HEAT TRANSFER AND PERFORMANCE CALCULATIONS

IN A ROTARY ENGINE

by

RAYMOND ANTHONY STANTEN

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Signature redacted

Signature of Author: ______________________________

Department of Mechanical Engineering
August 1987

Signature redacted

Certified by: ______________________________

Prof. John B. Heywood, Thesis Supervisor
Dept. of Mechanical Engineering

Signature redacted

Accepted by: ______________________________

Prof. Ain A. Sonin
Chairman, Dept. of Mechanical Engineering

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ABSTRACT

The Wankel stratified-charge combustion engine is a promising future powerplant for general aviation. The advantages of the engine include low weight, high specific power density and multifuel capability without a loss in performance.

Additional gains in performance may be possible if ceramics are used to insulate the rotary engine housing, thereby increasing engine wall temperatures and reducing heat transfer losses. To evaluate the feasibility of this insulation and its effect on performance, it was necessary to determine the local temperature and heat transfer profiles at specific locations within the engine. With this boundary condition information, a finite element calculation could be used to determine the temperature fluctuation and penetration depth. This data can then be used to calculate thermal stresses which are important in ceramic applications due to the high thermal resistivity and brittleness of ceramics relative to steel.

This paper describes the work which was done to determine the effect of different wall temperatures on engine performance, and toward determining the temperature fluctuations and penetration depth within the housing. A rotary engine cycle simulation program, along with data phasing routines, were used to determine the heat transfer boundary conditions. The combustion model used in the program was verified with actual rotary engine data and was found to agree well. The program was then altered to allow for a user-defined housing surface-temperature profile, which was necessary for ceramic considerations and was used in the cycle simulation heat transfer calculations. Due to the relatively high thermal resistivity of ceramics, temperature variations along the ceramic-lined engine walls will be much more pronounced and accordingly more important.

Thesis Supervisor: Professor John B. Heywood

Title: Professor of Mechanical Engineering
Director, Sloan Automotive Laboratory

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CHAPTER 1.0 INTRODUCTION

The type of fuel used by the general aviation industry, which is made up of smaller, propeller-driven aircraft, has been experiencing disproportionate price increases in comparison to other types of fuel, and is also becoming increasingly difficult to obtain in parts of the world and even in some areas of the United States. This is due to the fact that this type of fuel is produced in relatively small amounts when compared to the volume of fuel produced for the general automotive and commercial aviation sectors. It is also the most likely to experience cut-backs in production should another oil shortage occur.

This has given rise to research efforts by NASA and others toward developing an engine suitable for general aviation purposes, but which runs on a fuel whose economical and long-term supply is guaranteed. This means using one of the fuels which is already produced in great volumes, such as kerosine-type jet fuel, or "Jet - A."

The turbocharged, turbocompounded, direct-injection stratified charge Wankel rotary combustion engine is the most promising future powerplant of general aviation. Its advantages include high power density, multi-fuel capability, no balancing problems due to lack of reciprocating parts and low NO_x emissions. The problem which has traditionally plagued premixed spark-ignition Wankel engines has been poor fuel economy, however, the stratified-charge engine exhibits good fuel economy at part load.

A part of the research efforts conducted by NASA is the investigation into the heat transfer through the rotary engine housing (fig. 1), the resulting temperature distribution within the housing, and the temperature variation along the housing inner surface. This stems from an interest in thermal distortion of metallic cast housings, and in lining parts of the Wankel engine with ceramics, thereby reducing heat transfer losses and hopefully improving engine performance. Temperature fluctuations on the housing surface and the mean temperature distribution through the housing will be higher in a ceramic-lined engine than in a steel engine which makes the investigation into thermal stresses also important.

In order to do such calculations, a finite-element heat transfer analysis is usually performed. This requires heat-transfer boundary conditions to be known at locations on the inside of the housing. These include values of convective heat transfer coefficient and gas temperature at all the housing surface locations of interest, throughout the engine's operating cycle.

Wankel cycle simulation programs developed at the Sloan Automotive Laboratory at M.I.T. [1] can simulate the performance of the DISC Wankel engine under various operating conditions, giving overall cycle performance parameters and also physical/thermodynamic conditions within the chamber at any crankangle in the operating cycle. This includes the boundary conditions required for the above-mentioned finite element heat transfer analysis, namely the chamber gas temperature, heat transfer coefficient, and heat transfer rate to the housing. These values, however, are given as overall, average chamber conditions with respect to the moving rotor. As the rotor moves, the three chambers formed by the rotor and the housing also move. This means that each chamber "sees" a different part of the housing at each crankangle. In order to determine the chamber gas temperature, heat transfer coefficient, and heat transfer rate at each housing location, the output of the cycle simulation program needs to be transformed i.e., undergo a change of reference frame from the moving rotor to the stationary housing.

This paper describes, first, the work which has been done to calibrate the rotary engine cycle simulation routine with available rotary engine data, and second, the use of a data phasing routine to determine the thermodynamic/physcal properties of the working fluid at any housing location, thereby making the linking of the cycle simulation with a finite element analysis possible. In addition, the performance results from the cycle simulation routine, simulating the operation of a ceramic-lined engine and the operation of an equivalent all-metal engine, are presented and compared.

CHAPTER 2.0 DATA PHASING

2.1 ROTARY ENGINE GEOMETRY

To understand fully the modifications which must be made to the cycle simulation output data, it is essential to have a general understanding of the geometry of the Wankel rotary engine. The Wankel engine chamber is composed of three primary parts: the housing, rotor, and two side plates. The housing and side plates are analogous to the cylinder and cylinder head in a reciprocating IC engine, and the rotor is analogous to the piston. The rotor is shaped like an equilateral triangle with convex sides. The apexes of the rotor are always in contact with the inside wall of the housing thus forming the three working chambers.

Since the apexes of the rotor are always in contact with the inside walls of the housing, describing the motion of the rotor will also give the epitrochoidal shape of the housing. The rotor is mounted on an eccentric shaft that is centered at the geometric center of the housing. As the crankshaft rotates, an internal gear on the rotor-face mates with an external gear on the side plate and forces the rotor to revolve at one third of the angular speed of the crankshaft. The ratio of the radius of the internal gear on the rotor, to the radius of the external gear on the side plate is 3:2. With this arrangement, the geometric center of the rotor will rotate around the geometric center of the housing at a distance called the eccentricity, e. The distance from the center to the tip of the rotor is called the generating radius R. The ratio of these two radii (R/e) is known as the trochiod constant K, and usually ranges from six to eight. With the output/crank shaft rotating at angular speed ω, the rotor rotates at one-third that angular speed, or $\omega/3$ (fig. 2). The equations which then describe this motion, and therefore the epitrochoidal housing shape are

$$X = e \times \cos(\Theta) + R \times \cos(\Theta/3) \tag{2.1a}$$

$$Y = e \times \sin(\Theta) + R \times \sin(\Theta/3) \tag{2.1b}$$

where $\Theta = \omega t, \quad 0 \leq \Theta \leq 1080$

From the above equations, one can see (fig. 3) that a complete cycle for one chamber of the Wankel engine takes place over 1080 ($3 \times 360°$) crankangle degrees, unlike a reciprocating IC engine which requires only 720 ($2 \times 360°$) crankangle degrees. However, in the Wankel engine, there are two other chambers in different stages of their cycle, each being one-third of a cycle apart from the other two. This means that if one chamber is in its power stroke (combustion/expansion phase), the trailing chamber will be in its power phase 360 crankangle degrees later, and the third chamber will be in its power phase 720 crankangle degrees later. Therefore, in the Wankel engine, there occurs a torque pulse every 360 crankangle degrees, as opposed to 720 crankangle degrees in the reciprocating IC engine. This is a major reason why Wankel engines have a higher power/weight ratio than reciprocating IC engines.

2.2 REVIEW OF THE STRUCTURE OF THE DIRECT-INJECTION STRATIFIED CHARGE ROTARY ENGINE CYCLE SIMULATION ROUTINE

The simulation is separated into four distinct but sequential processes: intake, compression, combustion (and expansion), and exhaust. The intake process starts at the time that the intake port begins to open (TIPO) and continues until it is completely closed (TIPC). The compression process begins at TIPC and ends at timing of spark ignition (TSPARK). The combustion process is initiated with the spark occurrence and continues through expansion until the time that the exhaust port begins to open (TEPO). The exhaust process occurs from this point until the time that the intake port opens again, not until the exhaust port closing timing (TEPC). The cycle lasts exactly 1080 crankangle degrees.

The cycle simulation routine is a zero-dimensional model using the First Law of Thermodynamics and a specified combustion/fuel-energy release rate to predict engine operating characteristics. To solve the differential form of the First Law equation for the chamber contents, some basic assumptions were made. The chamber contents are assumed to be a homogeneous mixture of ideal gases. Therefore the contents have a single mean temperature. In addition, it was assumed that there are no pressure waves within the chamber so pressure is also uniform throughout the chamber. During intake and compression, chamber contents are characterized as a mixture of fresh charge (air and fuel vapor) and residual burned gas. The intake and exhaust manifolds are assumed to have uniform temperatures and pressures. During exhaust only burned product is assumed to be present in the chamber. Unburned gas that leaks into the exhaust chamber is treated as burned product and the amount of fuel energy lost is accounted for. During combustion, an algebraic expression defining the heat release rate for stratified charge engines is used (see Chapter 4).

The four engine processes, intake, compression, combustion-expansion, and exhaust are separated into different subroutines (fig. 4). The Main program controls, through ODERT (a numerical integrator), the program flow. That is, the Main decides when each engine process begins and ends or what, if any, numerical data should be printed out.

The program begins by supplying the required input data to Main. A calculation is then performed to initialize the thermodynamic state of the chamber gas at the beginning of the intake process. The intake subroutine is then called by ODERT. The intake subroutine then calls other subroutines where, for a given crankangle, the thermodynamic properties of the working fluid are evaluated, and the direction, composition, and magnitude of mass flow rates through the intake and exhaust ports are computed. The heat transfer rate, using the engine component temperatures defined by the user, is then calculated as is the rate of change of chamber volume. Next, leakage and crevice volume mass flow rates are calculated. Finally, the intake subroutine has enough information to evaluate the rates of change of pressure, temperature, work, heat transfer, and mass flow. The program control returns to ODERT which integrates the differential variables through time (crankangles) until program flow returns to Main. This interaction between Main and ODERT is repeated until each engine process is finished and the cycle is completed. The final values of chamber pressure, temperature, and mass are compared to the initialized values and if each property is within the error criteria, the calcualtion ends. However, if these criteria are not met, the program begins again with the final property values used as the initial quantities for the second iteration. Convergence occurs quickly, usually requiring three, or at most four iterations.

The following is a list of the files created by the cycle simulation routine and a description of their contents. The names of the files are the default names given when writing to that particular logical unit number.

FOR007.DAT	Values of chamber pressure, control flags, and housing surface temperature at each crankangle print interval.
FOR010.DAT	Leakage and crevice volume data. Values of chamber mass, lead crevice mass, lag crevice mass, lead leakage mass, lag leakage mass, lead crevice composition, and lag crevice composition at each crankangle print interval.
FOR013.DAT	Heat Transfer Data. Values of average cylinder gas velocity, heat flux to the rotor, side, and housing, and percentage of total heat transfer to the rotor, side, and housing at each crankangle print interval.
FOR016.DAT	Main output file. Repeats the values from the input file (FOR008.DAT) and then lists values of chamber pressure, gas temperature, integrated mass flux through the intake and exhaust ports, flow velocities through the intake and exhaust ports, burned fuel fraction, heat transfer rate, work, and two control flags at each crankangle print interval. In addition, gives overall cycle performance parameters.
FOR023.DAT	Same data as in FOR013.DAT except only for the third iteration.
FOR026.DAT	File used by the data transformation routine to determine boundary conditions at locations on the housing. Lists the third-iteration values of chamber pressure, gas temperature, total heat transfer, percent heat transfer to the housing, rotor, and side plates, and convective heat transfer coefficient at each crankangle print interval.

2.3 HOUSING POSITION DEFINITION

In order to determine gas temperatures and heat transfer rates at specific housing locations, there first must be a convention by which housing locations can be defined. The following convention is used here.

A housing location is defined by its angular position, ϕ, measured with respect to the center of the housing (location X = 0, Y = 0 from equations 2.1a/b). Position $\phi = 0°$ is defined to be the "waist" of the housing on the side opposite the intake and exhaust ports (fig. 5).

This corresponds to the housing position in the center of the chamber at crankangle zero. Phi increases in the direction of rotor and crankshaft rotation from $\phi = 0°$ to $\phi = 360°$. By this means, any position on the inside of the housing can be defined, and worked with.

2.4 CYCLE SIMULATION OUTPUT DATA MODIFICATION

Execution of the DISC Wankel cycle simulation program produces several output data files (see Chapter 2.2). These files contain overall cycle performance parameters and information concerning chamber conditions at each crankangle starting from TIPO (Time of Inlet Port Opening) to TIPO + 1080°, i.e. over the full engine cycle. The transforming of this data at a specified housing location is accomplished by a separate program which reads from the output data files. The parameters which are transformed are the chamber gas temperature, chamber pressure, heat transfer rate to the housing, and the convective heat transfer coefficient from the gas to the chamber walls.

During the operation of a Wankel rotary engine, the temperatures around the housing will vary with position and with time. One section of the housing will be hotter than others because the combustion and expansion process always occurs at the same section whereas the intake process , which is much cooler, will always take place at another section. In other words, any specific housing location is exposed to only one-third of the entire engine cycle that the rotor-faces are exposed to. Exactly which third of the cycle is dependent upon the particular location.

The transformation routine will, for a given housing location, determine which 360° of the full 1080° cycle (i.e. one-third) the given housing location will be exposed to, and transform the data to that location from TIPO to TIPO + 1080°. See Appendix C and figures 15, 16, 17, and 18 for a listing of the program which accomplishes this transformation, and an example of the "un-transformed" output from the cycle simulation routine and the transformed output produced by the transformation program.

2.5 ADDITIONAL MODIFICATIONS

The data transformation routine described above is, at present, a separate routine from the cycle simulation program. In order to link fully the cycle simulation program with a finite-element heat transfer analysis, the transformation routine should be made part of the cycle simulation so that the boundary conditions are automatically determined at specified housing locations. The format of these boundary values will have to be determined according to the input requirements of the finite-element analysis.

CHAPTER 3.0 CYCLE SIMULATION PROGRAM MODIFICATIONS

3.1 CALCULATION OF HEAT TRANSFER TO THE HOUSING

The heat transfer model used in the cycle simulation, is one of forced convection over a flat plate. The general equation used to calculate the heat transfer per area to the housing is

$$(dq/dt)_{hous} = h(T_{gas} - T_{hous})$$

The determination of the heat transfer coefficient is based on the empirical Nusselt-Reynolds number correlation [2] for turbulent internal flow, convective heat transfer, and the method proposed by Woshni [3] for reciprocating IC engines.

The housing, rotor, and sideplate temperatures are determined by the user input and are constant throughout the cycle. Because of the high thermal conductivity and large thermal capacity of steel, there is relatively little variation in surface temperature along the inside walls of the housing, so the calculation of heat transfer using a constant housing temperature throughout the engine cycle, is quite appropriate.

In the case of a ceramic-lined engine housing, however, the temperature variation will be much greater. This is due to the significantly lower thermal conductivity of ceramics, and the resulting "shallow" temperature penetration depth.

In order to simulate more accurately the operation of the Wankel engine with a ceramic-lined housing, the cycle simulation program has been modified so that the user may define a temperature profile around the housing. This is done by defining temperatures at specific crankangle locations. The program linearly interpolates the housing temperature at intermediate crankangle locations. In this way, the program user can represent more accurately the heat transfer conditions in a ceramic-lined engine.

3.2 USEFULNESS OF A USER-DEFINED HOUSING TEMPERATURE PROFILE

In order to model exactly the operation of a ceramic-lined engine, the temperature variation along the inside of the engine housing would first need to be determined. To accomplish this the cycle simulation routine would have to be linked to a finite-element heat transfer analysis. The cycle simulation program would then be run with some base conditions of housing temperature to determine heat transfer rates to the housing at numerous housing locations. From these initial results, the finite-element calculations would then use these heat transfer boundary conditions to arrive at a new set of location-specific housing temperatures. The cycle simulation routine would then re-calculate heat transfer rates based on the new housing temperature information which would, in turn, be used in the next finite element calculation to update the location-specific housing temperatures. This iterative routine should continue until the convergence of both the housing temperatures and heat transfer rates is achieved. This would then be the most accurate modeling of a ceramic-lined engine for the cycle simulation.

The capability of defining a housing temperature profile provides a quick, easy, and inexpensive way to accomplish the same task. The accuracy of this method is dependent only on the accuracy of the chosen profile, and its effect on the accuracy of the cycle simulation routine is unlikely to be very sensitive to minor discrepancies.

CHAPTER 4.0 COMBUSTION MODEL REVIEW

The combustion model provides, to the cycle simulation routine, the model by which the fuel-energy release rate is calculated. It should be a generalized model which is applicable to many different operating conditions such as engine speed, inlet pressure, and equivalence ratio. At the time when this combustion model was formulated, actual rotary engine operating data was not available for comparison. Therefore, it was not possible to calibrate the model to achieve greatest possible accuracy. Now, however, such engine data has been recorded and is available, making combustion model verification possible. Of interest is the accuracy of the overall general shape of the heat release profile and its possible variation under different operating conditions (i.e. engine load and speed)

4.1 BASIC COMBUSTION MODEL CHARACTERISTICS

The general shape of the combustion model used in the Wankel engine cycle simulation routine is a linear rise from zero at the prescribed spark timing to a maximum normalized fuel-energy release rate, followed by an exponential decay toward zero (fig. 6). The maximum normalized fuel-energy release rate was defined to be the maximum fuel-energy release rate during combustion, divided by the total released fuel energy per engine cycle.

The existing combustion model was formulated by Balles and Roberts [4]. Cycle-by-cycle cylinder pressure data, measured from a reciprocating stratified-charge engine, under two different speeds and different equivalence ratios, provided the data base for this work.

Reciprocating engine data was used because rotary engine data was not avialable at the time. A heat release analysis was then performed to convert the pressure data to fuel energy release rate. It was found that the curves of fuel energy release rate vs. crankangle, when normalized by the fuel heating value, were quite similar in all cases (fig. 7). Some of the trends noted were:

a) the start of positive heat release occured at the same crankangle location in relation to start of injection

b) the rise in heat release rate to a maximum value was linear and with approximately the same slope

c) the peak heat release rate occured at roughly the same crankangle location relative to the start of combustion

d) the decay in heat release rate was exponential in shape.

From these observations then, the combustion model was assumed to be a straight-line increase in fuel-energy release rate from zero, at the start of combustion, to a maximum value occuring at a constant number of crankangle degrees later. This was then followed by an exponential decrease, toward zero, in the fuel-energy release rate. The model was constrained during the exponentially decreasing energy-release portion so that the entire curve of the normalized fuel-energy rlease rate integrates to a value equal to the combustion efficiency. The combusiton efficiency is defined as the mass of fuel burned per cycle divided by the mass of fuel injected per cycle.

Use of the model requires four input variables from the user in the input file. These numbers are the start-of-combustion timing (Θ_s), the timing of maximum heat release rate (Θ_m), the normalized value of the maximum heat release rate ($(dQ/d\Theta)_m$), and the combustion efficiency

(η_c). The normalized heat release rate increases linearly from zero at the start of combustion, Θ_s, to the specified value of $(dQ/d\Theta)_m$ at Θ_m and then decays exponentially according to a calculated time constant, τ (fig. 8). The decay constant is dependent upon the three combustion variables and the combustion efficiency η_c. Therefore, the inputs to the model are:

Θ_s : crankangle of start of combustion (TSPARK)

Θ_m : crankangle of the peak normalized heat release rate (TMAX)

$(dQ/d\Theta)_m$: value of the peak normalized heat release rate as defined above (DQDTMAX)

η_c : combustion efficiency (XBSTOP).

In parametric form, the equations for the normalized fuel-energy release rate are, for Θ between Θ_s and Θ_m,

$$\frac{dQ}{d\Theta} = \left(\frac{dQ}{d\Theta}\right)_m [(\Theta - \Theta_s)/(\Theta_m - \Theta_s)] \tag{4.1}$$

and for Θ greater than Θ_m,

$$\frac{dQ}{d\Theta} = \left(\frac{dQ}{d\Theta}\right)_m \exp[\, - (\Theta - \Theta_m)/\tau \,] \tag{4.2}$$

where, according to the integral constraint,

$$\tau = [\eta_c / \left(\frac{dQ}{d\Theta}\right)_m - \frac{1}{2}(\Theta_m - \Theta_s)] \tag{4.3}$$

4.2 ANALYSIS OF ENGINE DATA

As mentioned above, the fuel-energy release profile for the rotary engine cycle simulation was formulated from data from a reciprocating engine. Now that heat release data from rotary engines is available, it is valuable to check the heat release model to see if any modifications are required.

It is of interest to compare the shape of the heat release model with the engine data and to study if there exist any "predictable" variations of the heat release profile with operating conditions. The questions to answer were:

a) Does the initial part of the profile increase approximately linearly?

b) How long (i.e. for how many crankangle degrees) does this linear rise last?

c) How closely does the decaying portion of the heat release resemble an exponential decay?

d) Does the maximum normalized heat release rate $(dQ/d\Theta)_m$ stay constant under different operating conditions, or does it vary? If it does vary, can the variation be expressed as a function of input variables?

The rotary engine data was taken by the Rotary Engine Division of John Deere Technologies International, Inc. and are presented in graphical form [5]. The data include curves of heat release vs. crankangle for two different displacement stratified-charge rotary engines operated at constant speed and under four different load conditions each. For each engine, under its maximum-load operating condition, there were also graphs of chamber pressure vs. crankangle. Each of the engines was turbocharged and intercooled. The two engines are referred to as the 580 engine (5.78 ℓ displacement) and the 070 engine (0.66 ℓ displacement) and have one rotor each. For further details about the engines and their operated conditions, see Table 1.

4.3 ENGINE DATA MANIPULATION AND COMPARISON

4.3.1 HEAT RELEASE SHAPE COMPARISON

Since the curves of heat release and chamber pressure vs. crankangle were only presented in graphical form, it would have been too subjective to make accurate comparisons with the cycle simulation's heat release model and pressure plots. Therefore these curves were digitized using a bit-pad digitizer in the Sloan Automotive Laboratory at M.I.T. From the data given with the heat release curves in ref. 5, there was no direct way of calculating the mass of fuel injected per cycle and hence no way of calculating the fuel energy released per cycle. This made the use of the bit-pad digitzer even more attractive because it calculated numerically the area under the heat release curves which gives the value of the total released fuel energy per cycle. This number is necessary to determine the normalized value of the maximum heat release rate for comparison with the heat release model.

In order to make the comparison in the shapes of the heat release curves as straight-forward, and accurate as possible, they were each normalized to unity (i.e. each curve was normalized to its respective maximum value of heat release rate) and shifted so that the location of their peak heat release rates were coincident at zero degrees crankangle (figs. 9 and 10).

4.3.2 INVESTIGATION OF MAX NORMALIZED HEAT RELEASE RATE

The maximum values of the heat release rates were read from the heat release curves and divided by their respective values of total released fuel energy per cycle which was obtained from the bit-pad digitizer (see Appendix A). This value is the normalized maximum heat release rate. For both engines, this normalized maximum heat release rate increased with load. A linear regression was used to find the best straight-line fit to these four data points for each engine. In addition, the correlation for each of these data sets was calculated using the equation [6],

$$\text{correlation} = \frac{\sum_{i=1}^{n}(x_i - \bar{x})(y_i - \bar{y})}{\left[\sum_{i=1}^{n}(x_i - \bar{x})^2 \sum_{i=1}^{n}(y_i - \bar{y})^2\right]^{1/2}} \tag{4.4}$$

For the 070 engine:

$$\left(\frac{dQ}{d\Theta}\right)_m = 2.43 \times 10^{-5}(\text{BMEP [kPa]}) + 1.50 \times 10^{-2} \tag{4.5}$$

correlation = 0.959

and for the 580 engine:

$$\left(\frac{dQ}{d\Theta}\right)_m = 1.69 \times 10^{-5}(\text{BMEP [kPa]}) + 2.35 \times 10^{-2} \tag{4.6}$$

correlation = 0.746

See figure 11 for a graphical representation of these functions.

Engine load, in the cycle simulation routine, however, is an output variable, not an input. Therefore, keeping in mind that it was sought to find a possible relation between the maximum normalized heat release rate, $(dQ/d\Theta)_m$, and some input variables, it was next reasoned that engine load (i.e. mean effective pressure) can be expressed as a function of inlet pressure multiplied by equivalence ratio. This was arrived at from the relationship

$$\text{mep} = \eta_f \eta_v Q_{HV} \rho_{inl} (F/A) \quad [7]$$

where η_f - fuel conversion efficiency

η_v - volumetric efficiency

Q_{HV} - heating value of the fuel

ρ_{inl} - inlet air density

F/A - fuel/air ratio

Since the inlet air density and fuel/air ratio scale linearly with the inlet pressure and equivalence ratio according to

$$\rho_{inl} = p_{inl}/(RT_{inl}) \quad \text{(inlet air temperature assumed constant)}$$

$$F/A = \Phi(F/A)_{stoich.}$$

and assuming that the variation in fuel conversion efficiency, and volumetric efficiency is relatively small, the relationship was found to be for the 070 engine:

$$\text{BMEP [kPa]} = 1130 \times (\Phi \times p_{inl}) - 170 \qquad (4.7)$$

correlation = 0.994

and for the 580 engine:

$$\text{BMEP [kPa]} = 1470 \times (\Phi \times p_{inl}) - 148 \qquad (4.8)$$

correlation = 0.999

See figure 12 for a graphical representation of these functions.

It was then possible to formulate an expression for the maximum normalized heat release rate as a function of the two input variables of equivalence ratio, and inlet pressure.

For the 070 engine:

$$\left(\frac{dQ}{d\Theta}\right)_m = 0.0282(\Phi \times p_{inl}) + 0.0103 \quad (4.9)$$

correlation = 0.979

And for the 580 engine:

$$\left(\frac{dQ}{d\Theta}\right)_m = 0.0246(\Phi \times p_{inl}) + 0.0211 \quad (4.10)$$

correlation = 0.736

See figure 13 for a graphical representation of these functions.

4.4 RESULTS OF COMBUSTION MODEL COMPARISON

From examining the unit-normalized and shifted heat release curves with the heat release model, it was concluded that the assumed shape of the heat release rate profile is, in fact, a good representation of the actual engine data.

The initial rise in heat release rate is well approximated by a straight-line increase. This is evident from the data from the 070 engine (fig. 9), and is especially true for the 580 engine data (fig. 10). The time, in crankangle degrees, to reach maximum rate of heat release appeared to be 23° for both engines and in each operating condition. The decaying portion of the heat release curves was also seen to be similar to an exponential decrease.

Overall, the heat release model in the rotary engine cycle simulation routine was found to be a valid representation of actual engine profiles. It would appear that the value of the maximum normalized heat release rate does vary with load, and two equations have been formulated to predict this variation, one for the small displacement, 070 series, engine and the other for the large displacement, 580 series, engine. Lacking more engine data under a wider variety of operating conditions, however, these equations have not been incorporated into the cycle simulation code, and are intended as a guide and reference for further investigations.

CHAPTER 5.0 PERFORMANCE CALCULATIONS

The rotary engine cycle simulation routine was modified to allow for a user-specified temperature distribution along the housing. As stated previously, this allows for the performance simulation of a ceramic-lined rotary engine. Because of research efforts into the use of rotary engines in light aircraft, it is of interest to determine the difference in the performance of ceramic-lined engines to regular all-metal engines. Aside from the potential performance gain through the use of ceramics, the fact that it could eventually lead to the development of engines without a cooling system makes their investiation worthwile.

The operating condition for a rotary engine in an aircraft at cruise power and at sea level which was used in the cycle simulation routine was obtained from a report by the Curtiss Wright and Cessna Aircraft Corporations for the NASA Lewis Research Center [8]. This described a 777 cm^3 displacement engine at 5000 RPM and equivalence ratio of 0.6. The engine was very similar in geometry to the pre-existant geometry of an OMC engine [1,9], so the geometry was not changed.

The effect on performance of the temperature of the three main engine components (housing, rotor, and side plates) was examined. This included a baseline "cool-all-around" case, and then cases in which individual engine components, and combinations of the three engine components were assigned "hot" temperatures. In addition, two cases were run with a user-defined housing temperature profile which simulated a ceramic-lined housing. Finally, high speed and low speed cases were run to examine the effect of speed on performance due to the expected increase in importance of heat transfer at the slow speed. In order to include the crevice effects in these runs, the crevice temperature was assigned equal to the rotor temperature. This is because the majority of the crevice volume is contained in the rotor [1,9].

A tabular summary of all the cases is to be found in Table 3.

5.1 CONSTANT HOUSING TEMPERATURES

Assuming a base average temperature for a metallic surface inside the engine chamber to be 425 K, it was reasoned that the rotor would be at a modestly higher temperature assumed to be 500 K. This is because the rotor is contained within the housing and side plates and therefore has no exterior surface exposed to the ambient air. This then, is the base, "cool-all-around", case temperature condition: housing and side plates at 425 K, and rotor at 500 K. The "hot" temperature was chosen to be 800 K. This is slightly cooler than the mean exhaust gas temperature for all of the cases.

5.2 USER-DEFINED HOUSING TEMPERATURE PROFILE

The inner-surface of the rotary engine housing is roughly equivalent to the cylinder head of a reciprocating engine. The main difference is that the entire cylinder head is exposed to every phase of the engine cycle whereas a specific location on the housing inner surface will only be exposed to (i.e. "see") one-third of the engine cycle. This means that the temperatures that any number of housing locations will be exposed to, will not be the same.

For an all-metal housing, this resulting difference in the average surface temperature between the hottest and coldest location is not critical. This is because steel has a high thermal conductivity which quickly transfers the heat energy away from the inner surface toward the cooling passages. A ceramic-lined engine would not behave the same. The thermal conductivity of ceramics is low relative to steel, classifying them as an insulating material. The difference in the average surface temperature between the hottest and coldest location would be much more pronounced and would accordingly have a greater impact on the performance of the engine.

This can be accounted for by defining a housing temperature profile where a temperature is assumed at selected crankangle locations on the housing (see Appendix B). The housing temperature is linearly interpolated for locations between two specified temperature locations. The specific shape of the profile was determined by following the same shape determined in reference 10 (fig. 14). The difference between the maximum and minimum housing temperature, however, was increased because the engine housing in reference 10 was metallic, in accordance with the reasons stated above.

The peak housing temperature was chosen to be 1200 K. This is approximately equal to the average gas temperature during the compression and combustion phase of the cycle. See Table 3 for a description of the operating conditions with a defined temperature profile.

5.3 COMPARISONS AND CONCLUSIONS

A listing of the cases and a selection of their respective cycle simulation output parameters appears in Tables 4 and 5.

From the results of the cycle simulation runs with constant housing temperatures, the effects of chamber-component temperature and engine speed can be observed. The convective heat transfer losses were highest in the "cool all around" case and, as expected, decreased steadily as the rotor, housing, both rotor and housing, and all three, rotor, housing and side plates, were, in turn, assigned the "hot" temperature. The crevice heat loss decreased significatly in the cases where the rotor was assigned the "hot" temperature, but showed little variation with the temperature of the housing or side-plates. This is, as stated earlier, because most of the crevice volumes are in the rotor and therefore the crevice temperature was set equal to the rotor temperature. The slower engine speed increased both the convective and crevice heat loss due to the greater time available for heat transfer. The increase in the convective heat loss at the slower engine speed was roughly 14% whereas the crevice heat loss increased approximately 45% due to the additional effect of the substantial increase in leakage through the crevices which occurs at slower speeds.

The volumetric efficiency decreased as engine components were assigned the "hot" temperature. This was due to the heating of the inlet air in the chamber during intake, which increased its specific volume and resulted in a diminished breathing capability. For the "cool all around" case, the volumetric efficiency decreased at the lower speed operating condition. This is due to the greater effect of reverse flow at the slower speed which will reduce volumetric efficiency, and to the greater time available for intake charge heating at the lower engine speed.

The variation in fuel-conversion efficiency due to temperature variation of the engine components, was not as noticeable as for the volumetric efficiency. There was a general increase in η_f for the cases with hot engine components. This is due to the decrease in convective heat transfer losses which result from higher chamber surface temperatures as reflected in Table 5. The effect of engine speed on the fuel-conversion efficiency was greater in the "cool all around" case. It decreased significatly at the slower speed due to the increase in heat transfer losses to the engine walls.

The indicated mean effective pressure decreased with increased engine wall temperatures and decreased at the slow speed condition. This is attributable to the variations in the volumetric and fuel-conversion efficiencies. Although the fuel-conversion efficiency increased with engine temperature, the voulumetric efficiency decreased by a greater amount. The percentage decrease in IMEP from the base, "cool all around" case to the "hot all around" case was approximately equal to the percentage decrease in the product of the fuel-conversion efficiency and the volumetric efficiency between the same two cases. This was expected from the relationship

$$mep = \eta_f \eta_v Q_{HV} \rho_{inl} (F/A) \quad [7]$$

The same trends were reflected in the results from the cases where a housing temperature profile was used. The convective and crevice heat losses decreased with increased rotor temperature as was expected. Higher engine temperatures resulted in decreased volumetric efficiencies but increased fuel-conversion efficiencies. Since the increase in fuel-conversion efficiency was not as large as the decrease in the volumetric efficiency, the indicated mean effective pressures decreased. The same effects of engine speeds were also evident. At the slow engine speed, the convected heat loss increased approximately 12% and the crevice heat loss displayed a significant increase of approximately 45%. The volumetric efficiency and the fuel-conversion efficiency decreased at the slower speed due to the same reasons stated for the constant housing temperature cases.

It is of interest to compare the cases with constant housing temperatures to the cases with the defined housing temperature profile. This is equivalent to comparing an all-metal engine to one with a ceramic-lined housing. The particular cases to compare are the "cool all around" with the "cool rotor", the two "hot rotor" cases, and the two cool, slow speed, cases. In all three instances the ceramic-lined engine exhibited higher fuel-conversion efficiencies with slightly lower volumetric efficiencies. This resulted in a net increase in the indicated mean effective pressures of approximately 2.5% with the additional bonus of a decrease in indicated specific fuel consumption by approximately 3%. These gains are attributable to the insulating effect of the ceramic material. Chamber wall temperatures will be higher which reduces the convected heat transfer losses throughout the cycle, but most importantly in the released fuel-energy during combustion and expansion. This reduction was on the order of 35%. In contrast, the crevice heat losses increased slightly (by about 4%). The overall decrease in total heat transfer was approximately four percent.

From these results, the use of ceramics to line the rotary engine housing, would result in performance gains with the added benefit of a decrease in fuel consumption. However, it appears that ceramic use on the rotor, while reducing heat transfer losses, also reduces volumetric efficiency resulting in an overall loss in mean effective pressure. These results definitly point to the need for further research into the use of ceramics in engine applications.

CHAPTER 6.0 FRAMEWORK FOR FINITE ELEMENT HOUSING TEMPERATURE DISTRIBUTION DETERMINATION

The boundary conditions at any location on the housing inner surface can be determined by the phasing routine listed in Appendix C. Included are also directions on how to use the phasing routine. With this, and a background boundary condition (such as a constant temperature defined to be equal to the coolant temperature), a transient finite element heat transfer analysis could be performed. At specific locations, a one-dimensional analysis could be performed, or, knowing the boundary conditions at many housing locations, a two-dimensional analysis could be performed. The specific boundary conditions to use would be the gas temperature and convective heat transfer coefficient. An analysis with these boundary condition values would return housing temperature values for use in the next cycle simulation. This would be iterated until convergence of the housing surface temperature profile.

With an initial guess of the housing surface temperature profile, the cycle simulation would be run to determine the gas temperature and heat transfer coefficient throughout the cycle. This output would then be transformed by the program in Appendix C to give the boundary conditions at the housing surface. This would be stored in a separate file in a format determined by the finite element routine. The finite element routine would then read the heat transfer data in this file and calculate the temperatures on the surface and within the housing. The surface temperatures should then be used to "update" the initial housing temperature profile. The cycle simulation would again be run using the new temperature profile to again determine the gas temperature and heat transfer coefficient throughout the cycle. This would again be transformed to give the boundary conditions at specific housing locations, and stored in a separate file. The finite element routine would again be run to calculate the surface and inner housing temperatures. The new temperature profile would be compared to the previous one, and if the differnce between the two would be within a user-prescribed criteria, the calculation would cease. If not, the temperature profile would again be "updated" and the cycle simulation routine would run again.

Of importance in the use of ceramics are the magnitudes of the surface temperature fluctuations, and the severity of the temperature gradients within the material. This is because thermal stesses may exceed the elastic limits of the material and cause failure. A transient finite element analysis could determine the thermal stresses if a very fine mesh is used at and near the surface. A program to generate the mode and element mesh for rotary engine housings has been developed at the Michigan Technological University [11, 12]. It is designed to be used with the I-DEAS CAD software package by the SDRC Corp. and with a minimum number of user inputs.

APPENDIX A

USE OF THE BIT-PAD DIGITIZER IN THE CALCULATION OF THE MAXIMUM NORMALIZED HEAT RELEASE RATE

The bit pad digitizer allowed for the digitizing of the curves of indicated fuel energy release rate in Joules per crankangle degree [J/CA] versus crankangle degree [CA]. This allowed the capability to manipulate the curves by changing their scale, normalizing them, graphing them with coincident maximum heat relwease rates, etc. The digitizer required the specification of two poinnts along the independent axis (axis of CA) and a scaling distance between them. This set the scale for the horizontal axis and was automatically taken as the scale for the vertical axis. The mouse was then used to trace the curve. Points were automatically read into the computer at evenly spaced intervals, the distance of which was specified by the user. After the complete curve was traced, the digitizer stored the data points into a file and the calculated area was given. Since the same scale of the horizontal axis is assumed for the vertical axis, the units of the calculated area was in crankangle degrees squared [CA^2]. This required a conversion factor between the horizontal and vertical axis. An equivalent length was measured on each exis to find out how many crankangle degrees equalled an equivalent length of Joules per crankangle degree [J/CA]. The conversion was then made from an area given in units of [CA^2] to the sought value in Joules. This is the total indicated released fuel energy per engine cycle. The maximum fuel energy release rate was then visually read from the graph and divided by this figure to give the normalized maximum fuel energy release rate (DQDTMAX). The table below summarizes the results for both engines and for each operating condition.

- 070 ENGINE -

BMEP [kPa]	DIGITIZED AREA [CA²]	CONVERSION FACTOR [J/CA²]	CORRECTED AREA [J]	MAXIMUM HEAT REL RATE [J/CA]	NORMALIZED MAX HEAT REL RATE [1/CA]
298	3591	0.3000	1077.15	22.5	0.0209
529	2880	0.5467	1574.30	44.5	0.0283
642	2362	0.6583	1555.25	51.0	0.0328
890	2361	0.9667	2282.69	80.0	0.0350

- 580 ENGINE -

BMEP [kPa]	DIGITIZED AREA [CA²]	CONVERSION FACTOR [J/CA²]	CORRECTED AREA [J]	MAXIMUM HEAT REL RATE [J/CA]	NORMALIZED MAX HEAT REL RATE [1/CA]
181	3157	1.1833	3736.12	92.0	0.0246
405	2928	2.1117	6182.31	177.0	0.0286
626	1864	4.6667	8696.60	360.0	0.0414
957	2169	6.4167	13920.2	500.0	0.0359

APPENDIX B

HOW TO DEFINE A TEMPERATURE PROFILE FOR USE IN THE CYCLE SIMULATION ROUTINE

A temperature profile is defined by the user by giving housing crankangle locations and assigning to each one a housing surface temperature. The housing surface temperatures for the in-between locations are linearly interpolated using the two specified temperatures of the neighboring specified locations. The housing locations follow the same convention as used for crankangle definition, starting from time of inlet port opening (TIPO) and going until TIPO + 1080° with zero crankangle being the point of minimum chamber volume. If the locations of the temperatures that the user wishes to define are in the convention described in Chapter 2.2, they must be transformed into the crankangle-equivalent locations in order for the intended temperature profile to be used in the program. A maximum of twenty temperature points may be defined.

In order for a temperature profile to be used by the cycle simulation program, the logical variable TEMPRO must be set equal to true (i.e. TEMPRO = .TRUE.) in the namelist, INPUT, inputfile whose default filename is FOR008.DAT. An example of this inputfile follows. With TEMPRO equal to true, the cycle simulation program will then read the namelist, PROFILE, inputfile which contains the crankangle and temperature pairs defining the temperature profile, and whose default filename is FOR009.DAT. An example of this input file, for the temperature profile used in the performance calculations, also follows.

The first variable in the PROFILE namelist is NSEGS. This must be set equal to an integer value equal to the number of points to which temperatures are to be defined. Following this is the real, dimensioned variable CSEGS to which NSEGS number of crankangle locations must be listed. These crankangle locations must be ordered in their occurance from TIPO to TIPO + 1080°. Following this, finally, is the real, dimensioned variable TSEGS to which NSEGS number of temperatures in degrees Kelvin must be listed according to their crankangle location listing in CSEGS.

Because of continuity, the housing temperature at TIPO must be equal to the housing temperature at TIPO + 1080° because these are actually the same location. The specified temperature profile will be automatically adjusted to meet this requirement in case the given profile does not. If, in the namelist, INPUT, inputfile (FOR008.DAT), t he logical variable TEMPRO is false (i.e. TEMPRO = .FALSE.), then the value of THOUSI (initial housing temperature) will be the assumed housing surface temperature throughout the cycle.

CYCLE SIMULATION PROGRAM INPUT FILE (FOR008.DAT)

```
$INPUT
     FIRE      = .TRUE.
     SPBURN    = .TRUE.
     TEMPRO    = .TRUE.
     FUELTP    = 1
     PHISTA    = 0.6
     ECCEN     = 1.5
     ROTRAD    = 10.5
     DEPTH     = 7.00
     VFLANK    = 35.00
     RPM       = 2500.
     TIPO      = -530.0
     TIPC      = -180.0
     TEPO      = 199.0
     TEPC      = 588.5
     TSPARK    = -3.0
     THIPO     = 120.0
     THEPO     = 40.0
     IPA       = 13.8
     EPA       = 6.5
     XBZERO    = 0.0003
     XBSTOP    = 0.995
     TMAX      = 20.0
     DQDTMAX   = 0.0415
     PATM      = 1.000
     TATM      = 300.0
     PIM       = 1.80
     TFRESH    = 300.0
     TEGR      = 300.0
     EGR       = 0.0
     PEM       = 1.30
     TROTOR    = 500.
     TSIDE     = 425.
     THOUSI    = 425.
     CONHT     = 0.037
     EXPHT     = 0.8
     TPRINT    = 1.0
     AREROT    = 1.E-4
     CIINTG    = 1.E-4
     CCINTG    = 1.E-4
     CBINTG    = 1.E-5
     CEINTG    = 1.E-4
     MXTRY     = 1
     REL       = 0.0002
     MAXITS    = 3
     MAXERR    = 0.03
     MAXTRY    = 10
     AREALK    = 0.01
     CREVOL    = 0.875
     TCREV     = 500.
     CON1      = 0.75
     CON2      = 0.324
$END
```

INPUT FILE FOR TEMPERATURE PROFILE DEFINITION (FOR009.DAT)

```
$PROFILE
    NSEGS = 8
    CSEGS = -410., -180., -85.,   50.,  205., 260., 350., 515.
    TSEGS =  435.,  525., 675., 1200., 1100., 550., 400., 420.
$END
```

or equivalently,

```
$PROFILE
    NSEGS=8
    CSEGS(1) = -410.
    CSEGS(2) = -180.
    CSEGS(3) =  -85.
      .   .      .
      .   .      .
      .   .      .
    CSEGS(8) =  515.
    TSEGS(1) =  435.
    TSEGS(2) =  525.
    TSEGS(3) =  675.
      .   .      .
      .   .      .
      .   .      .
    TSEGS(8) =  420.
$END
```

APPENDIX C

USE OF THE CYCLE SIMULATION OUTPUT PHASING PROGRAM

Execution of the rotary engine cycle simulation routine creates several output files (see Chapter 2.2). One of these output files contains a listing of chamber pressure, gas temperature, total heat transfer, and percentage of heat absorbed by the housing, rotor, and side plates, and heat transfer coefficient at each crankangle degree during the engine cycle. The default name given to this file by the cycle simulation routine, when writing to logical unit number 26, is FOR026.DAT.

In order to determine the thermodynamic and heat transfer conditions at a specific location on the housing surface, the data in this file (FOR026.DAT) needs to undergo a transformation from the reference frame of the rotor to the stationary reference frame of the housing . This is accomplished by a separate program called POSITION.FOR. This program, by default, reads from FOR026.DAT, but another name can be given if this file has been renamed. Upon execution of this program, it creates four graphable HDCOPY.PLT files of chamber pressure, gas temperature, total heat released and heat released to the housing, and heat transfer coefficient versus crankangle. In addition, a listing of these is written in a file called POSITION.OUT which could be used as the input to a finite element analysis.

The following is a listing of the program. See figures 15, 16, 17, and 18 for a graphical representation of the transformation of the output data.

```
      PROGRAM POSITION
C--------------------------------------------------------
C NOTE: It must be linked like "LIN POSITION,VTPLOT$/OPT"
C This program creates a HDCOPY.PLT plot of chamber gas
C temp., chamber press., heat transfer to the housing,
C or htrcoe at a given housing position over one complete
C cycle of a rotary engine.  It also creates a data file
C (POSITION.OUT) of these values.
C It reads the output file of the final iteration (FOR026.DAT)
C of the rotary engine cycle simulation (W****.FOR) if no
C other input filename is given.
C---------------------------------------------------------
      CHARACTER*40 XLAB, YLAB, INPUTF, OUT1
      CHARACTER ANS
      REAL DATAT(6,1100), GRAPH(6,1100)
      INTEGER  IXPLOT(6)
      DATA      INPUTF,          OUT1
     &    /'FOR026.DAT', 'POSITION.OUT'/
      DATA ECCEN,   R, TOLRNC, SICONV
     &     /  1.5,10.5,   0.01,  1.E-3/
      COMMON/GEOM/ECCEN,R
C
      WRITE(6,100)
  100 FORMAT(1X,'READ INPUT FILE NAME IF',
     &         ' OTHER THAN FOR026.DAT? (Y/N): '$)
      READ(5,101) ANS
  101 FORMAT(1A1)
C
      IF (ANS .EQ. 'N') THEN
         WRITE(6,102)
  102    FORMAT(1X,'READING FROM FOR026.DAT....')
      ELSE
         WRITE(6,103)
  103    FORMAT(1X,'TYPE NAME OF INPUT FILE FOR PHASING: '$)
         READ(5,104) INPUTF
  104    FORMAT(1A40)
      END IF
C
C Open the necessary files for input and output
      OPEN(UNIT=1, FILE=INPUTF, STATUS='OLD', READONLY)
      OPEN(UNIT=2, FILE=OUT1,   STATUS='NEW')
C
C Ask user what housing position he/she wishes to examine
      WRITE(6,150)
  150 FORMAT(1X,'HOUSING POSITION (0 - 360): '$)
      READ(5,*)POS
C
C Write the housing position in the output file
      WRITE(2,35) POS
   35 FORMAT(1X,'HOUSING POSITION= ',F5.1)
      WRITE(2,36)
   36 FORMAT(2X,'CA',4X,'PHASED CA',3X,'PRESSURE',4X,'GAS T',
     &        7X,'Q HOUS',8X,'Q TOTAL',8X,'HTRCOE',/,22X,'[atm]',
     &        6X,'[K]',11X,'[J/deg x 1.E3]',10X,'[W/m**2 K]')
```

```
C
C Read the first value in INPUTFile which is the print interval
      READ(1,119) TPRINT
  119 FORMAT(1F7.1,/,/)
C
C Start reading the CA and the corresponding chosen data values
      NCOUNT=1
  140 READ(1,120,END=130) CA, P, GAST, QTOTAL, QFRHO, QFRRO,
     &                    QFRSI, HTRCOE
C               CA        P         T         Q        QFR__      HTRCOE
  120 FORMAT(1F7.1,2X,F9.4,2X,F9.2,2X,F12.6,2X,3(F12.5,2X),F12.5)
C
C Skip the non-TPRINT integral values of CA
      IF (ABS(CA/TPRINT - IFIX(CA/TPRINT)) .GT. TOLRNC) GOTO 125
          DATAT(1,NCOUNT) = CA
          DATAT(2,NCOUNT) = P
          DATAT(3,NCOUNT) = GAST
          DATAT(4,NCOUNT) = QTOTAL
          DATAT(5,NCOUNT) = QTOTAL*QFRHO/100.
          DATAT(6,NCOUNT) = HTRCOE
          NCOUNT=NCOUNT + 1
  125 GOTO 140
  130 CONTINUE
C
C NLAST is the total number of data pairs which were read
      NLAST=NCOUNT - 1
      INC=1
      INDEX=0
      CA=-9.9999E10
C
C
C Start of loop to correlate values to housing position
C*****************************************************************
      DO 9000 I=1,NLAST
         GRAPH(1,I)=DATAT(1,I)
C
C Check for repeated values in output file (FOR026)
         IF( DATAT(1,I) .EQ. DATAT(1,I-1) ) INC=0
C
C Check for repeated values in phased values (CA)
   33    IF( CA .EQ. DATAT(1,INDEX+1) ) THEN
            INDEX=INDEX + 1
            GOTO 33
         END IF
C
         INDEX=INDEX + INC
C
         IF(INDEX .GT. NLAST) INDEX=1
         CA=DATAT(1,INDEX)
C
C Find where the apexes are...
C APEX1/2/3 are the angular geometric locations of the three apexes
         CALL APEX(CA,APEX1,APEX2,APEX3)
C
C Function subroutine to give the value of NJUMP.
```

```
C NJUMP is either 0, 1, or 2.
C NJUMP=0 Means that the housing position in-question is within
C         the two rotor apexes of the data file.
C NJUMP=1 Means that the housing position in-question is one chamber
C         ahead of the data file values.  Solution is to move ahead
C         in the reading of the data file values by 360 crank-angle
C         degrees (which is equivalent to moving one-chamber ahead).
C NJUMP=2 Means that the housing position in-question is two rotor-
C         chambers ahead of the data file values.  Solution is to move
C         ahead in the reading of the data file values by 720 CA
C         degrees (which is equivalent to moving two chambers ahead).
          NJUMP=NCHAMB(APEX1,APEX2,APEX3,POS)
C
          IF (CA .GT. ((DATAT(1,1) + 1080) - (NJUMP*360))) THEN
             CA=CA - (1080 - NJUMP*360)
             INDEX=INDEX - (NLAST - NJUMP*(NLAST/3))
          ELSE
             CA=CA + NJUMP*360
             INDEX=INDEX + NJUMP*(NLAST/3)
          END IF
C
C Find the exact index where DATAT(1,INDEX)=CA
  5       IF(JINT(CA - DATAT(1,INDEX)))10,30,20
 10       INDEX=INDEX - 1
          GOTO 5
 20       INDEX=INDEX + 1
          GOTO 5
 30       CONTINUE
C
C Calculate the increment.  Only necessary in the case where there is
C a skip in the FOR026.DAT file. (i.e. if there is output like:
C CA=33,34,35,37,.. the value of CA=36 is missing).
          INC = JNINT(( DATAT(1,I+1) - DATAT(1,I) )/ TPRINT)
          GRAPH(2,I)= DATAT(2,INDEX)
          GRAPH(3,I)= DATAT(3,INDEX)
          GRAPH(4,I)= DATAT(4,INDEX)
          GRAPH(5,I)= DATAT(5,INDEX)
          GRAPH(6,I)= DATAT(6,INDEX)
          WRITE(2,40) GRAPH(1,I), CA, (GRAPH(JJ,I), JJ=2,6)
   40     FORMAT(1F7.1,2X,F7.1,2X,F9.4,2X,F9.2,2X,F12.6,2X,
     &           F12.6,2X,F12.2)
C
 9000  CONTINUE
C*********************************************************************
C End of loop to correlate values.
C
       CALL GETXLAB (POS, XLAB)
       NPTS=NLAST
C
       YLAB= 'PRESSURE [ATM]'
       CALL PICTR (GRAPH, 6, XLAB//YLAB, XSCL, -2, NPTS, 1,
     &             00, 14, 1, 0.0, LOOK)
C
       YLAB= 'GAS TEMPERATURE [K]'
       CALL PICTR (GRAPH, 6, XLAB//YLAB, XSCL, -4, NPTS, 1,
```

```
     &              00, 14, 1, 0.0, LOOK)
C
       YLAB= 'H TRANS: TOTAL vs. HOUS [J/Deg x 1.E3]'
       CALL PICTR (GRAPH, 6, XLAB//YLAB, XSCL, -24, NPTS, 1,
     &              00, 14, 1, 0.0, LOOK)
C
       YLAB= 'HEAT TRANS. COEFF. [W/m**2 K]'
       CALL PICTR (GRAPH, 6, XLAB//YLAB, XSCL, -32, NPTS, 1,
     &              00, 14, 1, 0.0, LOOK)
C
C Close and save the data file
       CLOSE(UNIT=1,STATUS='KEEP')
       STOP
       END
C##########################################################################
C    End of main program
C##########################################################################

       SUBROUTINE APEX(CA,APEX1,APEX2,APEX3)
         COMMON/GEOM/ECCEN,R
         X(ANG1,ANG2)=-ECCEN*SIND(ANG1) + R*SIND(ANG2/3.)
         Y(ANG1,ANG2)=-ECCEN*COSD(ANG1) + R*COSD(ANG2/3.)
         SLUG(X,Y)=ATAN2D(X,Y) + 180.
         APEX1=SLUG(X(CA,CA       ),Y(CA,CA       ))
         APEX2=SLUG(X(CA,CA + 360.),Y(CA,CA + 360.))
         APEX3=SLUG(X(CA,CA + 720.),Y(CA,CA + 720.))
         RETURN
         END

       FUNCTION NCHAMB(APEX1,APEX2,APEX3,POS)
         LOGICAL HUNT(A,B,P)=((ABS(B-A) .LT. 180.) .AND.
     &           ((P .GE. A) .AND. (P .LE. B))) .OR.
     &           ((ABS(B-A) .GT. 180.) .AND. ((P .LE. B)
     &           .XOR. (P .GE. A)))
         NCHAMB=-1
         IF(LOGICAL HUNT(APEX1,APEX2,POS)) NCHAMB=2
         IF(LOGICAL HUNT(APEX2,APEX3,POS)) NCHAMB=0
         IF(LOGICAL HUNT(APEX3,APEX1,POS)) NCHAMB=1
         RETURN
         END

       SUBROUTINE GETXLAB(POS, XLAB)
         CHARACTER*40 XLAB
         CHARACTER*3  POSLAB
         CHARACTER    NUMBER(0:9)
         INTEGER      IDIG(3)
         DATA NUMBER/'0','1','2','3','4','5','6','7','8','9'/
         K= ININT(POS)
         DO 5 M=1,3
            IDIG(M)= K - (K/10)*10
            K=K/10
 5       CONTINUE
         POSLAB= NUMBER(IDIG(3))//NUMBER(IDIG(2))//NUMBER(IDIG(1))
         XLAB= 'HOUSING POSITION = '//POSLAB//'    CRANKANGLE'
         RETURN
         END
```

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TABLE 1

ENGINE GEOMETRIES

Engine Series	580	070
Displacement [cm^3]	5780.0	662.43
Eccentricity [cm]	3.175	1.542
Generating Radius [cm]	21.844	10.640
Rotor/Housing Width	15.875	7.711
Rotor Pocket Volume [cm^3]	355	
Intake Port	Peripheral	Peripheral
Port Timings [deg. ATC]		
TIPO	-627	-627
TIPC	-230	-230
TEPO	209	209
TEPC	611	611

TABLE 2

ENGINE OPERATING CONDITIONS

- 070 ENGINE -

BMEP [kPa]	RPM	Φ	P inl [atm abs.]	P exh [atm abs.]	DQDTMAX [1/J]
298	4000	0.378	1.081	1.057	0.0209
529	4000	0.528	1.147	1.047	0.0283
642	4000	0.618	1.214	1.067	0.0328
890	4000	0.675	1.358	1.074	0.0350

- 580 ENGINE -

BMEP [kPa]	RPM	Φ	P inl [atm abs.]	P exh [atm abs.]	DQDTMAX [1/J]
181	2400	0.222	1.054	1.113	0.0246
405	2400	0.305	1.190	1.145	0.0286
626	2400	0.380	1.372	1.218	0.0414
957	2400	0.474	1.593	1.278	0.0359

TABLE 3

CYCLE SIMULATION PERFORMANCE CASES

-CONSTANT HOUSING TEMPERATURE-

Descriptive Name	Case Name	RPM	Housing Temp [K]	Rotor Temp [K]	Side Plate Temp [K]
Cool All Around	A454	5000	425	500	425
Hot Rotor	A484	5000	425	800	425
Hot Housing	A854	5000	800	500	425
Hot Housing And Rotor	A884	5000	800	800	425
Hot All Around	A888	5000	800	800	800
Cool All Around And Slow Speed	A454SLOW	2500	425	500	425

-USER DEFINED HOUSING TEMPERATURE PROFILE-

Descriptive Name	Case Name	RPM	Housing Temp [K]	Rotor Temp [K]	Side Plate Temp [K]
Cool Rotor	AP54	5000	Profile	500	425
Hot Rotor	AP84	5000	Profile	800	425
Cool Rotor Slow Speed	AP54SLOW	2500	Profile	500	425
Hot Rotor Slow Speed	AP84SLOW	2500	Profile	800	425

TABLE 4: CYCLE SIMULATION PERFORMANCE RESULTS, PART 1

-CONSTANT HOUSING TEMPERATURE-

Descriptive Name	IMEP [kPa]	η_f[%]	η_v[%]	ISFC [g/kW•hr]	Mean Exhaust Temp [K]	Mass Of Fuel Inducted [g]
Cool All Around	1324	39.2	90.1	207	880	0.0436
Hot Rotor	1314	40.2	87.4	202	924	0.0422
Hot Housing	1294	39.4	87.4	206	912	0.0423
Hot Housing And Rotor	1284	40.4	84.8	201	964	0.0410
Hot All Around	1265	40.3	83.9	201	978	0.0405
Cool All Around And Slow Speed	1185	35.7	89.1	227	803	0.0429

-USER DEFINED HOUSING TEMPERATURE PROFILE-

Descriptive Name	IMEP [kPa]	η_f[%]	η_v[%]	ISFC [g/kW•hr]	Mean Exhaust Temp [K]	Mass Of Fuel Inducted [g]
Cool Rotor	1355	40.3	89.7	201	919	0.0435
Hot Rotor	1346	41.3	87.0	196	965	0.0421
Cool Rotor Slow Speed	1217	36.8	88.6	220	846	0.0426
Hot Rotor Slow Speed	1179	37.5	84.3	216	909	0.0406

TABLE 5: CYCLE SIMULATION PERFORMANCE RESULTS, PART 2

-CONSTANT HOUSING TEMPERATURE-

Descriptive Name	Conv. Heat Loss/Cycle [kJ]	Crevice Heat Loss/Cycle [kJ]	Total Heat Loss per Cycle / ($m_f \times Q_{LHV}$) [%]
Cool All Around	0.252	0.159	21.2
Hot Rotor	0.223	0.091	16.8
Hot Housing	0.181	0.167	18.5
Hot Housing And Rotor	0.151	0.094	13.5
Hot All Around	0.130	0.093	12.4
Cool All Around And Slow Speed	0.287	0.232	27.2

-USER DEFINED HOUSING TEMPERATURE PROFILE-

Descriptive Name	Conv. Heat Loss/Cycle [kJ]	Crevice Heat Loss/Cycle [kJ]	Total Heat Loss per Cycle / ($m_f \times Q_{LHV}$) [%]
Cool Rotor	0.172	0.168	17.6
Cool Rotor Slow Speed	0.191	0.242	23.2
Hot Rotor	0.141	0.094	12.4
Hot Rotor Slow Speed	0.156	0.137	16.3

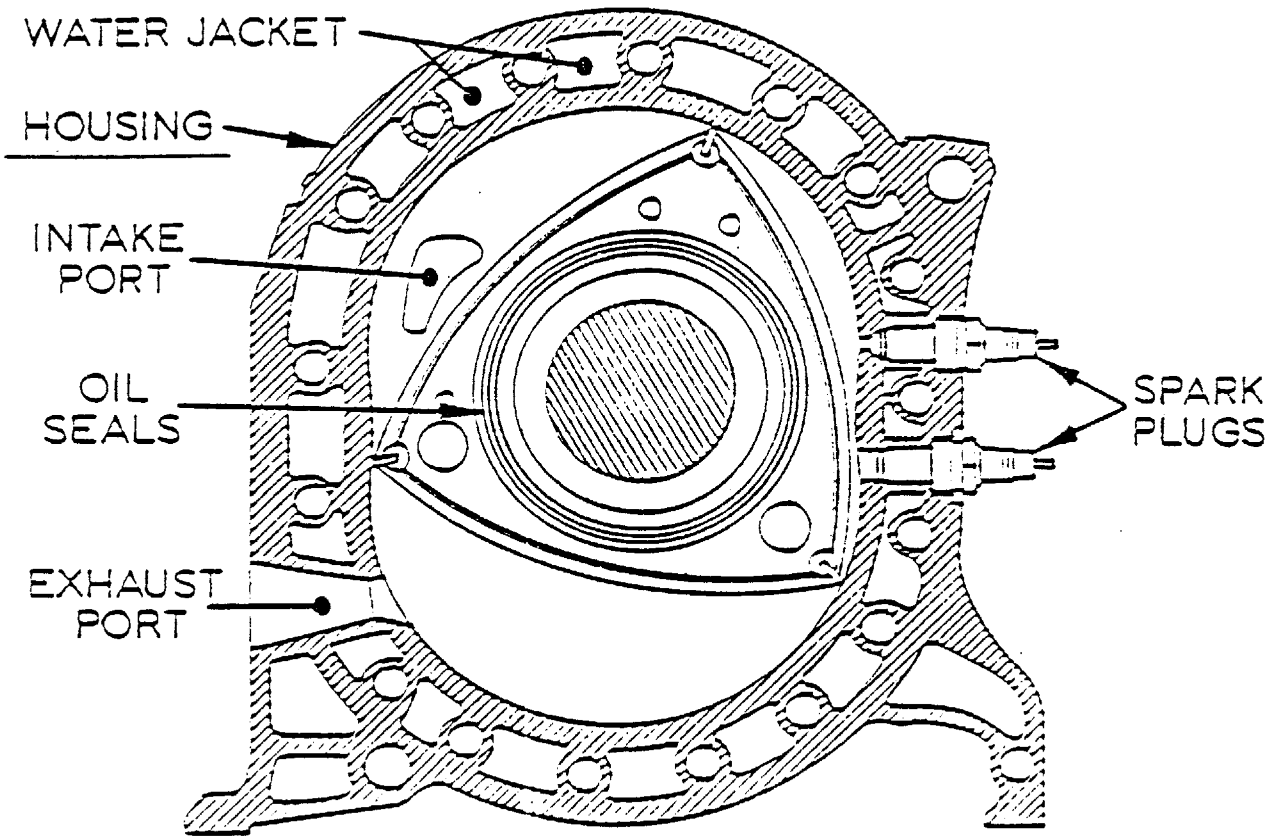
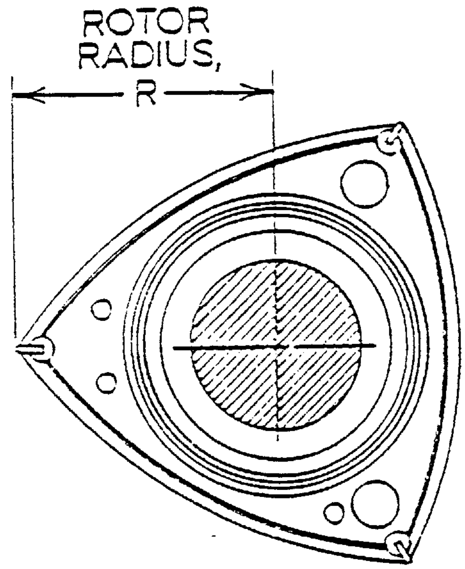


Figure 1. General Rotary Engine Arrangement



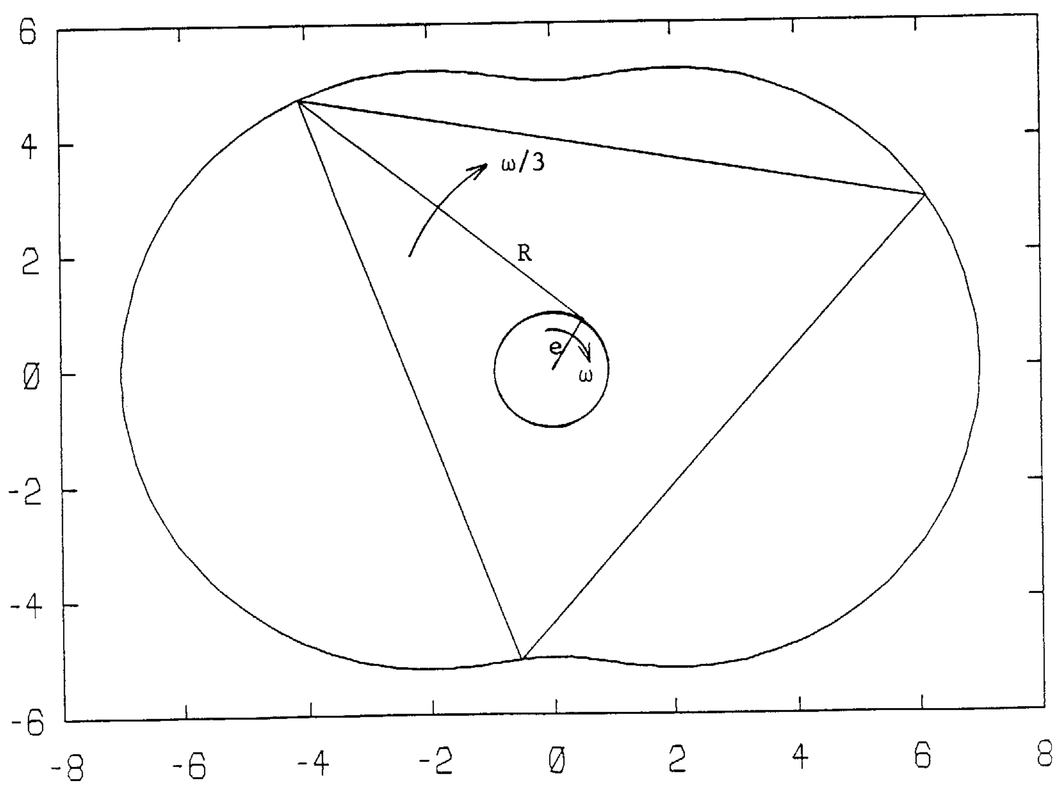


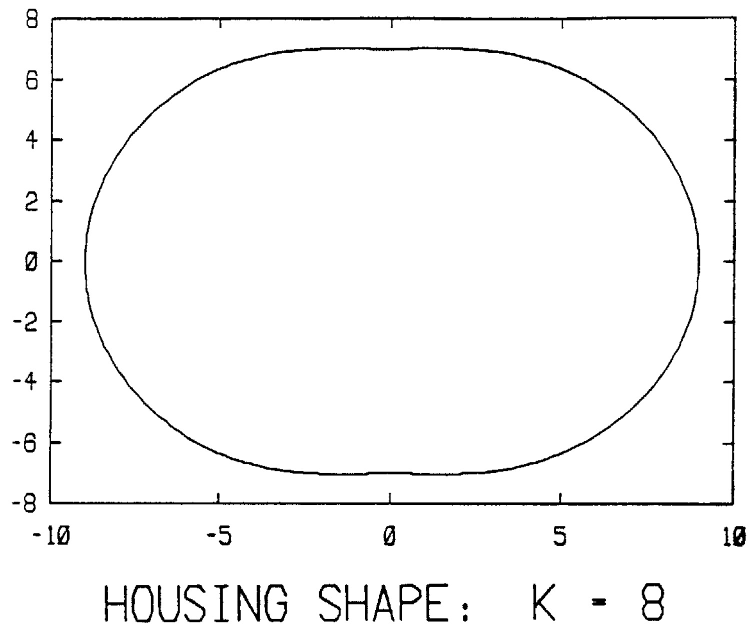
Figure 2. Rotary engine geometry. Rotor shape and epitrochoidal housing.

$$x = e \times \cos(\Theta) + R \times \cos(\Theta/3)$$

$$y = e \times \sin(\Theta) + R \times \sin(\Theta/3)$$

$$K = R/e$$

Typical Values of K: 6 to 8



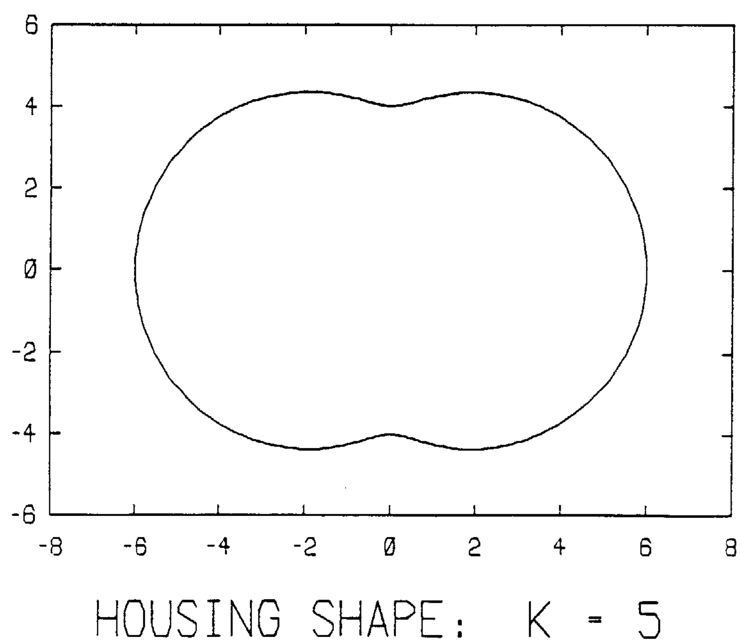


Figure 3. Equations for housing shape.
Effect of K on housing shape.

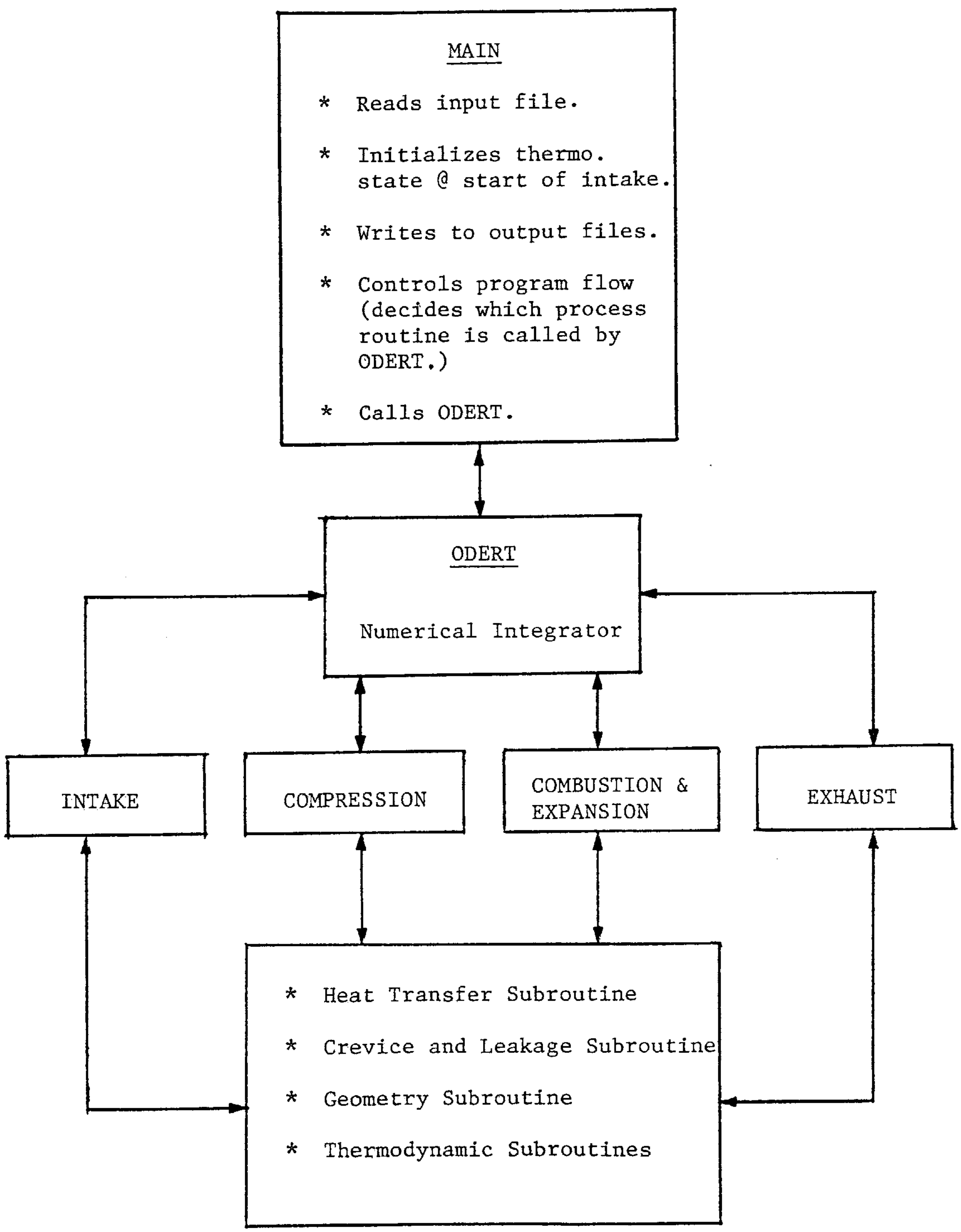


Figure 4. Flow chart for Wankel stratified-charge cycle simulation.

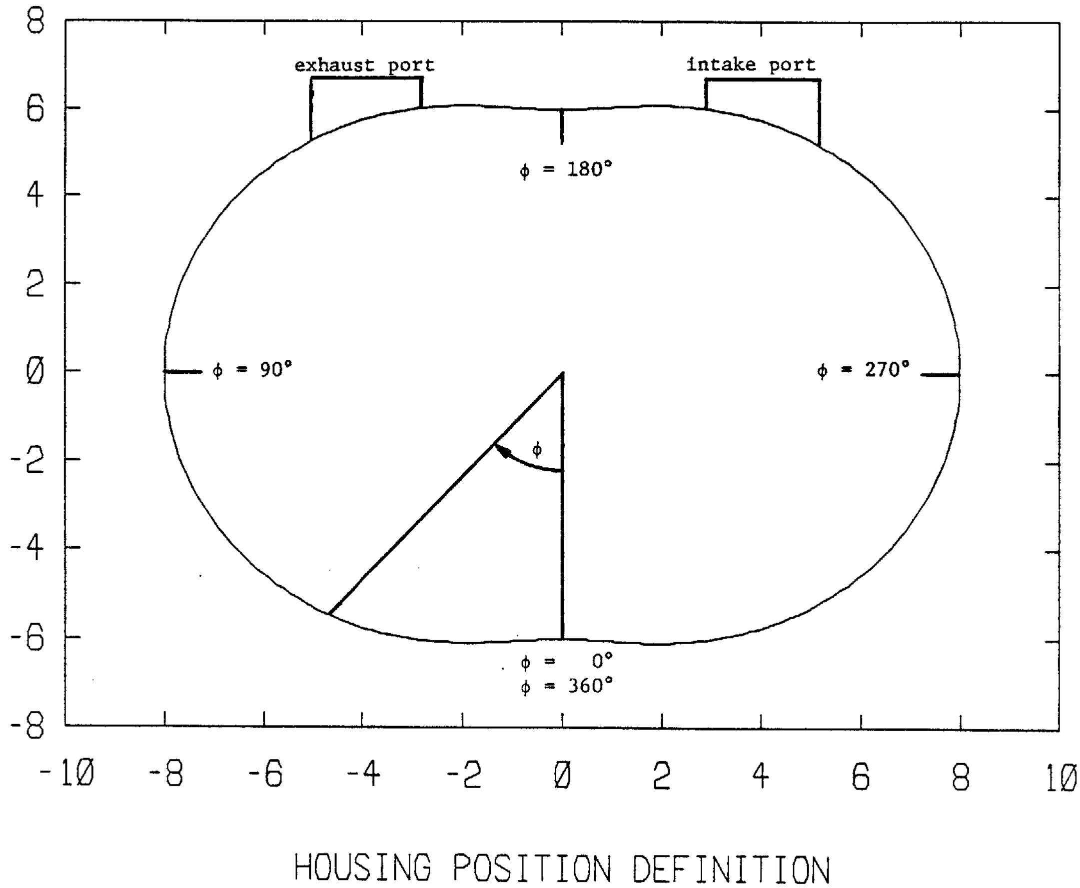


Figure 5. Convention for defining housing locations.

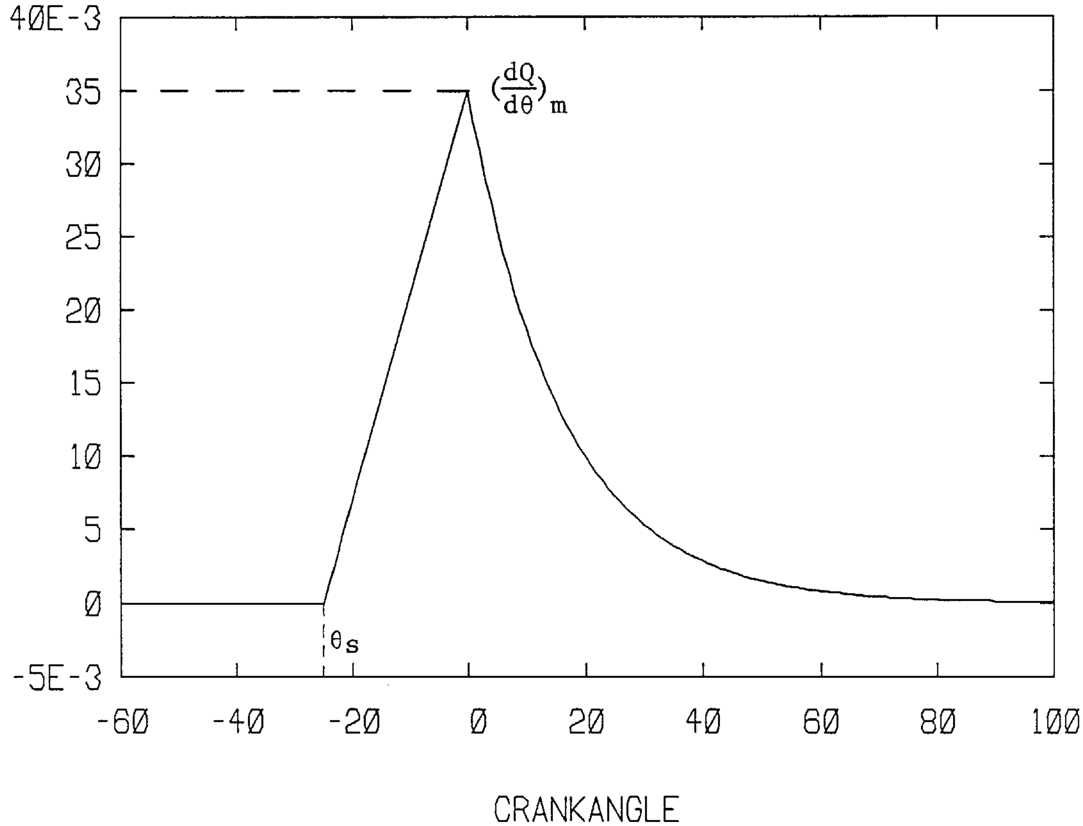


Figure 6. General shape of heat release model.

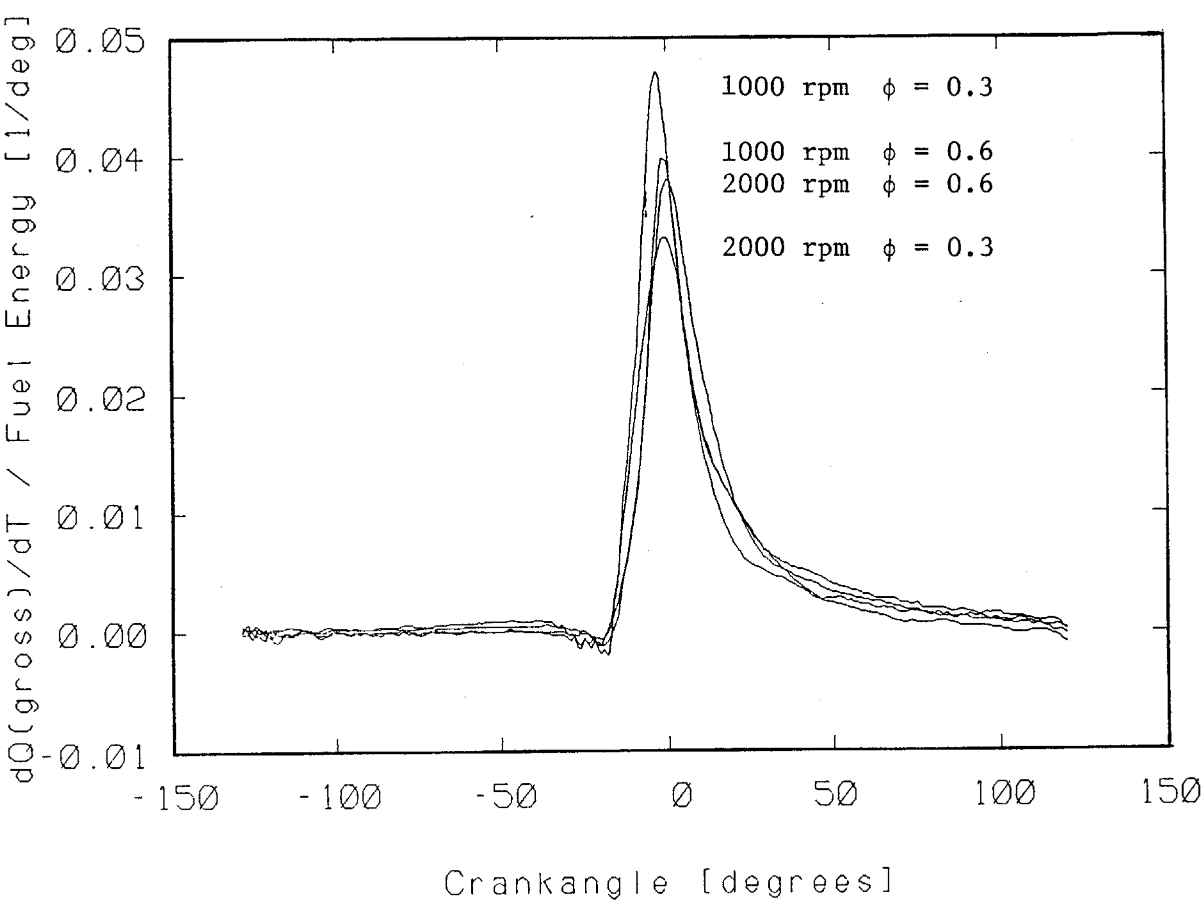


Figure 7. Normalized fuel-energy release rate as a function of crankangle from a reciprocating stratified-charge engine.

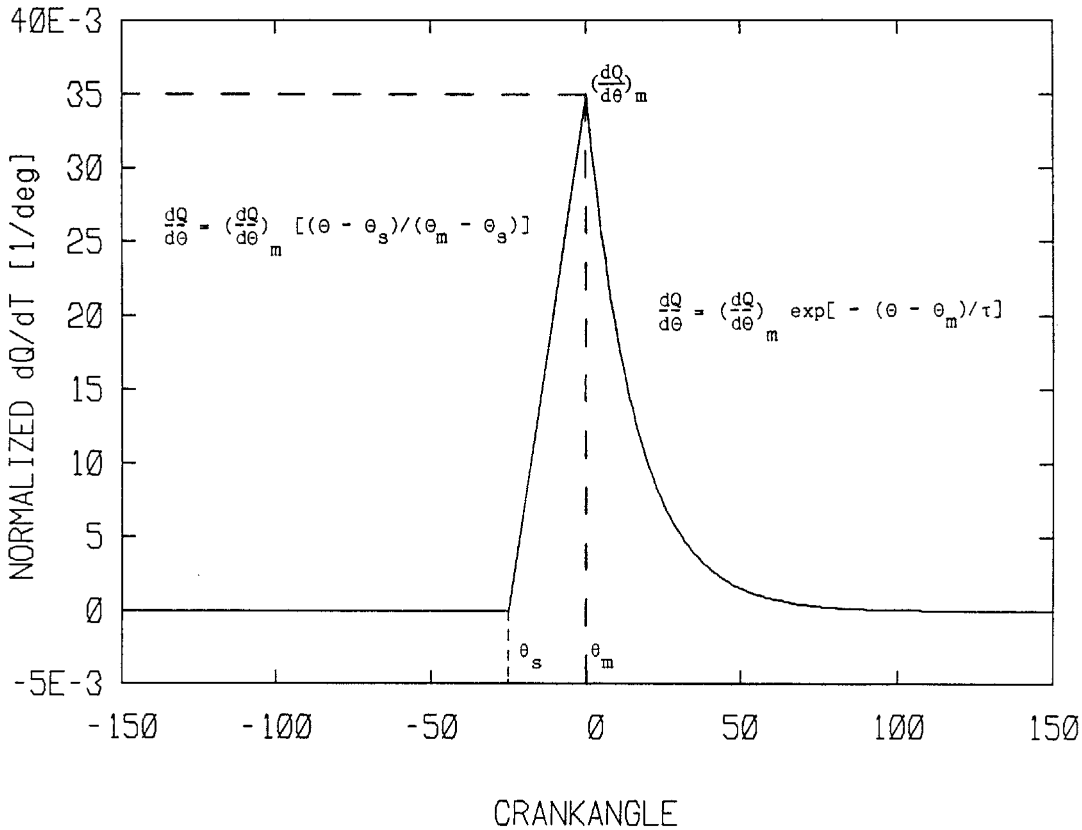


Figure 8. Model of normalized fuel-energy release rate for stratified-charge combustion.

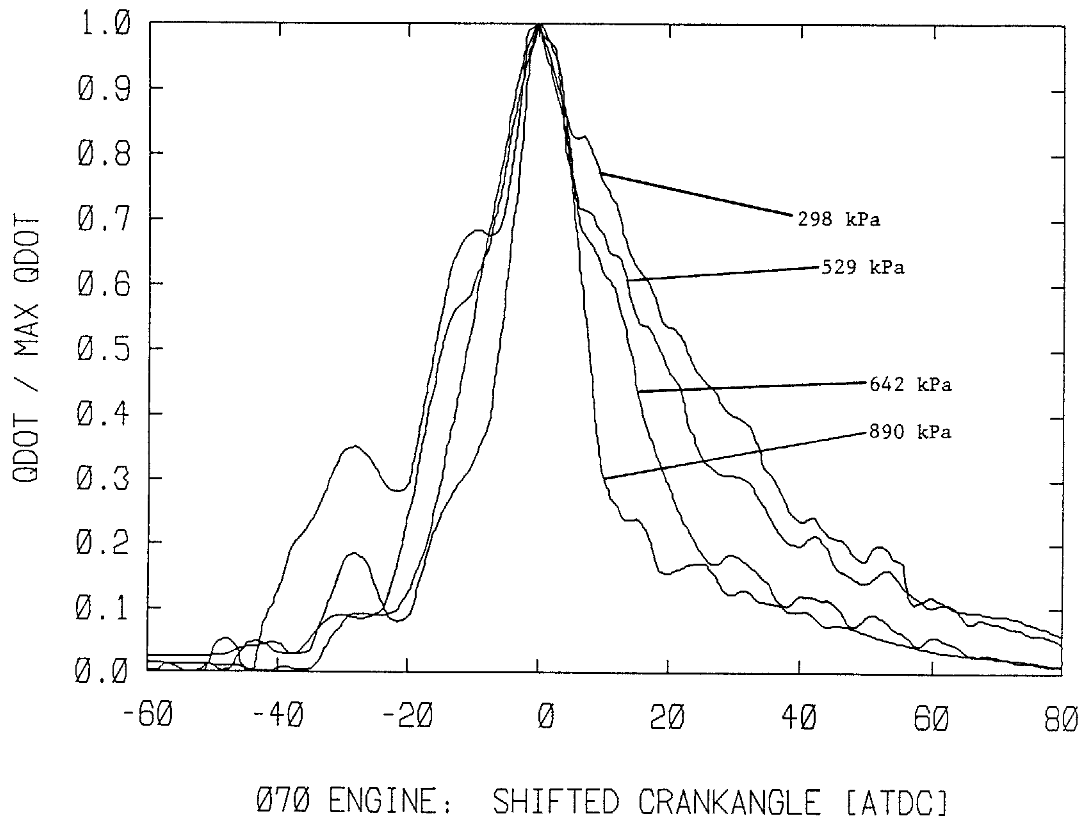


Figure 9. Unit normalized and shifted heat release curves from 070 engine used in shape comparison.

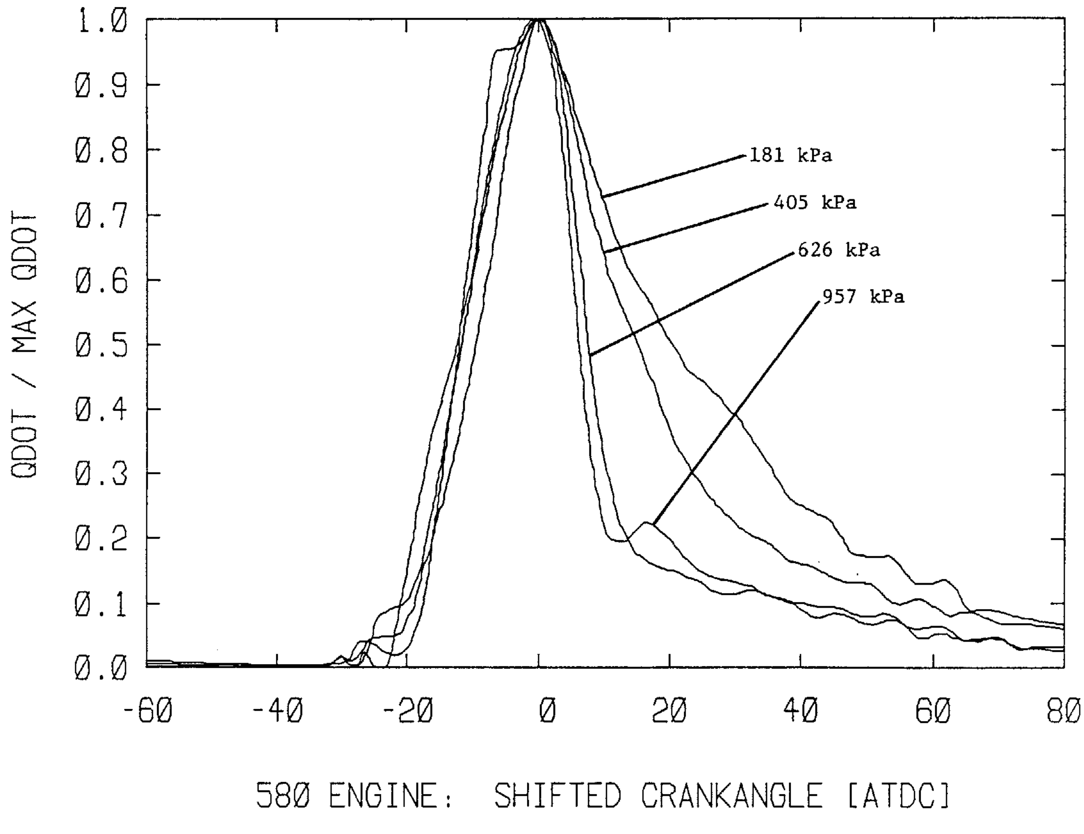


Figure 10. Unit normalized and shifted heat release curves from 580 engine used in shape comparison.

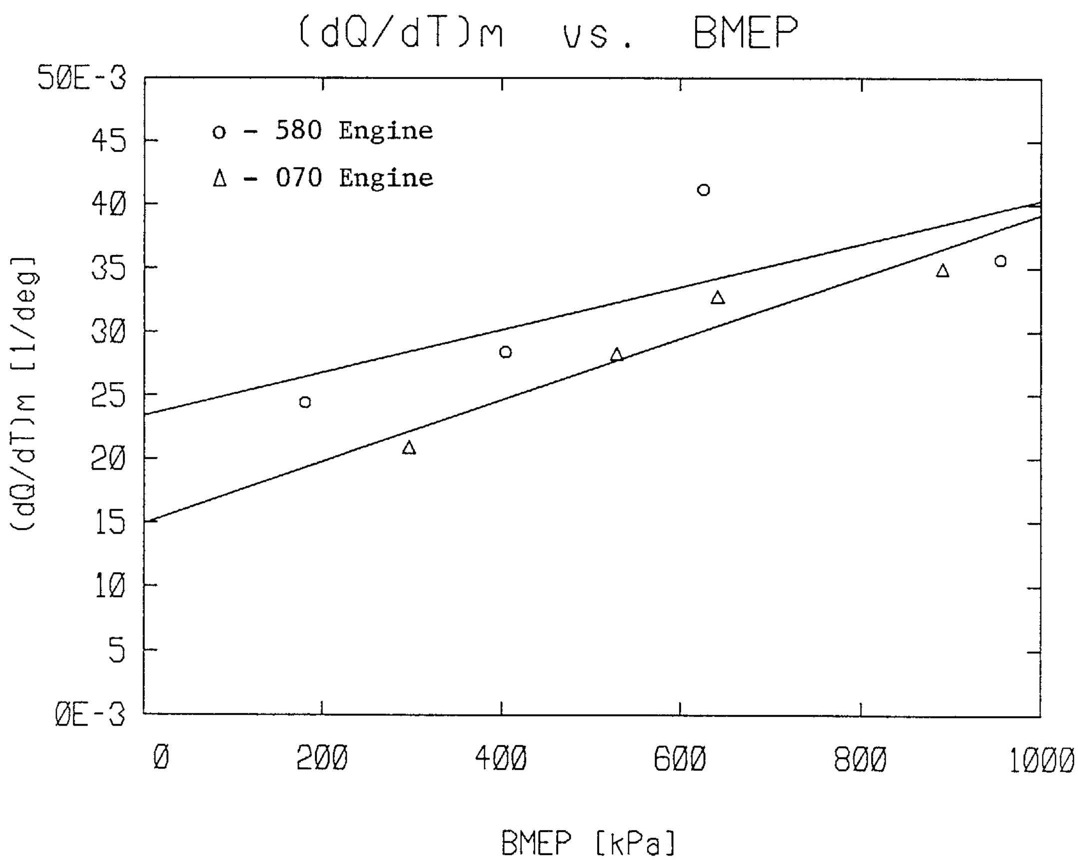


Figure 11. $(dQ/d\Theta)_m$ vs. BMEP for 580 end 070 engine.

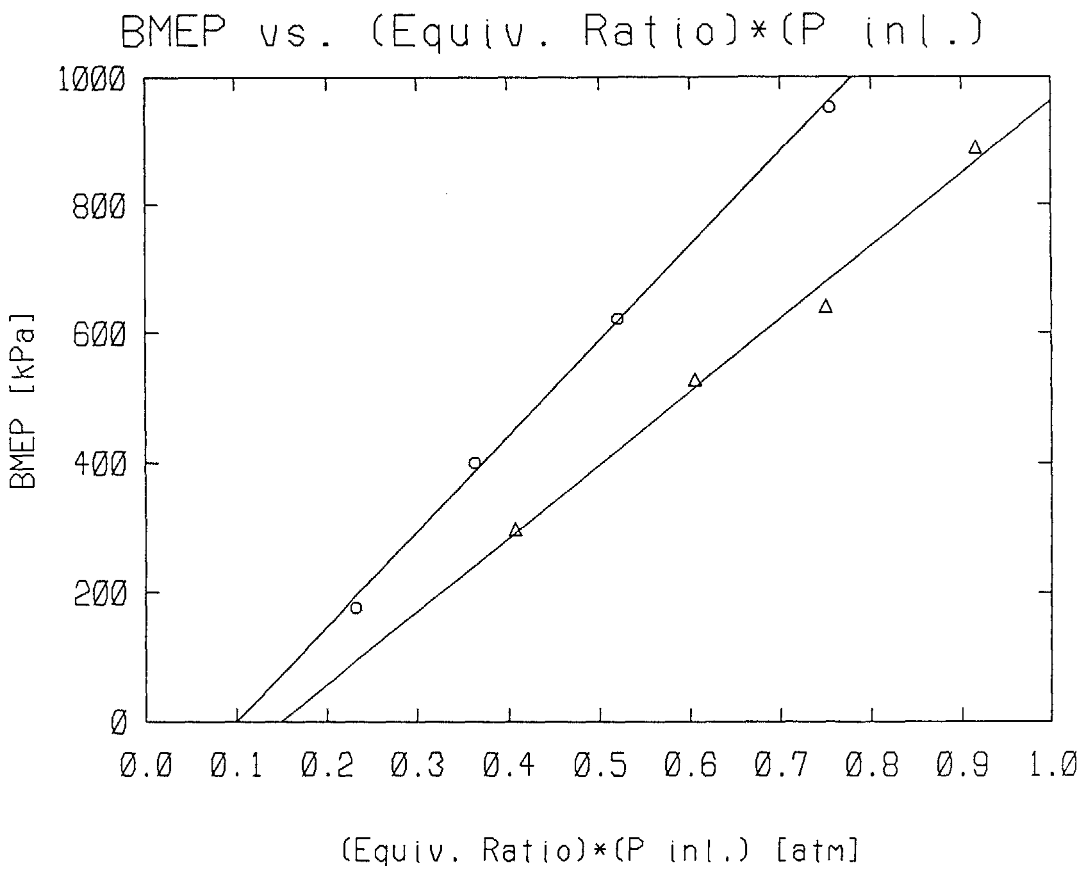


Figure 12. BMEP vs. ($\Phi \times P_{inl}$) for 580 and 070 engine.

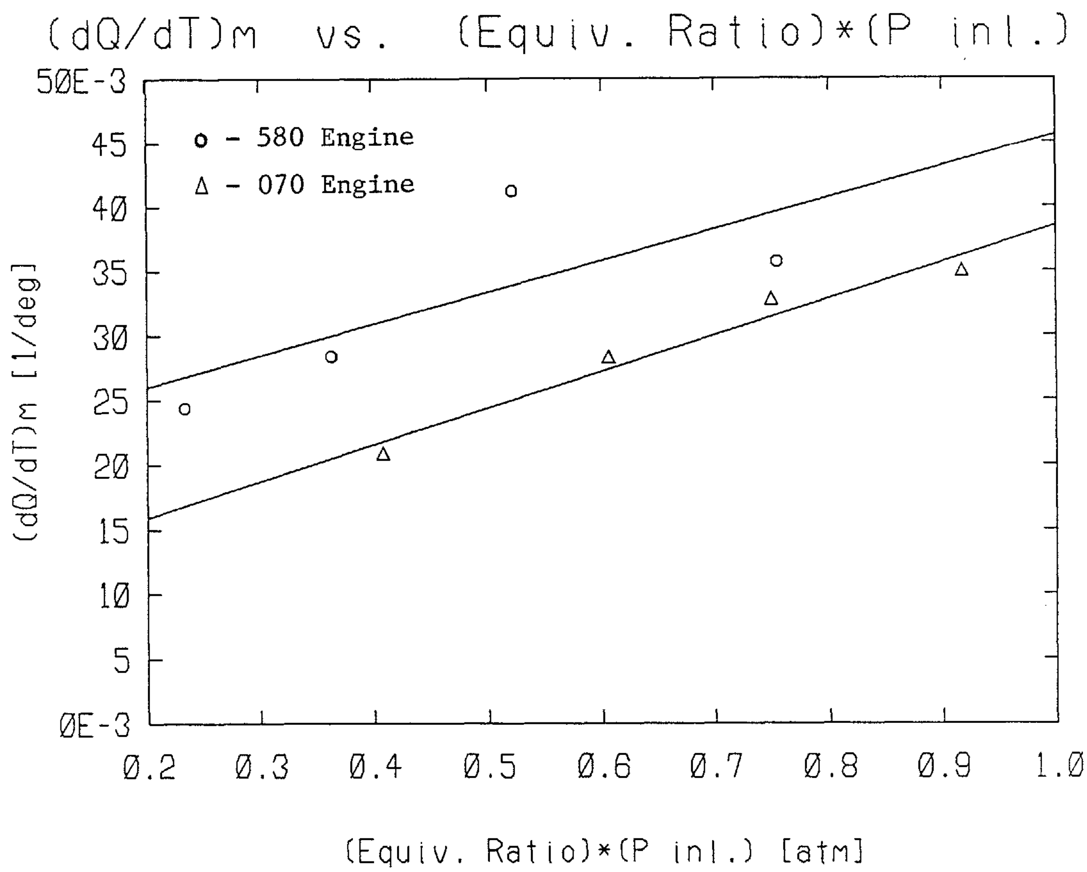


Figure 13. $(dQ/d\Theta)_m$ vs. $(\Phi \times P_{inl})$ for 580 and 070 engine.

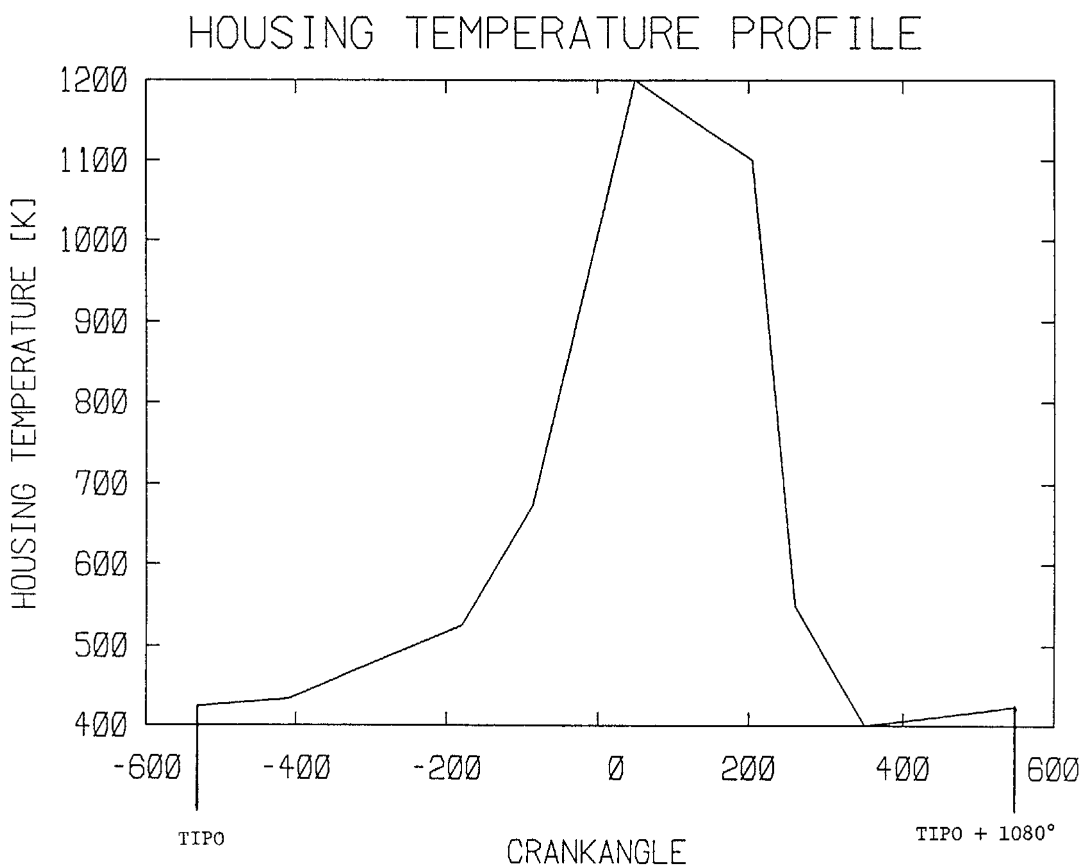


Figure 14. Housing temperature profile used in performance calculations.

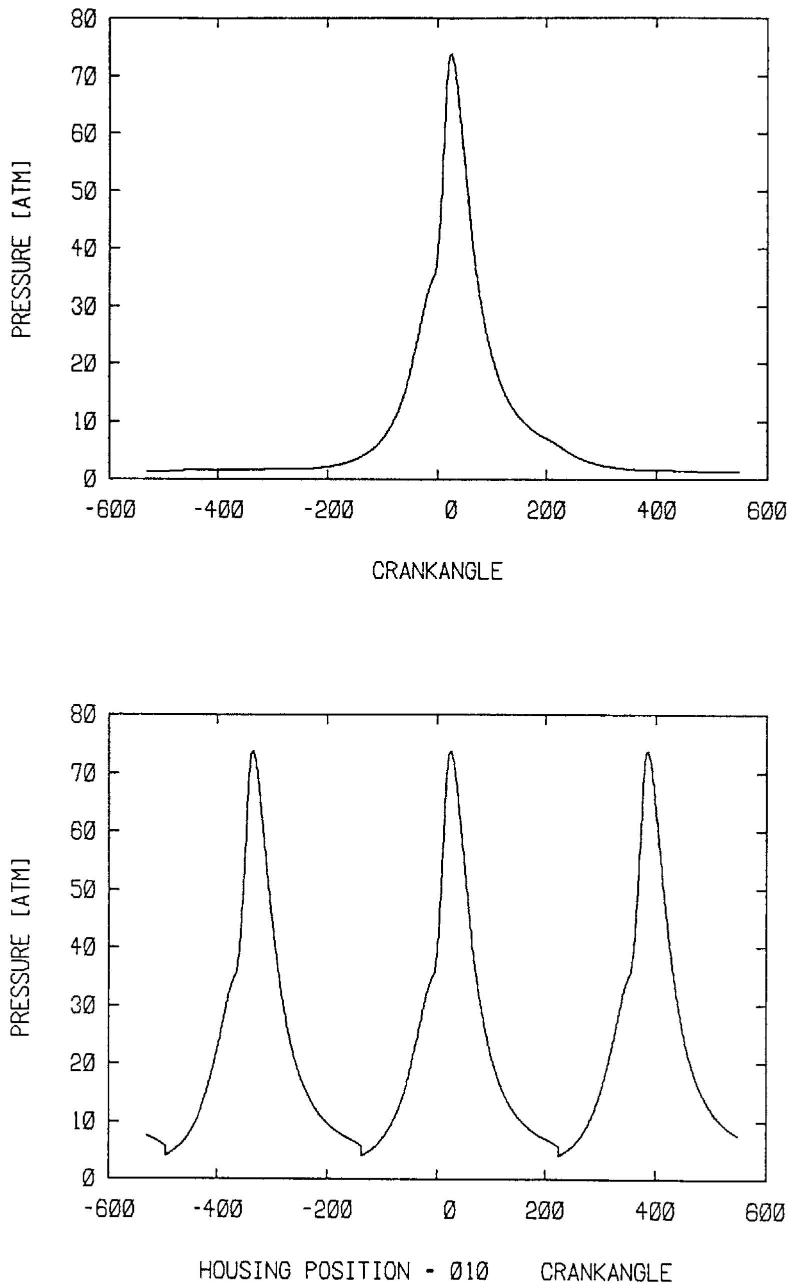


Figure 15. Chamber pressure with respect to the rotor and transformed with respect to housing location $\phi = 10°$

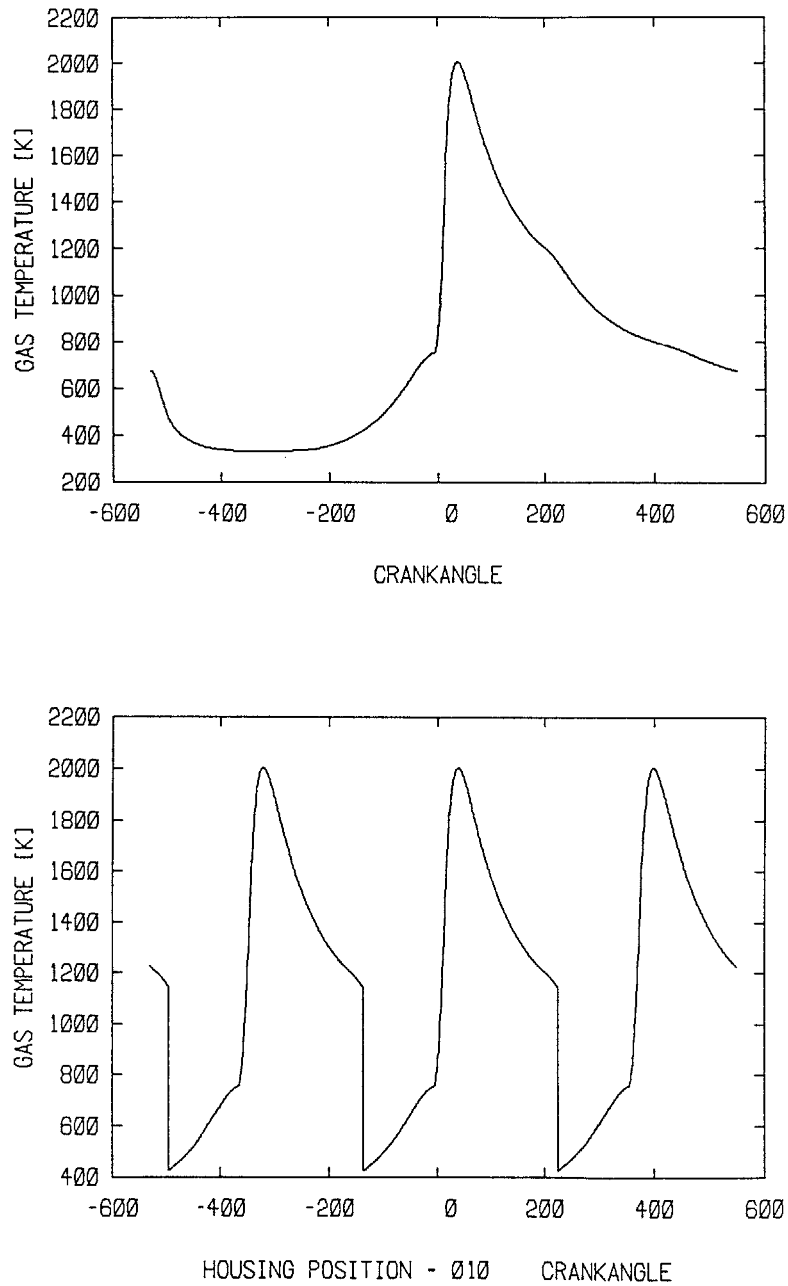


Figure 16. Gas temperature with respect to the rotor and transformed with respect to housing location $\phi = 10°$.

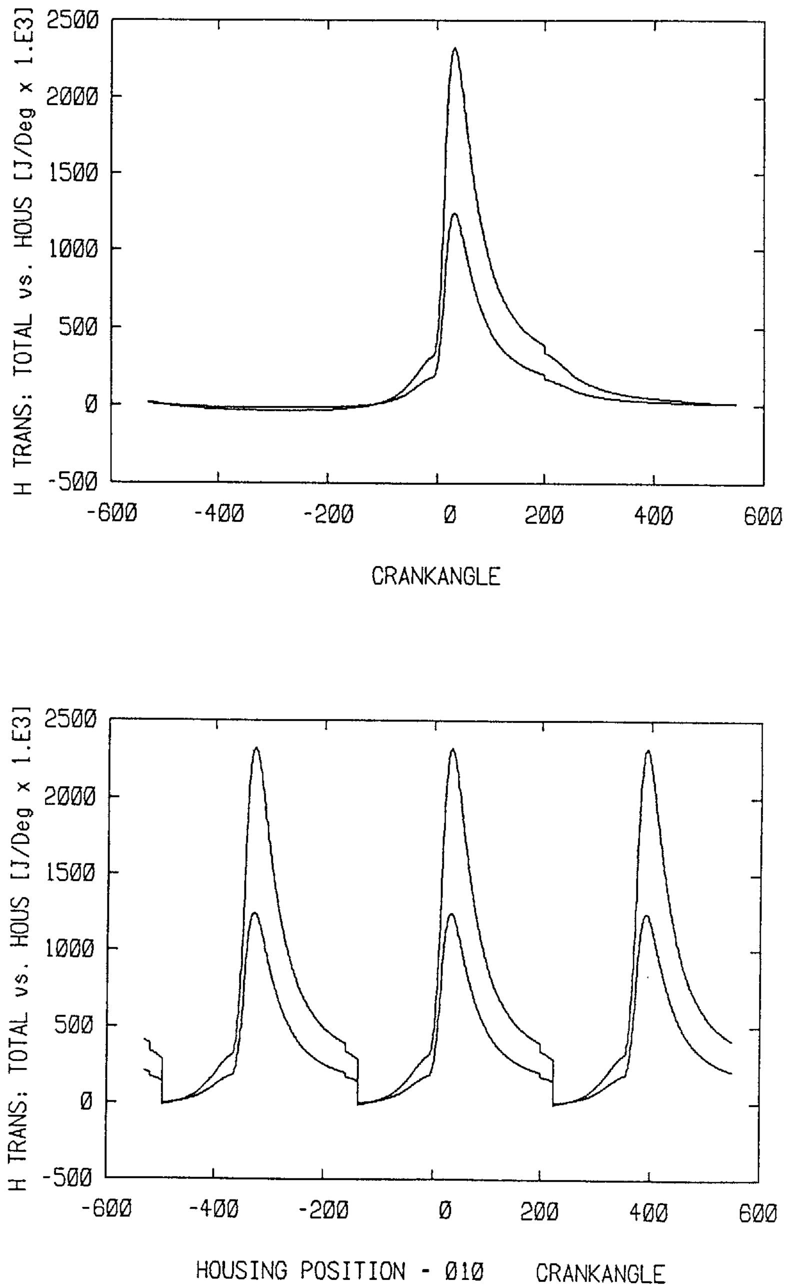


Figure 17. Heat transfer (total and to the housing) with respect to the rotor and transformed with respect to housing location ϕ = 10°.

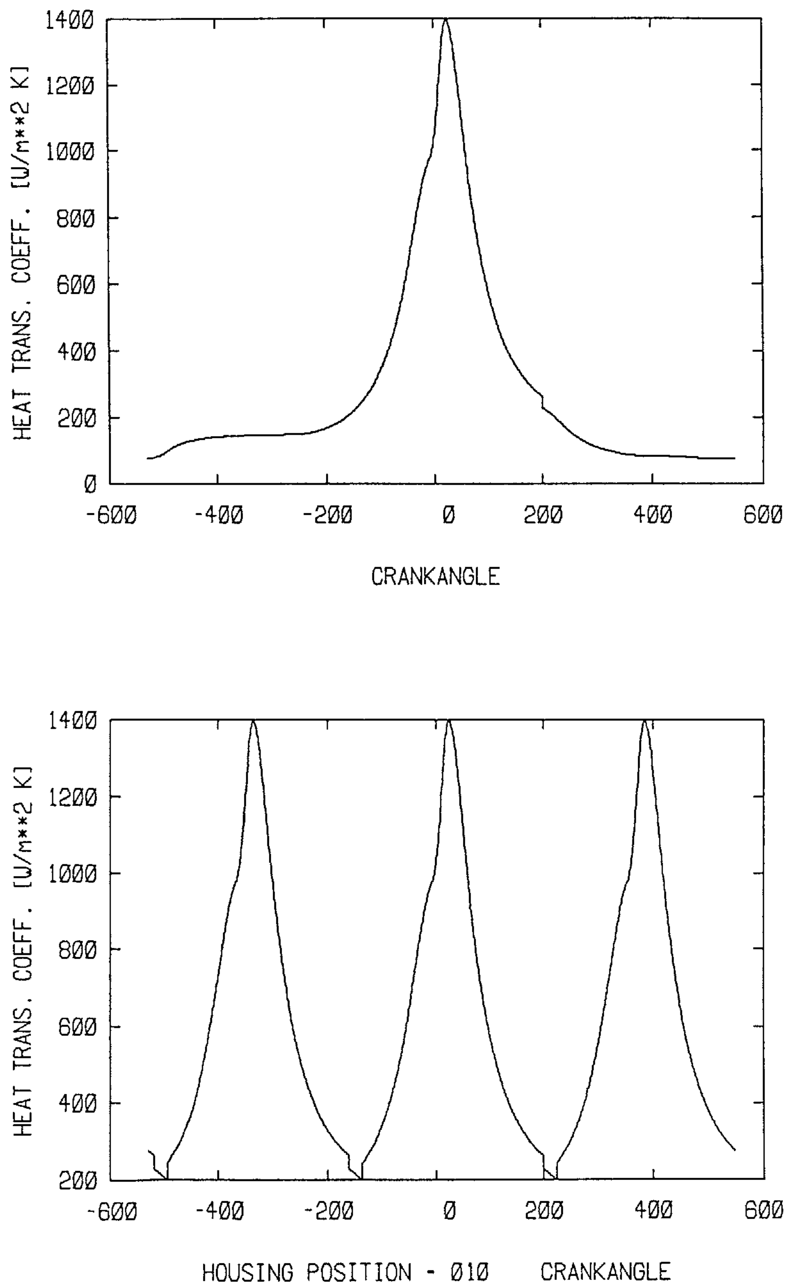


Figure 18. Heat transfer coefficient with respect to the rotor and transformed with respect to housing location $\phi = 10°$.