(10)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF HEAT TRANSFER - CYLINDER WALL (J/DEG)</td>
</tr>
<tr>
<td>DYP(11)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF CYLINDER TEMPERATURE (K/DEG)</td>
</tr>
<tr>
<td>DYP(12)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF CYLINDER PRESSURE (PA/DEG)</td>
</tr>
<tr>
<td>DYP(16)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF TOTAL WORK TRANSFER (J/DEG)</td>
</tr>
<tr>
<td>DYP(20)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF BURNED FUEL FRACTION (-)</td>
</tr>
</tbody>
</table>
FUEL FRACTION (1/DEG)

REMARKS
UNITS CHANGED TO S.I.

NOTE THAT RATES ARE IN TERMS OF REAL TIME UNTIL THE END
OF THE SUBROUTINE WHERE THEY ARE CONVERTED TO CRANK ANGLE
TIME (/DEGREE)

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
THERMO TRANSP CSAVDV DIFEQ FLAME

METHDO
SEE NASA REPORT

WRITTEN BY D. N. ASSANIS AND S. G. Poulos
EDITED BY D. N. ASSANIS AND R. M. Frank

SUBROUTINE CMBSTN (NEQN, DT, DY, DYP)
LOGICAL ANNAND
REAL*8 DT, DY(NEQN), DYP(NEQN)
REAL MDOTFU, MFUEL, MDORE, MDOIF, MDOTOT
REAL MW, KINVIS, MASS, MDOT, MDOTFR, MSTART, MACRSC
PARAMETER (PI = 3.141592654, SIGMA = 5.67E-8)

COMMON/ANNAND/ ANNAND
COMMON/RADN/ TCHTR, TRHTR, THTR, CHTRAP, CHTRAH, CHTRAW,
& RHTRAP, RHTRAH, RHTRAW
COMMON/FLAME/RTAIR, RPAIR, RTFLAM, RTRAD, EMIS
COMMON/PROFILE/ DTBRN, ALPHA, CSP1, CSP2, CSD1, CSD2
COMMON/CRAD/ CRAD
COMMON/FRATE/ FRATE
COMMON/PROFIL/ DTBRN, ALPHA, CSP1, CSP2, CSD1, CSD2
COMMON/TIMES/ TIVO, TEVC, TIVC, TINJ, TIGN, TEVO
COMMON/FUEL/ FUELTP, CX, HY, DEL, PSI, AFRASST, QLOWER, HFORM
COMMON/BURN/ FMIN
COMMON/HTCOTG/ HTCO(10), TGAS(10)

SAVE

DO 10 I = 1, 20
10 DYP(I) = 0.0
T = DT

FIND THERMODYNAMIC AND TRANSPORT PROPERTIES IN CYLINDER
FR = DY(20)
TCYL = DY(11)
PCYL = DY(12)
MFUEL = DY(4) * FMIN
CALL THERMO ( TCYL, PCYL, FR, H, CSUBP, CSUBT, CSUBF, & RHO, DRHODT, DRHODP, DRHODF, GAMMA, MW, ADUMY, BDUMY, CDUMY)
CALL TRANSP (TCYL, FR, GAMMA, CSUBP, DYNVIS, THRCON)
KINVIS = DYNVIS/RHO
MASS = MSTART + DY(1) - DY(2) + MFUEL

C FIND SURFACE AREAS AND VOLUME OF CHAMBER
CALL CSAVDV (T, ACW, VOLUME, DVDT)

TONDTB = (T - TIGN)/DTBRN
IF (TONDTB .GT. 1.0) TONDTS = 1.0
IF (TONDTB .GT. .15) GO TO 11

MDOPRE = CSP1 * CSP2 * TONDTB**(CSP1 -1.) * (1. - TONDTB**CSP1)**
& (CSP2 -1.)
GO TO 13

MDOPRE = 0.0

MDODIF = CSD1*CSD2 *(TONDTB**(CSD2-1.)) * EXP(-CSD1*TONDTB**CSD2)
MDOTOT= ALPHA * MDOPRE + (1. - ALPHA) * MDODIF
MDOTFU= DY(4) + FMIN

FBRATE = DYP(4)

DYP(20)= MDOTFU/MASS * (1.- FR)
YP20 = DYP(20)

MACRSC = VOLUME/(3.1415 * BORE * BORE/4.)
IF (MACRSC.GE.(BORE/2)) MACRSC = BORE/2.
DYP(6) = -.3307 * CBETA/MACRSC * DY(6) * SQRT(DY(7)/MASS)

& -.5443 * DY(7)/MACRSC * SQRT(DY(7)/MASS) + APPL

CHARACTERISTIC VELOCITY IN CYLINDER; (M/SEC).
CONSTR = CONRL/STROKE
SINTH = SIN( T*PI/180. )
COSTH = COS( T*PI/180. )
VONVPM = ABS( PI * SINTH * ( 1. + COSTH/SQRT( 4. * CONSTR*CONSTR
& - SINTH*SINTH ) )/2. )
VPMEAN = STROKE/(180. * ESPD)
VPISTO = VPMEAN + VONVPM
VMKE = SQRT( 2. * DY(6)/MASS )
UPRIME = SQRT( .666667 * DY(7)/MASS )
CVHTRN = SQRT( 0.25*VPISTO*VPISTO + VMKE*VMKE + UPRIME*UPRIME )

CALCULATE HEAT TRANSFER RATES
HTRCOE = CONHT*( (CVHTRN+MACRSC/KINVIS)**EXPH ) * THRCON/MACRSC
CHTPAP = HTRCOE * (TCYL - TWALL(1))
CHTPAH = HTRCOE * (TCYL - TWALL(2))
CHTPAW = HTRCOE * (TCYL - TCW)

C IF(ANNAND) THEN
TRAD = TCYL
C SIGMA IS STEPHAN-BOLTZMAN CONSTANT = 5.67E-8 W/M**2/K**4
CONRAD = CRAD * SIGMA
RTRAD = TRAD
GO TO 87
ENDIF

C CALL FLAME (TCYL, PCYL, TAIR, TFLAME, TRAD)
EMIS = 0.9 - 0.9 * (T-TIGN)/(TEVO-TIGN)
IF (EMIS .LT. 0.) EMIS = 0.
CONRAD = EMIS * SIGMA
RTAIR = TAIR
RPAIR = PCYL
RTFLAME = TFLAME
RTRAD = TRAD

C 87 TPSTON = TWALL(1)
THEAD = TWALL(2)
TRAD4 = TRAD**4
RHTPAP = CONRAD * (TRAD4 - TPSTON**4)
RHTPAH = CONRAD * (TRAD4 - THEAD**4)
RHTPAW = CONRAD * (TRAD4 - TCW**4)

C CALCULATE EFFECTIVE LINEAR HEAT TRANSFER COEFFICIENTS
(CONVECTIVE + RADIATIVE)

C
TGAS(1) = TCYL
TGAS(2) = TCYL
IF(TCYL.EQ.TPSTON) GO TO 92
HTCO(1) = HTRCOE + CONRAD * (TCYL**3 + TCYL**2 * TPSTON +
& TCYL + TPSTON**2 + TPSTON**3) +
& CONRAD * (TRAD4 - TCYL**4) / (TCYL - TPSTON)
92 CONTINUE

IF(TCYL.EQ.THEAD) GO TO 93
HTCO(2) = HTRCOE + CONRAD * (TCYL**3 + TCYL**2 * THEAD +
& TCYL + THEAD**2 + THEAD**3) +
& CONRAD * (TRAD4 - TCYL**4) / (TCYL - THEAD)
93 CONTINUE

IF(TCYL.EQ.TCW) GO TO 94
HLIN3 = HTRCOE + CONRAD * (TCYL**3 + TCYL**2 * TCW +
& TCYL + TCW**2 + TCW**3) +
& CONRAD * (TRAD4 - TCYL**4) / (TCYL - TCW)
94 CONTINUE

C
HTGPI = HTC(1) * TCYL
HTGHD = HTC(2) * TCYL
HTGCW = HLIN3 * TCYL

C CALCULATE CONVECTIVE AND RADIATIVE HEAT TRANSFER RATES
TO EACH COMPONENT SURFACE:

C
CHTRAP = CHTPAP * APSTON
CHTRAH = CHTPAH * AHEAD
CHTRAW = CHTPAW * ACW

C
RHTRAP = RHTPAP * APSTON
RHTRAH = RHTPAH * AHEAD
RHTRAW = RHTPAW * ACW
CALCULATE COMBINED HEAT TRANSFER RATES:

I) TO EACH COMPONENT; II) TOTALS

HTPAPI = CHTPAP + RHTPAP
HTPAHD = CHTPAH + RHTPAH
HTPACW = CHTPAW + RHTPAW

II) TOTALS

RHTPAP = CHTPAP + HTRAHD + HTRACW
RHTPAH = CHTPAH + HTRAH + HTRAHD
RHTPAW = CHTPAW + HTRAH + HTRACW

CALCULATE RATES OF CHANGE OF TEMPERATURE AND PRESSURE IN THE CYLINDER. THEN CALCULATE RATE OF DOING WORK.

\[
\begin{align*}
30 \phi_{\text{dot}} &= A_F R / (1. - F_R) \cdot \text{DYP}(20) \\
\phi &= F_R \cdot A_F R \cdot (1. - F_R) \\
\text{DYP}(11) &= (B_{\text{DUMY}}/A_{\text{DUMY}}) \cdot ((M_{\text{DOTFU}}/M_{\text{ASS}}) \cdot (1. - H/B_{\text{DUMY}}) \\
&+ \phi_{\text{dot}} \cdot \text{DVDT/VOLUME} - C_{\text{DUMY}}/B_{\text{DUMY}} \cdot \phi_{\text{dot}} \\
&+ (M_{\text{DOTFU}} \cdot H_{\text{FORM}} - TH_{\text{TRAN}})/(B_{\text{DUMY}} \cdot M_{\text{ASS}})
\end{align*}
\]

\[
\begin{align*}
\text{YP11} &= \text{DYP}(11) \\
\text{DYP}(12) &= \phi_{\text{dot}} \cdot \text{DYP}(11) \\
\text{DYP}(13) &= \text{DYP}(11) \cdot DRHODT/RHO \\
&- \phi_{\text{dot}} \cdot DRHODF/RHO + M_{\text{DOTFU}}/M_{\text{ASS}} \\
\text{YP12} &= \text{DYP}(12) \\
\text{DYP}(15) &= \text{PCYL} \cdot \text{DVDT} \\
\text{DYP}(8) &= \text{HTRAPI} \\
\text{DYP}(9) &= \text{HTRAHD} \\
\text{DYP}(10) &= \text{HTRACW}
\end{align*}
\]

CONVERT ALL TIME DERIVATIVES TO RATE PER CRANK ANGLE DEGREE.

DO 40 I = 1, 20

\[
\begin{align*}
40 & \text{DYP}(I) = \text{DYP}(I) \cdot \text{ESPD}
\end{align*}
\]

CALL DIFEQ (NEQN, T, DY(21), DYP(21))

RETURN

END
SUBROUTINE CMPRES

PURPOSE
CALCULATES THE TIME RATE OF CHANGE OF PRESSURE, TEMPERATURE, FUEL EQUIVALENCE RATIO, MASS, HEAT TRANSFER, WORK TRANSFER, MEAN KINETIC ENERGY, AND TURBULENT KINETIC ENERGY IN THE MASTER CYLINDER DURING COMPRESSION. ALSO USED TO PREDICT THE LENGTH OF THE IGNITION DELAY PERIOD.

USAGE
CALL CMPRES (NEQN, DT, DY, DYP)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>NEQN</td>
<td>YES</td>
<td>NO</td>
<td>NUMBER OF EQUATIONS TO BE INTEGRATED</td>
</tr>
<tr>
<td>DT</td>
<td>YES</td>
<td>NO</td>
<td>TIME (DEG)</td>
</tr>
<tr>
<td>DY(1)</td>
<td>YES</td>
<td>NO</td>
<td>MASS INDUCED INTO CHAMBER THROUGH INTAKE VALVE (KG)</td>
</tr>
<tr>
<td>DY(2)</td>
<td>YES</td>
<td>NO</td>
<td>MASS EXHAUSTED FROM CHAMBER THROUGH EXHAUST VALVE (KG)</td>
</tr>
<tr>
<td>DY(6)</td>
<td>YES</td>
<td>NO</td>
<td>MEAN KINETIC ENERGY IN CHAMBER (J)</td>
</tr>
<tr>
<td>DY(7)</td>
<td>YES</td>
<td>NO</td>
<td>TURBULENT KINETIC ENERGY IN CHAMBER (J)</td>
</tr>
<tr>
<td>DY(8)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - PISTON TOP (J)</td>
</tr>
<tr>
<td>DY(9)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - CYLINDER HEAD (J)</td>
</tr>
<tr>
<td>DY(10)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - CYLINDER WALL (J)</td>
</tr>
<tr>
<td>DY(11)</td>
<td>YES</td>
<td>NO</td>
<td>CYLINDER TEMPERATURE (K)</td>
</tr>
<tr>
<td>DY(12)</td>
<td>YES</td>
<td>NO</td>
<td>CYLINDER PRESSURE (PA)</td>
</tr>
<tr>
<td>DY(13)</td>
<td>YES</td>
<td>NO</td>
<td>ELAPSED/PREDICTED IGNITION DELAY (-)</td>
</tr>
<tr>
<td>DY(16)</td>
<td>YES</td>
<td>NO</td>
<td>TOTAL WORK TRANSFER (J)</td>
</tr>
<tr>
<td>DYP(1)</td>
<td>NO</td>
<td>YES</td>
<td>RATE AT WHICH MASS IS INDUCED THROUGH THE INTAKE VALVE (KG/DEG)</td>
</tr>
<tr>
<td>DYP(2)</td>
<td>NO</td>
<td>YES</td>
<td>RATE AT WHICH MASS IS EXHAUSTED THROUGH THE EXHAUST VALVE (KG/DEG)</td>
</tr>
<tr>
<td>DYP(6)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF MEAN KINETIC ENERGY (J/DEG)</td>
</tr>
<tr>
<td>DYP(7)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF TURBULENT KINETIC ENERGY (J/DEG)</td>
</tr>
<tr>
<td>DYP(8)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF HEAT TRANSFER - CYLINDER HEAD (J/DEG)</td>
</tr>
<tr>
<td>DYP(9)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF HEAT TRANSFER - PISTON TOP (J/DEG)</td>
</tr>
<tr>
<td>DYP(10)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF HEAT TRANSFER - CYLINDER WALL (J/DEG)</td>
</tr>
<tr>
<td>DYP(11)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF CYLINDER TEMPERATURE (K/DEG)</td>
</tr>
<tr>
<td>DYP(12)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF CYLINDER PRESSURE (PA/DEG)</td>
</tr>
<tr>
<td>DYP(13)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF ELAPSED/PREDICTED IGNITION DELAY FRACTION (1/DEG)</td>
</tr>
<tr>
<td>DYP(16)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF TOTAL WORK TRANSFER (J/DEG)</td>
</tr>
</tbody>
</table>
REMARKS
UNITS CHANGED TO S.I.
REVISED DOCUMENTATION
NOTES THAT DERIVATIVE UNITS ARE PER REAL TIME UNTIL
END OF SUBROUTINE WHERE THEY ARE CONVERTED TO PER DEGREE
SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
THERMO TRANSP CSAVDV DIFEQ

METHOD
SEE REPORT
WRITTEN BY D. N. ASSANIS AND S. G. POULOS
EDITED BY D. N. ASSANIS
SUBROUTINE CMPRES (NEQN, DT, DY, DYP)

REAL*8 DT, DY(NEQN), DYP(NEQN)
REAL MW, KINVIS, MASS, MDOT, MDOTFR, MSTART, MACRSC
PARAMETER (PI = 3.141592654)
COMMON/EPARAM/ BORE, STROKE, CONRL, CYLCA, CSATDC, CMRTIO, CLVTDC
COMMON/HTRC/ CONHT, EXPHT
COMMON/TEMPS/ NTEMP, TWALL(10), TSS(10), TSS2(10), TCW, TCW2
COMMON/DTDTH/ ESPD
COMMON/TURBU/ CBETA, MACRSC, UPRIME, VMKE, VPISTO
COMMON/RHMAS/ RHO, MASS, VOLUME, H, GAMMA
COMMON/AREAS/ AHEAD, APSTON
COMMON/HTCOTG/ HTCO(10), TGAS(10)

DO 10 I = 1, 20
10 DYP(I) = 0.0

T = DT

FIND THERMODYNAMIC AND TRANSPORT PROPERTIES IN CYLINDER

TCYL = DY(11)
PCYL = DY(12)
FR = DY(28)
CALL THERMO (TCYL, PCYL, FR, H, CSUBP, CSUBT, CSUBF, & RHO, DRHODP, DRHODT, GAMMA, MW, ADUMY, BDUMY, CDUMY)
CALL TRANSP (TCYL, FR, GAMMA, CSUBP, DYNVIS, THRON)
KINVIS = DYNVIS/RHO
MASS = MSTART + DY(1) - DY(2)

FIND SURFACE AREAS AND VOLUME OF CHAMBER

CALL CSAVDV (T, ACW, VOLUME, DVDT)
MACRSC = VOLUME/(3.1415 * BORE * BORE/4.)
IF (MACRSC.GE.(BORE/2.)) MACRSC = BORE/2.
DYP(6) = -.3307 * CBETA/MACRSC * DY(6) * SQRT(DY(7)/MASS)
ADD TURBULENCE AMPLIFICATION TERM FOR RAPID DISTORTION

\[ \text{RDOT} = \text{DRHODT} \cdot \text{YP11} + \text{DRHOOP} \cdot \text{YP12} \]

\[ \text{AMPL} = 2 \cdot \text{DY(7)/RHO} \cdot \text{RDOT} \]

\[ \text{DYP(7)} = 0.3387 \cdot \text{CBETA/MACRSC} \cdot \text{DY(6)} \cdot \text{SQRT(DY(7)/MASS)} \]

\[ & - \cdot 0.5443 \cdot \text{DY(7)/MACRSC} \cdot \text{SQRT(DY(7)/MASS)} + \text{AMPL} \]

CHARACTERISTIC VELOCITY IN CYLINDER; (M/SEC).

\[ \text{CONSTR} = \text{CONRL/STROKE} \]

\[ \text{SINTH} = \text{SIN} \left( \frac{\text{T} \cdot \text{PI}}{180} \right) \]

\[ \text{COSTH} = \text{COS} \left( \frac{\text{T} \cdot \text{PI}}{180} \right) \]

\[ \text{VONVPM} = \text{ABS} \left( \frac{\text{PI} \cdot \text{SINTH} \cdot \left( 1. + \frac{\text{COSTH}}{\text{SINTH}} \right)}{2} \right) \]

\[ \text{VPMEAN} = \frac{\text{STROKE}}{180 \cdot \text{ESPD}} \]

\[ \text{VPISTO} = \text{VPMEAN} \cdot \text{VONVPM} \]

\[ \text{VMKE} = \text{SORT} \left( 2 \cdot \frac{\text{DY(6)/MASS}}{} \right) \]

\[ \text{UPRIME} = \text{SORT} \left( 0.666667 \cdot \frac{\text{DY(7)/MASS}}{} \right) \]

\[ \text{CVHTRN} = \text{SORT} \left( 0.25 \cdot \text{VPISTO} + \text{VPISTO} + \text{VMKE} \cdot \text{VMKE} + \text{UPRIME} \cdot \text{UPRIME} \right) \]

CALCULATE HEAT TRANSFER RATES

\[ \text{HTRCOE} = \text{CONHT} \cdot \left( \frac{\text{CVHTRN} \cdot \text{MACRSC}}{} \right)^{\text{EXPHT}} \cdot \text{THRCON/MACRSC} \]

85 \[ \text{HTPAPI} = \text{HTRCOE} \cdot \left( \frac{\text{TCYL} - \text{TWALL}(1)}{} \right) \]

\[ \text{HTPAHD} = \text{HTRCOE} \cdot \left( \frac{\text{TCYL} - \text{TWALL}(2)}{} \right) \]

\[ \text{HTPACW} = \text{HTRCOE} \cdot \left( \frac{\text{TCYL} - \text{TCW}}{} \right) \]

\[ \text{THTRAN} = \text{HTRAPI} + \text{HTRAHD} + \text{HTRACW} \]

CALCULATE RATES OF CHANGE OF TEMPERATURE, PRESSURE, AND FUEL EQUIVALENCE RATIO IN THE CYLINDER.

THEN CALCULATE RATE OF DOING WORK.
DYP(9) = HTRAHD
DYP(10) = HTRACW

C IF(T.LT.TINJ) GO TO 32

C INTEGRATE PREDICTED CONTRIBUTIONS OF THE DURATION OF THE
C IGNITION DELAY PERIOD

31 ATPA = 1.01325E+5

C PRATM = PCYL/ATPA

202 TPRMS = 3.45 * EXP(2100./TCYL)/ PRATM**1.02

C TPRED = TPRMS/1000.

C DYP(13) = 1./TPRED

32 CONTINUE

C CONVERT ALL TIME DERIVATIVES TO RATE PER CRANK
C ANGLE DEGREE.

DO 40 I = 1, 20

40 DYP(I) = DYP(I) * ESPD

C CALL DIFEQ (NEQN, T, DY(21), DYP(21))

C DYP(54) = HTROE
DYP(55) = HTGPI
DYP(56) = HTROE
DYP(57) = HTGHD
DYP(58) = HTROE
DYP(59) = HTGOW

RETURN

END
SUBROUTINE CPROP

PURPOSE
TO CALCULATE THE SPECIFIC ENTHALPY OF THE PRODUCTS OF HC-AIR
COMBUSTION AT TEMPERATURES AND PressURES WHERE DISSOCIATION
OF THE PRODUCT GASES MAY BE IGNORED. THE DENSITY OF THE
PRODUCT GAS IS ALSO CALCULATED, AS ARE THE PARTIAL
DERIVATIVES OF BOTH OF THESE QUANTITIES WITH RESPECT TO
TEMPERATURE, PRESSURE, AND PHI.

USAGE
CALL CPROP (T, P, FR, ENTHLP, CSUBP, CSUBT, CSUBF,
& RHO, DRHODT, DRHODP, DRHODF)

DESCRIPTION OF PARAMETERS

GIVEN:

P : ABSOLUTE PRESSURE OF PRODUCTS (ATM)
T : TEMPERATURE OF PRODUCTS (DEG K)
FR : AVERAGE FUEL FRACTION
DELF : MOLAR C:H RATIO OF PRODUCTS
PSI : MOLAR N:O RATIO OF PRODUCTS

RETURNS:

ENTHLP: SPECIFIC ENTHALPY OF PRODUCTS (KCAL/G)

UNITS CHECKED FROM HIRES PAPER ON 9/26/83.
CSUBP : PARTIAL DERIVATIVE OF H WITH RESPECT TO T
(CAL/G-DEG K) - UNITS CHECKED FROM HIRES.
CSUBT : PARTIAL DERIVATIVE OF H WITH RESPECT TO P
(CAL/G)
CSUBF : PARTIAL DERIVATIVE OF H WITH RESPECT TO PHI
(CAL/G) - SAME UNITS WITH H
RHO : DENSITY OF THE PRODUCTS (G/CC)

DRHODT: PARTIAL DERIVATIVE OF RHO WITH RESPECT TO T
(G/CC-DEG K)
DRHODP: PARTIAL DERIVATIVE OF RHO WITH RESPECT TO P
(G/CC-ATM)
DRHODF: PARTIAL DERIVATIVE OF RHO WITH RESPECT TO PHI
(G/CC)

REMARKS

1) ENTHALPY DATUM STATE IS AT T = 0 ABSOLUTE WITH
   O2,N2,H2 GASEOUS AND C SOLID GRAPHITE
2) MULTIPLY ATM-CC BY 0.0242173 TO CONVERT TO CAL
3) MODIFIED VERSION OF MIKE MARTIN'S PROGRAM
   BY DENNIS ASSANIS (253-2453)
   ADDED PARTIAL DERIVATIVES OF ENTHALPY AND DENSITY
   WITH RESPECT TO EQUIVALENCE RATIO

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED
NONE

METHOD
SEE SAE PAPER BY HIRES ET AL (APPENDIX)
SEE MARTIN & HEYWOOD 'APPROXIMATE RELATIONSHIPS FOR THE
SUBROUTINE CPROP (T, P, FR, ENTHLP, CSUBP, CSUBT, CSUBF, 
& RHO, DRHODT, DRHOOP, DRHOOF)

LOGICAL RICH, LEAN

REAL*4 MBAR, K

DECLARE MCP

REAL MCP

DIMENSION A(6,6,2), X(6), DX(6)

DIMENSION A1(36), A2(36)

COMMON/FUEL/ FUELTP, CX, HY, DEL, PSI, AFRAST, QLOWER, HFORM

EQUIVALENCE (A1(1), A(1,1,1)), (A2(1), A(1,1,2))

INITIALIZE PARAMETERS, AND CHECK TO SEE IN WHAT TEMPERATURE RANGE WE ARE SO THAT THE CORRECT FITTED COEFFICIENTS WILL BE USED. FLAG TEMPERATURES TOO HIGH OR TOO LOW.

DATA A1/11.94033, 2.088581, -.47029, .037363, -.589447, -.97.1418,
6.139094, 4.60783, -.935609, 6.669498E-02, .0335801, -.56.62588,
7.099556, 1.275957, -.287745, .022356, -.1598696, -.27.73464,
5.555886, 1.787191, -.2881342, 1.951547E-02, .1611828, .76498,
7.865847, 8858719, -.031944, -.2.68708E-03, -.893455,
6.887771, 1.453404, -.329985, 2.561035E-02, -.1189462, -.331835,

DATA A2/4.737305, 16.65283, -.11.23249, 2.828001, 6.76782E-03,
1 -93.75793, 7.898572, -.2823519, 3.418708, -.1179013, 1.43629E-03,
2 -.57.08004, 6.97393, -.8238319, 2.942942, -.1.176239, 4.13249E-04,
3 -.27.19597, 6.991878, 1.617064, -.2182071, .2968197, -.1.625234E-02,
4 -.118189, 6.295715, 2.388387, -.6314788, -.3267433, 4.35925E-03,
5 -.103637, 7.92199, 1.295825, 3.20688, -.1.202212, 3.459738E-04,
6 -.013967,

PHI = FR * AFRAST / (1.-FR)

RICH = PHI .GT. 1.0

LEAN = .NOT. RICH

EPS = 4.*DEL/(1. + 4.*DEL)

IR = 1

IF (T .LT. 500.) IR = 2

GET THE COMPOSITION IN MOLES/MOLE OXYGEN

IF (RICH) GO TO 10

X(1) = EPS*PHI
X(2) = 2.*EPS*PHI
X(3) = 0.
X(4) = 0.
X(5) = 1. - PHI

DX(1) = EPS
DX(2) = 2.*EPS
DX(3) = 0.
DX(4) = 0.
DX(5) = 1.

GO TO 20

10 K = 3.5

ALPHA = 1. - K
BETA = (2.*(1.-EPS*PHI) + K*(2.*(PHI - 1.) + EPS*PHI))
GAMMA = 2.*K*EPS*PHI*(PHI - 1.)
C = ( -BETA + SQRT(BETA*BETA + 4.*ALPHA*GAMMA))/(2.*ALPHA)
X(1) = EPS*PHI - C
X(2) = 2.*(1. - EPS*PHI) + C
X(3) = C
X(4) = 2.*(PHI - 1.) - C
X(5) = 0.

C WORK-OUT DERIVATIVES DX(1) UNTIL DX(5) FOR RICH CASE
C AT A LATER DATE, AND REPLACE THE FOLLOWING ZEROES BY THE
C CORRECT EXPRESSIONS. NOT NECESSARY FOR MY CASE.

C

DX(1) = 0.
DX(2) = 0.
DX(3) = 0.
DX(4) = 0.
DX(5) = 0.

C

20 X(6) = PSI
20 DX(6) = 0.

C CONVERT COMPOSITION TO MOLE FRACTIONS AND CALCULATE AVERAGE MOLECULAR WEIGHT

C IF (LEAN) TMOLES = 1. + PSI + PHI*(1.-EPS)
C IF (RICH) TMOLES = PSI + PHI*(2.-EPS)
DO 30 J = 1, 6
X(J) = X(J)/TMOLES
30 CONTINUE

C

MBAR = ((8.*EPS + 4.)*PHI + 32. + 28.*PSI)/TMOLES

C MCP = (8.*EPS+4.)*PHI + 32. + 28.*PSI
DMCPDF = (8.*EPS+4.)

C

CALCULATE H, CP, CT, AND CF, AS IN WRITEUP,
USING FITTED COEFFICIENTS FROM JANAF TABLES.

C

ENTHLP = 0.
CSUBP = 0.
CSUBT = 0.
CSUBF = 0.
ST = T/1000.
DO 40 J = 1, 6
TH = (((A4(J,IR)/4.*ST + A3(J,IR)/3.)*ST + A2(J,IR)/2.*ST + A1(J,IR)))*ST
TCP = (( A4(J,IR)*ST + A3(J,IR) )*ST + A2(J,IR) )*ST + A1(J,IR)
TH = TH - A5(J,IR)/ST + A6(J,IR)
TCP = TCP + A5(J,IR)/ST**2
ENTHLP = ENTHLP + TH*X(J)
CSUBP = CSUBP + TCP*X(J)
CSUBF = CSUBF + 1./MCP * ( TH*DX(J) - DMCPDF/MCP*TH*X(J) )

40 CONTINUE

ENTHLP = ENTHLP/MBAR
CSUBP = CSUBP/MBAR

C

NOW CALCULATE RHO AND ITS PARTIAL DERIVATIVES
USING PERFECT GAS LAW

C
RHO = .012187*MBAR*P/T
DRHODT = -RHO/T
DRHODP = RHO/P

IF (RICH) GO TO 60

LEAN CASE

50 D = 1. + (1.-EPS)*PHI + PSI

GO TO 70

RICH CASE

60 D = (2.-EPS)*PHI + PSI

70 DMWDF = -MCP/D/D*(DDDF - D/MCP*DMCPDF)

RETURN

END
C******************************************************** VERSION 1.0 ********************************************************
C Oct 21, 1984
C
SUBROUTINE CSAVDV
C
PURPOSE
CALCULATES THE SURFACE AREA OF THE CYL. WALLS, AND THE VOLUME
AND TIME RATE OF CHANGE OF THE VOLUME OF THE COMBUSTION CHAMBER

USAGE
CALL CSAVDV (T, ACW, VOLUME, DVDT)

DESCRIPTION OF PARAMETERS

PARAMETER INPUT OUTPUT DESCRIPTION
T YES NO TIME (DEG)
ACW NO YES SURFACE AREA OF CYL. WALLS (M**2)
VOLUME NO YES VOLUME OF THE CHAMBER (M**3)
DVDT NO YES TIME RATE OF CHANGE OF VOLUME OF

CHAMBER (M**3/SEC)

REMARKS
UNITS CHANGED TO SI
SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
NONE

METHOD
SEE REPORT

WRITTEN BY D. N. ASSANIS
EDITED BY D. N. ASSANIS

SUBROUTINE CSAVDV (T, ACW, VOLUME, DVDT)
COMMONEPARAM/ BORE, STROKE, CONRL, CYLCA, CSATDC, CMRTIO, CLVTDC
COMMON/DTDTH/ ESPD

CONVERT ANGLE MEASURE FROM DEGREES TO RADIANS.
THETA = T * 0.0174533
COSC = 0.5 * STROKE * COS(THETA)
SINC = 0.5 * STROKE * SIN(THETA)
CI = SQRT( CONRL * CONRL - SINC * SINC )

CYLL: CYLINDER LENGTH FROM PISTON POSITION AT TDC

CYLL = CONRL + 0.5 * STROKE - COSC - CI
IF (CYLL LT. 0.0) CYLL = 0.0
ACW = 3.141593 * BORE * CYLL
VOLUME = CLVTDC + CYLCA * CYLL
DVDT = CYLCA * SINC * (1.0 + COSC/CI) * 0.0174533 / ESPD
RETURN
END
SUBROUTINE CYLPAR

PURPOSE
READS IN WALL CONSTRUCTION DATA GIVEN IN "CHEAT.DAT", FOR
UP TO 6 COMPONENTS WITH CYLINDRICAL COMPOSITE LAYERS, SUCH
AS THE CYLINDER LINER. ALSO, IT CALCULATES THE OVERALL
HEAT TRANSFER COEFFICIENT FOR EACH COMPONENT.

USAGE
CALL CYLPAR

DESCRIPTION OF PARAMETERS

PARAMETER INPUT OUTPUT DESCRIPTION

NCC YES NO TOTAL NUMBER OF COMPONENTS WITH
CYLINDRICAL COMPOSITE WALL STRUCTURES
CDIAM(I) YES NO INSIDE PARAMETER OF ITH COMPONENT (M)
NCLA(I) YES NO NUMBER OF MATERIAL LAYERS OF ITH
COMPONENT
CHCOOL(I) YES NO HEAT TRANSFER COEFFICIENT FROM THE
OUTSIDE WALL SURFACE OF ITH COMPONENT
TO THE COOLANT OR AMBIENT (W/M2/K)
CTCOOL(I) YES NO AMBIENT TEMPERATURE, COOLANT
TEMPERATURE, OR SPECIFIED OUTSIDE WALL TEMPERATURE
CTHIK(I,J) YES NO THICKNESS OF JTH LAYER OF ITH
COMPONENT (M)
CCOND(I,J) YES NO THERMAL CONDUCTIVITY OF JTH LAYER
OF ITH COMPONENT (W/M/K)
CUOVE(I) NO YES OVERALL HEAT TRANSFER COEFFICIENT
OF ITH COMPONENT (W/M**2/K)

REMARKS
FIRST ARRAY DIMENSION : COMPONENT DESCRIPTION
SECOND ARRAY DIMENSION : LAYER DESCRIPTION

SUBROUTINES AND FUNCTIONS REQUIRED
NONE

WRITTEN BY D. N. ASSANIS
EDITED BY D. N. ASSANIS

SUBROUTINE CYLPAR

DIMENSION CDIAM(6), NCLA(6), CTHIK(6,3), CCOND(6,3),
& CHCOOL(6), CTCOOL(6), CUOVE(6)

COMMON/NCLA/ NCLA, NCC
COMMON/CYLP/ CDIAM, CTHIK, CCOND, CHCOOL, CTCOOL, CUOVE

NAMELIST/ CHEAT/ CDIAM, NCLA, NCC, CTHIK, CCOND, CHCOOL, CTCOOL

READ (10, CHEAT)
CLOSE (UNIT = 10)
DO 10 I = 1, NCC
   C CALCULATE INSIDE RADIUS OF COMPONENT
   RADI = CDIAM(I) /2.
   C CALCULATE NET WALL THERMAL RESISTANCE
   SUM = 0.
   RAD1 = RADI
   DO 20 J = 1, NCLA(I)
      C SUM THERMAL RESISTANCE FOR EACH LAYER
      RAD2 = RAD1 + CTHIK(I,J)
      RES = LOG(RAD2/RAD1) /CCOND(I,J)
      SUM = SUM + RES
      RAD1 = RAD2
   20 CONTINUE
   C ADD THERMAL RESISTANCE AT outside WALL SURFACE
   IF (CHCOOL(I).NE. 0.0) SUM = SUM + 1. /RAD2 /CHCOOL(I)
   C CUOVE(I) = 1. /RADI /SUM
   10 CONTINUE
   RETURN
END
SUBROUTINE DATA(NEQN, DT, DY)

SUBROUTINE WRITTEN TO STORE DATA DURING SIM RUNS

REMARKS
THIS SUBROUTINE IS SET UP TO WRITE TO DIRECT ACCESS FILES
ON A VAX COMPUTER. IT IS STRICTLY OUTPUT AND CAN BE DELETED
FROM THE PROGRAM (IN STEPCA). OR, IT CAN BE MODIFIED TO
MEET OTHER OUTPUT REQUIREMENTS. MOST INFORMATION OF INTEREST
IN THE PROGRAM IS AVAILABLE IN DATA.FOR THROUGH COMMON BLOCKS.

******/ MARKERS ******/
8888 END OF CYCLE
9999 LAST ENTRY TO TABLE

***************

REAL*8 DT, DY(NEQN), TOUT
REAL MW, MMM, MSTART, MACRSC, MASS, MKESTA,
& MAXERR, MFINAL, MASSIN, MASSEX, KIL, MIL

INTEGER IROW, EROW, SIZC, SIZT, SIZPT, SIZ1, SIZ2,
& SIZ3
DIMENSION TABLIN(120, 2), TABLEX(120, 2)
PARAMETER (PI=3.1415927, T0=0, I1=1, I2=2)
PARAMETER (SIZC=6, SIZT=6, SIZPT=8, SIZ1=8, SIZ2=8, SIZ3=11)
PARAMETER (CEN=1.02, KIL=1.03, MIL=1.06, ERG=1.07,
& ATPA=1.01325E5, HGPA=0.2953E-3, PSIPA=6.8948E3,
& CSFC=1.644E-3, HPKW=0.7457 )
PARAMETER (NCV=6)

DIMENSION MASSIN(270), MASSEX(270),
& FCYLIN(270), FCYLEX(270),
& HCYLIN(270), HCYLEX(270)

DIMENSION NCLA(6)
DIMENSION PTW(10, 3, 51), NPLA(10)

COMMON/RADN/ TCHTR, TRHTR, THTR, CHTRAP, CHTRAH, CHTRAW,
& RHTRAP, RHTRAH, RHTRAW
COMMON/TOPO/ TOAIR, POAIR, GAMAIR
COMMON/FLAME/ TAIR, PAIR, TFLAME, TRAD, EMIS
COMMON/PFD/ PTW
COMMON/HTCOTG/ HTCO(10), TGAS(10)
COMMON/QSOL/ QWALL(10), QAVE(10), QSS(10)
COMMON/SUMIT/ ENGM(2), FMDOT(4), HDDOT(2)
COMMON/SUMITX/ ENGMX(NCV), FMDOTX(NCV), HDDOTX(NCV), QDOTX
COMMON/EF/ EN(8), HAL(8), FAL(8), TFLAG, TAL(8)
COMMON/WA/ YPM(2), YPH(2), YPF(2)
COMMON/FLOPRO/ EMFLO(NCV+1), EHFLLO(NCV+1), EFFLO(NCV+1)
COMMON/VALVE/ VIV, VEV
COMMON/B / CPM(2), HM(2), MWM(2), GM(2), RHOM(2)
COMMON/X / RTEMP(5), H(5), RMSS(5), RCORR(5)
COMMON/HWDF1F / ASP(3), PR(3), PRSS(5), DP(5), HI(5),
& TMAP(2), PTMAP(2), CMAP(2),
& CM(SIZC,SIZI,3), TM(SIZT,SIZZ,3), PTM(SIZPT,SIZZ,3),
& CPM(SIZC), TRPM(SIZPT), PTRPM(SIZPT), PSTD(3), TSTD(3)
COMMON/TEMDPS/ NTEMP, TWALL(10), TSS(10), TSS2(10), TOW, TOW2
COMMON/AREAS/ AHEAD, APSTON
COMMON/TURBU/ CBETA, MACRSC, UPRIME, WAKE, VPISTO
COMMON/HEATS/ CVHTRN, HTAROE, HTPAHO, HTPACW, HTAPI,
& HTRAHD, HTACW, THTRAN
COMMON/QP1/ EDIAM(3), EAREA(3), ECROSS(3), ETWALL(3), ECONHT(3)
COMMON/QP2/ EPRF(3), EVBLK(3), EREF(3), ENUF(3), EHTCOE(3),
& EQDOT(3)
COMMON/TURB/ TRBM
COMMON/DDTH/ ESPD
COMMON/EMGEOM/ EXLNG(NCV), EXDIAM(NCV), EXAREA(NCV),
& EXVOL(NCV), EXXAR(NCV)

DATA I, ITOLD /1,-45/

INITIALIZE CRANK ANGLE INTERVALS AND NUMBER OF CYCLES TO RECORD
FOR PLOTTING ROUTINE:
INTCA = 3
ICYCLS = 2
IT = NINT(DT)

CHECK FOR INTERVAL TO STORE DATA, IF NOT THE RIGHT INTERVAL, RET.
IF (MOD(IT,INTCA).NE.0.) RETURN

FIND NUMBER OF RECORDS TO BE KEPT
NREC = (720 / INTCA * ICYCLS) + ICYCLS

CHECK FOR START OF NEW CYCLE AND INSERT MARKER
IF ( IT .LT. ITOLD ) THEN
  WRITE (21,REC-I,FMT-51)
  WRITE (22,REC-I,FMT-51)
  WRITE (23,REC-I,FMT-51)
  I = I + 1
  IF (I.GT.NREC) I = 1
ENDIF
ITOLD = IT

TURBINE SPEED, INTAKE MANIFOLD PRESSURE, EXH MANIFOLD PRESS.
WRITE(21,REC=1,FMT=21)IT, DY(29), DY(23), DY(27), DY(26),
& DP(3)

MASS FLOWS AND HEAT TRANSFER (W)
Q1 = OWALL(3) / ESPD
Q2 = OWALL(4) / ESPD
Q3 = OWALL(5) / ESPD
WRITE(22,REC=1,FMT=21)IT, EMFLO(1), EMFLO(2), EMFLO(3),
& Q1, Q2

AVERAGED TEMP., PORT TEMP, PLENUM TEMP, ETC
WRITE(23,REC=1,FMT=21)IT, DY(11), DY(64), DY(72), DY(80),
& DY(88)

DT1 = TGAS(1) - TWALL(1)
DT2 = TGAS(3) - TWALL(3)
WRITE(25,REC=1,FMT=21)IT, DY(103), DY(104), DY(105), DY(106),
& DY(107)

FORMAT (I4,1X,5(E13.6,2X))
C INCREMENT RECORD, CHECK FOR END OF FILE, INSERT MARKER
122   I = I + 1
123   IF ( I .GT. NREC ) I=1
124   WRITE (21,REC=I,FMT=50)
125   WRITE (22,REC=I,FMT=50)
126   WRITE (23,REC=I,FMT=50)
127   WRITE (25,REC=I,FMT=50)
128 C
129   50 FORMAT('9999')
130   51 FORMAT('8888')
131 C
132   RETURN
133 END
SUBROUTINE DELH

PURPOSE

TO CALCULATE THE CHANGE IN SPECIFIC ENTHALPY OF A MIXTURE OF AIR AND PRODUCTS OF COMBUSTION ACROSS AN ADIABATIC SHAFT WORK DEVICE (COMPRESSOR OR TURBINE) GIVEN THE UPSTREAM AND DOWNSTREAM PRESSURES AND GAS TEMPERATURES.

USAGE

CALL DELH (T1, T2, P1, P2, F, EF, TOUT, H1, H2, DH)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>T1</td>
<td>YES</td>
<td>NO</td>
<td>UPSTREAM TEMPERATURE (K)</td>
</tr>
<tr>
<td>T2</td>
<td>YES</td>
<td>NO</td>
<td>DOWNSTREAM TEMPERATURE ESTIMATE (K)</td>
</tr>
<tr>
<td>P1</td>
<td>YES</td>
<td>NO</td>
<td>UPSTREAM PRESSURE (Pa)</td>
</tr>
<tr>
<td>P2</td>
<td>YES</td>
<td>NO</td>
<td>DOWNSTREAM PRESSURE (Pa)</td>
</tr>
<tr>
<td>F</td>
<td>YES</td>
<td>NO</td>
<td>UPSTREAM FUEL FRACTION</td>
</tr>
<tr>
<td>EF</td>
<td>YES</td>
<td>NO</td>
<td>COMPRESSOR EFFICIENCY OR TURBINE RECIPROCAL EFFICIENCY</td>
</tr>
<tr>
<td>TOUT</td>
<td>NO</td>
<td>YES</td>
<td>ACTUAL DOWNSTREAM TEMPERATURE (K)</td>
</tr>
<tr>
<td>H1</td>
<td>NO</td>
<td>YES</td>
<td>SPECIFIC ENTHALPY OF INLET FLOW (*)</td>
</tr>
<tr>
<td>H2</td>
<td>NO</td>
<td>YES</td>
<td>SPECIFIC ENTHALPY OF OUTLET FLOW (*)</td>
</tr>
<tr>
<td>DH</td>
<td>NO</td>
<td>YES</td>
<td>CHANGE IN SPECIFIC ENTHALPY FROM INLET TO OUTLET (*)</td>
</tr>
</tbody>
</table>

REMARKS

* UNITS ARE (J/KG)

SUBROUTINES AND FUNCTIONS REQUIRED

THERMO

METHOD


WRITTEN BY K. K. REPLOGLE
EDITED BY D. N. ASSANIS

SUBROUTINE DELH (T1, T2, P1, P2, F, EF, TOUT, H1, H2, DH)

DEFINE MEAN TEMPERATURE, MEAN PRESSURE AND PRESSURE RATIO:

\[ TM = (T1 + T2) / 2. \]
\[ PM = (P1 + P2) / 2. \]
\[ PR = P2 / P1 \]

CALCULATE MEAN SPECIFIC HEAT:
CALL THERMO (TM, PM, F, HM, CPM, X1, X2, X3, X4, X5, X6, 
& GAM, RMW, X9, X10, X11)

CALL THERMO (TM, PM, F, HM, CPM, X1, X2, X3, X4, X5, X6, 
& GAM, RMW, X9, X10, X11)

CALCULATE TEMPERATURE CHANGE:

\[ DT = \frac{T1}{EF} \times \left( PR^{\frac{(GAM-1)}{GAM}} - 1 \right) \]

CALCULATE OUTPUTS: INLET SPECIFIC ENTHALPY, EXIT TEMPERATURE, ENTHALPY CHANGE AND EXIT SPECIFIC ENTHALPY:

CALL THERMO (T1, P1, F, H1, CP1, Y1, Y2, Y3, Y4, Y5, Y6, 
& Y7, Y8, Y9, Y10, Y11)

TOUT = T1 + DT

DH = DT \times CPM

H2 = H1 + DH

RETURN

END
SUBROUTINE DERT1(FNEQN, Y, T, TOUT, RELERR, ABSERR, IFLAG, G, REROOT,
1 AEROOT, YY, WT, P, YP, YPOUT, PHI, ALPHA, BETA, SIG, V, W, GG, PHASE1, PSI,
2 X, HOLD, START, TOLD, DELSGN, GX, TROOT, NS, NORND, K, KOLD, ISNOLD)
C ***NAME CHANGED FROM DERT TO DERT1 TO AVOID A NAMING CONFLICT.
C ODERT MERELY ALLOCATES STORAGE FOR DERT TO RELIEVE THE USER OF THE INCONVENIENCE OF A LONG CALL LIST. CONSEQUENTLY DERT IS USED AS DESCRIBED IN THE COMMENTS FOR ODERT.
C CHANGES:
C NOTE THAT DERT1 IS MODIFIED FROM THE ORIGINAL VERSION TO INCLUDE THE NUMBER OF EQUATIONS BEING INTEGRATED (NEQN) IN THE CALLS TO THE PROCESS SUBROUTINES.
C IT HAS ALSO BEEN MODIFIED TO ELIMINATE ERRCHK DUE TO COMPATIBILITY PROBLEMS WITH IBM MACHINES.
C IMPLICIT REAL*8 (A-H,O-Z)
C CCCCC GENERIC
IMPLICIT INTEGER*4 (I-N)
LOGICAL STIFF, CRASH, START, PHASE1, NORND
DIMENSION Y(NEQN), YY(NEQN), WT(NEQN), PHI(NEQN,16), P(NEQN), YP(NEQN),
1 YPOUT(NEQN), PSI(12), ALPHA(12), BETA(12), SIG(13), V(12), W(12),
2 GG(13)
COMMON/MLDRT/SPACE(18)
EXTERNAL F, G
C C*******************************************************************************
C THE ONLY MACHINE DEPENDENT CONSTANT IS BASED ON THE MACHINE UNIT
C ROUNDOFF ERROR U WHICH IS THE SMALLEST POSITIVE NUMBER SUCH THAT
C 1.0+U.GT. 1.0 . U MUST BE CALCULATED AND FOURU=4.8+U INSERTED
C IN THE FOLLOWING STATEMENT BEFORE USING ODERT. THE SUBROUTINE
C MACHINE CALCULATES U. FOURU AND TWOU=2.0+U MUST ALSO BE
C INSERTED IN SUBROUTINE STEP BEFORE CALLING ODERT.
C*******************************************************************************
DATA FOURU/8.8E-16/
C*******************************************************************************
C THE CONSTANT MAXNUM IS THE MAXIMUM NUMBER OF STEPS ALLOWED IN ONE CALL TO ODERT. THE USER MAY CHANGE THIS LIMIT BY ALTERING THE FOLLOWING STATEMENT
DATA MAXNUM/500/
C
*** *** ***
C TEST FOR IMPROPER PARAMETERS
C IF(IABS(IFLAG).EQ.7) THEN
WRITE (7,*) 'ENTERED INTEGRATION SBRTN WITH IFLAG=7.'
STOP
ENDIF
IF(NEQN.LT.1) THEN

WRITE (7,*), 'NEQN MUST BE POSITIVE'
GOTO 10
ENDIF
IF(T .EQ. TOUT) THEN
WRITE (7,*), 'ENDPOINTS OF INTEGRATION INTERVAL MUST BE',
1 ' DISTINCT'
GOTO 10
ENDIF
IF(RELERR .LT. 0.0 .OR. ABSERR .LT. 0.0) THEN
WRITE (7,*), 'RELERR AND ABSERR MUST BE NON-NEGATIVE.'
GOTO 10
ENDIF
EPS = MAX(RELERR,ABSERR)
IF(EPS .LE. 0.0) THEN
WRITE (7,*), 'EITHER RELERR OR ABSERR MUST BE POSITIVE.'
GOTO 10
ENDIF
IF(REROOT .LT. 0.0 .OR. AEROOT .LT. 0.0) THEN
WRITE (7,*), 'REROOT AND AEROOT MUST BE NON-NEGATIVE.'
GOTO 10
ENDIF
IF(REROOT+AEROOT .LE. 0.0) THEN
WRITE (7,*), 'EITHER REROOT OR AEROOT MUST BE POSITIVE.'
GOTO 10
ENDIF
IF(IFLAG .EQ. 0) THEN
WRITE (7,*), 'INVALID INPUT FOR IFLAG.'
GOTO 10
ENDIF
ISN = ISIGN(1,IFLAG)
IFLAG = IABS(IFLAG)
IF(IFLAG .EQ. 1) GO TO 20
IF(T .NE. TOLD) THEN
WRITE (7,*), 'INPUT VALUE OF T MUST BE OUTPUT VALUE FROM',
1 ' PRECEEDING CALL.'
GOTO 10
ENDIF
IF(IFLAG .GE. 2 .AND. IFLAG .LE. 6) GO TO 15
IF(IFLAG .GE. 8 .AND. IFLAG .LE. 10) GO TO 15
WRITE (7,*), 'INVALID INPUT FOR IFLAG'
STOP
10 IFLAG = 7
RETURN

15 CONTINUE
IF (ISNOLD.LT.0 .OR. DELSIGN*(TOUT-T).LT.0.) GO TO 20
    C--- EVALUATE G AT EITHER TOUT (OUTPUT POINT THIS CALL) OR AT
    X (POINT TO WHICH INTERNAL INTEGRATION HAS ALREADY
    PROCEEDED), WHICHEREVER OCCURS FIRST.
    T2=X
    IF((X-T.GT.0.,AND.X-TOUT.GT.0.),OR.(X-T.LT.0.,AND.X-TOUT.LT.0.))
1 T2=TOUT
CALL INTRP(X,YY,T2,Y,YPOUT,NEQN,KOLD_PHI_PSI)
GOFT2=G(T2,Y,YPOUT)
C--- NOW EVALUATE AT T1=T
T1=T
CALL INTRP(X,YY,T1,Y,YPOUT,NEQN,KOLD_PHI_PSI)
GOFT1=G(T1,Y,YPOUT)
C--- NOW SEE IF A ROOT OF G OCCURS IN CLOSED INTERVAL (T1,T2).
IF( GOFT1.EQ.0. .OR. GOFT2.EQ.0.) GO TO 134
IF( SIGN(1.D0,GOFT1) * SIGN(1.D0,GOFT2) .LT. 0.D0 ) GO TO 134
GO TO 21
C ON EACH CALL SET INTERVAL OF INTEGRATION AND COUNTER FOR NUMBER OF
C STEPS. ADJUST INPUT ERROR TOLERANCES TO DEFINE WEIGHT VECTOR FOR
C SUBROUTINE STEP
C
20 T2=T
CALL F(NEQN,T2,Y,YPOUT)
GOFT2 = G(T2,Y,YPOUT)
21 CONTINUE
DEL = TOUT - T
ABSDEL = ABS(DEL)
TEND = T + 10.0*DEL
IF(ISN .LT. 0) TEND = TOUT
NOSTEP = 0
KLE4 = 0
STIFF = .FALSE.
RELEPS = RELERR/EPS
ABSEPS = ABSERR/EPS
IF(IFLAG .EQ. 1) GO TO 30
IF(ISNOLD .LT. 0) GO TO 30
IF(DELSGN*DEL > 0.0) GO TO 50
C START AND RESTART ALSO SET WORK VARIABLES X AND YY(*), STORE THE
C DIRECTION OF INTEGRATION, AND INITIALIZE THE STEP SIZE.
30 START = .TRUE.
X = T
TROOT = T
DO 40 L = 1,NEQN
40 YY(L) = Y(L)
DELSGN = SIGN(1.0D0,DEL)
H = SIGN(MAX(ABS(TOUT-X),FOURU*ABS(X)),TOUT-X)
C IF ALREADY PAST OUTPUT POINT, INTERPOLATE AND RETURN
50 CONTINUE
IF(ABS(X-T) .LT. ABSDEL) GO TO 60
CALL INTRP(X,YY,TOUT,Y,YPOUT,NEQN,KOLD PHI,PSI)
IFLAG = 2
T = TOUT
TOLD = T
ISNOLD = ISN
RETURN
C IF CANNOT GO PAST OUTPUT POINT AND SUFFICIENTLY CLOSE,
C EXTRAPOLATE AND RETURN
60 IF(ISN .GT. 0 .OR. ABS(TOUT-X) .GE. FOURU*ABS(X)) GO TO 80
H = TOUT - X
CALL F(NEQN,X,YY,YP)
DO 70 L = 1,NEQN
70 Y(L) = YY(L) + H*YP(L)
C *** NEXT STMNT ADDED BY LIENESCH TO ENSURE YPOT VALUES WILL ALWAYS BE
C *** AVAILABLE UNDER ANY CIRCUMSTANCES
CALL F(NEQN,X,Y,YPOUT)
IFLAG = 2
T = TOUT
TOLD = T
ISNOLD = ISN
RETURN
C TEST FOR TOO MUCH WORK
C
80 IF(NOSTEP .LT. MAXNUM) GO TO 100
IFLAG = ISN+4
IF(STIFF) IFLAG = ISN+5
DO 90 L = 1,NEQN
90 Y(L) = YY(L)
T = X
TOLD = T
ISNOLD = 1
RETURN
C LIMIT STEP SIZE, SET WEIGHT VECTOR AND TAKE A STEP
C
100 H = SIGN(MIN(ABS(H),ABS(TEND-X)),H)
DO 110 L = 1,NEQN
WT(L) = RELEPS*ABS(YY(L)) + ABSEPS
IF(WT(L) .LE. 8.0) GO TO 160
110 CONTINUE
CALL STEP1(FNEQN,YY,H,EPS,WTSTART,1
HOLD,K,KOLD,CRASH,PHI,PYPSI,
2 ALPHA,BETA,SIG,V,W,GG,PHASE1,NS,NORND)
C TEST FOR TOLERANCES TOO SMALL. IF SO, SET THE DERIVATIVE AT X
C BEFORE RETURNING
C
IF(.NOT.CRASH) GO TO 130
IFLAG = ISN+3
RELERR = EPS*RELEPS
ABSERR = EPS*ABSEPS
DO 120 L = 1,NEQN
YP(L) = PHI(L,1)
120 Y(L) = YY(L)
T = X
TOLD = T
ISNOLD = 1
RETURN
C AUGMENT COUNTER ON WORK AND TEST FOR STIFFNESS. ALSO TEST FOR A
C ROOT IN THE STEP JUST COMPLETED
C
130 NOSTEP = NOSTEP + 1
KLE4 = KLE4 + 1
IF(KOLD .GT. 4) KLE4 = 0
IF(KLE4 .GE. 50) STIFF = .TRUE.
T1=T2
GOFT1=GOFT2
T2=TOUT
C— Evaluate G at internal integration point X unless X is past TOUT
C— If X is past TOUT evaluate G at TOUT.
C IF X IS PAST TOUT EVALUATE G AT TOUT.
C
IF(ABS(X-T).LT.ABSDEL) T2=X
CALL INTRP(X,YY,T2,Y,YPOUT,NEQN,KOLD,PHI,PSI)
GOFT2=G(T2,Y,YPOUT)
IF(GOFT1.EQ.0 .OR. GOFT2.EQ.0) GO TO 134
IF(SIGN(1.D0,GOFT1)*SIGN(1.0D0,GOFT2) .LT.0.D0)GO TO 134
GO TO 50
C
C LOCATE ROOT OF G. INTERPOLATE WITH INTRP FOR SOLUTION AND
C DERIVATIVE VALUES
C 134   JFLAG=1
C— HERE ROOT IS BETWEEN T1 AND T2
B=T1
IF(GOFT1.EQ.0.)GO TO 150
B=T2
IF(GOFT2.EQ.0.)GO TO 150
C=T1
140 CALL ROOT(T,GT,B,C,REROOT,AEROOT,JFLAG)
IF(JFLAG .GT. 0) GO TO 150
IF( T.EQ.T1)GT-GOFT1
IF( T.EQ.T2)GT-GOFT2
IF( T.EQ.T1 .OR.T.EQ.T2)GO TO 140
CALL INTRP(X,YY,T,Y,POUT,NEQN,KOLD,PHI,PSI)
GT = G(T,Y,POUT)
GO TO 140
150 CONTINUE
IFLAG = JFLAG+7
IF(JFLAG .EQ. 2 .OR. JFLAG .EQ. 4) IFLAG = 8
IF(JFLAG .EQ. 3) IFLAG = 9
IF(JFLAG .EQ. 5) IFLAG = 10
IFLAG = IFLAG*ISN
CALL INTRP(X,YY,B,Y,POUT,NEQN,KOLD,PHI,PSI)
T = B
IF(ABS(T-TROOT) .LE. REROOT*ABS(T)+AEROOT) GO TO 50
TROOT = T
TOLD = T
ISNOLD = 1
RETURN
160 CALL ERRCHK(72,72HINODERT, PURE ABSOLUTE ERROR IMPOSSIBLE. USE N
C 10N-ZERO VALUE OF ABSERR.)
160 WRITE (7,*)'PURE ABSOLUTE ERROR IMPOSSIBLE. USE NON-ZERO VALUE',
1 'OF ABSERR.'
IFLAG = 6
RETURN
END
SUBROUTINE DIFEQ

PURPOSE
CALCULATES ANGULAR DERIVATIVES OF STATE VARIABLES, M, T, P AND F, FOR INTAKE MANIFOLD AND EXHAUST MANIFOLD, AND THE ANGULAR DERIVATIVE OF TURBOCHARGER SPEED. ALSO, PERFORMS TIME-AVERAGING OF VARIOUS SYSTEM VARIABLES. THE SUBROUTINE CAN HANDLE BOTH SIMPLE TURBOCHARGED AND TURBOCHARGED TURBOCOMPUNDED CASES.

USAGE
CALL DIFEQ (NEQN, T, DY(21), DYP(21))

DEFINITION OF PARAMETERS

PARAMETER INPUT OUTPUT DESCRIPTION

NEQN YES NO NUMBER OF EQUATIONS BEING INTEGRATED
DT YES NO CRANK ANGLE
DYDIF YES NO ARRAY OF SYSTEM VARIABLES
DYDPDIFF NO YES ARRAY OF SYSTEM VARIABLE DERIVATIVES

ALL THE ABOVE ARE DOUBLE PRECISION

Y NO NO SINGLE PRECISION WORKING ARRAY OF SYSTEM VARIABLES, INTERNAL TO DIFEQ
YDOT NO NO SINGLE PRECISION CALCULATED ARRAY OF ANGULAR DERIVATIVES OF SYSTEM VARIABLES, INTERNAL TO DIFEQ
YDOT1 NO NO STORAGE ARRAY FOR OLD VALUES OF YDOT FOR EXHAUST MANIFOLD CONVERGENCE CHECK

Y, DYDIF, YDOT, DYDPDIFF: EACH HAS ELEMENTS CORRESPONDING TO THE FOLLOWING SYSTEM VARIABLES

ELEMENT SYSTEM VARIABLE
(1) INTAKE MANIFOLD MASS (KG)
(2) INTAKE MANIFOLD TEMPERATURE (K)
(3) INTAKE MANIFOLD PRESSURE (PA)
(4) INTAKE MANIFOLD AVERAGE FUEL FRACTION
(5) AVE. EXHAUST MANIFOLD MASS (KG)
(6) AVE. EXHAUST MANIFOLD TEMPERATURE (K)
(7) AVE. EXHAUST MANIFOLD PRESSURE (PA)
(8) AVE. EXHAUST MANIFOLD FUEL FRAC.
(9) TURBOCHARGER SPEED (RPM * 1000.)
(10) POWER TURBINE CUMULATIVE WORK TRANSFER
(11) TOTAL MASS FLOWED FROM INTAKE MANIFOLD TO ALL CYLINDERS
(12) TOTAL MASS FLOWED FROM COMPRESSOR TO INTAKE MANIFOLD
(13) TOTAL MASS FLOWED FROM ALL CYLINDERS TO EXHAUST MANIFOLD
(14) TOTAL MASS FLOWED FROM EXHAUST MANIFOLD TO FIRST TURBINE
C EXHAUST MANIFOLD SUB-MODEL VARIABLES:

(43) EXHAUST PORT MASS (KG)

(44) EXHAUST PORT TEMPERATURE (K)

(45) EXHAUST PORT FUEL FRACTION

(46) TIME AVERAGED PORT TEMPERATURE

(47) MASS AVERAGED PORT TEMPERATURE

(48) INTEGRATED HEAT TRANSFER FOR PORT

(49) TIME AVERAGED HEAT TRANS COEF FOR PORT

(50) TIME AVERAGED PRODUCT OF PORT HEAT TRANSFER COEF. TIMES GAS TEMPERATURE

ADD INTEGER MULTIPLES OF 8 TO GET INDICES FOR VARIABLES FOR SECTIONS DOWNSTREAM OF THE PORT. ONLY THE HEAT TRANSFER VARIABLES (48+8*N, 49+8*N, &50+8*N) APPLY FOR THE TURBINE CONNECTING PIPE FOR THE TURBOCOMPONDED CASE.

REMARKS

NOTE THAT THE INDICES FOR DYDIF AND DYPDIF IS INCREASED BY 20 WHEN THE TRANSFER IS DONE BACK UP TO THE PROCESS SUBROUTINES (INTAKE, CMPRES, ETC.)

UNLIKE THE PROCESS SUBROUTINES WHICH ARE DONE IN REAL TIME AND TRANSLATED TO CRANK ANGLE TIME AT THE END OF THE SBRTN, DIFEQ IS IN CRANK ANGLE TIME (RATE OF CHANGE PER DEGREE) UNLESS OTHERWISE NOTED.

SUBROUTINES AND FUNCTIONS SUBPROGRAMS REQUIRED

THERMO DELH QMAN DPMAN
METHOD
SEE NASA DIESEL REPORT, 1984

WRITTEN BY D. N. ASSANIS, R.M. FRANK

SUBROUTINE DIFEQ (NEQNA,DYDIF,DYPDIF)

LOGICAL POWER, EXSUB
INTEGER SIZC, SIZPT, SIZT, SIZ1, SIZ2, SIZ3
PARAMETER (11=1, I2=2, I3=3, PI=3.1415927)
PARAMETER (SIZO=8, SIZ1=8, SIZT=6, SIZ2=8, SIZPT=6, SIZ3=11)
PARAMETER (NCV=6)
REAL MWM
REAL*8 DYDIF(NEQN-2), DYPDIF(NEQN-20)

DIMENSIONS OF Y, YDOT AND YDOT1 MUST BE AT LEAST NEQN-20
DIMENSION Y(90), YDOT(90), YDOT1(90)

DIMENSION EGA*MA(NCV), ECP(NCV), EMW(NCV)
DIMENSION EX1(NCV), EX2(NCV), EX3(NCV), EX4(NCV),
& EX5(NCV), EX6(NCV), EX7(NCV), EX8(NCV), EX9(NCV),
& EX10(NCV), EX11(NCV)
DIMENSION Q(NCV)
DIMENSION PHDOT(NCV)

COMMON/EXSUB/EXSUB
COMMON/POWER/POWER
COMMON/SUNIT/ENGM(2), FMOTO(2), HMOTO(2)
COMMON/B/CPM(2), HM(2), MMM(2), GM(2), RHOM(2)
COMMON/MA/YPM(2), YPH(2), YPF(2)

COMMON/SUNITX/ENMX(NCV), FMOTX(NCV), HMOTX(NCV), QDOTX
COMMON/B2/ERHO(NCV), DVNVS(NCV), EM(NCV), EFFR(NCV)
COMMON/FLOPRO/EMFLO(NCV+1), EHFLO(NCV+1), EFFLO(NCV+1)

COMMON/O/ERPM
COMMON/DTOTHE/ESPD
COMMON/F/DIF/ASP(3), PR(3), PRSS(5), DP(5), HI(5),
& TMAP(2), PTMAP(2), CMAP(2),
& CM(SIZC,SIZ1,3), TM(SIZT,SIZ2,3), PTM(SIZPT,SIZ3,3),
& CRPM(SIZC), TRPM(SIZT), PTRPM(SIZPT), PSTD(3), TSTD(3)
COMMON/I/PINLET
COMMON/K/RTEMP(5), H(5), RMSS(5), RCMR(5)
COMMON/FUEL/FUELTP, CX, HY, DEL, PSI, AFRAST, QLOWER, HFORM
COMMON/TCPAR/B(2)

COMMON/NUMEX/NUMEX, NUMEXT
COMMON/ENEX/ENLM(NCV), EXDIAM(NCV), EXAREA(NCV),
& EXVOL(NCV), EXVOL(NCV)
COMMON/TUBE/TRBM
COMMON/INFO2/IDIFCT, ILOPC
COMMON/HCTO/HCTO(10), TGAS(10)
DATA YDOT1, RHOM /90=0.1/

SAVE
IDIFCT = IDIFCT + 1

TRANSFER ARRAY TO SINGLE PRECISION

DO 10 I=1,NEQN-20
Y(I) = DYDIF(I)

FIND INTAKE MANIFOLD HEAT TRANSFER AND PRESSURE DROP:

YDOT(11) = ENGM(1)
CALL QMAN(Y(2), Y(3), Y(4), RHOM(1), I1, YDOT(11), Q1)
CALL DPMAN (I1, YDOT(11), RHOM(1), DP(2))

FIND COMPRESSOR DISCHARGE PRESSURE:
PRSS(1) = Y(3) + DP(2) + DP(1)

FIND COMPRESSOR MASS FLOW CHARACTERISTIC FROM MAP:
PR(1) = PRSS(1) / PINLET
ASP(1) = Y(9) * RCR(3)
CALL ICMAP(ASP(1), PR(1), CM, CRPM, SIZC, SIZ1, CMAP)

NOTE THAT IF AN INCOMPATIBLE INPUT SET OF SPEED AND PRESSURE RATIO IS SUPPLIED TO ICMAP, THE ROUTINE WILL RETURN A NEW SPEED THAT IS COMPATIBLE WITH THE INPUT PRESSURE RATIO, WHICH REMAINS UNCHANGED. THE NEW SPEED WILL BE USED IN THE FOLLOWING STEPS OF THE MATCHING PROCESS.

Y(9) = ASP(1) / RCR(3)

CONVERT FROM LB/MIN, CORRECTED, TO KG/DEG, ACTUAL:

RMASS(3) = CMAP(1)
CMAP(1) = RMASS(3) * RCR(2) * PINLET/PSTD(1) * RCR(3)

YDOT(12) = CMAP(1)

EVALUATE SPECIFIC ENTHALPY AND DISCHARGE TEMPERATURE FROM COMPRESSOR:

CALL DELH(RTEMP(1), RTEMP(2), PINLET, PRSS(1), 0., CMAP(2), & RTEMP(2), H(1), HC, H(3))

FIND MANIFOLD SPECIFIC ENTHALPY:

CALL THERMO (Y(2), Y(3), Y(4), HM(1), CPM(1), CM1, CM2, & RHOM(1), CM4, CM5, CM6, GM(1), MWM(1), CM9, CM10, CM11)

FIND INTERCOOLER DISCHARGE TEMPERATURE AND SPECIFIC ENTHALPY:

TMI = (RTEMP(2) + HI(3))/2.
CMIN = CMAP(1) / ESPD * CPI
HI(1) = 1. - EXP(-HI(5)/CMI)
HI(3) = RTEMP(2) - HI(1) * (RTEMP(2)-HI(2))

CALL THERMO (HI(3), Y(3), 0., HI(4), CPHI, DM1, DM2, DM3,
& DM4, DM5, DM6, DM7, DM8, DM9, DM10, DM11)  

C FIND EXHAUST MANIFOLD PRESSURE DROP  
YDOT(13) = ENG(2)  
CALL DPMAN (I2, YDOT(13), RHOMA, DP(3))  
C ASSIGN TURBINE INLET TEMPERATURE ACCORDING TO EXHAUST OPTION  
IF (EXSUB) THEN  
  IREF = 44 + 8*(NUMEX-1)  
  TURBIT = Y(IREF)  
ELSE  
  TURBIT = Y(6)  
ENDIF  
C IF (POWER) GO TO 20  
C FOLLOW THIS SECTION OF THE PROGRAM FOR A SIMPLE TURBOCHARGED CALCULATION (NO POWER TURBINE):  
C FIND TURBOCHARGER TURBINE INLET PRESSURE, OUTLET PRESSURE, AND HENCE PRESSURE RATIO:  
18 PRSS(2) = Y(7) - DP(3)  
PRSS(3) = PRSS(5)  
PR(2) = PRSS(2)/PRSS(3)  
C FIND TURBINE MAP FLOW AND EFFICIENCY FROM MAP:  
RCORR(4) = SQRT(TSTD(2)/TURBIT)  
ASP(2) = Y(9) * RCORR(4)  
CALL IPTMAP(ASP(2), PR(2), TM, TRPM, SIZT, SIZ2, TMAP)  
RMASS(4) = TMAP(1)  
C CONVERT FROM CORRECTED LBS/MIN MAP FLOW TO ACTUAL KG/DEG MASS FLOW:  
TRBM = RMASS(4) * RCORR(2) * PRSS(2) / PSTD(2) * RCORR(4)  
TEFF = TMAP(2)  
C YDOT(14) = TRBM  
C FIND TURBINE ENTHALPY CHANGE AND DISCHARGE TEMPERATURE:  
CALL DELH (TURBIT, RTEMP(3), PRSS(2), PRSS(3), Y(8), 1./TEFF,  
  RTEMP(3), HM(2), HT, H(4))  
C GO TO 40  
C FOLLOW THIS SECTION OF THE PROGRAM FOR A TURBOCHARGED TURBOCOMPONDED CALCULATION.  
20 CONTINUE  
DO 25 II = 1, 50  
C FIND FIRST TURBINE DOWNSTREAM PRESSURE AND EFFICIENCY  
C FROM TURBOCHARGER SPEED AND CORRECTED MASS FLOW:  
RCORR(4) = SQRT(TSTD(2)/TURBIT)  
ASP(2) = Y(9) * RCORR(4)  
CALL ITMAP (ASP(2), RMASS(4), TM, TRPM, SIZT, SIZ2, TMAP)  
PR(2) = TMAP(2)  
TEFF = TMAP(1)
PRSS(2) = Y(7) - DP(3)
PRSS(3) = PRSS(2) / PR(2)
PRSS(4) = PRSS(3) - DP(4)
PR(3) = PRSS(4) / PRSS(5)

CONVERT FROM CORRECTED LBS/MIN MASS FLOW TO ACTUAL KG/DEG MASS FLOW:
TRBM = RMASS(4) * RCORR(2) * PRSS(2) / PSTD(2) * RCORR(4)

FIND POWER TURBINE MASS FLOW RATE, GIVEN ENGINE SPEED, GEAR RATIO, AND POWER TURBINE PRESSURE RATIO.
RCORR(5) = SQRT(TSTD(3)/RTMP(4))
ASP(3) = ERPM * RCORR(1) * RCORR(5)
CALL IPTMAP (ASP(3), PR(3), PTRM, SIZPT, SIZ3, PTMAP)
RMASS(5) = PTMP(1)
PTRBM = RMASS(5) * RCORR(2) * PRSS(4) / PSTD(3) * RCORR(5)

IF (ABS(TRBM - PTRBM) .GT. 1.E-3 * PTRBM) GO TO 24
TRBM = PTRBM
GO TO 30

NOTE THAT THE NEW GUESS FOR THE TURBINE MASS FLOW IS MORE HEAVILY WEIGHTED TO THE OLD TURBINE MASS FLOW RATHER THAN THE POWER TURBINE MASS. THIS IMPROVES THE STABILITY OF THE MATCHING PROCESS AND REDUCES THE RUNNING TIME.

24 TRBM = 0.75*TRBM + 0.25*PTRBM
RMASS(4) = TRBM / RCORR(2) / PRSS(2) / RCORR(4) / PSTD(2)
25 CONTINUE

EQUATE THE ACTUAL TURBINE AND POWER TURBINE MASS FLOWS (ON A KG/DEG BASIS)
TRBM = PTRBM

30 YDOT(14) = TRBM

FIND TURBINE ENTHALPY CHANGE AND DISCHARGE TEMPERATURE:
CALL DELH (TURBIT, RTEMP(3), PRSS(2), PRSS(3), Y(8), 1./TEFF, & RTEMP(3), HM(2), HT, H(4))

FIND POWER TURBINE INLET TEMPERATURE:
CPTEMP = RTEMP(3)

CALCULATE HEAT TRANSFER AND PRESSURE DROP IN CONNECTING PIPE BETWEEN THE TWO TURBINES
CALL THERMO (CPTEMP, PRSS(3), Y(8), XX1, XX2, XX3, XX4, & RHOCP, XX6, XX7, XX8, XX9, XX10, XX11, XX12, XX13)

IF (EXSUB) THEN
IREF = NUMEX+1
CALL HTRATE (IREF, CPTEMP, PRSS(3), Y(8), TRBM, Q(IREF) )
ELSE
IREF = 3
CALL QMAN(CPTEMP, PRSS(3), Y(8), RHOCP, I3, TRBM, Q(3))
ENDIF
CALL DPMAN(I3, TRBM, RHOCP, DP(4))

DENTH = Q(IREF) / TRBM
HCP = HT - DENTH
CALL ITRATE (RTEMP(4), PRSS(3), Y(8), HCP, XX1, XX2, XX3, & XX4, XX5, XX6, XX7, XX8, XX9, XX10, XX11, XX12)

CALL ITRATE (RTEMP(4), PRSS(3), Y(8), HCP, XX1, XX2, XX3, & XX4, XX5, XX6, XX7, XX8, XX9, XX10, XX11, XX12)

FIND POWER TURBINE ENTHALPY CHANGE AND DISCHARGE TEMPERATURE:

CALL DELH (RTEMP(4), RTEMP(5), PRSS(4), PRSS(5), Y(8), & 1./PTMAP(2), RTEMP(5), HPT, HOUT, H(5))

CALCULATE POWER TURBINE INSTANTANEOUS POWER (J/KG)

NOTE NEGATIVE SIGN DUE TO THE DEFINITION OF ENTHALPY CHANGES IN DELH.

YDOT(10) = - YDOT(14) * H(5)

THIS SECTION IS COMMON FOR BOTH OPTIONS.

********* CALCULATE DERIVATIVES ******************

APPLY CONSERVATION OF MASS TO FIND INTAKE MANIFOLD MASS DERIVATIVE:

YDOT(1) = CMAP(1) - YDOT(11)

AY1 = YDOT(1) / Y(1)

FIND EQUIVALENCE RATIO DERIVATIVES:

FF = Y(4)

AA = - FMDOT(1) + YDOT(12) * (0. - FF)

FDOT = AA / Y(1)

YDOT(4) = FDOT

PHDOTI = FDOT * AFRAST / (1. - FF)**2

CALCULATE TEMPERATURE AND PRESSURE DERIVATIVES FOR INTAKE MANIFOLD:

H11 = HI(4) * CMAP(1)

HNETI = H11 - HMDOT(1) - Q1

YDOT(2) = (AY1*(CM10-HM(1)) - CM11*PHDOTI + HNETI/Y(1))/CM9

YDOT(3) = (AY1 * RHOM(1) - CM8 * PHDOTI - CM4 *YDOT(2))/CM5

FIND PROPERTIES OF AVERAGED EXHAUST MANIFOLD:

CALL THERMO (Y(6), Y(7), Y(8), HM(2), CPM(2), EXA1, EXA2, & RHOMA, EXA4, EXA5, EXA6, GM(2), MWM(2), EXA9,EXA10,EXA11)

RHOM(2) = RHOMA

EHMA = HM(2)

IF EXHAUST MANIFOLD SUB-VOLUMES ARE NOT SPECIFIED, SKIP OVER NEXT S

IF (.NOT.EXSUB) GOTO 50

Determine properties of exhaust manifold control volumes

DO 100 I = 1, NUMEX

ITEMP = 44 + (I-1)*8

IFR = ITEMP + 1
CALL THERMO (Y(ITEMP), Y(7), Y(IFR), EHM(I), ECP(I),
& EX1(I), EX2(I), ERHO(I), EX4(I), EX5(I), EX6(I),
& EGAMMA(I), EWM(I), EX9(I), EX10(I), EX11(I))
100 CONTINUE
C SET PROPERTIES OF GAS DOWNSTREAM OF EXHAUST VALVE
HM(2) = EHM(I)
CPM(2) = ECP(I)
RHOM(2) = ERHO(I)
GM(2) = EGAMMA(I)
MWM(2) = EWM(I)

C THE FOLLOWING SECTION IS ITERATED TO CONVERGE ON
C CONSISTENT CONDITIONS
DO 125 JJ=1,15
C
C FIND MASS STORAGE RATES IN CONTROL VOLUMES
DO 120 I=1, NUMEX
IMASS = 43 + (I-1)*8
ITEMP = IMASS + 1
IFR = ITEMP + 1
YDOT(IMASS) = Y(IMASS) * (EX5(I)*YDOT(7) + EX4(I)*YDOT(ITEMP)
& + EX6(I)*PHDOT(I)) / ERHO(I)
120 CONTINUE
C
C FIND FLOWS BETWEEN CONTROL VOLUMES
EMFLO(1) = YPM(2)
DO 130 I = 1, NUMEX
IMASS = 43 + (I-1)*8
EMFLO(I+1) = EMFLO(I) - YDOT(IMASS) + ENGMX(I)
130 CONTINUE
C
C SET PROPERTIES OF FLOW BETWEEN VOLUMES
EHFLO(1) = YPH(2)
EFFLO(1) = YPF(2)
C
DO 140 I = 2, NUMEX
IFR = 45 + (I-1)*8
IF (EMFLO(I)) 141, 141, 142
C
C REVERSE FLOW
141 EHFLO(I) = EHM(I)
EFFLO(I) = Y(IFR)
GOTO 140
C
C FORWARD FLOW
142 EHFLO(I) = EHM(I-1)
EFFLO(I) = Y(IFR-8)
C
140 CONTINUE
EHFLO(I) = EHM(I-1)
EFFLO(I) = Y(IFR)
C
**************
C DETERMINE HEAT TRANSFER IN EACH SECTION
DO 150 I = 1, NUMEX
ITEMP = 44 + (I-1)*8
IFR = ITEMP + 1
FLOAVE = (EMFLO(I) + ENGMX(I) + EMFLO(I+1)) / 2.
CALL HTRATE (I, Y(ITEMP), Y(7), Y(IFR), FLOAVE, Q(I))
CONTINUE

*************

DO 160 I = 1, NUMEX
   IMASS = 43 + (I-1)*8
   ITEMP = IMASS + 1
   IFR = ITEMP + 1
   IREF = I + 1
   APPL CONSERVATION OF FUEL FRACTION
   FF = Y(IFR)
   EFFR(I) = FF
   FDOT = ( EMFLO(I)*(EFFLO(I)-FF) - EMFLO(IFREF) * (EFFLO(IFREF)-FF) + FMDOTX(I) ) / Y(IMASS)
   YDOT(IFR) = FDOT
   PHDOT(I) = FDOT * AFRAST / (1 - FF)**2

CALCULATE TEMPERATURE DERIVATIVE

HNET = EMFLO(I)*EHFLO(I) + HMDOTX(I) - EMFLO(IFREF)*EHFLO(IFREF) - Q(I)
AY = YDOT(IMASS) / Y(IMASS)
YDOT(ITEMP) = (AY*(EX10(I)-EHM(I)) - EX11(I)*PHDOT(I) + HNET/Y(IMASS) ) / EX9(I)

160 CONTINUE

APPLY CONSERVATION OF MASS TO AVERAGED MANIFOLD
   YDOT(5) = YDOT(13) - TRBM
   AYA = YDOT(5) / Y(5)
   APPL CONSERVATION OF FUEL FRACTION
   FF = Y(8)
   FDOT = FMDOT(2) / Y(5)
   YDOT(8) = FDOT
   PHDOTA = FDOT * AFRAST / (1 - FF)**2
   CALCULATE TEMPERATURE AND PRESSURE DERIVATIVE FOR AVERAGED MANIFOLD:
   QAM = QDOTX
   DO 170 I = 1, NUMEX
   170 QAM = QAM + Q(I)
   HNETA = HMDOT(2) - TRBM*EHMA - QAM
   YDOT(6) = (AYA*(EXA10-EHMA) - EXA11*PHDOTA + HNETA/Y(5) ) / EXA9
   YDOT(7) = ( AYA*RHOMA - EXA6*PHDOTA - EXA4*YDOT(6) ) / EXA5
   CHECK CONVERGENCE OF EXHAUST MANIFOLD DERIVATIVES
   ENUM = 0
   EDEN = 0
   IST = 5
   DO 123 J=1,NUMEX+1
   IREF1 = IST + ((J-1)*8)
   DO 124 J2 = 1, 3
IREF = IREF1 + (J2-1)
DELTA = YDOT(IREF)-YDOT1(IREF)
ENUM = DELTA+DELTA + ENUM
EDEN = YDOT(IREF)+YDOT(IREF) + EDEN
YDOT1(IREF) = YDOT(IREF)

CONTINUE

IST = 35

123 CONTINUE
ILOPC = ILOPC + 1
IF (ENUM/EDEN .LT. 5.E-05) GOTO 126

125 CONTINUE

C
DEFINE DERIVATIVES TO INTEGRATE EXH. MANIFOLD HEAT TRANSFER
C

126 CONTINUE

DO 180 I = 1, NUMEX
ITEMP = 44 + (I-1)*8
IREF = I + 2
YDOT(ITEMP+2) = Y(ITEMP) / 720.
YDOT(ITEMP+3) = Y(ITEMP) * (EMFLO(I) + ENGMX(I))
YDOT(ITEMP+4) = Q(I)
YDOT(ITEMP+5) = HTCO(IREF)
YDOT(ITEMP+6) = HTCO(IREF) * Y(ITEMP)
TGAS(IREF) = Y(ITEMP)

180 CONTINUE

C
IF (.NOT. POWER) GOTO 60
ITEMP = 44 + NUMEX*8
YDOT(ITEMP+4) = Q(NUMEX+1)
YDOT(ITEMP+5) = HTCO(NUMEX+3)
YDOT(ITEMP+6) = HTCO(NUMEX+3) * CPTEMP

GOTO 60

**********************
FOR SINGLE EXHAUST MANIFOLD VOLUME CONTINUE:
C

50 CALL QMAN (Y(6), Y(7), Y(8), RHOMA, I2, TRBM, QAM)

C
APPLY CONSERVATION OF MASS TO MANIFOLD
YDOT(5) = YDOT(13) - TRBM
AYA = YDOT(5) / Y(5)

C
APPLY CONSERVATION OF FUEL FRACTION
FF = Y(8)
FDOT = FMDOT(2) / Y(5)
YDOT(8) = FDOT
PHDOTA = FDOT * AFRAST / (1 - FF)**2

C
CALCULATE TEMPERATURE AND PRESSURE DERIVATIVE FOR EXHAUST MANIFOLD:
HNETA = HMDOT(2) - TRBM*EHMA - QAM
YDOT(6) = (AYA*(EXA10-EHMA) - EXA11*PHDOTA + HNETA/Y(5) ) / EXAS

YDOT(7) = ( AYA*RHOMA - EXA6*PHDOTA - EXA4*YDOT(6) ) / EXAS

CONTINUE FOR BOTH EXHAUST MANIFOLD OPTIONS:
CALCULATE TURBOCHARGER SPEED DERIVATIVE:

\[ W_{DOT} = - (H(4) \cdot Y(14) + H(3) \cdot CMAP(1)) \cdot 9.E-4 / \pi^{2} \]

\[ Y_{DOT}(9) = (W_{DOT} / Y(9) - B(2) \cdot Y(9) \cdot ESPD) / B(1) \]

PERFORM TIME-AVERAGING OF PRESSURES, TEMPERATURES, AND TURBOMACHINERY VARIABLES.

\[ Y_{DOT}(15) = Y(3) / 720. \]
\[ Y_{DOT}(16) = Y(7) / 720. \]
\[ Y_{DOT}(17) = RTEMP(2) / 720. \]
\[ Y_{DOT}(18) = HI(3) / 720. \]
\[ Y_{DOT}(19) = Y(2) / 720. \]
\[ Y_{DOT}(20) = Y(8) / 720. \]
\[ Y_{DOT}(21) = RTEMP(3) / 720. \]
\[ Y_{DOT}(22) = RTEMP(5) / 720. \]
\[ Y_{DOT}(23) = HI(1) / 720. \]
\[ Y_{DOT}(24) = RMASS(3) / 720. \]
\[ Y_{DOT}(25) = TRBM / RRCORR(2) / PRSS(2) / RRCORR(4) \cdot PSTD(2) / 720. \]
\[ Y_{DOT}(26) = ASP(1) / 720. \]
\[ Y_{DOT}(27) = ASP(2) / 720. \]
\[ Y_{DOT}(28) = ASP(3) / 720. \]
\[ Y_{DOT}(29) = PR(2) / 720. \]
\[ Y_{DOT}(30) = CMAP(2) / 720. \]
\[ Y_{DOT}(31) = TEFF / 720. \]
\[ Y_{DOT}(32) = PTEM(2) / 720. \]
\[ Y_{DOT}(33) = PTTEM / RRCORR(2) / PRSS(4) / RRCORR(4) \cdot PSTD(3) / 720. \]
\[ Y_{DOT}(40) = RTEMP(4) / 720. \]

TRANSFER DERIVATIVES BACK INTO DOUBLE PRECISION ARRAY

DO 200 I=1, NEQN-29
\[ DYPDIF(I) = Y_{DOT}(I) \]
RETURN
END
SUBROUTINE DPMAN

PURPOSE
CALCULATES PRESSURE DROP ACROSS MANIFOLD

USAGE
CALL DPMAN (J, ENGM, RHOB, PDROP)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>J</td>
<td>YES</td>
<td>NO</td>
<td>INDEX (INTAKE: J-1, EXHAUST: J-2 CONNECTING PIPE: J-3)</td>
</tr>
<tr>
<td>ENGM</td>
<td>YES</td>
<td>NO</td>
<td>TOTAL ENGINE MASS FLOW RATE (KG/DEG)</td>
</tr>
<tr>
<td>RHOB</td>
<td>YES</td>
<td>NO</td>
<td>BULK DENSITY(KG/M**3)</td>
</tr>
<tr>
<td>PDROP</td>
<td>NO</td>
<td>YES</td>
<td>PRESSURE DROP (PA)</td>
</tr>
</tbody>
</table>

REMARKS
REPLACES PRESSURE DROP PORTION OF QDP IN THE ORIGINAL VERSION OF THE PROGRAM

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED

THERMO TRANSP

METHOD
PRESSURE DROP CALCULATED BASED ON EMPIRICAL DATA.
SEE ROSHENOW & CHOI, 'HEAT, MASS AND MOMENTUM TRANSFER'.

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EDITED BY D. N. ASSANIS & R.M. FRANK

SUBROUTINE DPMAN (J, ENGM, RHOB, PDROP)

COMMON/EMKT/ EMKT(3)
COMMON/DDTH/ ESPD
COMMON/QP1/ EDIAM(3), EAREA(3), ECROSS(3), ETWALL(3), ECONHT(3)
COMMON/QP2/ EPRF(3), EVBLK(3), EREF(3), ENUF(3), EHTCOE(3), & EQDOT(3)

EVBLK(J) = ENGM / ESPD / RHOB / ECROSS(J)

CALCULATE PRESSURE DROP:

PDROP = EMKT(J) * RHOB * EVBLK(J) * EVBLK(J) / 2.

RETURN
END
FUNCTION EQR

PURPOSE
TO CONVERT FUEL FRACTION INTO EQUIVALENCE RATIO

DESCRIPTION OF PARAMETERS
INPUT: FFR  FUEL FRACTION (THE RATIO OF MASS OF PRODUCTS
OF COMBUSTION TO TOTAL MASS OF AIR AND
COMBUSTION PRODUCTS)
OUTPUT: EQR  EQUIVALENCE RATIO (THE RATIO OF THE
ACTUAL FUEL-AIR RATIO TO THE STOICHIOMETRIC
FUEL-AIR RATIO)
COMMON: AFRAST  STOICHIOMETRIC AIR-FUEL RATIO

REMARKS
NONE

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
NONE

WRITTEN BY D.N. ASSANIS

FUNCTION EQR (FFR)

COMMON/FUEL/ FUELTP, CX, HY, DEL, PSI, AFRAST, QLOWER, HFORM

EQR = FFR * AFRAST / (1. - FFR)

RETURN
END
SUBROUTINE EXAUST

PURPOSE
C CALCULATES THE TIME RATE OF CHANGE OF PRESSURE, TEMPERATURE,
C FUEL EQUIVALENCE RATIO, MASS, HEAT TRANSFER, WORK TRANSFER,
C MEAN KINETIC ENERGY, AND TURBULENT KINETIC ENERGY IN THE
C MASTER CYLINDER DURING EXAUST.

USAGE
C CALL EXAUST (NEQN, DT, DY, DYP)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>NEQN</td>
<td>YES</td>
<td>NO</td>
<td>NUMBER OF EQUATIONS TO BE INTEGRATED</td>
</tr>
<tr>
<td>DT</td>
<td>YES</td>
<td>NO</td>
<td>TIME (DEG)</td>
</tr>
<tr>
<td>DY(1)</td>
<td>YES</td>
<td>NO</td>
<td>MASS INDUCTED INTO CHAMBER THROUGH INTAKE VALVE (KG)</td>
</tr>
<tr>
<td>DY(2)</td>
<td>YES</td>
<td>NO</td>
<td>MASS EXHAUSTED FROM CHAMBER THROUGH EXHAUST VALVE (KG)</td>
</tr>
<tr>
<td>DY(6)</td>
<td>YES</td>
<td>NO</td>
<td>MEAN KINETIC ENERGY IN CHAMBER (J)</td>
</tr>
<tr>
<td>DY(7)</td>
<td>YES</td>
<td>NO</td>
<td>TURBULENT KINETIC ENERGY IN CHAMBER (J)</td>
</tr>
<tr>
<td>DY(8)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - PISTON TOP (J)</td>
</tr>
<tr>
<td>DY(9)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - CYLINDER HEAD (J)</td>
</tr>
<tr>
<td>DY(10)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - CYLINDER WALL (J)</td>
</tr>
<tr>
<td>DY(11)</td>
<td>YES</td>
<td>NO</td>
<td>CYLINDER TEMPERATURE (K)</td>
</tr>
<tr>
<td>DY(12)</td>
<td>YES</td>
<td>NO</td>
<td>CYLINDER PRESSURE (PA)</td>
</tr>
<tr>
<td>DY(16)</td>
<td>YES</td>
<td>NO</td>
<td>TOTAL WORK TRANSFER (J)</td>
</tr>
<tr>
<td>DY(17)</td>
<td>YES</td>
<td>NO</td>
<td>TOTAL ENTHALPY EXHAUSTED (J)</td>
</tr>
<tr>
<td>DYP(1)</td>
<td>NO</td>
<td>YES</td>
<td>RATE AT WHICH MASS IS INDUCTED THROUGH THE INTAKE VALVE (KG/DEG)</td>
</tr>
<tr>
<td>DYP(2)</td>
<td>NO</td>
<td>YES</td>
<td>RATE AT WHICH MASS IS EXHAUSTED THROUGH THE EXHAUST VALVE (KG/DEG)</td>
</tr>
<tr>
<td>DYP(6)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF MEAN KINETIC ENERGY (J/DEG)</td>
</tr>
<tr>
<td>DYP(7)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF TURBULENT KINETIC ENERGY (J/DEG)</td>
</tr>
<tr>
<td>DYP(8)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF HEAT TRANSFER - CYLINDER HEAD (J/DEG)</td>
</tr>
<tr>
<td>DYP(9)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF HEAT TRANSFER - PISTON TOP (J/DEG)</td>
</tr>
<tr>
<td>DYP(10)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF HEAT TRANSFER - CYLINDER WALL (J/DEG)</td>
</tr>
<tr>
<td>DYP(11)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF CYLINDER TEMPERATURE (K/DEG)</td>
</tr>
<tr>
<td>DYP(12)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF CYLINDER PRESSURE (PA/DEG)</td>
</tr>
<tr>
<td>DYP(16)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF TOTAL WORK TRANSFER (J/DEG)</td>
</tr>
<tr>
<td>DYP(17)</td>
<td>NO</td>
<td>YES</td>
<td>RATE AT WHICH TOTAL ENTHALPY IS EXHAUSTED (J/DEG)</td>
</tr>
</tbody>
</table>
REMARKS

UNITS CHANGED TO SI

NOTE THAT DERIVATIVES ARE PER REAL TIME (/SEC) UNTIL END OF SUBROUTINE WHERE THEY ARE CONVERTED TO PER CRANKANGLE

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED

THERMO TRANSP MFLRT
CSAVDV VACDEX DIFEQ

METHOD

SEE REPORT

WRITTEN BY D. N. ASSANIS AND S. G. POULOS
EDITED BY D. N. ASSANIS

SUBROUTINE EXAUST (NEQN, DT, DY, DYP)

REAL*8 DT, DY(NEQN), DYP(NEQN)
REAL MW, MWEM, MMM, KINVIS, MASS, MDOT, MDOTFR, MSTART, MACRSC, MFUEL
PARAMETER (PI = 3.141592654)
COMMON/EPARAM/ BORE, STROKE, CONRL, CYLCA, CSATDC, CMRTIO, CLVTDC
COMMON/BURN/ FMIN
COMMON/HTRC/ CONHT, EXPHT
COMMON/TEMPS/ NTEMP, TWALL(10), TSS(10), TSS2(10), TCW, TCW2
COMMON/DTDTH/ ESPD
COMMON/MA/ YPM(2), YPH(2), YPF(2)
COMMON/MSTA/ MSTART
COMMON/TIMES/ TIVO, TEVC, TIVC, TINJ, TIGN, TEVO
COMMON/B/ CPM(2), HM(2), MMM(2), GM(2), RHOM(2)
COMMON/HEATS/ CVHTRN, HTRCOE, HTPAPI, HTPAHD, HTPACW, HTRAPI, & HTRAHD, HTRACW, THTRAN
COMMON/TURBU/ CBETA, MACRSC, UPRIME, VMKE, VPISTO
COMMON/VALVE/ VIV, VEV
COMMON/AREAS/ AHEAD, APSTON
COMMON/HTCOTG/ HTC(10), TGAS(10)
COMMON/PHASE/ EPHAO, EPHAC, IPHASE, REJ
COMMON/PGEOM/ NEVLV, EPLNG(3), PDIAH(2)

VEV = 0.0
FRAEV = 0.0

DO 11 I = 1, 20
11 DYP(I) = 0.0

T = DT
TCYL = DY(11)
PCYL = DY(12)

FIND THERMODYNAMIC AND TRANSPORT PROPERTIES IN CYLINDER

TEM = DY(28)
PEM = DY(27)
FEM = DY(28)
HEM = HM(2)
MWEM = MMM(2)
GEM = GM(2)
RHOEM = RHOM(2)
FR = DY(20)
MFUEL = DY(4) * FMIN
CALL THERMO (TCYL, PCYL, FR, H, CSUBP, CSUBT, CSUBF,
& RHO, DRHOOT, DRHODP, DRHODF, GAMA, MW, ADUMY, BDUMY, CDUMY)
CALL TRANSP (TCYL, FR, GAMMA, CSUBP, DYNVIS, THRCON)
KINVIS = DYNVIS/RHO
MASS = MSTART + DY(1) - DY(2) + MFUEL

C IS EXHAUST VALVE STILL OPEN ?
IPHASE = 4
IF (T .GE. TEVC) GO TO 50
YES IT IS.
ANY FLOW ACROSS IT ?
IF (PCYL - PEM) 30, 50, 40
YES, FLOW INTO CYLINDER.
FIND CD AND AREA FOR EXHAUST VALVE.

30 PR = PEM/PCYL
CALL VACDEX (T, AREA, CD)
IF (AREA .LE. 0.) GOTO 50
FIND MASS FLOW RATE.
CALL MFLRT (CD, AREA, PEM, MWEM, TEM, PCYL, GEM, FRAEV)
CALCULATE RATES DUE TO THIS FLOW.

DYP(2) = -FRAEV
VEV = -FRAEV/(RHOEM * AREA)
DYP(6) = .5 * FRAEV * VEV*VEV
DYP(20) = (FEM - DY(20)) * FRAEV / MASS
YPH(2) = HEM
YPF(2) = FEM
IPHASE = 2
GO TO 50
FLOW FROM CYLINDER INTO EXHAUST MANIFOLD.
FIND AREA AND CD FOR EXHAUST VALVE.

40 PR = PCYL/PEM
CALL VACDEX (T, AREA, CD)
IF (AREA .LE. 0.) GOTO 50
FIND MASS FLOW RATE.
CALL MFLRT (CD, AREA, PCYL, MW, TCYL, PEM, GAMMA, FRAEV)
CALCULATE RATES DUE TO THIS FLOW

DYP(2) = FRAEV
VEV = FRAEV/(RHO * AREA)
DYP(6) = -FRAEV * (DY(6)/MASS)
DYP(7) = -FRAEV * (DY(7)/MASS)
DYP(20) = 0.
YPH(2) = H
YPF(2) = FR

C FIND JET REYNOLDS NUMBER FOR EXHAUST PORT HEAT TRANSFER
REJ = VEV * PDIAM(1) * RHO / DYNVIS

C FIND SURFACE AREAS AND VOLUME OF CHAMBER

CALL CSADV (T, ACW, VOLUME, DVDT)
MACRSC = VOLUME/(3.1415 * BORE * BORE/4.)
IF (MACRSC .GE. (BORE/2.)) MACRSC = BORE/2.

DY(6) = DYP(6) -.3307 * CBETA/MACRSC * DY(6) * SQRT(DY(7)/MASS)
DYP(7) = DYP(7) + .3307 * CBETA/MACRSC * DY(6) * SQRT(DY(7)/MASS)
& -.5443 * DY(7)/MACRSC * SQRT(DY(7)/MASS)

MDOT = -DYP(2)

C CHARACTERISTIC VELOCITY IN CYLINDER (M/SEC)

CONSTR = CONRL/STROKE
SINTH = SIN( T*PI/180. )
COSTH = COS( T*PI/180. )
VONVPM = ABS( PI * SINTH * ( 1. + COSTH/SQRT( 4. * CONSTR*CONSTR
& - SINTH*SINTH ) )/2. )
VPMEAN = STROKE/(180. * ESPD)
VPISTO = VPMEAN * VONVPM
VMKE = SQRT( 2. * DY(6)/MASS )
UPRIME = SQRT( .666667 * DY(7)/MASS )

C ADD VELOCITY TERM DUE TO BLOWDOWN

VBDOWN = 4 * DYP(2)/ ( 3.1415925 * RHO * BORE * BORE )
IF ( DYP(2) .LT. 0.0 ) VBDOWN = 0.0
CVHTRN = SQRT( 0.25*VPISTO*VPISTO + VMKE*VMKE + UPRIME*UPRIME +
& VBDOWN*VBDOWN )

C CALCULATE HEAT TRANSFER RATES

HTRCOE = CONHT* ( (CVHTRN+MACRSC/KINVIS)**EXPHT ) * THRCON/MACRSC

HTPAPI = HTRCOE * ( TCYL - TWALL(1) )
HTPAHD = HTRCOE * ( TCYL - TWALL(2) )
HTPACW = HTRCOE * ( TCYL - TCW )

TGAS(1) = TCYL
TGAS(2) = TCYL
HTCO(1) = HTRCOE
HTCO(2) = HTRCOE
HLINS = HTRCOE

HTGPI = HTRCOE * TCYL
HTGHD = HTRCOE * TCYL
HTGCW = HTRCOE * TCYL

HTRAPI = APSTON * HTPAPI
HTRAHD = AHEAD * HTPAHD
HTRACW = ACW * HTPACW

THTRAN = HTRAPI + HTRAHD + HTRACW

C CALCULATE RATES OF CHANGE OF TEMPERATURE AND PRESSURE IN
C THE CYLINDER. THEN CALCULATE RATE OF DOING WORK.

60 DYP(11) = -(BDUMY/ADUMY)*( DYP(2)/MASS + DVDT/VOLUME +
& CDUMY/BDUMY * PHIDOT + THTRAN/(BDUMY*MASS) )

C DYP(12) = RHO/DRHODP * ( -DVDT/VOLUME - PHIDOT*DRHODF/RHO
& - DYP(11)*DRHODT/RHO - DYP(2)/MASS )

C DYP(16) = PCYL * DVDT

C DYP(8) = HTRAPI
DYP(9) = HTRAHD
DYP(10) = HTRACW

C DYP(17) = DYP(2) * H

C CONVERT ALL TIME DERIVATIVES TO RATE PER CRANK ANGLE DEGREE.

DO 70 I = 1, 20
70 DYP(I) = DYP(I) * ESPD
YPM(2) = DYP(2)

C CALL DIFEQ (NEQN, T, DY(21), DYP(21))

C DYP(54) = HTRCOE
DYP(55) = HTGPI
DYP(56) = HTRCOE
DYP(57) = HTGHD
DYP(58) = HTRCOE
DYP(59) = HTGCW

C RETURN
END
SUBROUTINE EXSUM

PURPOSE
CALCULATES TOTAL AND FUEL MASS FLOW RATES AND ENTHALPY FLUXES THAT ENTER EACH SECTION OF THE EXHAUST MANIFOLD. EXSUM IS ONLY USED FOR THE EXHAUST SUB-CONTROL VOLUME MODEL (EXSUB=TRUE).

METHOD
THE ALGORITHM SUMS THE MASS FLOW RATES (TOTAL AND FUEL), AND THE ENTHALPY FLUXES CONTRIBUTED BY OTHER SECTIONS OF THE EXHAUST EVERY INSTANT. THE MASS FLOW RATE AND ENTHALPY FLUX PROFILES OF THE MASTER VOLUME UPSTREAM OF A GIVEN NODE ARE STORED IN ARRAYS AND RETRIEVED WITH THE PROPER PHASE SHIFT IN ORDER TO DETERMINE THE ADDITION OF MASS AND ENTHALPY ALONG THE LINE OF THE EXHAUST MANIFOLD.

REMARKS
EXSUM CORRESPONDS WITH VLVSUM. HOWEVER, NOTE THAT IN EXSUM CALCULATES THE CONTRIBUTION FROM ALL SECTIONS EXCEPT THE MASTER SECTIONS WHEREAS VLVSUM INCLUDES THE FLOWS FROM ALL CYLINDERS (INCLUDING THE MASTER CYLINDER).

WRITTEN BY R.M. FRANK

UPDATE ARRAYS FOR FUTURE CALLS TO EXSUM

IREF = 1
QSEG = 0.
DO 10 I = 2, NUMEX
   QSEG = QSEG + QWALL(I)
   IF (NPIPE(I).EQ.1) GOTO 10
   XMASS(I) = EMFLO(I)
   XF(I) = EFLO(I)
   XH(I) = EHFLO(I)
   QLOSS(I) = QSEG
   QSEG = 0.
   IREF = IREF + 1
10 CONTINUE

ACCOUNT FOR MANIFOLD HEAT TRANSFER
QDOTX = 0.
IREF = 1
DO 20 I = 1, NUMEX
   IF (NPipe(I).EQ.1) GOTO 20
   NUMS = NUMSEG(I-1)
   IDEL = 720 / NUMS
   IT = IVLV
   DO 30 J = 1, NUMS-1
      IT = IT - IDEL
      IF (IT .LE. 0) IT = IT + 720
      QDOTX = QDOTX + QLOSS(IREF, IT)
   30 CONTINUE
   IREF = IREF + 1
20 CONTINUE

C SUM UP MASS FLOW RATES, ENTHALPY FLUXES, FUEL FRACTION
   FLUXES CONTRIBUTED BY EACH CONTROL VOLUME AT THAT INSTANT.
IREF = 1
DO 100 I = 1, NUMEX
   ENG = 0.
   FDOT = 0.
   HDOT = 0.
   IDEL = NSHFT(I)
   IT = IVLV
   C SUM FLOWS FOR OTHER SECTIONS
   NPIPE(I) IS THE NUMBER OF PIPES COMING TOGETHER AT A NODE
   NPIPE(I) = 1 INDICATES A STRAIGHT CONNECTION WITH NO
   ADDITIONAL PIPES COMING IN. THE FOLLOWING LOOP IS SKIPPED
   IF NPIPE(I) = 1.
   C DO 200 J = 1, NPIPE(I)-1
   C FIND SHIFTED TIME FOR EACH SECTION
   IT = IT - IDEL
   IF (IT .LE. 0) IT = IT + 720
   FLOW = XMASS(IREF, IT)
   C CHECK DIRECTION OF FLOW
   IF(FLOW) 40, 200.
   C INITIALIZE F AND H FOR FLOW OUT OF MANIFOLD
   F = 0.
   H = FLOW * EHM(I)
   GOTO 60
   C INITIALIZE F AND H FOR FLOW INTO MANIFOLD
   F = FLOW * (XF(IREF, IT) - EFFR(I))
   H = FLOW * XH(IREF, IT)
   40 C SUM FLOW
   FDOT = FDOT + F
   HDOT = HDOT + H
   ENG = ENG + FLOW
200 CONTINUE
   C ENGMX(I) = ENG
   FMDFX(I) = FDOT
   HMDFX(I) = HDOT
   IF (NPipe(I).NE.1) IREF = IREF + 1
121  100 CONTINUE
122       C
123         RETURN
124       END
SUBROUTINE EXWRIT

PURPOSE
WRITES OUTPUT FOR EXHAUST MANIFOLD AND THE CONNECTING PIPE BETWEEN THE TURBINES BOTH FOR INPUT PARAMETERS AND OPERATING OUTPUT FOR CURRENT RUN.

USAGE
CALL EXWRIT(NEQN, DY, ITIME)

REMARKS
EXWRIT IS WRITTEN IN TWO SECTIONS. THE FIRST CONTAINS THE OUTPUT STATEMENTS FOR THE INPUT DATA WHICH IS PRINTED OUT IN THE BEGINNING OF THE PROGRAM. THE SECOND SECTION DIRECTS THE OUTPUT BASED ON THE OPERATION OF THE ENGINE FOR THE CURRENT CYCLE SIMULATION. ITIME IS A FLAG USED TO INDICATE WHICH SECTION OF THE SUBROUTINE IS TO RUN.

SUBROUTINES AND FUNCTIONS REQUIRED
NONE

WRITTEN BY R.M. FRANK

SUBROUTINE EXWRIT(NEQN, DY, ITIME)

PARAMETER ( NCV = 8)
CHARACTER*16 MATL(NCV,3)
LOGICAL COM2, POWER
REAL*8 DT, DY(NEQN)
DIMENSION QTOT(NCV), QNORM(NCV), TMAVE(NCV), DTMAVE(NCV)

COMMON/POWER/ POWER
COMMON/EXNUM/ NUMEX, NUMEXT
COMMON/EXPAR/ NPIPE(NCV), NSHFT(NCV), NUMSEG(NCV)
COMMON/PEGOV/ NEXLAY(NCV), EXTHIK(NCV,3), XHCOOL(NCV), EXCOND(NCV,3), COM2, MATL
& EXCOND(NCV,3), COM2, MATL
COMMON/EGEOV/ EXLNG(NCV), EXDIAM(NCV), EXAREA(NCV),
& EXVOL(NCV), EXXAR(NCV)
COMMON/BEMSP/ BRAD(NCV), BANG(NCV)
COMMON/BEMP/ CBP(NCV), RLNG(NCV), ENCEE(NCV), VTOT, ATOT
COMMON/BMHT/ XTCOOL(NCV), UA(NCV)
COMMON/PARF/ PDELX(13), PCNUM(13), INNODE(103), PTHK(10),
& PCOND(10,3), PHEFF(10,3), PDIFU(10,3), PHCOOL(10),
& PTCOOL(10), PUOVE(10), PSURFA(10)
COMMON/QSOL/ QWALL(10), QAVE(1), QSS(10)
COMMON/TEMPS/ NTEMP, TWALL(10), TSS(10), TSS2(10), TCW, TCW2
COMMON/EXTINI/ EXTW(NCV)
COMMON/BURN/ FMIN
COMMON/FUEL/ FUELTP, CX, HY, DEL, PSI, AFRAST, QLOWER, HFORM
COMMON/D/ERPM
COMMON/EXHTEM/AVEREXT
COMMON/ICYL/ ICYL
C
C INITIALIZE OUTPUT FILE UNIT NUMBER
I=6
C
C IF (ITIME.NE.1) GOTO 1000
C
C ***********************************************************************
C
C OUTPUT FOR INPUT DATA AND DERIVED CONSTANTS
C
C ***********************************************************************
C
100 FORMAT ('/','WALL COMPOSITION OF PORT',/)
               & '/','--------------------------------',/)
               & '/','--------------------------------', /

WRITE (I,110)
110 FORMAT ('/','/','>>> EXHAUST MANIFOLD DESIGN PARAMETERS',/)
WRITE (I,125) NUMEX
125 FORMAT ('/','/','NUMBER OF SECTIONS TO EXHAUST (EXCLUDING TURBINE',
& ' & CONNECTING PIPE) - ',I1)
WRITE (I,126) (NPIPE(K), K=1,NUMEX)
126 FORMAT ('/','/','NPIPE - ',<NUMEX/(9X,I1))
WRITE (I,127) (NSHFT(K), K=1,NUMEX)
127 FORMAT ('/','/','NSHFT - ',<NUMEX/(7X,I3))
WRITE (I,128)
128 FORMAT ('/','/','PORT ******',/)
WRITE (I,130) NEVLV
130 FORMAT ('/','/','NUMBER OF EXHAUST VALVES PER CYLINDER - ',I1)
C
IF (NEVLV.NE.1) THEN
WRITE (I,140) PDIAM(1)
140 FORMAT ('/','/','DIAMETER OF INDIVIDUAL SECTIONS OF PORT = ',F5.3, 
& ' M')
WRITE (I,150) EPLNG(1), EPLNG(2)
150 FORMAT ('/','/','LENGTH OF INDIVIDUAL SECTIONS OF', 
& ' PORT = ',F6.3,', ',F6.3,' M')
ENDIF
C
WRITE (I,160)
160 FORMAT ('/','/','DIAMETER OF COMMON SECTION OF PORT = ',F5.3,' M')
WRITE (I,170) EPLNG(3)
170 FORMAT ('/','/','LENGTH OF COMMON SECTION OF PORT = ',F5.3)
C
WRITE (I,180)
180 FORMAT ('/','/','WALL COMPOSITION OF PORT')
WRITE (I,190)(K,K=1,NEXLAY(1))
190 FORMAT('0','/','LAYER',17X,5(I1,11X))
IF(MATL(1,1).NE.' ') WRITE (I,200)(MATL(1,K), K=1,NEXLAY(1))
200 FORMAT('0','/','MATERIAL',13X,5(A10,2X))
WRITE (I,210)(EXTHIK(1,K), K=1,NEXLAY(1))
210 FORMAT('0','/','THICKNESS',11X,5(F8.4,6X))
WRITE (I,220)(EXCOND(1,K),K=1,NEXLAY(1))
220 FORMAT('0','/','THERM. COND.',13X,5(F7.4,5X))
WRITE (I,222)(PDIFU(3,K),K=1,NEXLAY(1))
222 FORMAT('0','/','THERM. DIFFUSIVITY',5(E10.2,2X))
WRITE (I,224)(INNODE(3,K),K=1,NEXLAY(1))
224 FORMAT('0','/','NO. OF NODES ',5(3X,I2,7X))
C
DO 500 J = 2, NUMEX
WRITE (I,240) J
240 FORMAT (/,'0','****** EXHAUST VOLUME ','I2,', '******')
WRITE (I,250) EXDIAM(J), BRAD(J)
250 FORMAT ('0','INSIDE DIAMETER = ','F5.3,' & ' M',T65,'AVERAGE BEND RADIUS = ','F5.3, ' M')
WRITE (I,260) EXLNG(J), BANG(J)
260 FORMAT ('0','LENGTH = ','F6.3,' & ' M',T70,'TOTAL BEND ANGLE = ','F4.0, ' DEGREES')
IF (.NOT.COM2) THEN
WRITE (I,270) J
ELSEIF (J.EQ.2) THEN
WRITE (I,275)
ELSE
GOTO 500
ENDIF
270 FORMAT ('0','WALL COMPOSITION FOR VOLUME ','I2)
275 FORMAT ('0','WALL COMPOSITION OF EXHAUST MANIFOLD ')
WRITE (I,190)(K,K-1,NEXLAY(J))
WRITE (I,200)(MATL(J,K). K-i,NEXLAY(J))
WRITE (I,210)(EXTHIK(JK), K-1,NEXLAY(J))
WRITE (I,220)(EXCOND(JK), K-i,NEXLAY(J))
WRITE (I,222)(PDIFU(J+2,K),K-1 ,NEXLAY(J))
WRITE (I,224)(INNODE(J+2,K),K-1,NEXLAY(J))
500 CONTINUE
C
C IF (.NOT.POWER) GOTO 315
WRITE (I,600)
600 FORMAT (/,'****** TURBINE CONNECTING PIPE ******',/)
WRITE (I,610) EXDIAM(NUMEXT), BRAD(NUMEXT)
610 FORMAT ('0','INSIDE DIAMETER OF CONNECTING PIPE = ','F5.3,' & ' M',T65,'AVE. BEND RADIUS OF CONN.PIPE = ','F5.3, ' & ' M')
WRITE (I,620) EXLNG(NUMEXT), BANG(NUMEXT)
620 FORMAT ('0','LENGTH OF CONNECTING PIPE = ','F6.3,' & ' M',T70,'TOTAL BEND ANGLE OF CONN.PIPE = ','
 & ' F4.0, ' DEGREES')
WRITE (I,630)
630 FORMAT ('0','WALL COMPOSITION FOR CONNECTING PIPE')
WRITE (I,190)(K,K-1,NEXLAY(NUMEXT))
WRITE (I,200)(MATL(NUMEXT,K). K-i,NEXLAY(NUMEXT))
WRITE (I,210)(EXTHIK(NUMEXT,K), K-1,NEXLAY(NUMEXT))
WRITE (I,220)(EXCOND(NUMEXT,K), K-i,NEXLAY(NUMEXT))
C
315 CONTINUE
WRITE (I,320)
320 FORMAT (/,'****** BOUNDARY CONDITIONS ******',/)
IF (XHCOOL(1).EQ.e.) THEN
WRITE (I,329) XTCOOL(1)
329 FORMAT ('0','XTCOOL(1) = ',E10.3)
WRITE (I,336) ((K, XTCOOL(K)), K-2,NUMEX)
336 FORMAT ('0','XTCOOL(K) = ',E10.3, ' FOR K = 2, NUMEX')
IF (POWER) WRITE (I,331) XTCOOL(NUMEXT)
331 FORMAT ('0','XTCOOL(NUMEXT) = ',E10.3)
ELSE
WRITE (I,340) XTCOOL(1), XHCOOL(1)
340 FORMAT ('0','XTCOOL(1) = ',E10.3, ' XHCOOL(1) = ',E10.3)
WRITE (I,341) ((K, XTCOOL(K), XHCOOL(K)), K=2,NUMEX)
341 FORMAT ('0','XTCOOL(K) = ',E10.3, ' XHCOOL(K) = ',E10.3, ' FOR K = 2, NUMEX')
IF (POWER) WRITE (I,343) XTCOOL(NUMEXT), XHCOOL(NUMEXT)
ENDIF
FORMAT (' ', 'SPECIFIED OUTSIDE WALL SURFACE TEMPERATURE (K)', & '/6X, 'PORT (VOL 1) -- ', F5.0)
FORMAT (' ', 'CONTROL VOLUME', 'I2', -- ', F5.0)
FORMAT (' ', 'TURBINE CONNECTING PIPE -- ', F5.0)

FORMAT (' ', 'AMBIENT TEMPERATURE (K)', T50, 'OUTSIDE HEAT TRANSFER', & '/.6X. 'PORT (VOL 1) -- ', F5.0, T55, F7.1)
FORMAT (' ', 5X, 'CONTROL VOLUME', 'I2', -- ', F5.0, T55, F7.1)
FORMAT (' ', 'TURBINE CONNECTING PIPE -- ', F5.0, T55, F7.1)

WRITE (1,350) EXTW(1)
WRITE (1,330)((K, EXTW(K)), K=2,NUMEX)
WRITE (1,370) (K, K=2,NUMEX)
ELSE
WRITE (1,371) (K, K=2,NUMEX)
ENDIF

FORMAT ('/', T46, 4X, 'PORT', 8X, '<NUMEX-1>(C.V. I2,7X), 'CONN ', & 'PIPE')
FORMAT ('/', T46, 4X, 'PORT', 8X, '<NUMEX-1>(C.V. I2,7X))
WRITE (1,400)(EXXAR(K), K=1,NUMEXT)
& (E10.3,4X))
WRITE (1,410)(EXAREA(K), K=1,NUMEXT)
WRITE (1,420)(EXVOL(K), K=1,NUMEXT)
WRITE (1,430)(UA(K), K=1,NUMEXT)
WRITE (1,440)(ENCEE(K), K=1,NUMEXT)
WRITE (1,450)(RLNG(K), K=1,NUMEXT)
WRITE (1,460)(CBP(K), K=1,NUMEXT)
WRITE (1,480) VTOT, ATOT

WRITE (1,380) '****** COMPUTED CONSTANTS FOR EXHAUST'
& 'MANIFOLD ******')
IF (POWER) THEN
WRITE (1,370) (K, K=2,NUMEX)
ELSE
WRITE (1,371) (K, K=2,NUMEX)
ENDIF

FORMAT ('/', T46, 4X, 'PORT', 8X, '<NUMEX-1>(C.V. I2,7X), 'CONN ', & 'PIPE')
FORMAT ('/', T46, 4X, 'PORT', 8X, '<NUMEX-1>(C.V. I2,7X))
WRITE (1,400)(EXXAR(K), K=1,NUMEXT)
& (E10.3,4X))
WRITE (1,410)(EXAREA(K), K=1,NUMEXT)
WRITE (1,420)(EXVOL(K), K=1,NUMEXT)
WRITE (1,430)(UA(K), K=1,NUMEXT)
WRITE (1,440)(ENCEE(K), K=1,NUMEXT)
WRITE (1,450)(RLNG(K), K=1,NUMEXT)
WRITE (1,460)(CBP(K), K=1,NUMEXT)
WRITE (1,480) VTOT, ATOT
C
C     OUTPUT AT TERMINATION OF CYCLE
C
C     1000 CONTINUE
C     NUMEXT = NUMEX
C     IF (POWER) NUMEXT = NUMEX + 1
C     EFHTRT = ICYL * FMIN * QLOWER * ERPM / 120.
C     DO 1005 J = 1, NUMEXT
C          IREF = J + 2
C          ITEMP = 67 + (J-1)*8
C          QTOT(J) = QAVE(IREF) * EXAREA(J)
C          QNORM(J) = QTOT(J) * NUMSEG(J) / EFHTRT
C          TMAVE(J) = DY(ITEMP) * NUMSEG(J) / DY(34)
C     1005 CONTINUE
C     DTMAVE(1) = AVREXT - TMAVE(1)
C     DO 1006 J = 2, NUMEX
C          DTMAVE(J) = TMAVE(J-1) - TMAVE(J)
C     1006 CONTINUE
C     IF (POWER) DTMAVE(NUMEX+1) = DY(41) - DY(60)
C     DTTOT = AVREXT - TMAVE(NUMEX)
C     C
C     IF (POWER) THEN
C          WRITE (I,1010)
C     ELSE
C          WRITE (I,1011)
C     ENDIF
C     1010 FORMAT (///,'','>>> EXHAUST MANIFOLD AND TURBINE','
C                        & 'CONNECTING PIPE RESULTS',///)
C     1011 FORMAT (///,'','>>> EXHAUST MANIFOLD RESULTS',///)
C
C     WRITE (I,1020)
C     1020 FORMAT ('','HEAT TRANSFER DATA')
C     IF (POWER) THEN
C          WRITE (I,1030) (K, K-1.NUMEX)
C     ELSE
C          WRITE (I,1031) (K, K-1.NUMEX)
C     ENDIF
C     1030 FORMAT (',',T46,1X,<NUMEX>(1X,'C.V.', 12,
C                              7X),3X.'CONN PIPE')
C     1031 FORMAT (',',T46,1X,<NUMEX>(1X.'C.V.', 12,
C                              7X))
C
C     WRITE (I,1040) (QAVE(K), K=3,NUMEXT+2)
C     1040 FORMAT ('','4X,'AVE. HEAT FLUX OVER CYCLE (W/M**2)',
C              & T46,6(E11.3,3X))
C     WRITE (I,1050) (QTOT(K), K=1,NUMEXT)
C     1050 FORMAT ('','4X,'AVE. HEAT TRANSFER RATE FOR CONT VOL (W)',
C              & T46,6(E11.3,3X))
C
C     ILIM = 69 + ((NUMEXT-1)*8)
C     WRITE (I,1055) (DY(K), K=69,ILIM,8)
C     1055 FORMAT ('','4X,'AVE. HEAT TRANS COEF FOR CYCLE',
C              & ' (W/M**2/K)',T46,6(E11.3,3X))
C     C
C     WRITE (I,1060) (QNORM(K), K=1, NUMEXT)
C     1060 FORMAT ('','4X,'NORMALIZED HEAT TRANSFER RATE FOR SECT',
C              & T46,6(E11.3,3X))
C     C
C     WRITE (I,1070)
C     1070 FORMAT ('',' (TOTAL HEAT TRANSFER RATE/ENGINE').
& ' FUEL FLOW')
C WRITE (I,1090) (TWALL(K), K = 3, NUMEXT+2)
1090 FORMAT ('O', 'INSIDE WALL TEMPERATURE DATA (K)', & T46.6(F5.0,9X))
C WRITE (I,1100)
1100 FORMAT ('///, 'GAS TEMPERATURE DATA (K)')
WRITE (I,1110) (TMAVE(K), K = 1, NUMEX)
1110 FORMAT ('', 4X, 'MASS AVERAGED TEMPERATURE', & T46.6(F5.0,9X))
ILIM = 66 + ((NUMEX-1) * 8)
WRITE (I,1120) (DY(K), K = 66, ILIM, 8)
1120 FORMAT ('', 4X, 'TIME AVERAGED TEMPERATURE', & T46.6(F5.0,9X))
C WRITE (I,1130)
1130 FORMAT ('O', 'MASS AVERAGED TEMPERATURE DROP (K)')
WRITE (I,1139) DTMAVE(1)
1139 FORMAT ('', 'EXHAUST VALVE/PORT', T46, F4.0)
WRITE (I,1140) ((K-1,K,DTMAVE(K)), K = 2, NUMEX)
1140 FORMAT ('', 'BETWEEN CONTROL VOLUMES ', I1, ' & ', I1, T46, F4.0)
WRITE (I,1150) DTTOT
1150 FORMAT ('O', 'TOTAL TEMPERATURE DROP ', T46, F4.0)
C IF (.NOT. POWER) RETURN
DTCOMP = DY(41) - DY(60)
WRITE (I,1160) DTCOMP
1160 FORMAT ('O', 'TEMPERATURE DROP BETWEEN TURBINES', T46, F4.0)
RETURN
END
CFUNCTION FAHR

PURPOSE
CONVERTS TEMPERATURE IN DEGREES K TO DEGREES F.

PARAMETERS
INPUT: T   TEMPERATURE IN DEGREES KELVIN
OUTPUT: FAHR  TEMPERATURE IN DEGREES FAHRENHEIT

REMARKS
NONE

WRITTEN BY D.N. ASSANIS

FUNCTION FAHR (T)

FAHR = 9./5. * (T - 273.16) + 32.

RETURN
END
SUBROUTINE FILEDEF

C THIS SUBROUTINE OPENS FILES TO BE USED FOR DATA PLOTTING

LOGICAL EXSUB
COMMON/EXSUB/ EXSUB

C
OPEN (UNIT=3, FILE = 'COMAP.DAT', STATUS= 'OLD', READONLY)
OPEN (UNIT=4, FILE = 'TMAP.DAT', STATUS= 'OLD', READONLY)
OPEN (UNIT=5, FILE = 'PTMAP.DAT', STATUS= 'OLD', READONLY)
OPEN (UNIT=6, FILE = 'OUT1.DAT', STATUS= 'NEW')
OPEN (UNIT=7, FILE = 'TT:', STATUS= 'UNKNOWN')
OPEN (UNIT=8, FILE = 'FOR008', STATUS= 'OLD', READONLY)
OPEN (UNIT=12, FILE = 'FLOW.DAT', STATUS= 'NEW')
OPEN (UNIT=13, FILE = 'HEAT.DAT', STATUS= 'NEW')
OPEN (UNIT=14, FILE = 'RAD.DAT', STATUS= 'NEW')
OPEN (UNIT=15, FILE = 'RMAS.DAT', STATUS= 'NEW')

COUT
OPEN (UNIT=16, FILE = 'TITLE.DAT', STATUS= 'NEW')
OPEN (UNIT=75, FILE = 'TABLEX.DAT', STATUS= 'NEW')
OPEN (UNIT=76, FILE = 'TABLEX.DAT', STATUS= 'NEW')

IF EXHAUST MANIFOLD OPTION IS IN EFFECT, OPEN EXHAUST MANIFOLD
INPUT FILE
IF (EXSUB) OPEN (UNIT=9, FILE='FOR009', STATUS = 'OLD', READONLY)

DIRECT ACCESS FILES FOR PLOTTING ROUTINE (DATA.FOR)
open (unit=21, file = 'TURBOUT.dat' ,status= 'NEW',
1 ACCESS='DIRECT',FORM='FORMATTED',RECL=80,
2 RECORDTYPE='FIXED')
open (unit=22, file = 'ECHO.dat' ,status= 'NEW',
1 ACCESS='DIRECT',FORM='FORMATTED',RECL=80,
2 RECORDTYPE='FIXED')
open (unit=23, file = 'EXTEP.dat' ,status= 'NEW',
1 ACCESS='DIRECT',FORM='FORMATTED',RECL=80,
2 RECORDTYPE='FIXED')
open (unit=25, file = 'EXHTCO.dat' ,status= 'NEW',
1 ACCESS='DIRECT',FORM='FORMATTED',RECL=80,
2 RECORDTYPE='FIXED')

RETURN
END
SUBROUTINE FLAME

PURPOSE
CALCULATES APPARENT RADIANT TEMPERATURE BASED ON ADIABATIC
FLAME TEMPERATURE. THIS SUBROUTINE IS CALLED ONLY WHEN
THE NEW RADIATION MODEL OPTION IS SELECTED (I.E. NOT WITH
ANNAND'S MODEL OPTION)

USAGE
CALL FLAME (TG, P, TAIR, TFLAME, TRAD)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>TG</td>
<td>YES</td>
<td>NO</td>
<td>BULK GAS TEMPERATURE</td>
</tr>
<tr>
<td>P</td>
<td>YES</td>
<td>NO</td>
<td>BULK GAS PRESSURE</td>
</tr>
<tr>
<td>TAIR</td>
<td>NO</td>
<td>YES</td>
<td>TEMPERATURE OF AIR ZONES</td>
</tr>
<tr>
<td>TFLAME</td>
<td>NO</td>
<td>YES</td>
<td>ADIABATIC FLAME TEMPERATURE</td>
</tr>
<tr>
<td>TRAD</td>
<td>NO</td>
<td>YES</td>
<td>APPARENT RADIANT TEMPERATURE</td>
</tr>
</tbody>
</table>

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
NONE

METHOD
SEE D. N. ASSANIS, PH.D. THESIS

WRITTEN BY D. N. ASSANIS
EDITED BY D. N. ASSANIS

SUBROUTINE FLAME (TG, P, TAIR, TFLAME, TRAD)

COMMON/TOPO/ TO, PO, GAM

PARAMETER (ATPA=1.01325E5)

ESTIMATE AIR TEMPERATURE ASSUMING ADIABATIC COMPRESSION
FROM START OF IGNITION

TAIR = TO * (P / PO)**(GAM - 1.)/GAM
CALL THERMO (TAIR, P, 0. , X1, X2, X3, X4, X5, X6, X7, X8,
& GAM, X9, X10, X11, X12)

ESTIMATE ADIABATIC FLAME TEMPERATURE AS A FUNCTION OF
AIR TEMPERATURE AND PRESSURE, BASED ON CORRELATION
DERIVED BY CURVE-FITTING RESULTS OF NASA EQUILIBRIUM CODE.

PC = P /ATPA
IF(TG .GT. 800.) GO TO 20
TFLAME = (1. + 0.000249 * (TAIR - 650.)) *
& (2497.3 + 4.7521 * PC - 0.11065 *PC**2 + 0.000898 * PC**3)
GO TO 30

20 TFLAME = (1. + 0.0002317 * (TAIR - 950.)) *
& (2728.3 + 0.9306 * PC - 0.003233 * PC**2)

30 CONTINUE
ESTIMATE RADIANT TEMPERATURE AS THE MEAN OF THE GAS AND THE ADIABATIC FLAME TEMPERATURE.

TRAD = 0.5 * TFLAME + 0.5 * TG

RETURN

END
SUBROUTINE FUELDT

PURPOSE

THIS SUBROUTINE IS CALLED TO SET THE VALUES OF THE FUEL
RELATED PARAMETERS AT THE START OF PROGRAM EXECUTION. THE
ONLY INPUT REQUIRED IS THE FUEL TYPE. THE PARAMETERS WHICH
ARE SET ARE: I) THE ATOM RATIOS WHICH SPECIFY THE PROPERTIES
OF THE FUEL AIR MIXTURE
II) THE FUEL HEATING VALUE AND STOICHIOMETRIC FUEL/AIR RATIO;

USAGE

CALL FUELDT

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>FUELTP</td>
<td>YES</td>
<td>NO</td>
<td>FUEL TYPE</td>
</tr>
<tr>
<td>PSI</td>
<td>NO</td>
<td>YES</td>
<td>MOLE N2 TO O2 RATIO FOR AIR</td>
</tr>
<tr>
<td>CX</td>
<td>NO</td>
<td>YES</td>
<td># OF CARBON ATOMS/FUEL MOLECULE</td>
</tr>
<tr>
<td>HY</td>
<td>NO</td>
<td>YES</td>
<td># OF HYDROGEN ATOMS/FUEL MOLECULE</td>
</tr>
<tr>
<td>DEL</td>
<td>NO</td>
<td>YES</td>
<td>CARBON/HYDROGEN RATIO OF FUEL</td>
</tr>
<tr>
<td>QLOWER</td>
<td>NO</td>
<td>YES</td>
<td>LOWER HEATING VALUE OF THE FUEL (J/KG)</td>
</tr>
<tr>
<td>HFORM</td>
<td>NO</td>
<td>YES</td>
<td>ABSOLUTE ENTHALPY OF FUEL WRT. ITS CONSTITUENT ELEMENTS AT 0 K.</td>
</tr>
<tr>
<td>AFRAST</td>
<td>NO</td>
<td>YES</td>
<td>STOICHIOMETRIC AIR/FUEL RATIO</td>
</tr>
</tbody>
</table>

REMARKS

ONLY DIESEL FUEL AND ISOOCTANE ARE AVAILABLE
FOR USE AS FUELS.

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED

METHOD

SEE PURPOSE, ABOVE

WRITTEN BY D. N. ASSANIS
EDITED BY D. N. ASSANIS

SUBROUTINE FUELDT

INTEGER FUELTP

COMMON/FUEL/ FUELTP, CX, HY, DEL, PSI, AFRAST, QLOWER, HFORM

PSI = 3.78

IF (FUELTP .GT. 1) GO TO 10

OLLOWING DATA FOR DIESEL #2 FUEL (FUELTP = 1)

CX = 10.84
HY = 18.68
DEL = 10.84 / 18.68

UNITS: J/KG
QLOWER = 42.910E+6

NOTE THAT HFUEL = HF(298) + DHF(298-0)

HF(298) = -8.844E+6, DHF = 0.606E+6

VARIABLE HFORM DENOTES REALLY HFUEL REL TO ELEMENTS AT 0 K.

HFORM = -0.236E+6

AFRAST = 137.9 * (DEL+.25) / (12.*DEL +1.)

GO TO 20

FOLLOWING DATA FOR ISOOCTANE (FUELTP = 2)

10 CX = 8.0

HY = 18.0

DEL = 8.0/18.0

QLOWER = 44.39E+6

AFRAST = 15.11

20 RETURN

END
C*----------------------------------------------------- VERSION 1.0 *-----------------------------------------------------
C
C OCT 21, 1984
C
C FUNCTION GCMP
C CALLED BY 'ODER' TO CHECK FOR ROOTS DURING INTAKE, COMPRESSION
C (EXCLUDING IGNITION DELAY PERIOD), COMBUSTION, AND EXHAUST.
C THIS A DUMMY FUNCTION (GCMP ALWAYS >0)
C
C WRITTEN BY D.N. ASSANIS
C
FUNCTION GCMP (DT, DY, DYP)
REAL*8 DT, DY(1), DYP(1), GCMP
C GCMP =10.D0
C
RETURN
END
FUNCTION GIDEL
CALLED BY ODERT TO CALCULATE THE ROOT OF THE IGNITION DELAY FUNCTION, AND THUS PREDICT THE LENGTH OF THE IGNITION DELAY PERIOD.
WRITTEN BY D.N. ASSANIS

FUNCTION GIDEL (DT, DY, DYP)
REAL*8 DT, DY(1), DYP(1), GIDEL

DELFR = DY(13)
GIDEL = DELFR - 1.
RETURN
END
SUBROUTINE HPROP

PURPOSE
TO CALCULATE THE PROPERTIES OF THE PRODUCTS OF HYDROCARBON-
AIR COMBUSTION AS A FUNCTION OF TEMPERATURE AND PRESSURE,
USING AN APPROXIMATE CORRECTION FOR DISSOCIATION.
H AND RHO ARE CALCULATED AS FUNCTIONS OF P, T, AND PHI.
THE PARTIAL DERIVATIVES OF H AND RHO WITH RESPECT TO
P AND T ARE ALSO CALCULATED.

USAGE
CALL HPROP (T,P,FR, H,CP,CT,CF, RHO,DRHODT,DRHODP,DRHODF)

DESCRIPTION OF PARAMETERS
GIVEN:
P : ABSOLUTE PRESSURE OF PRODUCTS (ATM)
T : TEMPERATURE OF PRODUCTS (DEG K)
FR : AVERAGE FUEL FRACTION
DEL : MOLAR C:H RATIO OF PRODUCTS
PSI : MOLAR N:O RATIO OF PRODUCTS

RETURNS:
H : SPECIFIC ENTHALPY OF PRODUCTS (KCAL/G)
CP : PARTIAL DERIVATIVE OF H WITH RESPECT TO T
(CAL/G-DEG K)
CT : PARTIAL DERIVATIVE OF H WITH RESPECT TO P (CC/G)
CF : PARTIAL DERIVATIVE OF H WITH RESPECT TO PHI(KCAL/G)
RHO : DENSITY OF THE PRODUCTS (G/CC)
DRHODT : PARTIAL DERIVATIVE OF RHO WITH RESPECT TO T
(g/CC-DEG K)
DRHODP : PARTIAL DERIVATIVE OF RHO WITH RESPECT TO P
(g/CC-ATM)
DRHODF : PARTIAL DERIVATIVE OF RHO WITH RESPECT TO PHI
(g/CC)

REMARKS
1) ENTHALPY DATUM STATE IS AT T = 0 ABSOLUTE WITH
N2, H2 GASEOUS AND C SOLID GRAPHITE
2) MULTIPLY ATM-CC BY 0.0242173 TO CONVERT TO CAL
3) MODIFIED VERSION OF MIKE MARTIN'S PROGRAM
BY DENNIS ASSANIS (253-2453)
ADDED PARTIAL DERIVATIVES OF ENTHALPY AND DENSITY
WITH RESPECT TO EQUIVALENCE RATIO
SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED
CPROP

METHOD
SEE MARTIN AND HEYWOOD 'APPROXIMATE RELATIONSHIPS FOR THE
THERMODYNAMIC PROPERTIES OF HYDROCARBON-AIR COMBUSTION
PRODUCTS'

SUBROUTINE HPROP (T,P,FR, H, CP,CT,CF, RHO, DRHODT,DRHODP,DRHODF)
LOGICAL RICH, LEAN, NOTHOT, NOTWRM, NOTCLD
REAL MCP, MWT, K1, K2
COMMON/FUEL/ FUELTP, CX, HY, DEL, PSI, AFRAST, QLOWER, HFORM
INITIALIZE PARAMETERS USED IN THE CALCULATION

NOTE THAT R DENOTES THE UNIVERSAL GAS CONSTANT

GIVEN BY R = 1.9869 CAL / MOLE-DEG K

DATA R,ROVER2 /1.9869,0.99345/, PScale /2.42173E-2/
DATA TCOLD,THOT /1000.,1100./

PHI = FR * AFRAST / (1. - FR)
RICH = PHI .GE. 1.0
LEAN = .NOT. RICH
NOTHOT = T .LT. THOT
NOTCOLD = T .GT.TCOLD
NOTWRM = .NOT. (NOTCOLD .AND. NOTHOT)
EPS = (4.*DEL)/(1. + 4.*DEL)

USE SIMPLE ROUTINE FOR LOW TEMPERATURE MIXES

IF (NOTCOLD) GO TO 5
CALL CPROP (T,P,FR, H, CP,CT,CF, RHO, DRHODT,DRHODP,DRHODF)
RETURN

CALCULATE EQUILIBRIUM CONSTANTS FOR DISSOCIATION (Eqs. 3.9 & 3.10) (NOTE THAT THESE HAVE UNITS ATM**(.5) )

5 K1 = 5.819E-6 * EXP(0.8674*EPS + 35810./T)
K2 = 2.961E-5 * EXP(2.593*EPS + 28980./T)

CALCULATE A, X, Y, & U AS IN Eqs. 5.24, 3.6, 5.25, 3.7, 2.18, 2.19, & 3.8

C5 = 2. - EPS + PSI
A = (C5/(4.*P*K1*K1*EPS))**(.33333333)
C6 = EPS + 2.*C5
X = A*EPS*(3.*C5 + C6*A)/(3.*(1.+ 2.*A)*C5 + 2.*C6*A*A)

Z = ABS(1.-PHI)/X
If (LEAN) Y = X/SQRT(1. + 666667*Z + 1.33333333*(1.-PHI))
If (RICH) Y = X/(1.+.666667*Z + .3333333*Z*Z - .666667*(PHI-1.))
U = C5*(EPS - 2.*X)/(4.*K1*K2*P*X)

CALCULATE THE ENTHALPY OF FORMATION FOR THIS APPROXIMATE COMPOSITION AS IN Eqs. 3.21, 3.22, & 5.7. ALSO GET THE COEFFICIENTS FOR T & TV TERMS IN 3.15 USING 5.3 & 5.4

HF = 1000.*((121.5 + 29.59*EPS)*Y + 117.5*U)
HF = HF + (20372.*EPS - 114942.)*PHI
C1 = 7.*PSI + 5.*Y + 3.*U
C2 = 2.*(PSI - 3.*Y - U)

IF (LEAN) GO TO 10

RICH CASE

HF = HF + 1000.*((134.39 - 6.5/EPS)*(PHI - 1.)
C1 = 2. + 2.*(7.- 4.*EPS)*PHI + C1
C2 = 8. + 2.*(2.- 3.*EPS)*PHI + C2

GO TO 20
LEAN CASE

10 C1 = 7. + (9. - 8.*EPS)*PHI + C1
C2 = 2. + 2.* (5. - 3.*EPS)*PHI + C2

ADD IN TRANSLATIONAL, VIBRATIONAL, AND ROTATIONAL TERMS
TO GET TOTAL ENTHALPY, USING EQUATIONS 3.16, 5.6, 3.11, & 3.15

20 TV = (3256. - 2400.*EPS + 300.*PSI)/(1. - .5*EPS + .09*PSI)

EXPTVT = EXP(TV/T)
TVTIL = TV/(EXPTVT - 1.)
MCP = (8.*EPS + 4.)*PHI + 32. + 28.*PSI

NOTE MULTIPLICATION OF H BY 0.001 TO CONVERT UNITS
FROM CAL/G TO KCAL/G.
NOW, UNITS OF H ARE SAME IN HPROP AND CPROP.

H = 0.001*ROVER2*(C1*T + C2*TVTIL + HF)/MCP
CALCULATE THE AVERAGE MOLECULAR WEIGHT, AND GET DENSITY
BY USING THE PERFECT GAS LAW - EQUATIONS 3.12, 3.13, & 3.14

IF (LEAN) MWT = MCP/(1. + (1.- EPS)*PHI + PSI + Y + U)
IF (RICH) MWT = MCP/( (2.- EPS)*PHI + PSI + Y + U)

RHO = MWT*P*PSCALE/(R*T)
GET PARTIAL DERIVATIVES IF DESIRED
THE FOLLOWING USES IN ORDER EQUATIONS 5.8, 5.9, 5.32, 5.31, 5.30,
5.29, 5.28, & 5.26

C3 = (121.5 + 29.59*EPS)*1000.
C4 = 1.175E5

DUDTX = 64790.*U/(T*T)
DUDTX = -U/P
DUDXPT = -U*EPS/(X*(EPS - 2.*X))
DADTP = 23873.*A/(T*T)
DADPT = -A/(3.*P)

T5 = 3.*C5
DXDA = T5*EPS*(T5 + 2.*C6*A)/(T5*(1. + 2.*A) + 2.*C8*A*A)**2

FOLLOWING USES EQUATIONS 5.23, 5.19-5.22, 5.18-5.14, 5.12, & 5.13

IF (LEAN) DYDX = (Y*Y*Y)/(X*X*X) * (1. + Z + 1.333333* (1.-PHI))
IF (RICH) DYDX = (Y*Y)/(X*X)*(1. + 4.*Z/3. + Z*Z -2.*(PHI-1.)/3.)

Dyro = DYOX*DXDA*DADTP
DYOX = DYOX*DADTP
DUDTP = DUDTP*DADTP + DUDTPX
DUDPT = DUDPT*DADTP + DUDPTX

DHFDTP = C3*DYDTP + C4*DUDPT
DC2DTP = -2.*(3.*DYDTP + DUDPT)
DC1DPT = -3.*DYDPT + 3.*DUDPT
DHFDTP = C2*DYDTP + C4*DUDPT
DC2DTP = -2.*(3.*DYDTP + DUDPT)
DC1DTP = 5.*DYDTP + 3.*DUDTP

DTVDTP = (TVTIL*TVTIL)/(T*T)*EXPTVT

FOLLOWING USES Eqs. 5.10, & 5.11

CP = ROVER2/MCP*(C1 + T*DC1DTP + C2+DTVDTP + TVTIL*DC2DTP
& + DHFDTP)

CT = ROVER2/MCP*(T+DC1DPT + TVTIL*DC2DPT + DHFDPT)*PScale

DMCPDF = 8.* EPS + 4.

IF (RICH) GO TO 55

LEAN CASE

DYDF = 1./3. * (Y/X)**3 * (1.+2.*X)

DC1DF = (9.-8*EPS) + 5.*DYDF

DC2DF = 2.*(5.-3.*EPS) - 6.*DYDF

DHFDF = 20732.*EPS - 114942. + C3*DYDF

D = 1. + (1.-EPS)*PHI + PSI + Y + U

DDDF = (1.-EPS) + DYDF

GO TO 65

RICH CASE

55 DYDF = -2./3. * (Y/X)**2 * (1.+2.*X)

DC1DF = 2.*(7.-4.*EPS) + 5.*DYDF

DC2DF = 2.*(2.-3.*EPS) - 6.*DYDF

DHFDF = 20732.*EPS - 114942. + C3*DYDF - 6500./EPS

D = (2.-EPS)*PHI + PSI + Y + U

DDDF = (2.-EPS) + DYDF

NOTE MULTIPLICATION OF CF BY 0.001 TO CONVERT UNITS

FROM CAL/G TO KCAL/G, NEEDED BY THERMO.

NOW, UNITS OF CF ARE SAME IN HPROP AND CPROP.

NOTE ALSO THAT CF HAS SAME UNITS WITH ENTHALPY.

65 CF = 0.001 + ROVER2/MCP + ((DC1DF - C1/MCP*DMCPDF)*T +
& (DC2DF-C2/MCP*DMCPDF)*TVTIL + (DHFDF-HF/MCP*DMCPDF))

G = -MCP / (D*D)

DMMDT = G * (DYDTP + DUDTP)

DMMDP = G * (DYDTP + DUDTP)

DMMDF = -MCP/D*D* (DDDF - D/MCP*DMCPDF)

DRHODT = PScale * P * (DMMDT - MWT/T) / (R*T)

DRHODP = PScale * (MWT + P*DMMDP) / (R*T)

DRHODF = PScale * P * DMMDF / (R*T)

IF CALCULATING FOR AN INTERMEDIATE TEMPERATURE, USE A

WEIGHTED AVERAGE OF THE RESULTS FROM THIS ROUTINE AND

THOSE FROM THE SIMPLE ROUTINE

IF (NOTWRM) RETURN

CALL CPROP (T,P,FR, TH, TCP, TCT, TCF, TRHO, TDRT, TDRP, TDRF)
241 \[ W_1 = \frac{(T - TCOLD)}{(THOT - TCOLD)} \]
242 \[ W_2 = 1.0 - W_1 \]
243 \[ C \]
244 \[ H = W_1 \times H + W_2 \times TH \]
245 \[ RHO = W_1 \times RHO + W_2 \times TRHO \]
246 \[ CP = W_1 \times CP + W_2 \times TCP \]
247 \[ CT = W_1 \times CT + W_2 \times TCT \]
248 \[ CF = W_1 \times CF + W_2 \times TCF \]
249 \[ DRHODT = W_1 \times DRHODT + W_2 \times TDRT \]
250 \[ DRHODP = W_1 \times DRHODP + W_2 \times TDRP \]
251 \[ DRHODF = W_1 \times DRHODF + W_2 \times TDRF \]
252 \[ C \]
253 \[ RETURN \]
254 \[ END \]
SUBROUTINE HTRATE

PURPOSE
CALCULATES THE HEAT TRANSFER RATE THROUGH THE WALL
OF DIFFERENT SECTIONS OF THE EXHAUST MANIFOLD AND
THE CONNECTING PIPE BETWEEN THE TURBINES BASED
ON OPERATING CONDITIONS AND HEAT TRANSFER PARAMETERS
CALCULATED IN MANPAR.

USAGE
CALL HTRATE(ICV,TB,P,FFR,MDOT,QDOT)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>ICV</td>
<td>YES NO</td>
<td>FLAG GIVING SECTION OF EXHAUST MANIFOLD</td>
</tr>
<tr>
<td>TB</td>
<td>YES NO</td>
<td>GAS TEMPERATURE (K)</td>
</tr>
<tr>
<td>P</td>
<td>YES NO</td>
<td>STATIC PRESSURE (PA)</td>
</tr>
<tr>
<td>FFR</td>
<td>YES NO</td>
<td>FUEL FRACTION (-)</td>
</tr>
<tr>
<td>MDOT</td>
<td>YES NO</td>
<td>MASS FLOW RATE THROUGH SECTION (KG/DEG)</td>
</tr>
<tr>
<td>QDOT</td>
<td>NO YES</td>
<td>HEAT FLUX (JOULES/DEG)</td>
</tr>
<tr>
<td>NCV</td>
<td>YES NO</td>
<td>NUMBER OF CONTROL VOLUMES</td>
</tr>
</tbody>
</table>

ICV CONVENTION
1 - PORT
2 - RUNNER
NUMEX+1 - CONNECTING PIPE BETWEEN THE TURBINES

REMARKS
THIS SUBROUTINE IS PART OF SIM PROGRAM WITH MODIFIED HEAT TRANSFER MODELS FOR THE EXHAUST SYSTEM. IT IS ONLY CALLED FOR EXSUB = .TRUE.

SUBROUTINE AND FUNCTION PROGRAMS REQUIRED
THERMO TRANSP

METHOD
IN THE PORT SECTION, EMPIRICAL FORMULAS BASED ON WORK DONE BY G. CATON FOR THE HEAT TRANSFER IN THE EXHAUST PORT OF AN INTERNAL COMBUSTION ENGINE ARE USED. DOWNSTREAM OF THE PORT, THE HEAT TRANSFER RATE IS CALCULATED BASED ON AN EMPIRICAL FORMULA FOR TURBULENT PIPE FLOW CORRECTED FOR ENTRANCE EFFECTS AND HEAT TRANSFER ENHANCEMENT DUE TO PIPE BENDS.

WRITTEN BY: RICK FRANK
EDITED BY: RICK FRANK

SUBROUTINE HTRATE(ICV,TB,P,FFR,MDOT,QDOT)
PARAMETER (NCV = 6)
REAL MDOT, NCBP
COMMON/EMGEOM/ EXLNG(NCV),EXDIAM(NCV),EXAREA(NCV),
& EXVOL(NCV), EXXAR(NCV)
COMMON/EMPAR/ CBP(NCV), RLNG(NCV), ENCEE(NCV)
COMMON/PHASE/ EPHAO, EPHAC, IPHASE, REJ
COMMON/B2/ ERHO(NCV), DYNVIS(NCV), EMH(NCV), EFFR(NCV)
COMMON/HTCOTG/ HTCO(10), TGAS(10)
COMMON/TNPS/ NTEMP, TWALL(10), TSS(10), TSS2(10), TCW, TCW2
COMMON/QSOL/ QWALL(10), QAVE(10), QSS(10)
COMMON/HTCOTG/ HTCO(19), TGAS(16)
COMMON/TEMPS/ NTEMP, TWALL(16), TSS(10), TSS2(10), TCW, TCW2
COMMON/OSOL/ WALL(19), QAVE(16), QSS(18)
COMMON/DTDTH/ ESPD

C*** FOR NO HEAT TRANSFER
C IF (ICV.GT.-1) THEN
C QODT = 0.
C QWALL(ICV+2) = 0.
C HTCO(ICV+2) = 1.E-10
C RETURN
C ENDIF
C*** TW = TWALL(ICV+2)
IF (ICV.GT.1) GOTO 50
C ******** PORT SECTION ***************
C CHECK PHASE OF EXHAUST
IF (IPHASE.EQ.2 .OR. IPHASE.EQ.4) GOTO 50
C CHECK VLV ALMOST OPEN OR COMPLETELY CLOSED
C FIND FILM PROPERTIES
TF = (TB + TW)/2
CALL THERMO (TF, P, FFR, Z1, CPF, Z2, Z3, RHOF,
& Z4, Z5, Z6, GAMMAF, Z7, Z8, Z9, Z10)
CALL TRANSP (TF, FFR, GAMMAF, CPF, UF, TKF)
C CHECK IF VALVE IS IN OPENING PHASE OR CLOSING PHASE
IF (IPHASE.EQ.1) THEN
HTCOEF = 0.4 * REJ**0.6 * (TKF/EXDIAM(1))
ELSE
CLOSING PHASE
HTCOEF = 0.5 * REJ**0.5 * (TKF/EXDIAM(1))
ENDIF
C HEAT TRANSFER IN PORT (J/DEG)
QODT = HTCOEF * (TB - TW) * EXAREA(1) * ESPD
QWALL(3) = QODT
HTCO(3) = HTCOEF
RETURN
C ******** GENERAL SECTION - TURBULENT FLOW HEAT TRANSFER ****
C CALC FILM TEMPERATURE
TF = (TB + TW) / 2
C CALCULATE FILM PROPERTIES
CALL THERMO (TF, P, FFR, Z1, CPF, Z2, Z3, RHOF, Z4,
& Z5, Z6, GAMMAF, Z7, Z8, Z9, Z10)
CALL TRANSP (TF, FFR, GAMMAF, CPF, UF, TKF)
C CALCULATE FILM REYNOLDS NUMBER & PRANDTL NUMBER
FLO = MDOT
VELF = ABS(FLO) / ESPD / ERHO(ICV) / EXXAR(ICV)
REF = RHOF * VELF * EXDIAM(ICV) / UF
EPR = CPF * UF / TKF

C NUSSELT NUMBER CORRECTION FACTOR FOR BENT PIPE
C CORRECTION FOR BENT PIPE MODIFIED FROM MATH REPORT OF 8/85 TO
C USE CORRELATION BETTER MATCHED TO THE REYNOLDS NUMBERS IN THE
C EXHAUST SYSTEM.
NCBP = 1. + RLNG(ICV)*((21*CBP(ICV))/REF**0.15)

C INSIDE HEAT TRANSFER COEF. WITH CORRECTION FACTORS
C BASED ON TURBULENT PIPE FLOW CORRELATION WITH CORRECTIONS
C FOR ENTRANCE AND BENT PIPE EFFECTS
HTCOEF = NCBP * ENCEE(ICV) * (TKF/EXDIAM(ICV)) * 0.023 *
& (REF**0.8) * (EPR**0.3)

C HEAT TRANSFER (J/DEG)
QDOT = HTCOEF * (TB - TW) * EXAREA(ICV) * ESPD

C HTCO(ICV+2) = HTCOEF
QWALL(ICV+2) = QDOT

C RETURN
END
**Purpose**

Interpolates compressor map; given the value of two map variables, calculates the value of the two remaining map variables.

**Usage**

Call ICMAP (X, Y, AM, RPM, SIZ1, SIZ2, AMAP)

**Description of Parameters**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Input</th>
<th>Output</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>X</td>
<td>Yes</td>
<td>No</td>
<td>Corrected speed (thousands of RPM)</td>
</tr>
<tr>
<td>Y</td>
<td>Yes</td>
<td>No</td>
<td>Pressure ratio</td>
</tr>
<tr>
<td>AM</td>
<td>Yes</td>
<td>No</td>
<td>Map array (cm)</td>
</tr>
<tr>
<td>RPM</td>
<td>Yes</td>
<td>No</td>
<td>Speed array (thousands of RPM)</td>
</tr>
<tr>
<td>SIZ1</td>
<td>Yes</td>
<td>No</td>
<td>Dimension of RPM array and first</td>
</tr>
<tr>
<td>SIZ2</td>
<td>Yes</td>
<td>No</td>
<td>Second dimension of AM array</td>
</tr>
<tr>
<td>AMAP(1)</td>
<td>No</td>
<td>Yes</td>
<td>Corrected mass flow rate (lb/min)</td>
</tr>
<tr>
<td>AMAP(2)</td>
<td>No</td>
<td>Yes</td>
<td>Efficiency</td>
</tr>
</tbody>
</table>

**Remarks**

This subroutine replaces TCMAP for the turbocharger compressor.

**Subroutine and Function Subprograms Required**

None

**Method**

1) Given a corrected speed, a search of the RPM array is performed from the lowest speed value until a speed greater than the input speed is found. Using that greater speed and the speed just previous to that (the lesser speed), an interpolation parameter, "Q1", is calculated.
2) Using the speed interpolation parameter, values of pressure ratio at the input speed are calculated, until a pressure ratio greater than the input pressure ratio is found. Using the greater and lesser pressure ratios, a pressure ratio interpolation parameter, "Qj", is calculated.
3) Using the speed and pressure ratio interpolation parameters, calculate the mass flow rate and efficiency corresponding to the input values of speed and pressure ratio.

**Written by D. N. Assanis**

**Subroutine ICMAP (X, Y, AM, RPM, SIZ1, SIZ2, AMAP)**

**Notes**

- INTEGER SIZ2, SIZ1
- DIMENSION AMAP(2), RPM(SIZ1), AM(SIZ1,SIZ2,3), A(15)
- Search for RPM-value that is just greater than the given X, and stop search:
DO 10 I2 = 1, SIZ1
IF(RPM(I2).GT.X) GO TO 20
10 CONTINUE
C IF THE SEARCH WENT TO THE END OF THE SPEED ARRAY,
C SET I2 TO INDICATE THE HIGHEST SPEED:
C I2 = SIZ1
20 I1 = I2 - 1
C
C DEFINE X (SPEED) INTERPOLATION PARAMETER:
QI = (X - RPM(I1)) / (RPM(I2) - RPM(I1))
C SIMILARLY, SEARCH FOR MAP VALUE, INTERPOLATED BETWEEN
C HIGH AND LOW RPM CURVES, THAT IS JUST GREATER THAN
C THE GIVEN Y-VALUE:
C
DO 30 J2 = 2, SIZ2
C SET UP ARRAY OF INTERPOLATED VALUES OF PRESSURE RATIO
C A(J2) = QI * (AM(I2, J2, 3) - AM(I1, J2, 3)) + AM(I1, J2, 3)
C CHECK IF INTERPOLATED PRESSURE RATIO IS GREATER THAN
C INPUT VALUE:
C IF((A(J2).GT.Y).AND.(J2.NE.1)) GO TO 40
C 30 CONTINUE
C
IF THE SEARCH WENT TO END OF THE MAP ARRAY, THE GIVEN SPEED
IS TOO LOW FOR THE GIVEN PRESSURE RATIO. FIND THE LOWEST
POSSIBLE SPEED THAT IS COMPATIBLE WITH THAT PRESSURE RATIO,
AND RETURN THAT SPEED TO MAIN PROGRAM.
C 35 I2 = I2 + 1
C I1 = I1 + 1
C QI = (Y - AM(I1, SIZ2, 3)) / (AM(I2, SIZ2, 3) - AM(I1, SIZ2, 3))
C IF (I2 .EQ. SIZ1) GO TO 37
C 36 IF (QI.GT.1.0) GO TO 35
C 37 X = QI * (RPM(I2) - RPM(I1)) + RPM(I1)
C J2 = SIZ2
C A(J2) = Y
C 40 J1 = J2 - 1
C IF (J1.EQ.1) A(J1) = QI * (AM(I2, J1, 3) - AM(I1, J1, 3)) + AM(I1, J1, 3)
C DEFINE Y INTERPOLATION PARAMETER:
C QJ = (Y - A(J1)) / (A(J2) - A(J1))
C
DO 70 K = 1, 2
C FIND THE OTHER MAP-VARIABLE VALUES WHICH CORRESPOND
C TO THE INDICES OF THE HIGH AND LOW Y-VALUES:
C Z1 = QI * (AM(I2, J1, K) - AM(I1, J1, K)) + AM(I1, J1, K)
C Z2 = QI * (AM(I2, J2, K) - AM(I1, J2, K)) + AM(I1, J2, K)
C
AND INTERPOLATE BETWEEN EACH PAIR OF VALUES TO FIND
THE MAP VALUES WHICH CORRESPOND TO THE
ACTUAL MAP VALUES:

\[ \text{AMAP}(k) = qj \cdot (z_2 - z_1) + z_1 \]

CONTINUE

OUTPUT WARNING FLAG IF EDGE OF MAP IS USED
IT HAS BEEN ASSUMED FOR THIS FLAG THAT THE COMPRESSOR MAP
AS IT WAS INPUT HAS BEEN EXTRAPOLATED PAST THE MEASURED MAP RANGE
ONE EXTRA POINT.

IF (J2 .EQ. SIZ2) WRITE (6,*)'COMPRESSOR AT EDGE OF NORMAL MAP',
& ' RANGE - INTERP PARAM = ', QJ, '
& ' SPD = ', X, ' PRESS',
& ' RATIO = ', Y, ' MASS FLOW=' AMAP(1)
RETURN
END
SUBROUTINE INTAKE

PURPOSE
CALCULATES THE TIME RATE OF CHANGE OF PRESSURE, TEMPERATURE, FUEL EQUIVALENCE RATIO, MASS, HEAT TRANSFER, WORK TRANSFER, MEAN KINETIC ENERGY, AND TURBULENT KINETIC ENERGY IN THE MASTER CYLINDER DURING INTAKE.

USAGE
CALL INTAKE (NEQN, DT, DY, DYP)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>NEQN</td>
<td>YES</td>
<td>NO</td>
<td>NUMBER OF EQUATIONS BEING INTEGRATED</td>
</tr>
<tr>
<td>DT</td>
<td>YES</td>
<td>NO</td>
<td>TIME (DEG)</td>
</tr>
<tr>
<td>DY(1)</td>
<td>YES</td>
<td>NO</td>
<td>MASS INDUCTED INTO CHAMBER THROUGH INTAKE VALVE (KG)</td>
</tr>
<tr>
<td>DY(2)</td>
<td>YES</td>
<td>NO</td>
<td>MASS EXHAUSTED FROM CHAMBER THROUGH EXHAUST VALVE (KG)</td>
</tr>
<tr>
<td>DY(6)</td>
<td>YES</td>
<td>NO</td>
<td>MEAN KINETIC ENERGY IN CHAMBER (J)</td>
</tr>
<tr>
<td>DY(7)</td>
<td>YES</td>
<td>NO</td>
<td>TURBULENT KINETIC ENERGY IN CHAMBER (J)</td>
</tr>
<tr>
<td>DY(8)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - PISTON TOP (J)</td>
</tr>
<tr>
<td>DY(9)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - CYLINDER HEAD (J)</td>
</tr>
<tr>
<td>DY(10)</td>
<td>YES</td>
<td>NO</td>
<td>HEAT TRANSFER - CYLINDER WALL (J)</td>
</tr>
<tr>
<td>DY(11)</td>
<td>YES</td>
<td>NO</td>
<td>CYLINDER TEMPERATURE (K)</td>
</tr>
<tr>
<td>DY(12)</td>
<td>YES</td>
<td>NO</td>
<td>CYLINDER PRESSURE (PA)</td>
</tr>
<tr>
<td>DY(16)</td>
<td>YES</td>
<td>NO</td>
<td>TOTAL WORK TRANSFER (J)</td>
</tr>
<tr>
<td>DY(17)</td>
<td>YES</td>
<td>NO</td>
<td>TOTAL ENTHALPY EXHAUSTED (J)</td>
</tr>
<tr>
<td>DY(20)</td>
<td>YES</td>
<td>NO</td>
<td>BURNED FUEL FRACTION (-)</td>
</tr>
<tr>
<td>DYP(1)</td>
<td>NO</td>
<td>YES</td>
<td>RATE AT WHICH MASS IS INDUCTED THROUGH THE INTAKE VALVE (KG/DEG)</td>
</tr>
<tr>
<td>DYP(2)</td>
<td>NO</td>
<td>YES</td>
<td>RATE AT WHICH MASS IS EXHAUSTED THROUGH THE EXHAUST VALVE (KG/DEG)</td>
</tr>
<tr>
<td>DYP(6)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF MEAN KINETIC ENERGY (J/DEG)</td>
</tr>
<tr>
<td>DYP(7)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF TURBULENT KINETIC ENERGY (J/DEG)</td>
</tr>
<tr>
<td>DYP(8)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF HEAT TRANSFER - PISTON TOP (J/DEG)</td>
</tr>
<tr>
<td>DYP(9)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF HEAT TRANSFER - CYLINDER HEAD (J/DEG)</td>
</tr>
<tr>
<td>DYP(10)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF HEAT TRANSFER - CYLINDER WALL (J/DEG)</td>
</tr>
<tr>
<td>DYP(11)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF CYLINDER TEMPERATURE (K/DEG)</td>
</tr>
<tr>
<td>DYP(12)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF CYLINDER PRESSURE (PA/DEG)</td>
</tr>
<tr>
<td>DYP(16)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF TOTAL WORK TRANSFER (J/DEG)</td>
</tr>
<tr>
<td>DYP(17)</td>
<td>NO</td>
<td>YES</td>
<td>RATE AT WHICH TOTAL ENTHALPY IS EXHAUSTED (J/DEG)</td>
</tr>
<tr>
<td>DYP(20)</td>
<td>NO</td>
<td>YES</td>
<td>RATE OF CHANGE OF BURNED FUEL FRACTION (-)</td>
</tr>
</tbody>
</table>
FUEL FRACTION (1/DEG)

REMARKS

UNITS CHANGED TO S.I.


NOTE THAT UNITS FOR DERIVATIVES ARE PER REAL TIME UNTIL END

OF SUBROUTINE WHERE THEY ARE CONVERTED TO PER CRANK ANGLE

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED

THERMO TRANSP VACDIN MFLRT

CSAVDV VACDEX DIFEQ

METHOD

SEE REPORT

WRITTEN BY D. N. ASSANIS AND S. G. Poulos

EDITED BY D. N. ASSANIS & R. M. Frank

SUBROUTINE INTAKE (NEQN, DT, DY, DYP)

REAL*8 DT, DY(NEQN), DYP(NEQN)
REAL MM, MW, MMW, MWEM, KINVIS, MASS, MDOT, MSTART, MACRSC
COMMON/EPARAM/ BORE, STROKE, CONRL, CYLCA, CSATDC, CMRTIO, CLVTDC
COMMON/HTRC/ CONHT, EXPHT
COMMON/TEMPS/ NTEMP, TWALL(10), TSS(10), TSS2(10), TCW, TCW2
COMMON/DTDTH/ ESPD
COMMON/MA/ YPM(2), YPH(2), YPF(2)
COMMON/MSTA/ MSTART
COMMON/TIMES/ TIVO, TEVC, TIVC, TINJ, TIGN, TEVO
COMMON/B / CPM(2), HM(2), MWM(2), GM(2), RHOM(2)
COMMON/HEATS/ CVHTRN, HTRCOE, HTPAPI, HTPAHD, HTPACW, HTRAPI,
& HTRACW, HTRACW, HTTRAN
COMMON/TURBU/ CBETA, MACRSC, UPRIME, VAKE, VPISTO
COMMON/VALVE/ VIV, VEV
COMMON/RHMAS/ RHO, MASS, VOLUME, H, GAMMA
COMMON/AREAS/ AHEAD, APSTON
COMMON/FUEL/ FUELTP, CX, HY, DEL, PSI, AFRAST, QLOWER, HFORM
COMMON/EFFAR/ AREA, CD
COMMON/HTCOTG/ HTCO(10), TGAS(10)
COMMON/PHASE/ EPHAO, EPHAC, I PHASE, REJ
COMMON/PGEOM/ NEVLV, EPLNG(3), PDIAM(2)

VIV = 0.0
VEV = 0.0
FRAIV = 0.0
FRAEV = 0.0

T = DT
DO 20 I = 1, 20
20 DYP(I) = 0.0

PIM = DY(23)
GIM = GM(1)
MWIM = MMW(1)
RHOIM = RHOM(1)
FIM = DY(24)
TIM = DY(22)
HIM = HM(1)

PEM = DY(27)
GEM = GM(2)

MWEM = MMW(2)
RHOEM = RHM(2)

TEM = DY(26)

FEM = DY(28)

HEM = HM(2)

TCYL = DY(11)
PCYL = DY(12)

FIND THERMODYNAMIC AND TRANSPORT PROPERTIES IN CYLINDER

FR = DY(20)

CALL THERMO (TCYL, PCYL, FR, H, CSUBP, CSUBT, CSUBF, & RHO, DRHODT, DRHOOP, GAMMA, MW, ADUMY, BDUMY, CDUMY)

CALL TRANSP (TCYL, FR, GAMMA, CSUBP, DYNVIS, THRCNO)

KINVIS = DYNVIS/RHO

MASS = MSTART + DY(1) - DY(2)

FIND OUT IF INTAKE VALVE IS OPEN.

IF (T .GE. TIVC) GO TO 50

YES IT IS.

FIND OUT IF ANY MASS FLOWS ACROSS INTAKE VALVE.

IF (PIM - PCYL) 30. 50, 40

REVERSE FLOW PAST VALVE.

CALCULATE CD AND EFFECTIVE AREA.

30 PR = PCYL/PIM

CALL VACDIN (T, AREA, CD)

IF (AREA .LE. 0.) GOTO 50

CALCULATE MASS FLOW RATE FROM CYLINDER TO INTAKE MANIFOLD.

CALL MFLRT (CD, AREA, PCYL, MW, TCYL, PIM, GAMMA, FRAIV)

CALCULATE RATES DUE TO THIS FLOW.

DYP(1) = -FRAIV

VIV = -FRAIV/(RHO * AREA)

DYP(6) = -FRAIV * (DY(6)/MASS)

DYP(7) = -FRAIV * (DY(7)/MASS)

YPH(1) = H

YPF(1) = FR

GO TO 50

FLOW INTO CYLINDER.

CALCULATE CD AND AREA.

40 PR = PIM/PCYL

CALL VACDIN (T, AREA, CD)

IF (AREA .LE. 0.) GOTO 50

C
CALCULATE MASS FLOW RATE

CALL MFLRT (CD, AREA, PIM, MWIM, TIM, PCYL, GIM, FRAIV)

CALCULATE RATES DUE TO THIS FLOW

DYP(1) = FRAIV
DYP(28) = (FIM - DY(20)) * DYP(1)/MASS
VIV = FRAIV/(RHOIM * AREA)
DYP(6) = .5 * FRAIV * VIV*VIV
YPH(1) = HIM
YPF(1) = FIM

IS EXHAUST VALVE STILL OPEN?

IPHASE = 4
50 IF ((T + 720.) .GE. TEVC) GO TO 80

YES IT IS.
ANY FLOW ACROSS IT?

IF (PCYL - PEM) 60, 80, 70

YES, FLOW INTO CYLINDER.
FIND CD AND AREA FOR EXHAUST VALVE.

60 PR = PEM/PCYL

CONVERT TIME FOR EXHAUST VALVE
TEX = T + 720.
CALL VACDEX (TEX, AREA, CD)

FIND MASS FLOW RATE.

CALL MFLRT (CD, AREA, PEM, MWEM, TEM, PCYL, GEM, FRAEV)

CALCULATE RATES DUE TO THIS FLOW.

DYP(2) = -FRAEV
VEV = -FRAEV/(RHOEM * AREA)
DYP(6) = DYP(6) + .5 * FRAEV * VEV*VEV
DYP(28) = DYP(28) + ( FEM - DY(20) ) * FRAEV / MASS
YPH(2) = HEM
YPF(2) = FEM
IPHASE = 2
GO TO 80

FLOW FROM CYLINDER INTO EXHAUST MANIFOLD.
FIND AREA AND CD FOR EXHAUST VALVE.

70 PR = PCYL/PEM
TEX = T + 720.
CALL VACDEX (TEX, AREA, CD)

FIND MASS FLOW RATE.

CALL MFLRT (CD, AREA, PCYL, MM, TCYL, PEM, GAMMA, FRAEV)
C
C CALCULATE RATES DUE TO THIS FLOW.
C
DYP(2) = FRAEV
VEV = FRAEV/(RHO * AREA)
DYP(6) = DYP(6) - FRAEV * (DY(6)/MASS)
DYP(7) = DYP(7) - FRAEV * (DY(7)/MASS)
YPH(2) = H
YPF(2) = FR
C
C FIND JET REYNOLDS NUMBER FOR PORT HEAT TRANSFER
REJ = VEV + PDAM(1) * RHO / DYNVIS
C
C FIND SURFACE AREAS AND VOLUME OF CHAMBER
C
CALL CSAVDV (T, ACW, VOLUME, DVDT)
MACRSC = VOLUME/(3.1415 * BORE * BORE/4.)
IF (MACRSC .GE. (BORE/2.)) MACRSC = BORE/2.
DYP(6) = DYP(6) -.3367 * CBETA/MACRSC * DY(6) *
& SQRT( DY(7)/MASS)
DYP(7) = DYP(7) +.3367 * CBETA/MACRSC * DY(6) *
& SQRT( DY(7)/MASS)
& -.5443 * DY(7)/MACRSC * SQRT( DY(7)/MASS)
MDOT = DYP(1) - DYP(2)
C
C CHARACTERISTIC VELOCITY IN CYLINDER; (M/SEC).
C
PI = 3.141592654
CONSTR = CONRL/STROKE
SINH = SIN( T*PI/189. )
COSTH = COS( T*PI/189. )
VONVPM = ABS( PI + SINH * ( 1. + COSTH/SQRT( 4. * CONSTR*CONSTR
& - SINH+SINH ) )/2. )
VPMEAN = STROKE/(180. * ESPD)
VPISTO = VPMEAN * VONVPM
VMKE = SQRT( 2. * DY(6)/MASS )
UPRIME = SQRT( .666667 * DY(7)/MASS )
CVHTRN = SQRT( 0.25*VPISTO*VPISTO + VMKE*VMKE + UPRIME*UPRIME )
C
C CALCULATE HEAT TRANSFER RATES
C
HTRCOE = CONHT*(ABS(CVHTRN+MACRSC/KINVIS))**EXPHT / THRCON/MACRSC
C
85 HTPAPI = HTRCOE * ( TCYL - TWALL(1) )
HTPAHD = HTRCOE * ( TCYL - TWALL(2) )
HTPACW = HTRCOE * ( TCYL - TCW )
C
TGAS(1) = TCYL
TGAS(2) = TCYL
HTCO(1) = HTRCOE
HTCO(2) = HTRCOE
HLIN3 = HTRCOE
C
HTGPI = HTRCOE * TCYL
HTGMD = HTRCOE * TCYL
HTGCW = HTRCOE * TCYL
C
HTRAPI = APSTON * HTPAPI
HTRAHD = AHEAD * HTPAHD
HTRACW = ACW + HTPACW

THTRAN = HTRAPI + HTRAH + HTRACW

CALCULATE RATES OF CHANGE OF TEMPERATURE, PRESSURE, AND FUEL EQUIVALENCE RATIO IN THE CYLINDER.

THEN CALCULATE RATE OF DOING WORK.

PHIDOT = AFRAST/(1.-FR)/(1.-FR) • DYP(20)

PHI = FR • AFRAST / (1. - FR)

DYP(11) = (BDUMY/ADUMY)*((MDOT/MASS)*(1.-H/BDUMY) -
&DYDT/VOLUME +DYPT(1)*YPH(1) -DYP(2)*YPH(2) -
&TTRAN]/(BDUMY*MASS) - CDUMY/BDUMY • PHIDOT

DYP(12) = RHO/DRHDP • (-DYDT/VOLUME - DYP(11)*DRHODT/RHO
& - PHIDOT*DRHODF/RHO + MDOT/MASS)

DYP(16) = PCYL • DYDT

DYP(8) = HTRAPI
DYP(9) = HTRAH
DYP(10) = HTRACW

DYP(17) = DYP(2) • H

CONVERT ALL TIME DERIVATIVES TO RATE PER CRANK ANGLE DEGREE.

DO 100 I = 1, 20

100 DYP(I) = DYP(I) * ESPD

YPM(1) = DYP(1)
YPM(2) = DYP(2)

CALL DIFEQ (NEQN, T, DY(21), DYP(21))
THE METHODS IN SUBROUTINE STEP1 APPROXIMATE THE SOLUTION NEAR X
BY A POLYNOMIAL. SUBROUTINE INTRP APPROXIMATES THE SOLUTION AT
XOUT BY EVALUATING THE POLYNOMIAL THERE. INFORMATION DEFINING THIS
POLYNOMIAL IS PASSED FROM STEP1 SO INTRP CANNOT BE USED ALONE.

THIS CODE IS COMPLETELY EXPLAINED AND DOCUMENTED IN THE TEXT,
COMPUTER SOLUTION OF ORDINARY DIFFERENTIAL EQUATIONS, THE INITIAL
VALUE PROBLEM BY L. F. SHAMPINE AND M. K. GORDON.
FURTHER DETAILS ON USE OF THIS CODE ARE AVAILABLE IN *SOLVING
ORDINARY DIFFERENTIAL EQUATIONS WITH ODE, STEP, AND INTRP*,

THE USER PROVIDES STORAGE IN THE CALLING PROGRAM FOR THE ARRAYS IN
THE CALL LIST
DIMENSION Y(NEQN),YOUT(NEQN),YPOUT(NEQN),PHI(NEQN,16),PSI(12)
AND DEFINES
XOUT — POINT AT WHICH SOLUTION IS DESIRED.
THE REMAINING PARAMETERS ARE DEFINED IN STEP1 AND PASSED TO
INTRP FROM THAT SUBROUTINE.

Writing by L. F. Shampine and M. K. Gordon.

Abstract

The methods in subroutine STEP1 approximate the solution near X
by a polynomial. Subroutine INTRP approximates the solution at
XOUT by evaluating the polynomial there. Information defining this
polynomial is passed from STEP1 so INTRP cannot be used alone.

This code is completely explained and documented in the text,
computer solution of ordinary differential equations, the initial
Further details on use of this code are available in *SOLVING
ordinary differential equations with ODE, STEP, and INTRP*,

The user provides storage in the calling program for the arrays in
the call list
DIMENSION Y(NEQN),YOUT(NEQN),YPOUT(NEQN),PHI(NEQN,16),PSI(12)
and defines
XOUT — Point at which solution is desired.
The remaining parameters are defined in STEP1 and passed to
INTRP from that subroutine.
C OUTPUT FROM INTRP —
C YOUT(*) — SOLUTION AT XOUT
C YPOUT(*) — DERIVATIVE OF SOLUTION AT XOUT
C THE REMAINING PARAMETERS ARE RETURNED UNALTERED FROM THEIR INPUT
C VALUES. INTEGRATION WITH STEP1 MAY BE CONTINUED.
C
SUBROUTINE INTRP(X,Y,XOUT,YOUT,YPOUT,NEQN,KOLD PHI,PSI)
IMPLICIT REAL *8 (A-H,O-Z)
IMPLICIT INTEGER *4 (I-N)

CCCCC GENERIC
DIMENSION Y(NEQN),YOUT(NEQN),YPOUT(NEQN),PHI(NEQN,16),PSI(12)
DIMENSION G(13),W(13),RHO(13)
DATA G(1)/1.0/,RHO(1)/1.6/

HI = XOUT - X
KI = KOLD + 1
KIP1 = KI + 1

C INITIALIZE W(*) FOR COMPUTING G(*)
DO 5 I = 1,KI
  TEMP1 = I
  W(I) = 1.6/TEMP1
  TERM = 0.0
5 CONTINUE

C COMPUTE G(*)
DO 15 J = 2,KI
  JM1 = J - 1
  PSIJM1 = PSI(JM1)
  GAMMA = (HI + TERM)/PSIJM1
  ETA = HI/PSIJM1
  LIMIT1 = KIP1 - J
  DO 10 I = 1,LIMIT1
    W(I) = GAMMA*W(I) - ETA*W(I+1)
 10  CONTINUE
  G(J) = W(1)
  RHO(J) = GAMMA*RHO(JM1)

C INTERPOLATE
DO 20 L = 1,NEQN
  YPOUT(L) = 0.0
20  CONTINUE
DO 30 J = 1,KI
  I = KIP1 - J
  TEMP2 = G(I)
  TEMP3 = RHO(I)
  DO 25 L = 1,NEQN
    YOUT(L) = YOUT(L) + TEMP2*PHI(L,I)
 25  CONTINUE
  YPOUT(L) = YPOUT(L) + TEMP3*PHI(L,I)
30 CONTINUE
DO 35 L = 1,NEQN
  YOUT(L) = YOUT(L) + HI*YOUT(L)
35 RETURN
END
SUBROUTINE IPTMAP

PURPOSE
INTERPOLATES POWER TURBINE MAP (TURBOCOMPOUNDED CASE), OR THE
TURBINE MAP (TURBOCHARGED ONLY CASE); GIVEN THE VALUE OF TWO
MAP VARIABLES CALCULATES THE VALUE OF THE TWO REMAINING
MAP VARIABLES.

USAGE
CALL IPTMAP (X, Y, AM, RPM, SIZ1, SIZ2, AMAP)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT</th>
<th>OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>X</td>
<td>YES</td>
<td>NO</td>
<td>CORRECTED SPEED (THOUSANDS OF RPM)</td>
</tr>
<tr>
<td>Y</td>
<td>YES</td>
<td>NO</td>
<td>PRESSURE RATIO</td>
</tr>
<tr>
<td>AM</td>
<td>YES</td>
<td>NO</td>
<td>MAP ARRAY (PTM)</td>
</tr>
<tr>
<td>RPM</td>
<td>YES</td>
<td>NO</td>
<td>SPEED ARRAY (THOUSANDS OF RPM)</td>
</tr>
<tr>
<td>SIZ1</td>
<td>YES</td>
<td>NO</td>
<td>DIMENSION OF RPM ARRAY AND FIRST</td>
</tr>
<tr>
<td>SIZ2</td>
<td>YES</td>
<td>NO</td>
<td>SECOND DIMENSION OF AM ARRAY</td>
</tr>
<tr>
<td>AMAP(1)</td>
<td>NO</td>
<td>YES</td>
<td>CORRECTED MASS FLOW RATE (LB/MIN)</td>
</tr>
<tr>
<td>AMAP(2)</td>
<td>NO</td>
<td>YES</td>
<td>EFFICIENCY</td>
</tr>
</tbody>
</table>

REMARKS
THIS SUBROUTINE REPLACES TCMAP FOR THE POWER TURBINE.

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
NONE

METHOD
1) GIVEN A CORRECTED SPEED, A SEARCH OF THE RPM ARRAY IS
PERFORMED FROM THE LOWEST SPEED VALUE UNTIL A SPEED GREATER
THAN THE INPUT SPEED IS FOUND. USING THAT GREATER SPEED AND THE
SPEED JUST BEFORE THAT (THE LESSER SPEED), A SPEED
INTERPOLATION PARAMETER, "QI", IS CALCULATED.
2) USING THE SPEED INTERPOLATION PARAMETER, VALUES OF PRESSURE
RATIO AT THE INPUT SPEED ARE CALCULATED, UNTIL A PRESSURE RATIO
GREATER THAN THE INPUT PRESSURE RATIO IS FOUND. USING THE GREATER
AND LESSER PRESSURE RATIOS, A PRESSURE RATIO INTERPOLATION PARAMETER
"QJ" IS CALCULATED.
3) USING THE SPEED AND PRESSURE RATIO INTERPOLATION PARAMETERS,
THE MASS FLOW RATE AND EFFICIENCY CORRESPONDING TO THE
INPUT VALUES OF SPEED AND PRESSURE RATIO ARE CALCULATED.

WRITTEN BY D. N. ASSANIS

SUBROUTINE IPTMAP (X, Y, AM, RPM, SIZ1, SIZ2, AMAP)

INTEGER SIZ2, SIZ1
DIMENSION AMAP(2), RPM(SIZ1), AM(SIZ1,SIZ2,3), A(15)

SEARCH FOR RPM-VALUE THAT IS JUST GREATER THAN THE
GIVEN X, AND STOP SEARCH:
DO 10 I2 = 2, SIZ1
   IF(RPM(I2).GT.X) GO TO 20
10 CONTINUE
   IF THE SEARCH WENT TO THE END OF THE SPEED ARRAY,
   SET I2 TO INDICATE THE HIGHEST SPEED:
I2 = SIZ1
20 I1 = I2 - 1

DEFINE X (SPEED) INTERPOLATION PARAMETER:
QI = (X - RPM(I1)) / (RPM(I2) - RPM(I1))

SIMILARLY, SEARCH FOR MAP VALUE, INTERPOLATED BETWEEN
HIGH AND LOW RPM CURVES, THAT IS JUST GREATER THAN
THE GIVEN Y-VALUE:

DO 30 J2 = 2, SIZ2

SET UP ARRAY OF INTERPOLATED VALUES OF PRESSURE RATIO
A(J2) = QI * (AM(I2, J2, 3) - AM(I1, J2, 3)) + AM(I1, J2, 3)

CHECK IF INTERPOLATED PRESSURE RATIO IS GREATER THAN
INPUT VALUE:
IF((A(J2).GT.Y).AND.(J2.NE.1)) GO TO 40

30 CONTINUE

IF THE SEARCH WENT TO END OF THE MAP ARRAY, THE TURBINE IS CHOKED
RETURN THE VALUES CORRESPONDING TO CHOKED CONDITIONS AT THIS
SPEED. NOTE THAT WHEN THIS OCCURS, THE SIMULATION RESULTS MAY
BE IN ERROR. A WARNING FLAG IS PRINTED WHEN THE TURBINE APPEARS
TO BE CHOKED BASED ON THE MAPS INPUT.

J2 = SIZ2
Y = A(SIZ2)
WRITE (8,*) 'POWER TURBINE CHOKED (SEE SBRTN IPTMAP)'

DO 40 J1 = J2 - 1

DEFINE Y INTERPOLATION PARAMETER:
50 QJ = (Y - A(J1)) / (A(J2) - A(J1))

DO 70 K = 1, 2

FIND THE OTHER MAP-VARIABLE VALUES WHICH CORRESPOND
TO THE INDICES OF THE HIGH AND LOW Y-VALUES:
Z1 = QI * (AM(I2, J1, K) - AM(I1, J1, K)) + AM(I1, J1, K)
Z2 = QI * (AM(I2, J2, K) - AM(I1, J2, K)) + AM(I1, J2, K)

AND INTERPOLATE BETWEEN EACH PAIR OF VALUES TO FIND
THE MAP VALUES WHICH CORRESPOND TO THE ACTUAL MAP VALUES:
AMAP(K) = QJ * (Z2 - Z1) + Z1
70 CONTINUE
C
RETURN
END
SUBROUTINE ITMAP

PURPOSE
INTERPOLATES TURBINE MAP; GIVEN THE VALUE OF TWO MAP VARIABLES, CALCULATES THE VALUE OF THE TWO REMAINING MAP VARIABLES.

USAGE
CALL ITMAP (X, Y, AM, RPM, SIZ1, SIZ2, AMAP)

DESCRIPTION OF PARAMETERS

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<td>YES</td>
<td>NO</td>
<td>CORRECTED SPEED (THOUSANDS OF RPM)</td>
</tr>
<tr>
<td>Y</td>
<td>YES</td>
<td>NO</td>
<td>CORRECTED MASS FLOW RATE (LB/MIN)</td>
</tr>
<tr>
<td>AM</td>
<td>YES</td>
<td>NO</td>
<td>MAP ARRAY (TM)</td>
</tr>
<tr>
<td>RPM</td>
<td>YES</td>
<td>NO</td>
<td>SPEED ARRAY (THOUSANDS OF RPM)</td>
</tr>
<tr>
<td>SIZ1</td>
<td>YES</td>
<td>NO</td>
<td>DIMENSION OF RPM ARRAY AND FIRST DIMENSION OF AM ARRAY</td>
</tr>
<tr>
<td>SIZ2</td>
<td>YES</td>
<td>NO</td>
<td>SECOND DIMENSION OF AM ARRAY</td>
</tr>
<tr>
<td>AMAP(1)</td>
<td>NO</td>
<td>YES</td>
<td>EFFICIENCY</td>
</tr>
<tr>
<td>AMAP(2)</td>
<td>NO</td>
<td>YES</td>
<td>PRESSURE RATIO</td>
</tr>
</tbody>
</table>

REMARKS
THIS SUBROUTINE REPLACES TCMAP FOR THE TURBOCHARGER TURBINE.

WRITTEN BY D. N. ASSANIS
IF(RPM(I2).GT.X) GO TO 20
10 CONTINUE
C IF THE SEARCH WENT TO THE END OF THE SPEED ARRAY,
C SET I2 TO INDICATE THE HIGHEST SPEED:
C
I2 = SIZ1
20 I1 = I2-1
C DEFINE X (SPEED) INTERPOLATION PARAMETER:
QI = (X-RPM(I1)) / (RPM(I2)-RPM(I1))
SIMILARLY, SEARCH FOR MAP VALUE, INTERPOLATED BETWEEN
HIGH AND LOW RPM CURVES, THAT IS JUST GREATER THAN
THE GIVEN Y-VALUE:
DO 30 J2 = 2, SIZ2
SET UP ARRAY OF INTERPOLATED VALUES OF MASS FLOW RATE
A(J2) = QI*(AM(I2,J2,1)-AM(I1,J2,1)) + AM(I1,J2,1)
CHECK IF INTERPOLATED ARRAY VARIABLE (MASS FLOW RATE)
IS GREATER THAN INPUT VALUE:
IF((A(J2).GT.Y).AND.(J2.NE.I)) GO TO 40
30 CONTINUE
C IF THE SEARCH WENT TO END OF THE MAP ARRAY, THE TURBINE HAS
CHOKED RETURN VALUES CORRESPONDING TO THE CHOKED
CONDITIONS AT THIS POINT
J2 = SIZ2
Y = A(J2)
40 J1 = J2 - 1
C DEFINE Y INTERPOLATION PARAMETER:
QJ = (Y-A(J1)) / (A(J2)-A(J1))
DO 70 K = 1, 2
FIND THE OTHER MAP-VARIABLE VALUES WHICH CORRESPOND
TO THE INDICES OF THE HIGH AND LOW Y-VALUES:
Z1 = QI*(AM(I2,J1,K+1)-AM(I1,J1,K+1)) + AM(I1,J1,K+1)
Z2 = QI*(AM(I2,J2,K+1)-AM(I1,J2,K+1)) + AM(I1,J2,K+1)
AND INTERPOLATE BETWEEN EACH PAIR OF VALUES TO FIND
THE MAP VALUES WHICH CORRESPOND TO THE
ACTUAL MAP VALUES:
AMAP(K) = QJ*(Z2-Z1) + Z1
70 CONTINUE
RETURN
END
SUBROUTINE ITRATE

PURPOSE

THIS SUBROUTINE IS CALLED TO OBTAIN T GIVEN P, H, FR,
AND A GUESS FOR T. 'ITRATE' CALLS 'THERMO' WITH T
GUESS. 'THERMO' RETURNS WITH THE ENTHALPY CORRESPONDING TO THE
GIVEN TgueSS. THEN A NEW CORRECTED VALUE FOR TgueSS
IS CALCULATED BY USING THE DEFINITION OF CSUB P AND THE
KNOWN VALUES OF CORRECT H AND RETURNED HGUESS. THIS PRO-
CEDURE IS REPEATED AT MOST MAXTRY TIMES, OR FEWER TIMES
IF ACCURACY MAXERR IS ACHIEVED.

USAGE

CALL ITRATE ( TgueSS, P, FR, ENTHLP, CSUBP, CSUBT, CSUBF,
& RHO, DRHOOT, DRHOOP, DRHOOF, GAMMA, MW, ADUMY,
& BDUMY, CDUMY)

DESCRIPTION OF PARAMETERS

PARAMETER INPUT OUTPUT DESCRIPTION

TgueSS YES YES TEMPERATURE GUESS (K)

P YES NO PRESSURE (PA)

FR YES NO BURNED FUEL FRACTION (-)

ENTHLP YES NO ENTHALPY ON WHICH TO ITERATE (J/KG)

HGUESS NO NO ENTHALPY GUESS (J/KG)

CSUBP NO YES DH/DT @ CONSTANT P (J/KG-DEG K)

CSUBT NO YES DH/DP @ CONSTANT T (M**3/KG)

CSUBF NO YES DH/DPHI @ CONSTANT T,P (J/KG)

RHO NO YES DENSITY (KG/M**3)

DRHOOT NO YES PARTIAL OF RHO WITH RESPECT TO T

(KG/M**3-DEG K)

DRHOOP NO YES PARTIAL OF RHO WITH RESPECT TO P

(KG/M**3-PA)

DRHOOF NO YES PARTIAL OF RHO WITH RESPECT TO PHI

(KG/M**3)

MW NO YES MOLECULAR WEIGHT

GAMMA NO YES RATIO OF SPECIFIC HEATS

ADUMY NO YES DEFINED IN THERMO

BDUMY NO YES DEFINED IN THERMO

CDUMY NO YES DEFINED IN THERMO

REMARKS

UNITS CHANGED TO S.I.

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED

THERMO

METHOD

SEE PURPOSE, ABOVE

WRITTEN BY D. N. ASSANIS AND S. G. Poulos

EDITED BY D. N. ASSANIS

SUBROUTINE ITRATE ( TgueSS, P, FR, ENTHLP, CSUBP, CSUBT, CSUBF,
& RHO, DRHOOT, DRHOOP, DRHOOF, GAMMA, MW, ADUMY,
& BDUMY, CDUMY)
C REAL MW, MAXERR
C COMMON/ITRLIM/ MAXTRY, MAXERR
C
DO 10 I = 1, MAXTRY
   CALL THERMO (TGUESS, P, FR, HGUESS, CSUBP, CSUBT, CSUBF, RHO,
                 & DRHODT, DRHOOP, DRHODF, GAMMA, MW, ADUMY, BDUMY, CDUMY)
   TOLD = TGUESS
   TGUESS = TOLD + (ENTHLP - HGUESS)/CSUBP
   IF( ABS((TGUESS - TOLD)/TGUESS) .LE. MAXERR ) GO TO 20
10 CONTINUE
C
20 CALL THERMO (TGUESS, P, FR, HGUESS, CSUBP, CSUBT, CSUBF, RHO,
                 & DRHODT, DRHOOP, DRHODF, GAMMA, MW, ADUMY, BDUMY, CDUMY)
C
RETURN
END
SUBROUTINE MANPAR

PURPOSE
   CALCULATES NECESSARY HEAT TRANSFER PARAMETERS FOR EXHAUST
   MANIFOLD AND CONNECTING PIPE BETWEEN THE TURBINES FROM
   INPUT FILE. MANPAR IS ONLY CALLED FOR EXSUB=.TRUE.

USAGE
   CALL MANPAR

ARRAY INDICES:
   FIRST DIMENSION:
      1 - PORT
      2 - RUNNER
      3 - NEXT VOLUME DOWNSTREAM OF RUNNER
      ...
   NUMEX+1 - CONNECTING PIPE BETWEEN TURBINES

SECOND DIMENSION:
   1 - INSIDE LAYER
   2 - MIDDLE LAYER
   3 - OUTSIDE LAYER

DEFINITION OF PARAMETERS

PARAMETER       INPUT       OUTPUT       DESCRIPTION
NCV             NO          NO          DIMENSION PARAMETER FOR EXHAUST
NUMEX           YES         YES         NUMBER OF SECTIONS IN EXHAUST
               EXCLUDING TURB CONNECTING PIPE
NUMEXT          NO          YES         NUMBER OF SECTIONS IN EXHAUST
               INCLUDING TURB CONN PIPE
NPIPE(I)        YES         YES         NUMBER OF PIPES COMING
               TOGETHER AT Ith NODE
NSHFT(I)        YES         YES         PHASE SHIFT OF PIPES COMING
               INTO NODE (FOR NPIPE>1)
NEVLV           YES         YES         NUMBER OF EXHAUST VLVS/CYL
               (1 OR 2)
EPLNG(3)        YES         YES         LENGTH OF INDIVIDUAL AND COMMON
               PORT SECTIONS (M)
PDIAM(2)        YES         YES         PORT DIAMETER -
               INDIVIDUAL AND COMMON SECTIONS
EXLNG(NCV)      YES         YES         SECTION CENTERLINE LENGTH (M)
EXDIAM(NCV)     YES         YES         SECTION INSIDE DIAMETER (M)
EXAREA(NCV)     NO          YES         SECTION INSIDE SURFACE AREA
               (M**2)
EXXAR(NCV)      NO          YES         CROSS SECTIONAL AREA (M**2)
EXVOL(NCV)      NO          YES         SECTION VOLUME (M**3)
BRAD(NCV)       YES         YES         BEND RADIUS OF SECTION (M)
BANG(NCV)       YES         YES         TOTAL BEND ANGLE OF SECTION
               (DEG)
NEXLAY(NCV)     YES         YES         NUMBER OF COMPOSITE LAYERS
MATL(NCV,3)     YES         YES         MATERIAL OF Ith LAYER (CHARACTER
               STRING)
EXCOND(NCV,3)   YES         YES         THERMAL CONDUCTIVITY OF
               EACH LAYER (W/M*K)
EXTHI(K(NCV,3) YES YES THICKNESS OF EACH LAYER (M)
XHCOOL(NCV) YES YES OUTSIDE HEAT TRANSFER COEF. (W/M**2*K)
XTCOOL(NCV) YES YES AMBIENT TEMPERATURE OR OUTSIDE WALL SURFACE TEMPERATURE (K)
(IF XHCOOL=0)
EXTW(NCV) YES YES INSIDE WALL TEMPERATURE (K)
UA(NCV) NO YES NET WALL HEAT CONDUCTIVITY (INCLUDING OUTSIDE H)
CBP(NCV) NO YES DIAMETER RATIO FOR BENT PIPE
ENCEE(NCV) NO YES NUSSELT CORRECTION FACTOR FOR ENTRANCE EFFECTS
EPHAO YES NO EXHAUST VALVE CRANKANGLE TO BEGIN FULL OPEN PHASE FOR PORT HEAT TRANS
EPHAC YES NO EXH VLV CRANKANGLE TO BEGIN VALVE CLOSING PHASE FOR HEAT TRANSFER

REMARKS

THE USER SPECIFIES THE EXHAUST MANIFOLD CONFIGURATION USING NUMEX, NPIPE(I), AND NSHFT(I). NUMEX DEFINES THE NUMBER OF DIFFERENT SECTIONS OF THE EXHAUST MANIFOLD (EG PORT+RUNNER+PLENUM => NUMEX=3). NUMEX MUST BE LESS THAN OR EQUAL TO NCV FOR THE NON-TURBOCHARGED CASE, LESS THAN NCV FOR THE TURBOCOMPounded CASE. NCV IS CURRENTLY 6. SEARCH PROGRAM FOR NCV=6 TO LOCATE PARAMETER STATEMENTS FOR NCV.

NPIPE(I) SPECIFIES THE NUMBER OF PIPES COMING INTO A NODE. NPIPE=1 SHOULD BE USED FOR DIVISIONS IN THE MANIFOLD THAT ARE NOT JOINTS. THE PRODUCT OF NPIPE(I), I=1, NUMEX MUST BE EQUAL TO THE NUMBER OF CYLINDERS IN THE ENGINE. NPIPE(I) MUST EQUAL 1.

NSHFT(I) GIVES THE RELATIVE PHASE SHIFT OF ADDITIONAL PIPES COMING INTO A JOINT (NPIPE.GT.1). NSHFT IS BASED ON THE FIRING ORDER OF THE ENGINE. THE PHASE SHIFT IS RELATIVE TO THE TIMING OF THE MASTER CYLINDER. SEE SUBROUTINE EXSUM FOR THE USE OF THESE PARAMETERS.

THE STORAGE ARRAYS QLOSS, XMMASS, XF, & XH HAVE BEEN dimensioned FOR UP TO TWO JOINTS WITH NPIPE(1) GREATER THAN ONE (IE TWO PLACES WHERE EXHAUST PIPES COME TOGETHER). TO INCREASE THIS DIMENSION COMMON BLOCKS /QX/ AND /ARRAYX/ MUST BE REDIMENSIONED IN SIM, MANPAR, EXSUM, AND VLVSUM (QX ONLY). THE DO LOOP BELOW TO INITIALIZE XMMASS, ETC. SHOULD ALSO HAVE THE LIMIT INCREASED ACCORDINGLY.

WALL COMPOSITION AND HEAT TRANSFER PARAMETERS ARE
ASSUMED TO BE THE SAME FOR ALL PARTS OF THE PORT
REGARDLESS OF THE NUMBER OF EXHAUST VALVES.
FOR THE CASE OF ONLY ONE EXHAUST VALVE PER CYLINDER, EXLN(1)
AND EXDIAM(1) SHOULD BE USED TO SPECIFY THE PORT GEOMETRY.
FOR THE TURBOCOMPUNDED CASE, THE TURBINE CONNECTING PIPE
GEOMETRY AND WALL MATERIALS SHOULD BE SPECIFIED WITH THE
SAME ARRAYS AS FOR THE EXHAUST MANIFOLD WITH THE INDEX
EQUAL TO NUMEX+1.
PROVISION IS MADE TO AVOID HAVING TO INPUT THE MATERIAL
FOR ALL SECTIONS OF THE MANIFOLD. IF THE CONDUCTIVITY
OF THE THIRD SECTION IS 0., THE SAME WALL COMPOSITION
IS USED AS WAS SPECIFIED FOR CONTROL VOLUME 2 (RUNNER).
THE WALL COMPOSITION IS REPEATED FOR ANY OTHER
MANIFOLD VOLUMES DOWNSTREAM OF C.V. 2. THE TURBINE
CONNECTING PIPE MUST BE SPECIFIED REGARDLESS.
THREE LAYERS MUST BE SPECIFIED FOR THE TRANSIENT
CALCULATION. TO ACCOMPLISH THIS, IT MAY BE NECESSARY
TO DIVIDE ONE LAYER UP INTO TWO OR THREE LAYERS WITH
THE SAME THERMAL PROPERTIES AND A TOTAL THICKNESS
EQUAL TO THE ORIGINAL LAYER.
EPHAO AND EPHAC ARE USED TO SPECIFY THE VALVE PHASE
FOR THE PORT HEAT TRANSFER. EPHAO IS THE CRANK ANGLE
DURING THE OPENING PHASE OF THE EXHAUST VALVE WHEN
THE FLOW CAN BE CONSIDERED AS TURBULENT PIPE
FLOW. EPHAC IS THE EXHAUST VALVE CRANK ANGLE WHEN
THE HEAT TRANSFER IS BASED ON THE VALVE CLOSING PHASE
CORRELATION BOTH SHOULD BE NORMALY SET FOR L/D = 0.2.
EPHAO = 0.0 RESULTS IN THE PROGRAM USING TURBULENT
PIPE FLOW FOR PORT HEAT TRANSFER THROUGHOUT THE COMPLETE
CYCLE. SEE VACDEX FOR USE OF EPHAO & EPHAC AND HTRATE FOR
THE DIFFERENT PHASE CORRELATIONS.
BRAD AND BANG ARE NOT USED FOR THE PORT HEAT TRANSFER
SUBROUTINES REQUIRED
NONE
WRITTEN BY: RICK FRANK
EDITED BY: RICK FRANK
SUBROUTINE MANPAR
PARAMETER (NCV = 6, PI = 3.1416)
CHARACTER*10 MATL(NCV,3)
LOGICAL COM2, POWER
DIMENSION DIFU(NCV,3), IENODE(NCV,3)
DIMENSION RES(NCV,3)
COMMON/POWER/ POWER
COMMON/EXNUM/ NUMEX, NUMEXT
COMMON/EXPAR/ NPIPE(NCV), NSHFT(NCV), NUMSEG(NCV)
COMMON/PGEOM/NEVLV, EPLNG(3), PDIAM(2)
COMMON/EHALL/ NEXLAY(NCV), EXTHIK(NCV,3), XHCOOL(NCV),
& EXCOND(NCV,3), COM2, MATL
COMMON/EMGEOM/ EXLNG(NCV), EXDIAM(NCV), EXAREA(NCV),
& EXVOL(NCV), EXXAR(NCV)
COMMON/EMWALL/ NEXLAY(NCV), EXTHIK(NCV,3), MATL
COMMON/EMHT/ XTCOOL(NCV), UA(NCV)
COMMON/EXTINI/ EXTW(NCV)
COMMON/ICYL/ ICYL
COMMON/PHASE/ EPHAO, EPHAC, IENODE
NAMELIST/EXHMAN/ NUMEX, NPIPE, NSHFT, NEVLV,
& EPHAO, EPHAC, EPLNG, PDIAM, EXLNG, EXDIAM, MATL,
& BRAD, BANG, NEXLAY, EXCOND, EXTHIK, XHCOOL,
& XTCOOL, EXTW, DIFU, IENODE
COMMON/ARRAYX/ XMASS(2,720), XF(2,720), X(2,720)
COMMON/PARF/PDELX(10,3), PCNUM(10,3), INNODE(10,3), PTHIK(10,3),
& PCOND(10,3), PHEFF(10,3), PDIFFU(10,3), PHCOOL(10), PTCOOL(10),
& PUOVE(10), PSURFA(10)
COMMON/NPLA/ NPLA(10), NPC
COMMON/PERFCT/ PERI, FACT, DELT
COMMON/TEMPS/ NTEMP, TWALL(10), TSS(1e), TSS2(10), TCW, TCW2
DEFAULT VALUES FOR INITIAL WALL TEMPERATURES:
EXTW(1) = 450
EXTW(2) = 400
EXTW(3) = 400
EXTW(4) = 400
EXTW(5) = 400
ENTER INPUT LIST
READ(9,EXHMAN)
CLOSE (UNIT = 9)
COM2 = .FALSE.
ECHO WALL MATERIAL AND HEAT TRANSFER PARAMETERS FOR
OTHER SECTIONS IF NOT SPECIFIED OTHERWISE.
IF (EXCOND(3,1).EQ.0) THEN
  DO 12 J = 3, NUMEX
    DO 11 I = 1, NEXLAY(2)
      EXCOND(J,I) = EXCOND(2,I)
      EXTHIK(J,I) = EXTHIK(2,I)
      MATL(J,I) = MATL(2,I)
      DIFU(J,I) = DIFU(2,I)
    11 CONTINUE
    NEXLAY(J) = NEXLAY(2)
    XTCOOL(J) = XTCOOL(2)
    XHCOOL(J) = XHCOOL(2)
  12 CONTINUE
ENDIF
BENT PIPE CORRELATION NOT APPLIED TO PORT
BANG(1) = 0.
NUMEXT= NUMEX
IF (POWER) NUMEXT = NUMEXT + 1

C ******** GENERAL SECTION **************
DO 15 I = 1, NUMEXT
   RADI = EXDIAM(I) / 2
   EXAREA(I) = EXLNG(I) * EXDIAM(I) * PI
   EXXAR(I) = RADI * RADI * PI
   EXVOL(I) = EXXAR(I) * EXLNG(I)
C
CALCULATE NET THERMAL RESISTANCE THROUGH WALL
SUM = 0
RADI = RADI
SUM THERMAL RESISTANCE
DO 20 J = 1, NEXLAY(I)
   RAD2 = RADI + EXTHIK(I, J)
   RES(I, J) = LOG (RAD2/RADI)/(EXCOND(I, J))
   SUM = SUM + RES(I, J)
   RADI = RAD2
20 CONTINUE
ADD THERMAL RESISTANCE AT OUTSIDE SURFACE OF MANIFOLD
IF (XHCool(I).NE. 0.0)
   SUM = SUM + (1 / (RAD2 * XHCool(I)))
   UA(I) = 2 * PI * EXLNG(I) / SUM
   CHECK FOR STRAIGHT PIPE
   IF (BRAD(I).EQ.0..OR.BANG(I).EQ.O)
      CBP(I) = 0
   ELSE
      CBP(I) = EXDIAM(I) / (2 * BRAD(I))
   ENDIF
   IF PERCENTAGE BENT IS GREATER THAN LENGTH OF PIPE SET
   IF (RLNG(I).GT.1.) RLNG(I) = 1
   ELSE
      RLNG(I) = 0
   ENDIF
   IF (RLNG(I).GT.1.) RLNG(I) = 1
   ELSE
      RLNG(I) = 0
   ENDIF
   ACCOUNT FOR DIVIDED SECTION OF PORT FOR 2 EXHAUST VALVES
   PDIAM(2) = EXDIAM(1)
   EPLNG(3) = EXLNG(1)
   IF (NEVLV .NE. 2) GOTO 24
   PLNG = (EPLNG(1) + EPLNG(2)) / 2.
   PAREA = 2. * PI * PLNG * PDIAM(1)
   PXAREA = PI * PDIAM(1) * PDIAM(1) / 2.
   PVOL = PXAREA * PLNG
   UA(I) = UA(I) * (EXAREA(1) + PAREA) / EXAREA(1)
EXAREA(1) = EXAREA(1) + PAREA
PRATIO = PAREA / EXAREA(1)
EXXAR(1) = PRATIO*PXAREA + (1-PRATIO)*EXXAR(1)
EXLNG(1) = EXLNG(1) + PLNG
EXDIAM(1) = PRATIO*PDIAM(1) + (1-PRATIO)*PDIAM(2)
EXVOL(1) = EXVOL(1) + PVOL

C COMPUTE NUSSELT NUMBER CORRECTION FACTOR FOR ENTRANCE EFFECTS

TLNG = 0
DO 25 I = 1, NUMEX
   IF (EXLNG(I).EQ.ZERO) GOTO 25
   TLNG2 = TLNG + EXLNG(I)
   ENCEE(I) = 3.1 / EXLNG(I) * EXDIAM(I)**0.3 * (TLNG2**0.7 - TLNG**0.7)
   IF (ENCEE(I).LT.1.) ENCEE(I) = 1.0
   TLNG = TLNG2
25 CONTINUE

IF (POWER) & ENCEE(NUMEXT) = 3.1*(EXDIAM(NUMEXT)/EXLNG(NUMEXT))**0.3

C TRANSFER VARIABLES INTO GLOBAL ARRAYS
C USED IN PFDIF FOR TRANSIENT TEMPERATURE CALCS.

DO 45 I = 1, NUMEX
   IREF = I + 2
   DO 55 J = 1, NEXLAY(I)
   PTHIK(IREF,J) = EXTHIK(I,J)
   PCOND(IREF,J) = EXCOND(I,J)
   INNODE(IREF,J) = IENODE(I,J)
   IF (INNODE(IREF,J) .LT. 3) INNODE(IREF,J) = 3
   PDELX(IREF,J) = EXTHIK(I,J)/INNODE(IREF,J) - 1.
   PHEFF(IREF,J) = PCOND(IREF,J) / PDELX(IREF,J)
   PCNUM(IREF,J) = PDELX(IREF,J)**2 / (DIFU(I,J)*DELT)
   PDIFU(IREF,J) = DIFU(I,J)
55 CONTINUE

PHCOOL(IREF) = XHCOOL(I)
PTCOOL(IREF) = XTCOOL(I)
PSURFA(IREF) = EXAREA(I)
PUOVE(IREF) = UA(I) / EXAREA(I)
TWALL(IREF) = EXTW(I)
45 CONTINUE

IF (POWER) NUMSEG(NUMEXT) = 1
NUMSEG(NUMEX) = 1
DO 60 I = NUMEX-1, 1, -1
   NUMSEG(I) = NPIPE(I+1) * NUMSEG(I+1)
60 CONTINUE

IF (NUMSEG(1) .NE. ICYL) GOTO 200

VTOT = 0.
ATOT = 0.
DO 100 I = 1, NUMEX
   VTOT = VTOT + NUMSEG(I)*EXVOL(I)
   ATOT = ATOT + NUMSEG(I)*EXAREA(I)
100 CONTINUE

C SET STORAGE ARRAYS TO ZERO

DO 110 I = 1, 2
DO 110 J = 1, 720
    XMASS(I,J) = 0.
    XF(I,J) = 0.
    XH(I,J) = 0.
110 CONTINUE
RETURN
C
200 CONTINUE
   WRITE(6,*) 'ERROR - NUMBER OF EXHAUST',
   &       ' PORTS DOES NOT MATCH NUMBER OF CYLINDERS - CHECK EXHAUST',
   &       ' INPUT.'
   WRITE(6,*) '(SEE SUBROUTINE MANPAR)
STOP
C
END
SUBROUTINE MFLRT

PURPOSE
CALCULATES MASS FLOW RATE THROUGH AN ORIFICE.

USAGE
CALL MFLRT (CD, AREA, PO, MW, TO, PS, GAMMA, FLRT)

DESCRIPTION OF PARAMETERS

PARAMETER INPUT OUTPUT DESCRIPTION

CD   YES  NO  DISCHARGE COEFFICIENT
AREA YES  NO  AREA OF RESTRICTION (M**2)
PO   YES  NO  UPSTREAM PRESSURE (PA)
PS   YES  NO  DOWNSTREAM PRESSURE (PA)
MW   YES  NO  MOLECULAR WEIGHT (KG/KMOLE)
TO   YES  NO  UPSTREAM TEMPERATURE (K)
GAMMA YES  NO  RATIO OF SPECIFIC HEATS, CP/CV
FLRT NO  YES  MASS FLOW RATE (KG/S)

REMARKS
UNITS CHANGED TO SI

SUBROUTINE AND FUNCTION SUBPROGRAM REQUIRED
NONE

METHOD
FLOW THROUGH THE ORIFICE IS TREATED AS ONE-DIMENSIONAL, QUASI-STEADY, AND ISENTROPIC (MODIFIED BY A DISCHARGE COEFFICIENT)

WRITTEN BY S. H. MANSOURI AND D. N. ASSANIS
EDITED BY D. N. ASSANIS

SUBROUTINE MFLRT (CD, AREA, PO, MW, TO, PS, GAMMA, FLRT)

REAL MW

FLRT = 0.0
IF (PO .EQ. PS) GO TO 20
GI = 1.0/GAMMA
SUM = GAMMA * MW/TO
PROD = 1./91.1767 * CD * AREA * PO * SQRT(SUM)
C RATIO = PS/PO
CRIT = ( 2./(GAMMA + 1.) )**( GAMMA/(GAMMA - 1.) )
C CHECK IF FLOW IS CHOKED
C IF (RATIO .LT. CRIT) GO TO 10
C SUBSONIC FLOW
C
SUN = 2./(GAMMA - 1.) * ( RATIO**GI + GI ) - RATIO**GI
FLRT = PROD * SQRT(SUN)
GO TO 20

CHOKED FLOW

FLRT = PROD * CRIT**((0.5 * (1.0 + GI))

RETURN

END
SUBROUTINE ODERT INTEGRATES A SYSTEM OF NEQN FIRST ORDER
ORDINARY DIFFERENTIAL EQUATIONS OF THE FORM
\[ \frac{dy(i)}{dt} = f(t,y(1),\ldots,y(n)) \]
\[ y(1) \] GIVEN AT T.
THE SUBROUTINE INTEGRATES FROM T IN THE DIRECTION OF TOUT UNTIL
IT LOCATES THE FIRST ROOT OF THE NONLINEAR EQUATION
\[ g(t,y(1),\ldots,y(n),y(1),\ldots,y(n)) = 0. \]
UPON FINDING THE ROOT, THE CODE RETURNS WITH ALL PARAMETERS IN THE
CALL LIST SET FOR CONTINUING THE INTEGRATION TO THE NEXT ROOT OR
THE FIRST ROOT OF A NEW FUNCTION G. IF NO ROOT IS FOUND, THE
INTEGRATION PROCEEDS TO TOUT. AGAIN ALL PARAMETERS ARE SET TO
CONTINUE.

THE DIFFERENTIAL EQUATIONS ARE ACTUALLY SOLVED BY A SUITE OF CODES,
DERT1, STEP1, AND INTRP. ODERT ALLOCATES VIRTUAL STORAGE IN
THE WORK ARRAYS WORK AND IWORK AND CALLS DERT1. DERT1 IS A
SUPERVISOR WHICH DIRECTS THE INTEGRATION. IT CALLS ON STEP1 TO
ADVANCE THE SOLUTION AND INTRP TO INTERPOLATE THE SOLUTION AND
ITS DERIVATIVE. STEP1 USES A MODIFIED DIVIDED DIFFERENCE FORM OF
THE ADAMS PECE FORMULAS AND LOCAL EXTRAPOLATION. IT ADJUSTS THE
ORDER AND STEP SIZE TO CONTROL THE LOCAL ERROR PER UNIT STEP IN A
GENERALIZED SENSE. NORMALLY EACH CALL TO STEP1 ADVANCES THE
SOLUTION ONE STEP IN THE DIRECTION OF TOUT. FOR REASONS OF
EFFICIENCY ODERT INTEGRATES BEYOND TOUT INTERNALLY, THOUGH
NEVER BEYOND T+10*(TOUT-T), AND CALLS INTRP TO INTERPOLATE THE
SOLUTION AND DERIVATIVE AT TOUT. AN OPTION IS PROVIDED TO STOP
THE INTEGRATION AT TOUT BUT IT SHOULD BE USED ONLY IF IT IS
IMPOSSIBLE TO CONTINUE THE INTEGRATION BEYOND TOUT.

AFTER EACH INTERNAL STEP, DERT1 EVALUATES THE FUNCTION G AND
CHECKS FOR A CHANGE IN SIGN IN THE FUNCTION VALUE FROM THE
PRECEDING STEP. SUCH A CHANGE INDICATES A ROOT LIES IN THE
INTERVAL OF THE STEP JUST COMPLETED. DERT1 THEN CALLS SUBROUTINE
ROOT TO REDUCE THE BRACKETING INTERVAL UNTIL THE ROOT IS
DETERMINED TO THE DESIRED ACCURACY. SUBROUTINE ROOT USES A
COMBINATION OF THE SECANT RULE AND BISECTION TO DO THIS. THE
SOLUTION AND DERIVATIVE VALUES REQUIRED ARE OBTAINED BY
INTERPOLATION WITH INTRP. THE CODE LOCATES ONLY THOSE ROOTS
FOR WHICH G CHANGES SIGN IN (T,TOUT) AND FOR WHICH A
BRACKETING INTERVAL EXISTS. IN PARTICULAR, IT WILL NOT DETECT A
ROOT AT THE INITIAL POINT T.

THE CODES STEP1, INTRP, ROOT, AND THAT PORTION OF DERT1
WHICH DIRECTS THE INTEGRATION ARE EXPLAINED AND DOCUMENTED IN
THE TEXT, COMPUTER SOLUTION OF ORDINARY DIFFERENTIAL EQUATIONS, THE
INITIAL VALUE PROBLEM, BY L. F. SHAMPINE AND M. K. GORDON.

DETAILS OF THE USE OF ODERT ARE GIVEN IN SAND-75-0211.

C******************************************************************************
C THE PARAMETERS FOR ODERT ARE
C******************************************************************************
C F — SUBROUTINE F(T,Y,YP) TO EVALUATE DERIVATIVES YP(I) = DY(I)/DT
C NEQN — NUMBER OF EQUATIONS TO BE INTEGRATED
C Y(*) — SOLUTION VECTOR AT T
C T — INDEPENDENT VARIABLE
C TOUT — ARBITRARY POINT BEYOND THE ROOT DESIRED
C RELERR,ABSERR — RELATIVE AND ABSOLUTE ERROR TOLERANCES FOR LOCAL
C ERROR TEST. AT EACH STEP THE CODE REQUIRES
C \abs(\text{LOCAL ERROR}) \leq \text{ABS}(Y) \times \text{RELERR} + \text{ABSERR}
C FOR EACH COMPONENT OF THE LOCAL ERROR AND SOLUTION VECTORS
C IFLAG — INDICATES STATUS OF INTEGRATION
C WORK,IWORK — ARRAYS TO HOLD INFORMATION INTERNAL TO THE CODE
C WHICH IS NECESSARY FOR SUBSEQUENT CALLS
C G — FUNCTION OF T, Y(*), YP(*) WHOSE ROOT IS DESIRED.
C REROOT, AEROOT — RELATIVE AND ABSOLUTE ERROR TOLERANCES FOR
C ACCEPTING THE ROOT. THE INTERVAL CONTAINING THE ROOT IS
C REDUCED UNTIL IT SATISFIES
C 0.5 \times \abs(\text{LENGTH OF INTERVAL}) \leq \text{REROOT} \times \abs(\text{ROOT}) + \text{AEROOT}
C WHERE ROOT IS THAT ENDPOINT YIELDING THE SMALLER VALUE OF
C G IN MAGNITUDE. PURE RELATIVE ERROR IS NOT RECOMMENDED
C IF THE ROOT MIGHT BE ZERO.
C******************************************************************************
C FIRST CALL TO ODERT —
C******************************************************************************
C THE USER MUST PROVIDE STORAGE IN HIS CALLING PROGRAM FOR THE
C ARRAYS IN THE CALL LIST,
C Y(NEQN), WORK(100+21*NEQN), IWORK(5)
C AND DECLARE F, G IN AN EXTERNAL STATEMENT. HE MUST SUPPLY THE
C SUBROUTINE F(T,Y,YP) TO EVALUATE
C \frac{DY(I)}{DT} = YP(I) = F(T,Y(I),\ldots,Y(NEQN))
C AND THE FUNCTION G(T,Y,YP) TO EVALUATE
G = G(T,Y(1),...,Y(NEQN),YP(1),...,YP(NEQN)).
NOTE THAT THE ARRAY YP IS AN INPUT ARGUMENT AND SHOULD NOT BE
COMPUTED IN THE FUNCTION SUBPROGRAM. FINALLY THE USER MUST
INITIALIZE THE PARAMETERS
NEQN — NUMBER OF EQUATIONS TO BE INTEGRATED
Y(*) — VECTOR OF INITIAL CONDITIONS
T — STARTING POINT OF INTEGRATION
TOUT — ARBITRARY POINT BEYOND THE ROOT DESIRED
RELERR,ABSErr — RELATIVE AND ABSOLUTE LOCAL ERROR TOLERANCES
FOR INTEGRATING THE EQUATIONS
IFLAG — +1,-1. INDICATOR TO INITIALIZE THE CODE. NORMAL INPUT
IS +1. THE USER SHOULD SET IFLAG=-1 ONLY IF IT IS
IMPOSSIBLE TO CONTINUE THE INTEGRATION BEYOND TOUT .
REROOT,AEROOT — RELATIVE AND ABSOLUTE ERROR TOLERANCES FOR
COMPUTING THE ROOT OF G

C ALL PARAMETERS EXCEPT F, G, NEQN, TOUT, REROOT AND AEROOT MAY BE
ALTERED BY THE CODE ON OUTPUT SO MUST BE VARIABLES IN THE CALLING
PROGRAM.
C***********************************************************************
C OUTPUT FROM ODERT —
C***********************************************************************
C NEQN — UNCHANGED
C Y(*) — SOLUTION AT T
C T — LAST POINT REACHED IN INTEGRATION. NORMAL RETURN HAS
C T = TOUT OR T = ROOT
C TOUT — UNCHANGED
C RELERR,ABSErr — NORMAL RETURN HAS TOLERANCES UNCHANGED. IFLAG=3
C SIGNALS TOLERANCES INCREASED
C IFLAG = 2 — NORMAL RETURN. INTEGRATION REACHED TOUT
C = 3 — INTEGRATION DID NOT REACH TOUT BECAUSE ERROR
C TOLERANCES TOO SMALL. RELERR , ABSErr INCREASED
C APPROPRIATELY FOR CONTINUING
C = 4 — INTEGRATION DID NOT REACH TOUT BECAUSE MORE THAN
C 500 STEPS NEEDED
C = 5 — INTEGRATION DID NOT REACH TOUT BECAUSE EQUATIONS
C APPEAR TO BE STIFF
C = 6 — INTEGRATION DID NOT REACH TOUT BECAUSE SOLUTION
C VANISHED MAKING PURE RELATIVE ERROR IMPOSSIBLE.
C MUST USE NON-ZERO ABSErr TO CONTINUE
C = 7 — INVALID INPUT PARAMETERS (FATAL ERROR)
C = 8 — NORMAL RETURN. A ROOT WAS FOUND WHICH SATISFIED
C THE ERROR CRITERION OR HAD A ZERO RESIDUAL
C = 9 — ABNORMAL RETURN. AN ODD ORDER POLE OF G WAS
C FOUND.
C =10 — ABNORMAL RETURN. TOO MANY EVALUATIONS OF G WERE
C REQUIRED (AS PROGRAMMED 500 ARE ALLOWED.)
C THE VALUE OF IFLAG IS RETURNED NEGATIVE WHEN THE INPUT
C VALUE IS NEGATIVE AND THE INTEGRATION DOES NOT REACH
C TOUT , I.E., -3,...,-6,-8,-9,-10.
C WORK(*),IWORK(*) — INFORMATION GENERALLY OF NO INTEREST TO THE
C USER BUT NECESSARY FOR SUBSEQUENT CALLS
C REROOT,AEROOT — UNCHANGED
C***********************************************************************
C SUBROUTINE ODERT RETURNS WITH ALL INFORMATION NEEDED TO CONTINUE
C THE INTEGRATION. IF THE INTEGRATION DID NOT REACH TOUT AND THE
C USER WANTS TO CONTINUE, HE JUST CALLS AGAIN. IF THE INTEGRATION
REACHED TOUT, THE USER NEED ONLY DEFINE A NEW TOUT AND CALL
AGAIN. THE OUTPUT VALUE OF IFLAG IS THE APPROPRIATE INPUT VALUE
FOR SUBSEQUENT CALLS. THE ONLY SITUATION IN WHICH IT SHOULD BE
ALtered IS TO STOP THE INTEGRATION INTERNALLY AT THE NEW TOUT.
I.E., CHANGE OUTPUT IFLAG=2 TO INPUT IFLAG=2. ONLY THE ERROR
TOLERANCES AND THE FUNCTION G MAY BE CHANGED BY THE USER BEFORE
CONTINUING. ALL OTHER PARAMETERS MUST REMAIN UNCHANGED. A NEW
FUNCTION G IS DETECTED AUTOMATICALLY.

SUBROUTINE ODERT(F,NEQN,Y,T,TOUT,RELERR,ABSErr,IFLAG,WORK,IWORk,
G,REROOT,AEROOT)
IMPLICIT REAL*8 (A-H,O-Z)
IMPLICIT INTEGER*4 (I-N)

LOGICAL START,PHASE1,NORND
DIMENSION Y(NEQN),WORK(1570),IWORK(5)
EXTERNAL F,G
DATA IALPHA,IBETA,ISIG,IV,IW,IGG,IPHASE,IPSI,IX,IH,IHOLD,ISTart,
1 ITOLD,IDELENs,IGX,ITROOT/1,13,25,38,50,62,75,76,88,89,90,91,
2 92,93,94,95/
IYY = 100
IWT = IYY + NEQN
IP = IWT + NEQN
IYP = IP + NEQN
IYPout = IYP + NEQN
IPHI = IYPout + NEQN
IF(IABS(IFLAG) .EQ. 1) GO TO 1
START = WORK(ISTART) .GT. 0.0
PHASE1 = WORK(IPHASE) .GT. 0.0
NORND = IWORK(2) .NE. -1
CALL DERT1(F,NEQN,Y,T,TOUT,RELERR,ABSErr,IFLAG,G,REROOT,AEROOT,
1 WORK(IYy),WORK(IWT),WORK(IP),WORK(IYP),WORK(IYPout),WORK(IPHI),
2 WORK(IALPHA),WORK(IBETA),WORK(ISig),WORK(IV),WORK(IW),WORK(IGG),
3 PHASE1,WORK(IPSI),WORK(IX),WORK(IH),WORK(IHOLD),START,
4 WORK(ITOLD),WORK(IDELEN),WORK(IGX),WORK(ITROOT),IWORK(1),
5 NORND,IWORK(3),IWORK(4),IWORK(5))
WORK(ISTART) = -1.0
IF(START) WORK(ISTART) = 1.0
WORK(IPHASE) = -1.0
IF(PHASE1) WORK(IPHASE) = 1.0
IWORK(2) = -1
IF(NORND) IWORK(2) = 1
RETURN
END
SUBROUTINE PARFIN

PURPOSE
READS IN WALL CONSTRUCTION DATA, GIVEN IN "PHEAT.DAT", FOR
THE PISTON AND THE CYLINDER HEAD. ALSO, CALCULATES THE
COURANT NUMBER FOR EACH MATERIAL LAYER AND THE OVERALL
HEAT TRANSFER COEFFICIENT FOR EACH COMPONENT.

USAGE
CALL PARFIN

DESCRIPTION OF PARAMETERS

PARAMETER INPUT OUTPUT DESCRIPTION

NPC YES NO TOTAL NUMBER OF COMPONENTS WITH
PARALLEL COMPOSITE WALL STRUCTURES
NPLA(I) YES NO NUMBER OF MATERIAL LAYERS OF
IITH COMPONENT
PHCOOL(I) --- --- HEAT TRANSFER COEFFICIENT FROM THE
OUTSIDE WALL SURFACE OF IITH COMPONENT
PTHIK(I,J) --- --- TO THE COOLANT OR AMBIENT (W/M2/K)
PTCOOL(I) YES NO AMBIENT TEMPERATURE, COOLANT TEMPERATURE,
OR SPECIFIED OUTSIDE WALL TEMPERATURE

P(THI)(J) YES NO THICKNESS OF JTH LAYER OF IITH
COMPONENT (M).
PCOND(I,J) YES NO THERMAL CONDUCTIVITY OF JTH LAYER
OF IITH COMPONENT (W/M/K)
PDIFU(I,J) YES NO THERMAL DIFFUSIVITY OF JTH LAYER OF
IITH COMPONENT (M2/SEC)
INNODE(I,J) YES NO NUMBER OF NODES PLACED IN JTH LAYER
OF IITH COMPONENT
FACT YES NO FRACTION OF NODES OF FIRST LAYER PLACED
WITHIN THE PENETRATION DEPTH

REMARKS
FIRST ARRAY DIMENSION : COMPONENT DESCRIPTION
SECOND ARRAY DIMENSION : LAYER DESCRIPTION

THREE LAYERS MUST BE INPUT FOR THE TRANSIENT WALL TEMPERATURE
CASE

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
NONE

WRITTEN BY D. N. ASSANIS
EDITED BY D. N. ASSANIS

SUBROUTINE PARFIN

COMMON/PARF/PDEIX(10,3),PCNUM(10,3),INNODE(10,3),PTHIK(10,3),
PCOND(10,3),PHEFF(10,3),PDIFU(10,3),PHCOOL(10),PTCOOL(10),
& PUOUE(10), PSURFA(10)
COMMON/PERFCT/ PERI, FACT, DLT
COMMON/DDT/DSPD
COMMON/NPLA/ NPLA(10), NPC
NAMELIST/PHEAT/ FACT, NPC, NPLA, PTCOOL, PHCOOL, PTHIK, PCOND, & PDIFU, INNODE

C
READ (11, PHEAT)
CLOSE (UNIT =11)
C
DELT = ESPD
C
DO 10 I = 1, NPC
SUM = 0.
DO 20 J = 1, NPLA(I)
PDEIX(I,J) = PTHIK(I,J)/(INNODE(I,J)-1.)
PCNUM(I,J) = PDEIX(I,J)**2 / (PDIFU(I,J)*DELT)
PHEFF(I,J) = PCOND(I,J) / PDEIX(I,J)
SUM = SUM + PTHIK(I,J)/PCOND(I,J)
20 CONTINUE
IF (PHCOOL(I) .NE.0.) SUM = SUM + 1./PHCOOL(I)
PUOVE(I) = 1./SUM
10 CONTINUE
C
RETURN
END
SUBROUTINE PFDIF

PURPOSE
  Calculates the periodic temperature distribution within each layer of any component having composite walls (piston and cylinder head). The method is based on an explicit finite difference scheme, suitably modified to handle optimally-spaced discrete nodes within the first material layer of each component.

USAGE
  CALL PFDIF

DESCRIPTION OF PARAMETERS

PARAMETER INPUT OUTPUT DESCRIPTION
INDEX YES NO INDEX DESCRIBING THE COMPONENT WALL
  1: PISTON
  2: CYLINDER HEAD
  3: EXHAUST PORT
  4...: OTHER EXHAUST SECTIONS
TWALL NO YES INSTANTANEOUS WALL SURFACE TEMPERATURE

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
NONE

REMARKS
  This subroutine is set up for three layers of material. Even for single layer walls, they should be input as a composite of three layers.

METHOD
  See D. N. Assanis, Ph.D. Thesis

WRITTEN BY D. N. Assanis
EDITED BY D. N. Assanis

SUBROUTINE PFDIF

DIMENSION PTW(10,3,51), TW(3,51), TNW(3,51)
DIMENSION DELX(3), CNUM(3), NNODE(3), THIK(3), COND(3),
& HEFF(3), DIFU(3)

COMMON/PTW/PTW
COMMON/PCON/PDELX(10,3),PCNUM(10,3),INNODE(10,3),PTHIK(10,3),
& PCOND(10,3), PHEFF(10,3), PDIFU(10,3), PTCOOL(10),
& PUCOOL(10), PSURFA(10)
COMMON/HTCOG/HTCO(10), TGAS(10)
COMMON/PERCFT/PRECFT, PERIOD, FACT, DELT
COMMON/QSOL/QWALL(10), QAVE(10), QSS(10)
COMMON/DDTH/ESPD
COMMON/NPLA/ NPLA(10), NPC
COMMON/TMPS/NTEMP, TWALL(10), TSS(10), TSS2(10), TCW, TCW2

DECL = ESPD

LOOP FOR NUMBER OF SURFACES FOR TRANSIENT CALCULATION
DO 100 INDEX = 1, NTEMP
C DEFINE DUMMY VARIABLES
HTRCOE = HTCO(INDEX)
TG = TGAS(INDEX)
HCOOL = PHCOOL(INDEX)
TCOOL = PTCOOL(INDEX)
UOVE = PUOVE(INDEX)
QSOL = QSS(INDEX)
STW = TSS(INDEX)
CDO 88 J = 1, 3
DELX(J) = PDELX(INDEX,J)
CNUM(J) = PCNUM(INDEX,J)
NNODE(J) = INNODE(INDEX,J)
THIK(J) = PTHIK(INDEX,J)
COND(J) = PCOND(INDEX,J)
HEFF(J) = PHEFF(INDEX,J)
DIFU(J) = PDIFU(INDEX,J)
CDO 86 K = 1, NNODE(J)
TW(J,K) = PTW(INDEX,J,K)
C C CALCULATE PARAMETERS REQUIRED TO OPTIMIZE GRID SPACING
C SKIN = SORT(DIFU(1)*PERIOD)
ALPHA = 1./THIK(1)
GAMMA = (SKIN*ALPHA - FACT)/(SKIN-THIK(1))/FACT
C NNODE1 = NNODE(1)
NNODE2 = NNODE(2)
NNODE3 = NNODE(3)
C C CALCULATE TEMPERATURES AT INTERIOR NODES. NOTE TRANSFORMATION
INTRODUCED IN FIRST LAYER TO MAP VARIABLY-SPACED NODES INTO
CORRESPONDING UNIFORMLY-SPACED NODES.
CIF (NNODE1.EQ.2) GO TO 15
DO 10 I = 2, (NNODE1 - 1)
YMID = ((I-1.))/(NNODE1-1.)
XMID = (1.-YMID)/ (GAMMA*YMID - ALPHA)
DYDXM = (ALPHA - GAMMA)/(GAMMA*XMID+1.)**2
C YLEF = (I-1.5)/(NNODE1-1.)
XLEF = (1.-YLEF)/ (GAMMA*YLEF - ALPHA)
DYDXL = (ALPHA - GAMMA)/(GAMMA*XLEF+1.)**2
C YRIT= (I-0.5)/(NNODE1-1.)
XRIT = (1.-YRIT)/ (GAMMA*YRIT - ALPHA)
DYDXR = (ALPHA - GAMMA)/(GAMMA*XRIT+1.)**2
C TNW(1,I)= TW(1,I)+(((TW(1,I+1)-TW(1,I))*DYDXR-(TW(1,I)-
&W(1,I-1))*DYDXL)*DIFU(1)+DELT=DYDXM*(NNODE1-1.)**2
10 CONTINUE
15 CONTINUE
CIF (NNODE2.EQ.2) GO TO 25
DO 20 I = 2, (NNODE2 - 1)
TNW(2,I)= (TW(2,I-1) + TW(2,I+1) +
& (CNUM(2)-2.)*TW(2,I))/CNUM(2)
20 CONTINUE
25 CONTINUE
C
IF (NNOD3.EQ.2) GO TO 28
DO 26 I = 2, (NNOD3 - 1)
  TNW(3,I) = (TW(3,I-1) + TW(3,I+1) +
& (CNUM(3)-2.)*TW(3,I))/CNUM(3)
26 CONTINUE
28 CONTINUE
C
UPDATE INTERIOR NODES
C
IF (NNOD1.EQ.2) GO TO 35
DO 30 I = 2, (NNOD1 - 1)
  TW(1,I) = TNW(1,I)
30 CONTINUE
35 CONTINUE
C
IF (NNOD2.EQ.2) GO TO 45
DO 40 I = 2, (NNOD2 - 1)
  TW(2,I) = TNW(2,I)
40 CONTINUE
45 CONTINUE
C
IF (NNOD3.EQ.2) GO TO 60
DO 50 I = 2, (NNOD3 - 1)
  TW(3,I) = TNW(3,I)
50 CONTINUE
C
CALCULATE WALL TEMPERATURE AT GAS SIDE:
DYDX = (ALPHA-GAMMA)/(1.-GAMMA*THIK(1))**2
FF = DYDX * (NNOD1-1.) * COND(1) / HTRCOE
FG = FF * HTRCOE / 2.
TW(1,1) = ((4.*TW(1,2) - TW(1,3))*FG - QSOL +
& HTRCOE*(TG-STW)) / (3.*FG + HTRCOE)
C
CALCULATE TEMPERATURE AT INTERFACE BETWEEN FIRST AND
SECOND LAYER:
FF = (NNOD2-1.)/(NNOD1-1.) / THIK(2) * COND(2)/COND(1)/DYDX
TW(1,NNOD1) = (2.*TW(1,NNOD1-1) - 0.5*TW(1,NNOD1-2) -
& FF*(0.5*TW(2,3) - 2.*TW(2,2)))/1.5/(1.+FF)
C
TW(2,1) = TW(1,NNOD1)
C
CALCULATE TEMPERATURE AT INTERFACE BETWEEN SECOND AND
THIRD LAYER:
FF = (NNOD3-1.)/(NNOD2-1.) * THIK(2)/THIK(3) * COND(3)/COND(2)
TW(2,NNOD2) = (2.*TW(2,NNOD2-1) - 0.5*TW(2,NNOD2-2) -
& FF*(0.5*TW(3,3) - 2.*TW(3,2)))/1.5/(1.+FF)
C
TW(3,1) = TW(2,NNOD2)
C
CALCULATE WALL TEMPERATURE AT COOLANT SIDE:
IF (HCOOL .EQ. 0.) GO TO 70
FG = HEFF(3)/ 2.
TW(3,NNOD3) = (4.*TW(3,NNOD3-1) - TW(3,NNOD3-2))*FG +
& (3.*FG + HCOOL)
GO TO 75
C
70 TW(3,NNOD3) = 0.0
C 75 T WALL(INDEX) = TW(1,1) + STW
C DO 988 J = 1, J
   DO 988 K = 1, NNODE(J)
   PTW(INDEX,J,K) = TW(J,K)
988 CONTINUE
C 100 CONTINUE
RETURN
END
SUBROUTINE QEND(NEQN, DY)

PURPOSE
QEND IS CALLED BY SIM TO CALCULATE AVERAGE HEAT TRANSFER
RATES AND NEW STEADY-STATE INSIDE WALL SURFACE TEMPERATURES
BASED ON THESE TEMPERATURES. QEND IS CALLED ONCE AT THE
END OF EACH CYCLE.

USAGE
CALL QEND(NEQN, DY)

SUBROUTINES AND FUNCTIONS REQUIRED
NONE

WRITTEN BY R.M. FRANK

SUBROUTINE QEND(NEQN, DY)

REAL*8 DY(NEQN)
LOGICAL EXSUB

COMMON/ITERAS/ ITERAS, ISTEDY
COMMON/EXSUB/ EXSUB
COMMON/EXNUM/ NUMEX, NUMEXT
COMMON/PARF/ PDELX(10,3),PCNUM(10,3),INNODE(10,3),PTHIK(10,3),
PCOND(10,3), PHEFF(10,3), PDIFU(10,3), PHCOOL(10), PTCOOL(10),
PUOVE(10), PSURFA(10)
COMMON/CYLP/ CDIAM(6), CTHIK(6,3), CCOND(6,3), CHCOOL(6),
& CTCOOL(6), CUOVE(6)
COMMON/AREAS/ AHEAD, APSTON
COMMON/DTOTH/ ESPD
COMMON/TEMPS/ NTEMP, TWALL(10), TSS(10), TSS2(10), TCW, TCW2
COMMON/QLSOL/ QWALL(10), QAVE(10), QSS(10)

CONVERT AVERAGED HEAT TRANSFER COEF. AND H.T. COEF. * GAS TEMP
DY(54) = DY(54)/720.
DY(55) = DY(55)/720.
DY(56) = DY(56)/720.
DY(57) = DY(57)/720.
DY(58) = DY(58)/720.
DY(59) = DY(59)/720.

CALCULATE APPARENT STEADY-STATE WALL TEMPERATURES,
BASED ON WALL CONDUCTIVITY AND AVERAGE HEAT TRANSFER
PARAMETERS
TSS2(1) = (PUOVE(1)*PTCOOL(1) + DY(55))/(PUOVE(1)+DY(54))
TSS2(2) = (PUOVE(2)*PTCOOL(2) + DY(57))/(PUOVE(2)+DY(56))

TCW2 = (CUOVE(1)*CTCOOL(1) + DY(59))/(CUOVE(1)+DY(58))

CALCULATE AVERAGE HEAT FLUX FROM THE GAS TO THE SURFACE
OF THE PISTON AND THE HEAD:
QAVE(1) = DY(8) /APSTON /ESPD /720.
QAVE(2) = DY(9) /AHEAD /ESPD /720.

CONTINUE ONLY FOR EXHAUST MANIFOLD SUB-DIVIDED CASE
IF (.NOT.EXSUB) RETURN

C LOOP FOR NUMBER OF CONTROL VOLUMES IN EXHAUST SYSTEM
DO 10 I = 1, NUMEXT
  IREF = I + 2
  IHTC = 69 + (I-1)*8
  DY(IHTC) = DY(IHTC) / 720.
  DY(IHTC+1) = DY(IHTC+1) / 720.
  TSS2(IREF) = (PUOVE(IREF)*PTCOOL(IREF) + DY(IHTC+1)) /
            (PUOVE(IREF) + DY(IHTC))
  QAVE(IREF) = DY(IHTC-1) / PSURFA(IREF) / ESPD / 720.
10 CONTINUE
C
RETURN
END
SUBROUTINE QMAN

PURPOSE
CALCULATES HEAT TRANSFER FROM MANIFOLD

USAGE
CALL QMAN (TB, P, FFR, RHOB, J, ENGM, Q)

DESCRIPTION OF PARAMETERS

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>INPUT/OUTPUT</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>TB</td>
<td>YES/NO</td>
<td>MANIFOLD GAS TEMPERATURE (K)</td>
</tr>
<tr>
<td>P</td>
<td>YES/NO</td>
<td>MANIFOLD AVERAGE PRESSURE (PA)</td>
</tr>
<tr>
<td>FFR</td>
<td>YES/NO</td>
<td>MANIFOLD FUEL FRACTION (-)</td>
</tr>
<tr>
<td>RHOB</td>
<td>YES/NO</td>
<td>BULK DENSITY (KG/M**3)</td>
</tr>
<tr>
<td>J</td>
<td>YES/NO</td>
<td>INDEX (INTAKE: J-1, EXHAUST: J-2, CONNECTING PIPE: J-3)</td>
</tr>
<tr>
<td>ENGM</td>
<td>YES/NO</td>
<td>TOTAL ENGINE MASS FLOW RATE (KG/DEG)</td>
</tr>
<tr>
<td>Q</td>
<td>YES/NO</td>
<td>HEAT TRANSFER (W)</td>
</tr>
</tbody>
</table>

REMARKS
QMAN REPLACES THE HEAT TRANSFER PORTION OF QDP USED IN THE ORIGINAL VERSION OF THE PROGRAM

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
THERMO TRANSP

METHOD
HEAT TRANSFER CALCULATED FROM CORRELATION OF REYNOLDS, PRANDTL, AND NUSSELT NUMBERS FOR TURBULENT FORCED CONVECTION IN CIRCULAR TUBES.
SEE ROSENOH & CHOI, 'HEAT, MASS AND MOMENTUM TRANSFER'.

WRITTEN BY D. N. ASSANIS
EDITED BY D. N. ASSANIS & R. M. FRANK

SUBROUTINE QMAN (TB, P, FFR, RHOB, J, ENGM, Q)

COMMON/DDTH/ ESPD
COMMON/QP1/ EDIAM(3), EAREA(3), ECROSS(3), ETWALL(3), ECONHT(3)
COMMON/QP2/ EPRF(3), EVBLK(3), EREF(3), ENUF(3), EHTCOE(3), EQDOT(3)

IF (ENG .GT. 0.) GO TO 20
Q = 0.
RETURN

DEFINE WALL AND FILM TEMPERATURES:
20 TW = ETWALL(J)
TF = (TB + TW) / 2.

CALCULATE FILM DENSITY, SPECIFIC HEAT AND GAMMA:
CALL THERMO (TF, P, FFR, HF, CPF, Z1, Z2, RHOF, Z3, Z4, Z5, GAMF, Z6, Z7, Z8, Z9)
CALL TRANSP (TF, FFR, GAMF, CPF, UF, TKF)

CALCULATE BULK VELOCITY, FILM REYNOLDS NUMBER,
C FILM PRANDTL NUMBER:
EVBLK(J) = ENGM / ESPD / RHOB / ECROSS(J)
EREF(J) = EVBLK(J) * EDIAM(J) * RHOF / UF
EPRF(J) = CPF * UF / TKF

C CALCULATE NUSSELT NUMBER, HEAT TRANSFER COEFFICIENT
ENUF(J) = ECONHT(J) * EREF(J)**.8 * EPRF(J)**.3
EHTCOE(J) = ENUF(J) * TKF / EDIAM(J)

C CALCULATE HEAT TRANSFER RATE:
EQDOT(J) IS HEAT TRANSFER FROM FLUID TO WALL
EQDOT(J) = EHTCOE(J) * EAREA(J) * (TB - TW)

C CONVERT TO J/DEG:
Q = EQDOT(J) * ESPD

RETURN
END
SUBROUTINE RESULT

PURPOSE
WRITES CRANK-ANGLE BY CRANK-ANGLE RESULTS OF THE CYCLE SIMULATION TO THE APPROPRIATE OUTPUT FILES. THESE ARE RESULTS WHICH ARE RECORDED DEPENDING ON THE ENGINE PROCESS (SEE PROCESS LOOPS), AND RESULTS WHICH ARE RECORDED THROUGHOUT THE CYCLE (SEE COMMON SECTION).

USAGE
CALL RESULT (INDEX, NEQN, DT, DY)

DESCRIPTION OF PARAMETERS

PARAMETER INPUT OUTPUT DESCRIPTION
INDEX YES NO DETERMINES WHICH SET OF RESULTS ARE TO BE RECORDED

REMARKS
INDEX=1 MEANS THAT INTAKE LOOP RESULTS RECORDED
INDEX=2 MEANS THAT CMPRES LOOP RESULTS RECORDED
INDEX=3 MEANS THAT CMBSN LOOP RESULTS RECORDED
INDEX=4 MEANS THAT EXAUST LOOP RESULTS RECORDED

SUBROUTINE AND FUNCTION SUBPROGRAMS REQUIRED
NONE

WRITTEN BY D. N. ASSANIS
EDITED BY D. N. ASSANIS

SUBROUTINE RESULT (INDEX, NEQN, DT, DY, IFLAG)

LOGICAL SPTEMP, ANNAND, EXSUB

INTEGER SIZC, SIZT, SIZPT, SIZ1, SIZ2, SIZ3

REAL*8 DT, DY(NEQN)
REAL MMM, MCRSC, MASS, MASSIN, MASSEX, KIL, MIL

DIMENSION PTW(10,3,51)

PARAMETER (PI=3.1415927)
PARAMETER (SIZC=6, SIZT=6, SIZPT=6, SIZ1=8, SIZ2=8, SIZ3=11)
PARAMETER (CEN = 1.E2, KIL = 1.E3, MIL = 1.E0, ERG = 1.E7,
& ATPA = 1.01325E5)

DIMENSION MASSIN(270), MASSEX(270),
& FCYLIN(270), FCYLEX(270),
& HCYLIN(270), HCYLEX(270)

COMMON/SPTEMP/ SPTEMP
COMMON/ANNAND/ ANNAND
COMMON/EXSUB/ EXSUB
COMMON/RADN/ TCHTR, TRHTR, THTR, CHTRAP, CHTRAH, CHTRAW,
& RHTRAP, RHTRAH, RHTRAW
C
 COMMON/FLAME/ TAIR, PAIR, TFLAME, TRAD, EMIS
C
 COMMON/PFD/ PTW
 COMMON/HOTCOTG, HTCO(10), TGAS(10)
 COMMON/PARF/PDELX(10,3), PCNUM(10,3), NNODE(10,3), PTHIK(10,3),
 & PCOND(10,3), PHEFF(10,3), PDIFU(10,3), PHCOOL(10), PTOOL(10),
 & PUOVE(10), PSURFA(10)
C
 COMMON/ITERAS/ ITERAS, ISTDY
 COMMON/PRINT/ TPRINT, TSCREEN
C
 COMMON/D /ERPM
 COMMON/DOUTH/ ESPD
 COMMON/ARRAY/ MASSIN, MASEX, FCYLIN, FCYLEX, HCYLIN, HCYLEX
 COMMON/RHMS/ RHO, MASS, VOLUME, HH, GAMMA
 COMMON/VALVE/ VIV, VEV
 COMMON/B /CPM(2), HM(2), MM(2), GM(2), RHOM(2)
 COMMON/K/RTEMP(5), H(5), RMASS(5), RCORR(5)
 COMMON/NEWDIIF /ASP(3), PR(3), PRSS(5), DP(5), HI(5),
 & TMAP(2), PTMAP(2), CMAP(2),
 & CM(SIZC,SIZ1,3), TM(SIZT,SIZ2,3), PTM(SIZPT,SIZ3,3),
 & CRPM(SIZC), TRPM(SIZT), PTRPM(SIZPT), PSTD(3), TSTD(3)
 COMMON/FBRATE/ FBRATE
 COMMON/TURBU/ CBETA, MACRSC, UPRIME, VMAKE, VFISTO
 COMMON/HEATS/ CVHTRN, HTROCE, HTPAPI, HTPAHD, HTPACW, HTRAPI,
 & HTRASH, HTRACW, HTRAN
 COMMON/OPZ2/ EPRF(3), EVBLK(3), EREF(3), ENUF(3), ETHCOE(3),
 & EQDOT(3)
 COMMON/AREAS/ AHEAD, APSTON
C
 T = DT
C
 IF(INDEX .EQ. 1) GO TO 10
 IF(INDEX .EQ. 2) GO TO 20
 IF(INDEX .EQ. 3) GO TO 30
 IF(INDEX .EQ. 4) GO TO 40
C
C---------------------------------------- INTAKE ----------------------------------------
C
 10 CONTINUE
 11 IF (AMOD(T, TSCREEN).EQ.0.0) WRITE (7,881)
 12 & DT, DY(12)/ATPA, DY(11), IFLAG
 13 IF (AMOD(T, TPRINT).EQ.0.0) WRITE(6,4210) DT, DY(12)/ATPA,
 14 & DY(11),DY(1)*KIL,DY(2)*KIL, EQR(DY(28)), DY(23)/ATPA,
 15 & DY(22), DY(27)/ATPA, DY(26), IFLAG, DY(29)
 16 GO TO 50
C
C---------------------------------------- COMPRESSION LOOP ----------------------------------------
C
C----------------------------------------
C
 20 CONTINUE
 21 IF (AMOD(T, TSCREEN).EQ.0.0) WRITE(7,882) DT,
 22 & DY(12)/ATPA, IFLAG
 23 IF (AMOD(T, TPRINT).EQ.0.0) WRITE(6,4211) DT, DY(12)/ATPA,
 24 & DY(11), EQR(DY(28)),DY(23)/ATPA, DY(22), DY(27)/ATPA,
 25 & DY(26), IFLAG, DY(29)
 26 GO TO 50
C ************************************************************
C COMBUSTION LOOP
C ************************************************************
C 30 CONTINUE
C IF (.NOT. ANAND .AND. AMOD(T,TPRINT).EQ.0.)
& WRITE(14,8210) T, DY(12)/ATPA, TAIN, DY(11), TFLAME,
& TRAD, EMIS, TRHR/KIL, THTR/KIL, RHTRAP/APSTON/KIL
C IF (AMOD(T, TSCREEN).EQ.0.) WRITE(7,883) DT, DY(12)/ATPA,
& DY(4), IFLAG
C PRINT OUTPUT EVERY CRANK ANGLE FOR FIRST PART OF COMBUSTION
IF (DT .LT. 380. .OR. AMOD(T, TPRINT).EQ.0.0)
& WRITE(6,4212) DT, DY(12)/ATPA,
& DY(11), DY(4), EQR(DY(20)), DY(23)/ATPA, DY(22),
& DY(27)/ATPA, DY(26), IFLAG, DY(29)
GO TO 50
C EXHAUST LOOP
C 40 CONTINUE
C IF (AMOD(T, TSCREEN).EQ.0.) WRITE(7,882)
& DT, DY(12)/ATPA, IFLAG
IF (AMOD(T, TPRINT).EQ.0.0) WRITE(6,4213) DT, DY(12)/ATPA,
& DY(11), DY(4), EQR(DY(20)), DY(23)/ATPA, DY(22),
& DY(27)/ATPA, DY(26), IFLAG, DY(29)
C COMMON SECTION FOLLOWS
C 50 CONTINUE
C WRITE TRANSIENT WALL TEMPERATURE PROFILES
IF (.NOT. SPTEMP .AND. ITERAS.GT.ISTEDY .AND.
& AMOD(T, TPRINT).EQ.0.0) THEN
WRITE(36,991) T, (PTW(1,1.INOD), INOD-1,MIN(15,INNODE(1,1)),1)
WRITE(37,991) T, (PTW(2,1.INOD), INOD-1,MIN(15,INNODE(2,1)),1)
IF (EXSUB) THEN
WRITE(38,991) T, (PTW(3,1,INOD), INOD - 1,MIN(15,INNODE(3,1)),2)
WRITE(39,991) T, (PTW(4,1,INOD), INOD - 1,MIN(15,INNODE(4,1)),1)
WRITE(48,991) T, (PTW(5,1,INOD), INOD - 1,MIN(15,INNODE(5,1)),2)
ENDIF
ENDIF
C WRITE TURBULENCE FLOW OUTPUT
IF (AMOD(T,TPRINT).EQ.0.) WRITE(12,6210)
& T, DY(6), DY(7), VIV, VEV, VMKE, UPRIME, CVHTRN, MACRSC*CEN
C WRITE CYLINDER HEAT TRANSFER OUTPUT
IF (AMOD(T,TPRINT).EQ.0.) WRITE(13,7210)
& T, HTRCOE, HTPAPI/KIL, HTPAHD/KIL, HTPACW/KIL, THTRAN/KIL
C
C*****************************************************************************
C FORMAT STATEMENTS FOLLOW
C*****************************************************************************
C
C 881 FORMAT (1H, 2X, 'CA = ', F6.2, 10X, 'P = ', F10.5, 9X, 'T = ', F10.2,
C & 8X, 'IFG = ', I2)
C
C 882 FORMAT (1H, 2X, 'CA = ', F6.2, 10X, 'P = ', F10.5, 28X, 'IFG = ', I2)
C
C 883 FORMAT (1H, 2X, 'CA = ', F6.2, 10X, 'P = ', F10.5, 9X, 'XB = ', F9.6,
C & 5X, 'IFG = ', I2)
C
C 991 FORMAT (/1X, F4.0, 15(F8.3))
C
C 4210 FORMAT (1X, F8.1, 2X, F9.4, 2X, F9.2, 2X, F9.5, 2X, F9.5,
C
C 4211 FORMAT (1X, F8.1, 2X, F9.4, 2X, F9.2, 24X,
C
C 4212 FORMAT (1X, F8.1, 2X, F9.4, 2X, F9.2, 10X, F6.4, 8X, F9.5,
C
C 4213 FORMAT (1X, F8.1, 2X, F9.4, 2X, F9.2, 13X, F9.5, 2X, F9.5,
C
C 6210 FORMAT (5X, F7.1, 4X, F9.5, 5X, F9.5, 4X, F10.3, 3X, F10.3, 3X,
C & F10.3, 3X, F10.3, 4X, F10.3, 4X, F10.3)
C
C 7210 FORMAT (5X, F7.1, 3X, F10.1, 3X, F11.3, 3X, F11.3, 3X, F11.3, 3X,
C & F10.3)
C
C 8210 FORMAT (5X, F7.1, 2X, F10.1, 3X, F10.1, 3X, F10.1, 3X, F10.1,
C & 3X, F10.1, 2X, F10.3, 3X, F10.3, 3X, F10.3, 3X, F10.3)
C
C RETURN
C END
ROOT COMPUTES A ROOT OF THE NONLINEAR EQUATION \( F(X) = 0 \)
WHERE \( F(X) \) IS A CONTINUOUS REAL FUNCTION OF A SINGLE REAL
VARIABLE \( X \). THE METHOD USED IS A COMBINATION OF BISECTION
AND THE SECANT RULE.

NORMAL INPUT CONSISTS OF A CONTINUOUS FUNCTION \( F \) AND AN
INTERVAL \( (B, C) \) SUCH THAT \( F(B) \cdot F(C) \leq 0 \). EACH ITERATION
FINDS NEW VALUES OF \( B \) AND \( C \) SUCH THAT THE INTERVAL \( (B, C) \) IS
SHRUNK AND \( F(B) \cdot F(C) \leq 0 \). THE STOPPING CRITERION IS
\[
\text{ABS}(B-C) \cdot 2 \cdot 0 \cdot (\text{RELERR} + \text{ABS}(B) + \text{ABSERR})
\]
WHERE \( \text{RELERR} \) = RELATIVE ERROR AND \( \text{ABSERR} \) = ABSOLUTE ERROR ARE
INPUT QUANTITIES. SET THE FLAG \( IFLAG \), POSITIVE TO INITIALIZE
THE COMPUTATION. \( A \), \( B \), \( C \) AND \( IFLAG \) ARE USED FOR BOTH INPUT AND
OUTPUT, THEY MUST BE VARIABLES IN THE CALLING PROGRAM.

IF \( 0 \) IS A POSSIBLE ROOT, ONE SHOULD NOT CHOOSE \( \text{ABSERR} = 0.0 \).

THE OUTPUT VALUE OF \( B \) IS THE BETTER APPROXIMATION TO A ROOT
AS \( B \) AND \( C \) ARE ALWAYS REDEFINED SO THAT \( \text{ABS}(F(B)) \cdot \text{ABS}(F(C)) \).

TO SOLVE THE EQUATION, ROOT MUST EVALUATE \( F(X) \) REPEATEDLY. THIS
IS DONE IN THE CALLING PROGRAM. WHEN AN EVALUATION OF \( F \) IS
NEEDED AT \( T \), ROOT RETURNS WITH \( IFLAG \) NEGATIVE. EVALUATE \( FT = F(T) \)
AND CALL ROOT AGAIN. DO NOT ALTER IFLAG.

WHEN THE COMPUTATION IS COMPLETE, ROOT RETURNS TO THE CALLING
PROGRAM WITH IFLAG POSITIVE.

\( IFLAG = 1 \) IF \( F(B) \cdot F(C) \lt 0 \) AND THE STOPPING CRITERION IS MET.

\( IFLAG = 2 \) IF A VALUE \( B \) IS FOUND SUCH THAT THE COMPUTED VALUE
\( F(B) \) IS EXACTLY ZERO. THE INTERVAL \( (B, C) \) MAY NOT
SATISFY THE STOPPING CRITERION.

\( IFLAG = 3 \) IF \( \text{ABS}(F(B)) \) EXCEEDS THE INPUT VALUES \( \text{ABS}(F(B)), \text{ABS}(F(C)) \). IN THIS CASE IT IS LIKELY THAT \( B \) IS CLOSE
TO A POLE OF \( F \).

\( IFLAG = 4 \) IF NO ODD ORDER ROOT WAS FOUND IN THE INTERVAL. A
LOCAL MINIMUM MAY HAVE BEEN OBTAINED.

\( IFLAG = 5 \) IF TOO MANY FUNCTION EVALUATIONS WERE MADE.
(AS PROGRAMMED, 500 ARE ALLOWED.)

THIS CODE IS A MODIFICATION OF THE CODE ZEROIN WHICH IS COMPLETELY
EXPLAINED AND DOCUMENTED IN THE TEXT, NUMERICAL COMPUTING, AN
INTRODUCTION BY L. F. SHAMPINE AND R. C. ALLEN.

SUBROUTINE ROOT(T, FT, B, C, RELERR, ABSERR, IFLAG)

IMPLICIT REAL*8 (A-H, O-Z)

COMMON/MLDRT/AACBSAE, FAFC, FCEX, IC, KOUNT, RELERR

THE ONLY MACHINE DEPENDENT CONSTANT IS BASED ON THE MACHINE UNIT
* ROUNDOFF ERROR \( U \) WHICH IS THE SMALLEST POSITIVE NUMBER SUCH THAT *
C* 1.0+U .GT. 1.0 . U MUST BE CALCULATED AND INSERTED IN THE
C* FOLLOWING DATA STATEMENT BEFORE USING ROOT. THE ROUTINE MACHIN
C* CALCULATES U .
C***************************************************************************
DATA U /2.2E-16/
C***************************************************************************
C
C IF(FLAG.LT.0.0) GO TO 100
RE=MAX(RELERR,U)
AE=MAX(ABSERR,0.0D0)
IC=0
ACBS=ABS(B-C)
A=C
T=A
IFLAG=1
RETURN
100 IFLAG=ABS(FLAG)
GO TO (200,300,400),IFLAG
200 FA=FT
T=B
IFLAG=2
RETURN
300 FB=FT
FC=FA
KOUNT=2
FX=MAX(ABS(FB),ABS(FC))
GO TO 1
400 FB=FT
IF(FB.EQ.0.0) GO TO 9
KOUNT=KOUNT+1
IF(SIG(1.0D0,FB).NE.SIGN(1.0D0,FC))GO TO 1
C=A
FC=FA
1 IF(ABS(FC).GE.ABS(FB))GO TO 2
C C INTERCHANGE B AND C SO THAT ABS(F(B)).LE.ABS(F(C)).
C
C
A=B
FA=FB
B=C
FB=FC
C=A
FC=FA
2 CMB=0.5*(C-B)
ACMB=ABS(CMB)
TOL=RE+ABS(B)+AE
C
C TEST STOPPING CRITERION AND FUNCTION COUNT.
C
C
IF(ACMB.LE.TOL)GO TO 8
IF(KOUNT.GE.500)GO TO 12
C
C CALCULATE NEW ITERATE IMPLICITLY AS B+P/Q
C WHERE WE ARRANGE P.GE.0. THE IMPLICIT
C FORM IS USED TO PREVENT OVERFLOW.
C
P=(B-A)*FB
Q=FA-FB
IF(P.GE.0.0)GO TO 3
F=P
Q\rightarrow Q

UPDATE A, CHECK IF REDUCTION IN THE SIZE OF BRACKETING INTERVAL IS SATISFACTORY. IF NOT, BISECT UNTIL IT IS.

3 A=B
 FA=FB
 IC=IC+1
 IF(IC.LT.4)GO TO 4
 IF(8.0*ACMB.GE.ACBS)GO TO 6
 IC=0
 ACBS=ACMB

TEST FOR TOO SMALL A CHANGE.

4 IF(P.GT.ABS(Q)*TOL)GO TO 5

INCREMENT BY TOLERANCE.

B=B+SIGN(TOL,CMB)
 GO TO 7

ROOT OUGHT TO BE BETWEEN B AND (C+B)/2.

5 IF(P.GE.CMB*Q)GO TO 6

USE SECANT RULE.

B=B+P/Q
 GO TO 7

USE BISECTION.

6 B=0.5*(C+B)

HAVE COMPLETED COMPUTATION FOR NEW ITERATE B.

7 T=B
 IFLAG=3
 RETURN

FINISHED. SET IFLAG.

8 IF(SIGN(1.0D0,FB).EQ.SIGN(1.0D0,FC))GO TO 11
 IF(ABS(FB).GT.FX)GO TO 10
 IFLAG=1
 RETURN

9 IFLAG=2
 RETURN

10 IFLAG=3
 RETURN

11 IFLAG=4
 RETURN

12 IFLAG=5
 RETURN

END
NOTE THAT STEP1 HAS BEEN MODIFIED FROM THE ORIGINAL VERSION TO INCLUDE
THE NUMBER OF EQUATIONS TO BE INTEGRATED (NEQN) IN CALL STATEMENTS TO
PROCESS SUBROUTINES.

SUBROUTINE STEP1 IS NORMALLY USED INDIRECTLY THROUGH SUBROUTINE
ODE. BECAUSE ODE SUFFICES FOR MOST PROBLEMS AND IS MUCH EASIER
TO USE, USING IT SHOULD BE CONSIDERED BEFORE USING STEP1 ALONE.

SUBROUTINE STEP1 INTEGRATES A SYSTEM OF NEQN FIRST ORDER ORDINARY
DIFFERENTIAL EQUATIONS ONE STEP, NORMALLY FROM X TO X+H, USING A
MODIFIED DIVIDED DIFFERENCE FORM OF THE ADAMS PECE FORMULAS. LOCAL
EXTRAPOLATION IS USED TO IMPROVE ABSOLUTE STABILITY AND ACCURACY.
THE CODE ADJUSTS ITS ORDER AND STEP SIZE TO CONTROL THE LOCAL ERROR
PER UNIT STEP IN A GENERALIZED SENSE. SPECIAL DEVICES ARE INCLUDED
TO CONTROL ROUNDOFF ERROR AND TO DETECT WHEN THE USER IS REQUESTING
TOO MUCH ACCURACY.

THIS CODE IS COMPLETELY EXPLAINED AND DOCUMENTED IN THE TEXT,
COMPUTER SOLUTION OF ORDINARY DIFFERENTIAL EQUATIONS, THE INITIAL
VALUE PROBLEM BY L. F. SHAMPINE AND M. K. GORDON.

FURTHER DETAILS ON USE OF THIS CODE ARE AVAILABLE IN *SOLVING
ORDINARY DIFFERENTIAL EQUATIONS WITH ODE, STEP, AND INTRP*,

THE PARAMETERS REPRESENT —
F — SUBROUTINE TO EVALUATE DERIVATIVES
NEQN — NUMBER OF EQUATIONS TO BE INTEGRATED
Y(*) — SOLUTION VECTOR AT X
X — INDEPENDENT VARIABLE
H — APPROPRIATE STEP SIZE FOR NEXT STEP. NORMALLY DETERMINED BY
CODE
EPS — LOCAL ERROR TOLERANCE
WT(*) — VECTOR OF WEIGHTS FOR ERROR CRITERION
START — LOGICAL VARIABLE SET .TRUE. FOR FIRST STEP, .FALSE.
OTHERWISE
HOLD — STEP SIZE USED FOR LAST SUCCESSFUL STEP
K — APPROPRIATE ORDER FOR NEXT STEP (DETERMINED BY CODE)
KOLD — ORDER USED FOR LAST SUCCESSFUL STEP
CRASH — LOGICAL VARIABLE SET .TRUE. WHEN NO STEP CAN BE TAKEN,
.FALSE. OTHERWISE.
YP(*) — DERIVATIVE OF SOLUTION VECTOR AT X AFTER SUCCESSFUL
STEP

THE ARRAYS PHI, PSI ARE REQUIRED FOR THE INTERPOLATION SUBROUTINE
INTRP*. THE ARRAY P IS INTERNAL TO THE CODE. THE REMAINING NINE
VARIABLES AND ARRAYS ARE INCLUDED IN THE CALL LIST ONLY TO ELIMINATE
LOCAL RETENTION OF VARIABLES BETWEEN CALLS.

INPUT TO STEPI
FIRST CALL —

THE USER MUST PROVIDE STORAGE IN HIS CALLING PROGRAM FOR ALL ARRAYS
IN THE CALL LIST, NAMELY

DIMENSION Y(NEQN),WT(NEQN),PHI(NEQN,16),P(NEQN),YP(NEQN),PSI(12),
1 ALPHA(12),BETA(12),SIG(13),V(12),W(12),G(13)

— — — **NOTE**

THE USER MUST ALSO DECLARE START, CRASH, PHASE1 AND NORND
LOGICAL VARIABLES AND F AN EXTERNAL SUBROUTINE, SUPPLY THE
SUBROUTINE F(X,Y,YP) TO EVALUATE
DY(I)/DX = YP(I) = F(X,Y(1),Y(2),....Y(NEQN))
AND INITIALIZE ONLY THE FOLLOWING PARAMETERS.
NEQN — NUMBER OF EQUATIONS TO BE INTEGRATED
Y(*) — VECTOR OF INITIAL VALUES OF DEPENDENT VARIABLES
X — INITIAL VALUE OF THE INDEPENDENT VARIABLE
H — NOMINAL STEP SIZE INDICATING DIRECTION OF INTEGRATION
AND MAXIMUM SIZE OF STEP. MUST BE VARIABLE
EPS — LOCAL ERROR TOLERANCE PER STEP. MUST BE VARIABLE
WT(*) — VECTOR OF NON-ZERO WEIGHTS FOR ERROR CRITERION
START — .TRUE.

STEP1 REQUIRES THAT THE L2 NORM OF THE VECTOR WITH COMPONENTS
LOCAL ERROR(L)/WT(L) BE LESS THAN EPS FOR A SUCCESSFUL STEP. THE
ARRAY WT ALLOWS THE USER TO SPECIFY AN ERROR TEST APPROPRIATE
FOR HIS PROBLEM. FOR EXAMPLE,
WT(L) = 1.0 SPECIFIES ABSOLUTE ERROR,
= ABS(Y(L)) ERROR RELATIVE TO THE MOST RECENT VALUE OF THE
L-TH COMPONENT OF THE SOLUTION,
= ABS(YP(L)) ERROR RELATIVE TO THE MOST RECENT VALUE OF
THE L-TH COMPONENT OF THE DERIVATIVE,

\[ C_{\text{MAX}}(\text{WT}(L),|\text{ABS}(Y(L))|) \] ERROR RELATIVE TO THE LARGEST

MAGNITUDE OF L-TH COMPONENT OBTAINED SO FAR,

\[ \text{ABS}(Y(L))\times\text{RELERR}/\text{EPS} + \text{ABSERR}/\text{EPS} \] SPECIFIES A MIXED

RELATIVE-ABSOLUTE TEST WHERE RELERR IS RELATIVE

ERROR, ABSERR IS ABSOLUTE ERROR AND EPS =

\[ \text{MAX}(\text{RELERR},\text{ABSERR}) \] .

SUBSEQUENT CALLS —

SUBROUTINE STEP1 IS DESIGNED SO THAT ALL INFORMATION NEEDED TO

CONTINUE THE INTEGRATION, INCLUDING THE STEP SIZE H AND THE ORDER

K, IS RETURNED WITH EACH STEP. WITH THE EXCEPTION OF THE STEP

SIZE, THE ERROR TOLERANCE, AND THE WEIGHTS, NONE OF THE PARAMETERS

SHOULD BE ALTERED. THE ARRAY WT MUST BE UPDATED AFTER EACH STEP

TO MAINTAIN RELATIVE ERROR TESTS LIKE THOSE ABOVE. NORMALLY THE

INTEGRATION IS CONTINUED JUST BEYOND THE DESIRED ENDPOINT AND THE

SOLUTION INTERPOLATED THERE WITH SUBROUTINE INTRP. IF IT IS

IMPOSSIBLE TO INTEGRATE BEYOND THE ENDPOINT, THE STEP SIZE MAY BE

REDUCED TO HIT THE ENDPOINT SINCE THE CODE WILL NOT TAKE A STEP

LARGER THAN THE H INPUT. CHANGING THE DIRECTION OF INTEGRATION,

I.E., THE SIGN OF H, REQUIRES THE USER SET START = .TRUE. BEFORE

CALLING STEP1 AGAIN. THIS IS THE ONLY SITUATION IN WHICH START

SHOULD BE ALTERED.

OUTPUT FROM STEP1

SUCCESSFUL STEP —

THE SUBROUTINE RETURNS AFTER EACH SUCCESSFUL STEP WITH START AND

CRASH SET .FALSE. . X REPRESENTS THE INDEPENDENT VARIABLE

ADVANCED ONE STEP OF LENGTH HOLD FROM ITS VALUE ON INPUT AND Y

THE SOLUTION VECTOR AT THE NEW VALUE OF X . ALL OTHER PARAMETERS

REPRESENT INFORMATION CORRESPONDING TO THE NEW X NEEDED TO

CONTINUE THE INTEGRATION.

UNSUCCESSFUL STEP —

WHEN THE ERROR TOLERANCE IS TOO SMALL FOR THE MACHINE PRECISION,

THE SUBROUTINE RETURNS WITHOUT TAKING A STEP AND CRASH = .TRUE. .

AN APPROPRIATE STEP SIZE AND ERROR TOLERANCE FOR CONTINUING ARE

ESTIMATED AND ALL OTHER INFORMATION IS RESTORED AS UPON INPUT

BEFORE RETURNING. TO CONTINUE WITH THE LARGER TOLERANCE, THE USER

JUST CALLS THE CODE AGAIN. A RESTART IS NEITHER REQUIRED NOR

DESIRABLE.

SUBROUTINE STEP1(F,NEQN,Y,X,H,EPS,WT,START,

1 HOLD,K,KOLD,CRASH,PHI,P,YP,PSI,

1 ALPHA,BETA,SIG,V,W,G,PHASE1,NS,NORND)

C* THE ONLY MACHINE DEPENDENT CONSTANTS ARE BASED ON THE MACHINE UNIT *
181 C* ROUND OFF ERROR U WHICH IS THE SMALLEST POSITIVE NUMBER SUCH THAT *
182 C* 1.0+U .GT. 1.0 . THE USER MUST CALCULATE U AND INSERT *
183 C* TWOU=2.0*U AND FOURU=4.0*U IN THE DATA STATEMENT BEFORE CALLING *
184 C* THE CODE. THE ROUTINE MACHIN CALCULATES U . *
185 DATA TWOU,FOURU/4.4E-16,8.8E-16/
186
187 C*********************************************************************
188 DATA TWO/2.0,4.0,8.0,16.0,32.0,64.0,128.0,256.0,512.0,1024.0, *
189 1 2846.0,4096.0,8192.0/
190 DATA GSTR/0.500,0.0833,0.0417,0.0264,0.0188,0.0143,0.0114,0.00936, *
191 1 0.00789,0.00679,0.00592,0.00524,0.00468/
192
193 C*** BEGIN BLOCK 0 ***
194 C CHECK IF STEP SIZE OR ERROR TOLERANCE IS TOO SMALL FOR MACHINE *
195 C PRECISION. IF FIRST STEP, INITIALIZE PHI ARRAY AND ESTIMATE A *
196 C STARTING STEP SIZE. *
197 C***
198 C IF STEP SIZE IS TOO SMALL, DETERMINE AN ACCEPTABLE ONE *
199 C
200 CRASH = .TRUE.
201 IF(ABS(H) .GE. FOURU*ABS(X)) GO TO 5
202 H = SIGN(FOURU*ABS(X),H)
203 RETURN
204 5 P5EPS = 0.5*EPS
205 IF ERROR TOLERANCE IS TOO SMALL, INCREASE IT TO AN ACCEPTABLE VALUE *
206 C
207 ROUND = 0.0
208 DO 10 L = 1,NEQN
209 10 ROUND = ROUND + (Y(L)/WT(L))**2
210 ROUND = TWOU*SQRT(ROUND)
211 IF(P5EPS .GE. ROUND) GO TO 15
212 EPS = 2.0*ROUND*(1.0 + FOURU)
213 RETURN
214 15 CRASH = .FALSE.
215 G(1) = 1.0
216 G(2) = 0.5
217 SIG(1) = 1.0
218 IF(.NOT.START) GO TO 99
219
220 C INITIALIZE. COMPUTE APPROPRIATE STEP SIZE FOR FIRST STEP *
221 CALL F(NEQN,X,Y,YP)
222 SUM = 0.0
223 DO 20 L = 1,NEQN
224 PHI(L,1) = YP(L)
225 PHI(L,2) = 0.0
226 20 SUM = SUM + (YP(L)/WT(L))**2
227 SUM = SQRT(SUM)
228 ABISH = ABS(H)
229 IF(EPS .LT. 16.0*SUM+H+H) ABISH = 0.25*SQRT(EPS/SUM)
230 H = SIGN(MAX(ABISH,FOURU*ABS(X)),H)
231 HOLD = 0.0
232 K = 1
233 KOLD = 0
234 START = .FALSE.
235 PHASE1 = .TRUE.
236 NORND = .TRUE.
IF(PSEPS .GT. 100.0*ROUND) GO TO 99
NORND = .FALSE.
DO 25 L = 1, NEQN
25 PHI(L,15) = 0.0
99 IFAIL = 0
C *** END BLOCK 0 ***
C *** BEGIN BLOCK 1 ***
C COMPUTE COEFFICIENTS OF FORMULAS FOR THIS STEP. AVOID COMPUTING
C THOSE QUANTITIES NOT CHANGED WHEN STEP SIZE IS NOT CHANGED.
C ***

100 KP1 = K+1
KP2 = K+2
KM1 = K-1
KM2 = K-2
C NS IS THE NUMBER OF STEPS TAKEN WITH SIZE H, INCLUDING THE CURRENT
C ONE. WHEN K.LT.NS, NO COEFFICIENTS CHANGE
C
IF(H .NE. HOLD) NS = 0
IF (NS.LE.KOLD) NS = NS+1
NSP1 = NS+1
IF (K .LT. NS) GO TO 110
DO 105 I = NSP1, K
IM1 = I-1
TEMP2 = PSI(IM1)
PSI(IM1) = TEMP1
BETA(I) = BETA(IM1)*PSI(IM1)/TEMP2
TEMP1 = TEMP2 + H
ALPHA(I) = H/TEMP1
REALI = I
SIG(I+1) = REALI*ALPHA(I)*SIG(I)
105 SIG(I+1) = REALI*ALPHA(I)*SIG(I)
110 PSI(K) = TEMP1
C COMPUTE COEFFICIENTS G(*)
C INITIALIZ V(*) AND SET W(*)
C
IF(NS .GT. 1) GO TO 120
DO 115 IQ = 1, K
TEMP3 = IQ*(IQ+1)
V(IQ) = 1.0/TEMP3
115 W(IQ) = V(IQ)
GO TO 140
C IF ORDER WAS RAISED, UPDATE DIAGONAL PART OF V(*)
C
120 IF(K .LE. KOLD) GO TO 130
TEMP4 = K*KP1
V(K) = 1.0/TEMP4
NSM2 = NS-2
IF(NSM2 LT. 1) GO TO 130
DO 125 J = 1,NSM2
I = K-J
125 V(I) = V(I) - ALPHA(J+1)*V(I+1)
C C UPDATE V(*) AND SET W(*)
C
130 LIMIT1 = KP1 - NS
TEMP5 = ALPHA(NS)
DO 135 IQ = 1,LIMIT1
V(IQ) = V(IQ) - TEMP5*V(IQ+1)
135 W(IQ) = V(IQ)
G(NSP1) = W(1)
C
COMPUTE THE G(*) IN THE WORK VECTOR W(*)
C
140 NSP2 = NS + 2
IF(KP1 .LT. NSP2) GO TO 199
DO 150 I = NSP2,KP1
LIMIT2 = KP2 - I
TEMP6 = ALPHA(I-1)
DO 145 IQ = 1,LIMIT2
W(IQ) = W(IQ) - TEMP6*W(IQ+1)
145 G(I) = W(1)
199 CONTINUE
C *** END BLOCK 1 ***
C
*** BEGIN BLOCK 2 ***
C PREDICT A SOLUTION P(*), EVALUATE DERIVATIVES USING PREDICTED
C SOLUTION, ESTIMATE LOCAL ERROR AT ORDER K AND ERRORS AT ORDERS K,
C K-1, K-2 AS IF CONSTANT STEP SIZE WERE USED.
C ***
C
CHANGE PHI TO PHI STAR
C
IF(K .LT. NSP1) GO TO 215
DO 210 I = NSP1,K
TEMP1 = BETA(I)
DO 205 L = 1,NEQN
PHI(L,I) = TEMP1*PHI(L,I)
205 CONTINUE
210 CONTINUE
C PREDICT SOLUTION AND DIFFERENCES
C
215 DO 220 L = 1,NEQN
PHI(L,KP2) = PHI(L,KP1)
PHI(L,KP1) = 0.0
220 P(L) = 0.0
DO 230 J = 1,K
I = KP1 - J
IP1 = I+1
TEMP2 = G(I)
DO 225 L = 1,NEQN
P(L) = P(L) + TEMP2*PHI(L,I)
225 PHI(L,I) = PHI(L,I) + PHI(L,IP1)
230 CONTINUE
IF(NORND) GO TO 240
DO 235 L = 1,NEQN
TAU = H*P(L) - PHI(L,15)
P(L) = Y(L) + TAU
PHI(L,16) = (P(L) - Y(L)) - TAU
GO TO 250
DO 245 L = 1,NEQN
P(L) = Y(L) + H*P(L)
250 XOLD = X
X = X + H
ABSH = ABS(H)
CALL F(NEQN,X,P,Y)

C ESTIMATE ERRORS AT ORDERS K,K-1,K-2
C
ERKM2 = 0.0
ERKM1 = 0.0
ERK = 0.0
DO 265 L = 1,NEQN
TEMP3 = 1.0/WT(L)
TEMP4 = YP(L) - PHI(L,1)
IF(KM2) = 265,260,255
ERKM2 = ERKM2 + ((PHI(L,KM1)+TEMP4)*TEMP3)**2
ERKM1 = ERKM1 + ((PHI(L,K)+TEMP4)*TEMP3)**2
ERK = ERK + (TEMP4*TEMP3)**2
IF(KM2) = 260,275,270
ERKM2 = ABSH*SIG(KM1)*GSTR(KM2)*SQRT(ERKM2)
ERKM1 = ABSH*SIG(K)*GSTR(KM1)*SQRT(ERKM1)
ERK = TEMP5*SIG(KP1)*GSTR(K)
KNEW = K

C TEST IF ORDER SHOULD BE LOWERED
C
IF(KM2) = 299,290,285
IF(MAX(ERKM1,ERKM2) .LE. ERK) KNEW = KM1
GO TO 299
IF(ERK .LE. 0.5*ERK) KNEW = KM1

C TEST IF STEP SUCCESSFUL
C
IF(ERR .LE. EPS) GO TO 400
*** END BLOCK 2 ***
C
C THE STEP IS UNSUCCESSFUL. RESTORE X, PHI(*,*), PSI(*).
C IF THIRD CONSECUTIVE FAILURE, SET ORDER TO ONE. IF STEP FAILS MORE
C THAN THREE TIMES, CONSIDER AN OPTIMAL STEP SIZE. DOUBLE ERROR
C TOLERANCE AND RETURN IF ESTIMATED STEP SIZE IS TOO SMALL FOR MACHINE
C PRECISION.
C
C RESTORE X, PHI(*,*), PSI(*)
C
PHASEI = .FALSE.
X = XOLD
DO 310 I = 1,K
TEMP1 = 1.0/BETA(I)
IP1 = I+1
DO 305 L = 1,NEQN
PHI(L,1) = TEMP1*(PHI(L,1) - PHI(L,IP1))
CONTINUE

IF(K .LT. 2) GO TO 320
DO 315 I = 2, K
315 PSI(I-1) = PSI(I) - H
C
C ON THIRD FAILURE, SET ORDER TO ONE. THEREAFTER, USE OPTIMAL STEP SIZE
C
320 IFAIL = IFAIL + 1
325 TEMP2 = 0.5
330 IF(IFAIL = 3) 335, 330, 325
335 IF(P5EPS .LT. 0.25*ERK) TEMP2 = SQRT(P5EPS/ERK)
330 KNEW = 1
335 H = TEMP2*H
340 K = KNEW
345 IF(ABS(H) .GE. FOURU*ABS(X)) GO TO 340
350 EPS = EPS + EPS
360 RETURN
340 GO TO 100
C *** END BLOCK 3 ***
343 C *** BEGIN BLOCK 4 ***
345 C THE STEP IS SUCCESSFUL. CORRECT THE PREDICTED SOLUTION, EVALUATE THE DERIVATIVES USING THE CORRECTED SOLUTION AND UPDATE THE DIFFERENCES. DETERMINE BEST ORDER AND STEP SIZE FOR NEXT STEP.
348 C ***
349 400 KOLD = K
405 HOLD = H
C
C CORRECT AND EVALUATE
C
405 TEMP1 = H*G(KP1)
410 IF(NORND) GO TO 410
DO 405 L = 1, NEQN
405 RHO = TEMP1*(YP(L) - PHI(L,1)) - PHI(L,16)
410 Y(L) = P(L) + RHO
415 PHI(L,15) = ( Y(L) - P(L) ) - RHO
420 GO TO 420
410 DO 415 L = 1, NEQN
415 Y(L) = P(L) + TEMP1*(YP(L) - PHI(L,1))
420 CALL F(NEQN,X,Y,YP)
C
C UPDATE DIFFERENCES FOR NEXT STEP
C
420 DO 425 L = 1, NEQN
425 PHI(L,KP1) = YP(L) - PHI(L,1)
425 PHI(L,KP2) = PHI(L,KP1) - PHI(L,KP2)
430 DO 435 I = 1, K
435 PHI(L,I) = PHI(L,I) + PHI(L,KP1)
435 CONTINUE
C
C ESTIMATE ERROR AT ORDER K+1 UNLESS:
C IN FIRST PHASE WHEN ALWAYS RAISE ORDER,
C ALREADY DECIDED TO LOWER ORDER,
C STEP SIZE NOT CONSTANT SO ESTIMATE UNRELIABLE
C
C ERKP1 = 0.0
IF(KNEW .EQ. KM1 .OR. K .EQ. 12) PHASE1 = .FALSE.

IF(PHASE1) GO TO 450
IF(KNEW .EQ. KM1) GO TO 455
IF(KP1 .GT. NS) GO TO 460
DO 440 L = 1, NEQN

440  ERKP1 = ERKP1 + (PHI(L,KP2)/WT(L))**2

ERKP1 = ABSH*GSTR(KP1)*SQRT(ERKP1)

C USING ESTIMATED ERROR AT ORDER K+1, DETERMINE APPROPRIATE ORDER

FOR NEXT STEP

IF(K .GT. 1) GO TO 445
IF(ERKP1 .GE. 0.5*ERK) GO TO 460
GO TO 450

445  IF(ERKM1 .LE. MIN(ERK,ERKP1)) GO TO 455
IF(ERKP1 .GE. ERK .OR. K .EQ. 12) GO TO 460

HERE ERKP1 .LT. ERK .LT. AMAX1(ERKM1,ERKM2) ELSE ORDER WOULD HAVE BEEN LOWERED IN BLOCK 2. THUS ORDER IS TO BE RAISED

RAISE ORDER

450  K = KP1
ERK = ERKP1
GO TO 460

LOWER ORDER

455  K = KM1
ERK = ERKM1

WITH NEW ORDER DETERMINE APPROPRIATE STEP SIZE FOR NEXT STEP

460  HNEW = H + H
IF(PHASE1) GO TO 465
IF(P5EPS .GE. ERK*TW(K+1)) GO TO 465
HNEW = H
IF(P5EPS .GE. ERK) GO TO 465
TEMP2 = K+1
R = (P5EPS/ERK)**(1.0/TEMP2)
HNEW = ABSH*MAX(0.5D0,MIN(0.9D0,R))
HNEW = SIGN(MAX(HNEW,FOURU*ABS(X)),H)

465  H = HNEW
RETURN

END
SUBROUTINE STEPCA

PURPOSE
STEPCA IS CALLED BY THE MAIN PROGRAM. IT ADVANCES THE
SOLUTION ONE TIME STEP BY CALLING THE INTEGRATION
SUBROUTINE ODERT AND THE NECESSARY SUPPORT ROUTINES.

USAGE
CALL STEPCA(F, NEQN, DY, DT, TOUT, IFLAG, WORK, IWORK,
& G, IPROC)

DEFINITION OF PARAMETERS
F       DUMMY ARGUMENT FOR THE PROCESS IN PROGRESS (INTAKE,
        COMPRESSION, COMBUSTION, OR EXHAUST)
NEQN    NUMBER OF EQUATIONS BEING INTEGRATED
DY      ARRAY OF SYSTEM VARIABLES
DT      CRANK ANGLE
TOUT    INTEGRATION LIMIT
IFLAG   INTEGRATION FLAG RETURNED FROM ODERT
WORK    ARRAY USED BY ODERT FOR INTEGRATION
IWORK   " " " " " " " " " "
G       DUMMY ARGUMENT FOR THE FUNCTION TO BE USED FOR THE
        ROOT REQUIRED BY ODERT (GIDEL OR GCM)
IPROC   INDEX FOR PROCESS IN PROGRESS

REMARKS
STEPCA CARRIES OUT THOSE STEPS THAT ARE REPEATED FOR
EACH PROCESS IN THE MAIN PROGRAM. IT'S MAIN PURPOSE
IS TO CALL THOSE SUBROUTINES THAT ARE CALLED ONCE FOR
EACH TIME STEP. ORIGINALLY THESE STEPS WERE REPEATED
IN EACH PROCESS SECTION OF THE MAIN PROGRAM.

SUBROUTINES AND FUNCTIONS REQUIRED
ODERT   VLVSUM   EXSUM
PFDIF   DATA     RESULT

WRITTEN BY R.M. FRANK

SUBROUTINE STEPCA(F, NEQN, DY, DT, TOUT, IFLAG, WORK, IWORK,
& G, IPROC)
REAL*8 DT, DY(NEQN), TOUT, RELERR, ABSERR, REROOT, AEROOT,
& WORK(2410), IWORK(5)
REAL MASSIN(270), MASSEX(270),
& FCYLIN(270), FCYLEX(270),
& HCYLIN(270), HCYLEX(270)
EXTERNAL F,G
LOGICAL SPTEMP, EXSUB
COMMON/SPTEMP/ SPTEMP
COMMON/EXSUB/ EXSUB
COMMON/ITERAS/ ITERAS, ISTEDY
COMMON/ARRAY/ MASSIN, MASSEX, FCYLIN, FCYLEX, HCYLIN, HCYLEX
COMMON/B/ CPM(2), HM(2), MMM(2), GM(2), RHOM(2)
COMMON/TIMES/ TIVO, TEVC, TIVC, TINJ, TIGN, TEVO
COMMON/VLVOPN/ IINOPN, IEXOPN
COMMON/INTERR/ RELERR, ABSERR, REROOT, AEROOT

CALL INTEGRATION SUBROUTINE

IFAIL = 0

CALL ODERT (F, NEQN, DY, DT, TOUT, RELERR, ABSERR,
 & IFLAG, WORK, IWORK, G, REROOT, AEROOT)

IF (IFLAG.EQ.2) GOTO 20
IF (IFLAG.EQ.8) RETURN

INTEGRATION DID NOT REACH TOUT
IFAIL = IFAIL + 1
IF (IFAIL.GT.5) GOTO 50
GOTO 10

CONTINUE FOR NORMAL INTEGRATION:

CALL VLVSUM FOR TOTAL INTAKE FLOW FROM OTHER CYLINDERS AND ARRAY UPDATE
IT = NINT(DT)
IIN = IT - NINT(TIVO)
CALL VLVSUM (IIN, 1, DY(24), HM(1), MASSIN, FCYLIN, HCYLIN, & IINOPN)

CALL VLVSUM FOR TOTAL EXHAUST FLOW FROM OTHER CYLINDERS AND ARRAY UPDATE
IEX = IT - NINT(TEVO)
IF (IPROC .NE. 4) IEX = IEX + 20
CALL VLVSUM (IEX, 2, DY(28), HM(2), MASSEX, FCYLEX, HCYLEX, & IEXOPN)

FOR CASE WITH SUBDIVIDED EXHAUST MANIFOLD SUM FLOWS FROM OTHER EXHAUST MANIFOLD SECTIONS AND UPDATE STORAGE ARRAYS
IF (EXSUB) CALL EXSUM(IEX)

WRITE TO OUTPUT FILE
CALL RESULT(IPROC, NEON, DT, DY, IFLAG)

WRITE TO PLOT FILES
CALL DATA(NEQN, DT, DY)

IF(SPTEP .OR. ITERAS.LE.ISTEDY) RETURN

FIND TRANSIENT TEMPERATURES
CALL PFDIF
RETURN

***************
INTEGRATION FAILURE - DUMP CONTENTS OF DY TO OUTPUT FILE

WRITE(7,1000)
WRITE(6,1000)
WRITE(7,1100) (I, DY(I), I=1,NEQN)
WRITE(6,1100) (I, DY(I), I=1,NEQN)

1000 FORMAT (///, ' INTEGRATION FAILURE - CHECK PROGRAM', & ' AND INPUT ',//, & ' DUMP OF ARRAY DY FOLLOWS')
1100 FORMAT (///,1X,3('DY(','I2,') = ',E11.4,5X))
SUBROUTINE THERMO

PURPOSE
'THERMO' is called by the 4 process routines and by 'MAIN' and returns with the required thermodynamic properties in each case. It calls 'HPROP', and then calculates from the returned data any additional properties of interest. THERMO also converts all values to units that are consistent with those used in the rest of the program.

USAGE
CALL THERMO ( TEMP, P, FR, ENTHLP, CSUBP, CSUBT, CSUBF, RHO, DRHODT, DRHOOP, DRHODF, GAMMA, MW, ADUMY, BDUMY, CDUMY)

DESCRIPTION OF PARAMETERS

PARAMETER INPUT OUTPUT DESCRIPTION

TEMP YES NO TEMPERATURE (K)
P YES NO PRESSURE (PA)
FR YES NO BURNED FUEL FRACTION
ENTHLP NO YES ENTHALPY (J/KG)
CSUBP NO YES PARTIAL OF H WITH RESPECT TO T (J/KG-DEG K)
CSUBT NO YES PARTIAL OF H WITH RESPECT TO P (M**3/KG)
CSUBF NO YES PARTIAL OF H WITH RESPECT TO PHI (J/KG)
RHO NO YES DENSITY (KG/M**3)
DRHODT NO YES PARTIAL OF RHO WITH RESPECT TO T (KG/M**3-DEG K)
DRHOOP NO YES PARTIAL OF RHO WITH RESPECT TO P (KG/M**3-PA)
DRHODF NO YES PARTIAL OF RHO WITH RESPECT TO PHI (KG/M**3)
MW NO YES MOLECULAR WEIGHT
GAMMA NO YES RATIO OF SPECIFIC HEATS
ADUMY NO YES SEE ASSIGNMENT STATEMENTS BELOW (J/KG-DEG K)
BDUMY NO YES SEE ASSIGNMENT STATEMENTS BELOW (J/KG)
CDUMY NO YES SEE ASSIGNMENT STATEMENTS BELOW (J/KG)

REMARKS
UNITS CHANGED TO SI

SUBROUTINES AND FUNCTION SUBPROGRAMS REQUIRED
HPROP

METHOD
SEE PURPOSE, ABOVE

WRITTEN BY D. N. ASSANIS
EDITED BY D. N. ASSANIS
SUBROUTINE THERMO (TEMP, P, FR, ENTHLP, CSUBP, CSUBT, CSUBF, RHO, 
& DRHODT, DRHODP, DRHODF, GAMMA, MW, ADUMY, BDUMY, CDUMY)

REAL MW

PRES = P/1.01325E+5
CALL HPROP (TEMP, PRES, FR, ENTHLP, CSUBP, CSUBT, CSUBF, 
& RHO, DRHODT, DRHODP, DRHODF)

CONVERT TO UNITS NEEDED IN MAIN PROGRAM:

ENTHLP = ENTHLP * 4.184E+6
CSUBP = CSUBP * 4.184E+3
CSUBT = CSUBT / 1000.
CSUBF = CSUBF * 4.184E+6

RHO = RHO * 1000.
DRHODT = DRHODT * 1000.
DRHODP = DRHODP / 101.325
DRHODF = DRHODF * 1000.

CALCULATE GAS CONSTANT, MOLECULAR WEIGHT, DUMMY VARIABLES:

ADUMY = CSUBP + ( DRHODT/DRHODP )*( 1./RHO - CSUBT )
BDUMY = (1. - RHO * CSUBT) / DRHODP
CDUMY = CSUBF + ( DRHODF/DRHODP )*( 1./RHO - CSUBT )

R = P / (RHO * TEMP)
MW = 8.3145E+3/R
GAMMA = CSUBP/( CSUBP - R )

RETURN
END
SUBROUTINE TRANSP

PURPOSE
CALCULATES DYNAMIC VISCOSITY AND THERMAL CONDUCTIVITY
OF BURNED PRODUCTS

USAGE
CALL TRANSP (TEMP, FR, GAMMA, CP, DYNVIS, THRCON)

DESCRIPTION OF PARAMETERS
PARAMETER INPUT OUTPUT DESCRIPTION

TEMP YES NO TEMPERATURE (K)
FR YES NO AVERAGE FUEL FRACTION
GAMMA YES NO RATIO OF SPECIFIC HEATS
CP YES NO HEAT CAPACITY AT CONSTANT PRESSURE
DYNVIS NO YES OF COMBUSTION PRODUCTS (J/KG-DEG K)
THRCON NO YES THERMAL CONDUCTIVITY OF

REMARKS
UNITS CHANGED TO S.I.

METHOD
SEE S. H. MANSOURI AND J. B. HEYWOOD, "CORRELATIONS FOR THE
VISCOSITY AND PRANDTL NUMBER OF HYDROCARBON-AIR COMBUSTION
PRODUCTS", COMBUSTION SCIENCE AND TECHNOLOGY, 1980, VOL. 23,
PP. 251-256.

WRITTEN BY S. H. MANSOURI AND D. N. ASSANIS
EDITED BY D. N. ASSANIS

SUBROUTINE TRANSP (TEMP, FR, GAMMA, CP, DYNVIS, THRCON)
COMMON/FUEL/ FUELP, CX, HY, DEL, PSI, AFRAST, QLOWER, HFORM

PHI = FR * AFRAST / (1.-FR)
DYNVIS = 3.3E-7 * (TEMP**.7)/((-1.0 + 0.027 * PHI)
PRNDTL = 0.05 + 4.2 * (GAMMA - 1.0) - 6.7 * (GAMMA - 1.0)
& (GAMMA - 1.0)
THRCON = DYNVIS * CP/PRNDTL
IF ((PHI .LE. 1.0) .OR. (TEMP .LE. 1500.)) RETURN
PRNDTL = PRNDTL/(1.0 + 1.5E-8 + PHI + PHI * TEMP * TEMP)
THRCON = DYNVIS * CP/PRNDTL
RETURN
END
SUBROUTINE VACDEX

PURPOSE
CALCULATES EFFECTIVE AREA OF EXHAUST VALVE BY INTERPOLATING VALUES GIVEN IN TABLE.

WRITTEN BY D.N. ASSANIS

REMARKS
AREA CONVERSION FROM SQUARE FEET TO SQUARE METERS DONE IN MAIN PROGRAM.

EPHAO AND EPHAC ARE VALVE TIMINGS FOR DETERMINING THE VALVE PHASE FOR THE EXHAUST PORT HEAT TRANSFER. SEE SUBROUTINE MANPAR FOR INPUT.

SUBROUTINE VACDEX (X, AREA, CD)
INTEGER IROW, EROW
DIMENSION TABLEX (120,2)
COMMON/TABSIZ/ IROW, EROW
COMMON/TABLEX/ TABLEX
COMMON/PHASE/ EPHAO, EPHAC, IPHASE, REJ

IPHASE = 4
IF (X .LT. TABLEX(1,1) .OR. X .GT. TABLEX(EROW,1)) RETURN

DO 10 I2 = 2, EROW
   IF (TABLEX(I2,1) .GT. X) GO TO 20
10 CONTINUE
I2 = EROW
I1 = I2 - 1

Q = (X - TABLEX(I1,1)) / (TABLEX(I2,1) - TABLEX(I1,1))
AREA = TABLEX(I1,2) + Q * (TABLEX(I2,2) - TABLEX(I1,2))

DISCHARGE COEFFICIENT:
CD = 1.0

DETERMINE PHASE OF VALVE OPENING FOR PORT HEAT TRANSFER
IF (EPHAO.EQ.0.) RETURN
IF (X.LT.EPHAO) THEN
   IPHASE = 1
ELSEIF (X.LT.EPHAC) THEN
   IPHASE = 2
ELSE
   IPHASE = 3
ENDIF

RETURN
END
SUBROUTINE VACDIN

PURPOSE

CALCULATES EFFECTIVE AREA OF INTAKE VALVE

BY INTERPOLATING TABLE INPUT.

REMARKS

AREA CONVERSION FROM SQ FEET TO SQ METERS DONE IN MAIN PROGRAM

WRITTEN BY D.N. ASSANIS

SUBROUTINE VACDIN (X, AREA, CD)

INTEGER IROW, EROW

COMMON/TABSIZ/ IROW, EROW
COMMON/TABLIN/ TABLIN(126,2)

IF (X .LT. TABLIN(1,1) .OR. X .GT. TABLIN(IROW,1)) RETURN

DO 10 I2 = 2, IROW

IF (TABLIN(I2,1) .GT. X) GO TO 20

CONTINUE

I2 = IROW

I1 = I2 - 1

Q = (X - TABLIN(I1,1)) / (TABLIN(I2,1) - TABLIN(I1,1))

AREA = TABLIN(I1,2) + Q * (TABLIN(I2,2) - TABLIN(I1,2))

CD = 1.0

RETURN

END
SUBROUTINE VLVSUM

PURPOSE
CALCULATES TOTAL AND FUEL MASS FLOW RATES AND ENTHALPY FLUXES
THAT ENTER INTAKE AND EXHAUST MANIFOLDS AND STORES CURRENT FLOW
RATES FOR FUTURE CALLS TO VLVSUM.

METHOD
THE ALGORITHM SUMS THE MASS FLOW RATES (TOTAL AND FUEL), AND
THE ENTHALPY FLUXES CONTRIBUTED BY EACH CYLINDER EVERY INSTANT.
THE MASS FLOW RATE AND ENTHALPY FLUX PROFILES OF THE MASTER
CYLINDER DURING THE INTAKE AND THE EXHAUST PROCESS ARE GENERATED
BY THE SIMULATION AND STORED IN APPROPRIATE ARRAYS (MASSIN, MASSEX,
FCYLIN, FCYLEX, HCYLIN, HCYLEX).
THE PROFILES OF THE OTHER CYLINDERS ARE ASSUMED TO BE ECHOES OF
THE MASTER CYLINDER PROFILES, SHIFTED BY THE APPROPRIATE
PHASE ANGLES.

DEFINITION OF PARAMETERS

PARAMETER INPUT OUTPUT DESCRIPTION
IVLV YES NO TRANSFORMED TIME BASED ON DEGREES
AFTER VALVE OPENS
IMAN YES NO FLAG FOR MANIFOLD (INTAKE=1, EXHAUST=2)
FFR YES NO FUEL FRACTION OF MANIFOLD
HM YES NO SPECIFIC ENTHALPY OF MANIFOLD
AMASS YES YES STORAGE ARRAY FOR VALVE MASS FLOW
AF YES YES SAME FOR FUEL FRACTION OF FLOW
AH YES YES SAME FOR SPEC ENTHALPY OF FLOW
IOPN YES NO INTERVAL THAT VALVE IS OPEN (IN DEGREES)

REMARKS
VLVSUM REPLACES MASUM IN PREVIOUS VERSIONS OF THE PROGRAM
WRITTEN BY D. N. ASSANIS & R.M. FRANK

SUBROUTINE VLVSUM (IVLV, IMAN, FFR, HM, AMASS, AF, AH,
& IOPN)

PARAMETER (NCV=6)
DIMENSION AMASS(1), AF(1), AH(1)
LOGICAL EXSUB
COMMON/ICYL/ ICYL
COMMON/MA/ YPM(2), YPH(2), YPF(2)
COMMON/SUMIT/ ENGM(2), FMDOT(2), HMDOT(2)

UPDATE STORAGE ARRAYS FOR FUTURE CALLS TO VLVSUM
IF (IVLV .GT. IOPN) GOTO 5
AMASS(IVLV) = YPM(IMAN)
AF(IVLV) = YPF(IMAN)
AH(IVLV) = YPH(IMAN)

SUM UP MASS FLOW RATES, ENTHALPY FLUXES, FUEL FRACTION
C FLUXES CONTRIBUTED BY EACH CYLINDER AT THAT INSTANT:

5 IFLAG = -1
6 IF (IMAN.EQ.1) IFLAG = 1

10 ENG = 0.
FDOT = 0.
HDOT = 0.
IDEL = 720/ICYL
IT = IVLV

C SUM FLOWS FOR OTHER CYLINDERS
DO 20 I = 1, ICYL
C FIND SHIFTED TIME FOR EACH CYLINDER
IT = IT - IDEL
IF (IT .LE. 0) IT = IT + 720
C IF THE CRANK ANGLE LIES BEFORE OR AFTER THE CYLINDER'S
C PROCESS PERIOD, MOVE ON TO NEXT CYLINDER.
C
IF (IT.GT.IOPN) GO TO 20

C FLOW = AMASS(IT)
C INITIALIZE F AND H FOR FLOW INTO MANIFOLD
F = FLOW * (AF(IT) - FFR)
H = FLOW * AH(IT)

C CHECK FOR FLOW OUT OF MANIFOLD
IF(FLOW+IFLAG.LT.0) GOTO 15

C INITIALIZE F FOR FLOW OUT OF MANIFOLD
F = 0.
C FLOW OUT OF AVERAGE EXHAUST MANIFOLD CONTROL VOL BASED
C ON RECORDED ENTHALPY, FLOW OUT OF INTAKE MANIFOLD BASED
C ON CURRENT h.
C
H = FLOW * HM

C SUM FLOW
15 FDOT = FDOT + F
HDOT = HDOT + H
ENG = ENG + FLOW

20 CONTINUE
C
ENGM(IMAN) = ENG
FMDOT(IMAN) = FDOT
HMDOT(IMAN) = HDOT
C
RETURN
END
APPENDIX D

SAMPLE INPUT FILES FOR COMPUTER CODE
General Input File

Filename: INPUT.DAT
Unit Number: 8
Read in: Main Program

$INPUT

TITLE = 'SAMPLE RUN FOR THESIS PRINTOUT'
ANNAND = .FALSE.
SPDEL = .FALSE.
SPTEMP = .FALSE.
TRANS = .TRUE.
POWER = .TRUE.
EXSUB = .TRUE.

TINJ = 346.
TIGN = 349.
ERPM = 1600.
FMIN = 0.197
PATM = 0.954
TATM = 302.8
PINLET = 0.954
RTEMP(1) = 302.6
ELNG = 0.7, 0.638, 0.5
EDIAM = 0.1, 0.1, 0.13
HI = 9, 305., 305., 0., 1250.
RCORR(1) = 0.017
TSC = 1.08

CBETA = 1.5
CFACTR = 1.05
CRAD = 2.0
CONHT = 0.045
ECONHT = 0.035, 0.0, 0.0
ETWALL = 305., 750., 600.
DP = 3600., 1000., 1000., 1000., 5000.
EMKT(2) = 0.5
PRSS = 0., 0., 0., 1.3E5, 1.05E5

PSTART = 3.0
TSTART = 900.
PHISTA = 0.4
MKESTA = 2.5E-3
TKESTA = 1.1E-3
Y0(2) = 300.
Y0(3) = 230.E3
Y0(4) = 0.0
Y0(6) = 900.
Y0(7) = 290.E3
Y0(8) = 0.0336
Y0(9) = 60.

TPRINT = 10.
TSCREEN = 10.
AREROT = 1.E-4
CIINTG = 4.E-4
CCINTG = 4.E-4
CBINTG = 4.E-4
CEINTG = 4.E-4
MAXITS = 10
MINITS = 2

DELQ1 = 0.01
DELM1 = 0.002
DELQ2 = 0.005
DELM2 = 0.002

$END
**EXHAUST MANIFOLD SET-UP FOR NORMAL HEAT TRANSFER**

```plaintext
$EXHMAN
NUMEX = 3
NPIPE = 1,1,6
NSHFT = 0,0,120

NEVLY = 2
EPHAO = 544
EPHAC = 683

EPLNG(1)= 0.1
EPLNG(2)= 0.05
EXLNG(1)= 0.075
EXLNG(2)= 0.3
EXLNG(3)= 0.127
EXLNG(4)= 0.5

BRAD(2) = 0.05
BRAD(3) = 0.05
BRAD(4) = 0.3

NEXLAY(1) = 3
NEXLAY(2) = 3
NEXLAY(4) = 1

EXCOND(1,1)= 54.4
EXCOND(1,2)= 54.4
EXCOND(1,3)= 54.4
EXCOND(2,1)= 54.4
EXCOND(2,2)= 54.4
EXCOND(2,3)= 54.4
EXCOND(4,1)= 54.4
EXTHIK(1,1)= 0.002
EXTHIK(1,2)= 0.004
EXTHIK(1,3)= 0.002
EXTHIK(2,1)= 0.002
EXTHIK(2,2)= 0.004
EXTHIK(2,3)= 0.002
EXTHIK(4,1)= 0.005

XTCOOL(1) = 365.
XTCOOL(2) = 322.
XTCOOL(4) = 322.

EXTW(1) = 400.
EXTW(2) = 700.
EXTW(3) = 700.
EXTW(4) = 600.

DIFU(1,1)= 1.57E-5
DIFU(1,2)= 1.57E-5
DIFU(1,3)= 1.57E-5
DIFU(2,1)= 1.57E-5
DIFU(2,2)= 1.57E-5
DIFU(2,3)= 1.57E-5
DIFU(4,1)= 1.57E-5

$END
```
Cylinder Liner Geometry and Thermal Properties Input File

Filename: CHEAT.DAT
Unit Number: 18
Read in: CYLPAR

$CHEAT
NCC = 1
CDIAM(1) = 0.11
NCLA(1) = 1
CHCOOL(1) = 0.
CTCOOL(1) = 370.
CTHIK(1,1) = 7.E-3
CCOND(1,1) = 54.4
$END

Piston and Head Geometry and Thermal Properties Input File

Filename: PHEAT.DAT
Unit Number: 11
Read in: PARFIN

$PHEAT
NPC = 2
FACT = 0.6
NPLA(1) = 3
PTCOOL(1) = 440.
PHCOOL(1) = 0.
INNODE(1,1) = 15
INNODE(1,2) = 5
INNODE(1,3) = 5
PTHIK(1,1) = 1.5E-3
PTHIK(1,2) = 7.E-3
PTHIK(1,3) = 3.E-3
PCOND(1,1) = 54.4
PCOND(1,2) = 54.4
PCOND(1,3) = 54.4
PDIFU(1,1) = 1.57E-5
PDIFU(1,2) = 1.57E-5
PDIFU(1,3) = 1.57E-5
NPLA(2) = 3
PTCOOL(2) = 440.
PHCOOL(2) = 0.
INNODE(2,1) = 15
INNODE(2,2) = 5
INNODE(2,3) = 5
PTHIK(2,1) = 1.5E-3
PTHIK(2,2) = 7.E-3
PTHIK(2,3) = 3.E-3
PCOND(2,1) = 54.4
PCOND(2,2) = 54.4
PCOND(2,3) = 54.4
PDIFU(2,1) = 1.57E-5
PDIFU(2,2) = 1.57E-5
PDIFU(2,3) = 1.57E-5
$END
Intake Valve Flow Area Profile

Filename: TABLIN.DAT
Unit Number: 75
Read in: MAIN PROGRAM

Crankangle, Valve Open Area (ft**2)
-11., .0000000
-6., .0007920
-1., .0022120
4., .0028850
9., .0059450
14., .0077140
19., .0091950
24., .0108060
29., .0124590
34., .0140720
39., .0152660
44., .0163960
49., .0176110
54., .0173840
59., .0177170
64., .0178440
69., .0179970
74., .0180890
79., .0181600
84., .0182110
89., .0182410
94., .0182500
99., .0182380
104., .0182040
109., .0181480
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Exhaust Valve Flow Area Profile

Filename: TABLEX.DAT
Unit Number: 76
Read in: MAIN PROGRAM

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Unit Number: 3
Read in: Main Program

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Turbocharger Turbine Map Input File

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Unit Number: 4  
Read in: Main Program

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Power Turbine Map Input File

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Unit Number: 5
Read in: Main Program

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37.5, 0.812, 1.405
40., 0.807, 1.47
45., 0.77, 1.62
47.5, 0.741, 1.715
50., 0.703, 1.83
52.5, 0.667, 1.92
45.
  0., 0., 1.
20., 0.40, 1.19
25., 0.52, 1.258
30., 0.641, 1.325
35., 0.75, 1.41
37.5, 0.778, 1.47
40., 0.793, 1.54
45., 0.801, 1.72
47.5, 0.789, 1.804
50., 0.753, 1.91
52.5, 0.72, 1.995
45.
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20., 0.35, 1.23
25., 0.471, 1.391
30., 0.595, 1.373
35., 0.7, 1.471
37.5, 0.733, 1.54
40., 0.755, 1.62
45., 0.79, 1.813
47.5, 0.795, 1.9
50., 0.79, 2.005
52.5, 0.762, 2.095
APPENDIX E

SAMPLE OUTPUT FILE FOR COMPUTER CODE
M.I.T. TURBOCOMPRESSOR DIESEL ENGINE CYCLE SIMULATION

>>>>> INPUT DATA

>>>>> OPERATING MODE

SAMPLE RUN FOR THESIS PRINTOUT

PREDICTED IGNITION DELAY OPTION
PREDICTED QUASI-STEADY WALL TEMPERATURES
FLAME RADIATION MODEL
EXHAUST MANIFOLD SUB-CONTROL VOLUME MODEL

>>>>> OPERATING CONDITIONS

FUEL USED IS DIESEL #2
ENGINE SPEED - ERPM = 1600.0 RPM
INJECTION TIMING - TINJ = 346.0 DEG CA
FUEL INJECTED CYL/CYCLE = 0.1970 G
TOTAL FUELING RATE = 2.0828 LB/MIN

COMPRESSOR INLET PRESSURE -PINLET- = 0.9540 ATM
COMPRESSOR INLET TEMPERATURE -RTEMP(1)- = 302.60 K

ATMOSPHERIC PRESSURE -PATM- = 0.9540 ATM
ATMOSPHERIC TEMPERATURE -TATM- = 302.80 K

>>>>> ENGINE DESIGN PARAMETERS

NUMBER OF CYLINDERS - ICYL = 6

CYLINDER BORE = 13.970 CM
CRANKSHAFT STROKE = 15.240 CM
CONNECTING ROD LENGTH - CONRL = 30.480 CM

COMPRESSION RATIO = 14.500

DISPLACED VOLUME = 2335.972 CC
CLEARANCE VOLUME - CLVTDC = 173.035 CC
ENGINE DISPLACEMENT = 14.016 LT

FRICITION CONSTANT 2 - CFR2 = 7.000
FRICITION CONSTANT 3 - CFR3 = 0.000

INTAKE VALVE OPENS - TIVO = -11.0 DEG CA
INTAKE VALVE CLOSES - TIVC = 212.0 DEG CA
EXHAUST VALVE OPENS - TEVO = 505.0 DEG CA
EXHAUST VALVE CLOSES - TEVC = 736.0 DEG CA
INTAKE MANIFOLD DIMENSIONS

LENGTH (M) 0.700000
DIAMETER (M) 0.100000
CROSS-SECTIONAL AREA (M²) 7.8539820E-03
INTERNAL SURFACE AREA (M²) 0.2199115
VOLUME (LT) 5.497787

EXHAUST MANIFOLD DESIGN PARAMETERS

NUMBER OF SECTIONS TO EXHAUST (EXCLUDING TURBINE CONNECTING PIPE) - 3
NPIPE - 1 1 6
NSHFT - 0 0 120

PORT

NUMBER OF EXHAUST VALVES PER CYLINDER - 2

DIAMETER OF INDIVIDUAL SECTIONS OF PORT = 0.036 M
LENGTH OF INDIVIDUAL SECTIONS OF PORT = 0.100, 0.050 M

DIAMETER OF COMMON SECTION OF PORT = 0.046 M
LENGTH OF COMMON SECTION OF PORT = 0.075

WALL COMPOSITION OF PORT

LAYER | 1 | 2 | 3
MATERIAL | STEEL | STEEL | STEEL
THICKNESS | 0.0020 | 0.0040 | 0.0020
THERM. COND. | 54.4000 | 54.4000 | 54.4000
THERM. DIFFUSIVITY | 0.16E-04 | 0.16E-04 | 0.16E-04
NO. OF NODES | 15 | 5 | 5
***** EXHAUST VOLUME 2 *****

INSIDE DIAMETER = 0.046 M
LENGTH = 0.300 M

WALL COMPOSITION OF EXHAUST MANIFOLD

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<tr>
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<td>STEEL</td>
<td>STEEL</td>
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<tr>
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<td>0.0040</td>
<td>0.0020</td>
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<td>NO. OF NODES</td>
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AVG. BEND RADIUS = 0.050 M
TOTAL BEND ANGLE = 90. DEGREES

***** EXHAUST VOLUME 3 *****

INSIDE DIAMETER = 0.060 M
LENGTH = 0.127 M

WALL COMPOSITION OF EXHAUST MANIFOLD

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<td>THERM. COND.</td>
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AVG. BEND RADIUS = 0.050 M
TOTAL BEND ANGLE = 45. DEGREES

***** TURBINE CONNECTING PIPE *****

INSIDE DIAMETER OF CONNECTING PIPE = 0.127 M
LENGTH OF CONNECTING PIPE = 0.500 M

WALL COMPOSITION FOR CONNECTING PIPE

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<td>THERM. COND.</td>
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AVG. BEND RADIUS OF CONN. PIPE = 0.300 M
TOTAL BEND ANGLE OF CONN. PIPE = 90. DEGREES

***** BOUNDARY CONDITIONS *****

<table>
<thead>
<tr>
<th>AMBIENT TEMPERATURE (K)</th>
<th>OUTSIDE HEAT TRANSFER COEFFICIENT (W/M**2/K)</th>
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</thead>
<tbody>
<tr>
<td>PORT (CONT VOL 1) - 365.</td>
<td>6540.0</td>
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<tr>
<td>CONTROL VOLUME 2 - 322.</td>
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</tr>
<tr>
<td>CONTROL VOLUME 3 - 322.</td>
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<tr>
<td>TURBINE CONN PIPE - 322.</td>
<td>56.8</td>
</tr>
</tbody>
</table>
INITIAL GUESSES FOR INSIDE WALL TEMPERATURE (K)

PORT (VOL 1) - 400.
CONTROL VOLUME 2 - 700.
CONTROL VOLUME 3 - 700.
TURBINE CONNECTING PIPE - 600.

****** COMPUTED CONSTANTS FOR EXHAUST MANIFOLD ******

<table>
<thead>
<tr>
<th></th>
<th>PORT</th>
<th>C.V. 2</th>
<th>C.V. 3</th>
<th>CONN PIPE</th>
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</thead>
<tbody>
<tr>
<td>INSIDE CROSS-SECTIONAL AREA (M**2)</td>
<td>0.189E-02</td>
<td>0.166E-02</td>
<td>0.283E-02</td>
<td>0.127E-01</td>
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<tr>
<td>INSIDE SURFACE AREA (M**2)</td>
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<td>0.434E-01</td>
<td>0.239E-01</td>
<td>0.199E+00</td>
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<tr>
<td>VOLUME (M**3)</td>
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<td>0.499E-03</td>
<td>0.359E-03</td>
<td>0.633E-02</td>
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<tr>
<td>EFFECTIVE CONDUCTIVITY - UA (W/K)</td>
<td>116.02</td>
<td>3.29</td>
<td>1.71</td>
<td>12.16</td>
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<tr>
<td>CORRECTION FACTOR FOR ENTRANCE EFFECTS</td>
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<td>1.26</td>
<td>1.14</td>
<td>2.05</td>
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<tr>
<td>FRACTION OF PIPE BENT</td>
<td>0.000</td>
<td>0.262</td>
<td>0.309</td>
<td>0.942</td>
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<tr>
<td>CBP - RATIO OF PIPE TO BEND DIAM</td>
<td>0.000</td>
<td>0.460</td>
<td>0.600</td>
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TOTAL VOLUME OF EXHAUST MANIFOLD = 0.501E-02 M**3
TOTAL INSIDE SURFACE AREA = 0.451E+00 M**2

>>> TURBOMACHINERY DATA

<p>| | | | | |</p>
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<td>T/C INERTIA (KG-M**2)</td>
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<tr>
<td>T/C DAMPING (KG-M**2/S)</td>
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</table>
| P.TURBINE TRANSMISSION EFFIC. | 0.9000000 | P.TURBINE GEAR RATIO 1.7000001E-02 :
| COMPRESSOR EFF. SCALE FACTOR - CSC | 1.000000 |
| COMP MASS FLOW SCALE FACTOR - CMSC | 1.000000 |
| TURBINE EFF. SCALE FACTOR - TSC  | 1.000000 |
| TURB MASS FLOW SCALE FACTOR - TMSC | 1.000000 |
| POWER TURB EFF. SCALE FACTOR- PTSC | 1.000000 |
| PWR TURB MASS FLOW SCL FACTOR- PTMSC | 1.000000 |
SYSTEM PRESSURE LOSSES

SPECIFIED PRESSURE DROPS (PA):
COMPRESSOR EXIT - INTERCOOLER INLET - DP(1)  3600.000
POWER TURBINE EXIT - ATMOSPHERIC - DP(5)   5000.000

SPECIFIED LOSS FACTORS:
INTERCOOLER INLET - INTAKE - EMKT(1)  1.000000
EXHAUST MANIFOLD - TURBINE INLET - EMKT(2) 0.5000000
TURBINE CONNECTING PIPE EMKT(3)   1.000000

HEAT TRANSFER AND TURBULENCE PARAMETERS

HEAT TRANSFER CONSTANT(CHAMBER) - CONHT = 0.0450
HEAT TRANSFER CONSTANT(INT MAN) - ECONHT(1) = 0.0350
HEAT TRANSFER EXponent - EPHHT = 0.8000

INITIAL PISTON TEMPERATURE - TPSTON = 600.00 K
INITIAL CYL HEAD TEMPERATURE - THEAD = 600.00 K
INITIAL CYL WALL TEMPERATURE - TCW = 450.00 K
INT. MANIFOLD WALL TEMPERATURE - ETWALL(1) = 305.00 K
TURBULENT DISSIPATION CONSTANT - CBETA = 1.5000
### WALL CONDUCTION MODELS

#### PISTON WALL STRUCTURE

<table>
<thead>
<tr>
<th>Layer</th>
<th>1</th>
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<tbody>
<tr>
<td>Thickness (m)</td>
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<td>0.00700</td>
<td>0.00300</td>
</tr>
<tr>
<td>Thermal Conductivity (W/mK)</td>
<td>54.400</td>
<td>54.400</td>
<td>54.400</td>
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<tr>
<td>Thermal Diffusivity (m²/Sec)</td>
<td>0.157E-04</td>
<td>0.157E-04</td>
<td>0.157E-04</td>
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<tr>
<td>Number of Nodes</td>
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<tr>
<td>Nodes Within Skin Depth</td>
<td>9</td>
<td></td>
<td></td>
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<tr>
<td>Courant Number</td>
<td>7.019</td>
<td>1872.612</td>
<td>343.949</td>
</tr>
<tr>
<td>Outside Wall Temperature (K)</td>
<td>440.0</td>
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<tr>
<td>Overall Conductivity (W/mK)</td>
<td>4730.4</td>
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#### CYL. HEAD WALL STRUCTURE

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<td>Thickness (m)</td>
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<td>0.00300</td>
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<tr>
<td>Thermal Conductivity (W/mK)</td>
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<td>54.400</td>
<td>54.400</td>
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<tr>
<td>Thermal Diffusivity (m²/Sec)</td>
<td>0.157E-04</td>
<td>0.157E-04</td>
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<tr>
<td>Number of Nodes</td>
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<td>Nodes Within Skin Depth</td>
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<td>Courant Number</td>
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<tr>
<td>Outside Wall Temperature (K)</td>
<td>440.0</td>
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<tr>
<td>Overall Conductivity (W/mK)</td>
<td>4730.4</td>
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<tr>
<td>Layer</td>
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<tr>
<td>Inside Diameter (m)</td>
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<tr>
<td>Thickness (m)</td>
<td>0.00700</td>
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<tr>
<td>Thermal Conductivity (W/m/K)</td>
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<td>Outside Wall Temperature (K)</td>
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<tr>
<td>DELQ1</td>
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<tr>
<td>DELQ2</td>
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## OUTPUT DATA

### ENGINE CRANK-ANGLE BY CRANK-ANGLE RESULTS

### START OF INTAKE PROCESS

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<th>P (ATM)</th>
<th>TEMP (K)</th>
<th>MIN (G)</th>
<th>MEX (G)</th>
<th>PHI (-)</th>
<th>PIM (ATM)</th>
<th>TIM (K)</th>
<th>PEM (ATM)</th>
<th>TEM (K)</th>
<th>IFG (-)</th>
<th>SPEED (K RPM)</th>
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<td>PHI (-)</td>
<td>PIM (ATM)</td>
<td>TIM (K)</td>
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>>>>>> START OF EXHAUST PROCESS
<table>
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<tr>
<th>Component</th>
<th>Pressure (ATM)</th>
<th>Pressure (IN-HG)</th>
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<tr>
<td>Compressor Inlet</td>
<td>0.9540000</td>
<td>28.54489</td>
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<td>Compressor Discharge</td>
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<td>Intake Manifold</td>
<td>2.452003</td>
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<td>Exhaust Manifold</td>
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<td>Turbine Inlet</td>
<td>2.826206</td>
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<td>Turbine Discharge</td>
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<td>Power Turbine Inlet</td>
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<th>Component</th>
<th>Temperature (K)</th>
<th>Temperature (F)</th>
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<td>Compressor Inlet</td>
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<td>Compressor Discharge</td>
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<td>Engine Exhaust</td>
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<td>Turbine Inlet</td>
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TURBINE EXHAUST 718.2698 833.1977
POWER TURBINE INLET 711.3785 820.7933
P. TURBINE EXHAUST 666.5048 740.6206

>>>>> INTERCOOLER DATA
INTERCOOLER EFFECTIVENESS 0.9289910
COOLANT INLET TEMPERATURE 305.0000
INTERCOOLER "A+U" (W/K) 1250.000

>>>>> TURBOCHARGER DATA

COMPRESSOR TURBINE P.TURBINE

MAP FLOW (LB/MIN) 62.84204 21.46060 38.70645
MAP SPEED (KRPM) 61.23917 65.15362 30.98639
PRESSURE RATIOS: 2.613800 1.913834 1.459602
EFFICIENCIES: 0.8119020 0.7089130 0.7790174

>>>>> FINAL STEADY-STATE CYLINDER TEMPERATURES

PISTON TEMPERATURE (K) = 494.
HEAD TEMPERATURE (K) = 494.
LINER TEMPERATURE (K) = 407.
### Exhaust Manifold and Turbine Connecting Pipe Results

#### Heat Transfer Data

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<tr>
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<th>C.V. 1</th>
<th>C.V. 2</th>
<th>C.V. 3</th>
<th>Conn Pipe</th>
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</thead>
<tbody>
<tr>
<td>Ave. Heat Flux Over Cycle (W/M²)</td>
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<td>0.302E+05</td>
<td>0.301E+05</td>
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<td>Ave. Heat Transfer Rate for Cont Vol (W)</td>
<td>0.410E+04</td>
<td>0.131E+04</td>
<td>0.721E+03</td>
<td>0.387E+04</td>
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<td>Ave. Heat Trans Coef for Cycle (W/M²/K)</td>
<td>0.353E+03</td>
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<td>Normalized Heat Transfer Rate for Sect</td>
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<td>0.107E-02</td>
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*Normalized Heat Transfer Rate = (Total Heat Transfer Rate/Engine Fuel Flow)*

#### Inside Wall Temperature Data (K)

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<th>780.</th>
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#### Gas Temperature Data (K)

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<th>Mass Averaged Temperature</th>
<th>Time Averaged Temperature</th>
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<td></td>
<td>838.</td>
<td>768.</td>
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<tr>
<td></td>
<td>829.</td>
<td>791.</td>
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<td>816.</td>
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#### Mass Averaged Temperature Drop (K)

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<th>Exhaust Valve/Port</th>
<th>Between Control Volumes 1 &amp; 2</th>
<th>Between Control Volumes 2 &amp; 3</th>
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<td>38.</td>
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#### Total Temperature Drop

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#### Temperature Drop Between Turbines

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### Diesel Engine Performance Results

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<td>Volumetric Efficiency; (%)</td>
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<td>224.2</td>
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<td>Based on: Intake / ATM</td>
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<tr>
<td>Pumping Mean Eff. Pressure (ATM, PSI) : PMEP</td>
<td>-0.76</td>
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<tr>
<td>Gross Ind. Mean Eff. Pressure (ATM, PSI) : IMEP</td>
<td>16.97</td>
<td>249.41</td>
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<td>Friction Mean Eff. Pressure (ATM, PSI) : FMEP</td>
<td>1.48</td>
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<td>Brake Mean Eff. Pressure (ATM, PSI) : BMEP</td>
<td>14.73</td>
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<td>Gross Indicated S.F.C. (G/KW/HR, LB/HP/HR) : ISFC</td>
<td>176.551</td>
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<td>Brake S.F.C. (G/KW/HR, LB/HP/HR) : BSFC</td>
<td>203.413</td>
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<td>Gross Indicated Thermal Efficiency; (%)</td>
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<tr>
<td>Net Indicated Thermal Efficiency; (%)</td>
<td>45.4</td>
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### Cylinder Brake Thermal Efficiency

\[
\text{CYLINDER BRAKE THERMAL EFFICIENCY; (\%)} \quad 41.2
\]

\[
\frac{\text{(CYL. HEAT TRANSFER PER CYCLE)}}{\text{(MASS OF FUEL TIMES LHV)}} \quad (\%) \quad 10.9
\]

\[
\text{MEAN EXHAUST TEMPERATURE; (K)} \quad 875.8
\]

### Total System Performance Results

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<td>Power Turbine Work Per Cycle (KJ)</td>
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<td>Total Heat Input Per Cycle (KJ)</td>
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<td>Diesel Brake Power (KW, HP): ENBHP</td>
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<td>Power Turb. Brake Power (KW, HP): PTBHP</td>
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OVERALL BRAKE S.F.C.
(G/KW/HR, LB/HP/HR) : BSFC
188.304 0.310

OVERALL BRAKE THERMAL
EFFICIENCY; (%) 44.6

>>> CYLINDER MASS SUMMARY

MASS IN CYLINDER AT TIVO = 0.29087 G
MASS IN CYLINDER AT TIVC = 6.07788 G
MASS OF AIR INDUCTED = 5.80077 G
MASS OF FUEL INJECTED = 0.19700 G

>>> CYLINDER HEAT & WORK TRANSFERS

HEATI = -0.071721 KJ (TIVO - 180)
WORKI = 0.540347 KJ

HEATC = 0.061034 KJ (180 - TIGN)
WORKC = -2.669233 KJ

HEATCE = 0.888516 KJ (180 - 540)
WORKCE = 4.016975 KJ

HEATE = 0.101893 KJ (540 - TIVO)
WORKE = -0.721317 KJ
>>> CYLINDER ENTHALPY BALANCE

INITIAL ENTHALPY /CYL/ CYCLE = -0.16725 KJ
TOTAL ENTHALPY IN /CYL/ CYCLE = 1.72643 KJ
TOTAL ENTHALPY OUT/CYL/ CYCLE = -2.98772 KJ
TOTAL HEAT LOSS / CYL / CYCLE = 0.91869 KJ
IND. WORK OUTPUT / CYL /CYCLE = 3.83601 KJ
BRAKE WORK OUTPUT /CYL /CYCLE = 3.48650 KJ
RESIDUAL ENTHALPY /CYL/ CYCLE = -0.16731 KJ
NET ENERGY GAIN / CYL/CYCLE = 0.04048 KJ
(ENERGY GAIN)/(H IVC) = 2.59624 %
(ENERGY GAIN)/(MFUEL*LHV) = 0.47887 %

>>> CYLINDER MASS BALANCE

TOTAL INTAKE VALVE MASS FLOW (G/CYCLE) = 5.0008
FUEL FLOW (G/INJ) = 0.1970
TOTAL EXHAUST VALVE MASS FLOW (G/CYCLE) = 5.9986

NET CHANGE IN MASS (G/CYCLE) = -0.0009
NORMALIZED CHANGE IN MASS (% OF EXHAUST MASS FLOW) = -0.0146 %
>>> OVERALL MASS BALANCE

TOTAL COMPRESSOR MASS FLOW (G/CYCLE) = 34.7043
TOTAL FUEL FLOW (G/CYCLE) = 1.1820
TOTAL TURBINE MASS FLOW (G/CYCLE) = 35.8890

NET CHANGE IN MASS (G/CYCLE) = -0.0027

NORMALIZED CHANGE IN MASS (% OF TURBINE MASS FLOW) = -0.0074 %

*** THE CHANGE IN MASS INCLUDES BOTH THE ERROR INTRODUCED BY THE INTEGRATION
AND THE CHANGE IN MASS IN THE ENGINE IF THE PROGRAM HAS NOT CONVERGED EXACTLY

>>> COMBUSTION SUMMARY

IGNITION DELAY PERIOD = 4.413 DEG CA
IGNITION TIMING = 350.413 DEG CA
BURN DURATION = 125.000 DEG CA
WEIGHTING FACTOR = 0.145
PREMIXED CONSTANT 1 = 2.095
PREMIXED CONSTANT 2 = 5000.000
DIFFUSION CONSTANT 1 = 23.196
DIFFUSION CONSTANT 2 = 2.304
CFACCTR = 1.050

AVERAGE NUMBER OF CALLS TO DIFEQ/CYCLE THIS RUN: 1437
AVERAGE TIMES THRU LOOP FOR EXHAUST LOOP PER CALL TO DIFEQ: 1.464793