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Analysis of Combustion Chamber Design Variables in Planetary Rotary Engine

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ANALYSIS OF COMBUSTION CHAMBER DESIGN VARIABLES IN PLANETARY ROTARY ENGINE

by

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Sathyanarayanan Arumugam.

ANALYSIS OF COMBUSTION CHAMBER DESIGN VARIABLES IN PLANETARY ROTARY ENGINE

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This research entails examining the Hopkins Rotor Mechanism (HRM) for the perspective of commercial viability. Factors that were studied herein include basic operating characteristics, combustion effects, dimensional changes and behaviors unique to the HRM that may impose limits to commercial viability were also examined.

To compare specific characteristics of the HRM engine, in a comparable four-stroke mode, hypothetical 200, 400 and 600 cubic centimeter (cc) single cylinder reciprocating engines, each with compression ratio of 10, 15 and 20 were used as a basis for comparison. For similar outputs this resulted in the use of a 100, 200, 300 cc HRM configuration each with compression ratio of 10, 15 and 20. The geometric variables and combustion variables of The HRM combustion chamber is analyzed and compared with the baseline reciprocating engine.

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CHAPTER I

INTRODUCTION

Overview of Hopkins Rotor Mechanism

The Hopkins Rotor Mechanism (HRM) is a unique engine concept. It has prospective advantages over other IC engines in current use (Reciprocating Rotary spark ignition (S.I) and compression ignition (C.I) engines) in terms of specific power, specific weight and specific bulk. In some respects the HRM could challenge turbine engines in terms of the above characteristics.

The simple design concept and light weight of the HRM engine has potential to make it equal to or lower in cost than other IC engines in current use. The simplicity of this design can be expected to result in reduced maintenance. If it is to be a viable alternative in other respects, such as fuel economy (overall engine efficiency) or emissions, the HRM could be expected to match values of current IC engines in use.

The name ‘planetary rotary engine’ was chosen by Masterson in his sealing patent to identify an entire class of engines, which he described as ‘having three or more rotors that are radially displaced from the center’ and ‘rotate together to alternatively increase and decrease the volume of a chamber defined by the rotors’.

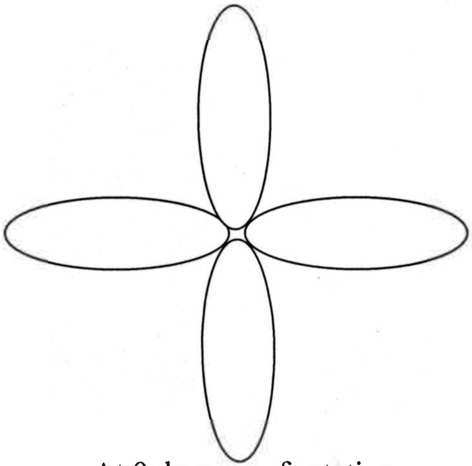
Unique Feature of the HRM Engine

The feature that differentiates HRM engine from other internal combustion engines is that it has one central combustion chamber surrounded by three or more rotors. During firing the combustion pressure is exerted on one side of each rotor, generating the power. The rotors run synchronously in the same direction, forming and reforming the combustion chamber. The combustion pressure is contained by each rotor maintaining line contact with adjacent rotors. This class of engines thus features rotors rotating on fixed axes, enclosing an interior space whose volume varies as the rotors revolve.

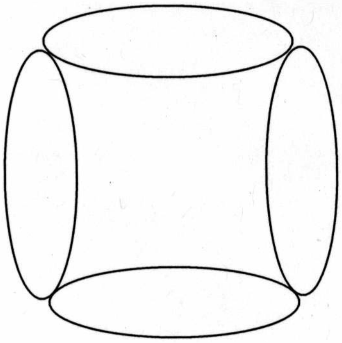
Rotors and Combustion Chamber

When a minimum number of three rotors is used, their cross-sectional shape is a 'rounded' triangle, and the space enclosed within them (the working volume) will alternate from minimum (when the rotor tips face inward) to a maximum (when the rotor flanks face inward) after 60 degrees of rotation. This will occur three times during one complete revolution of each rotor.

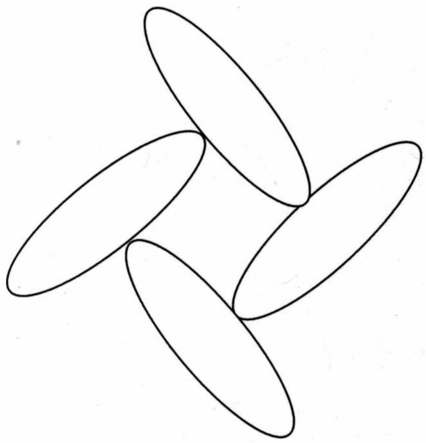
The use of four rotors may represent the optimum for the HRM class of engines. For this design the rotors are oval in cross section. The volume they enclose will vary twice during each revolution, from a minimum when the rotor tips face towards the engine centerline to a maximum, after 90 degrees of rotation, when the flanks face towards the engine centerline. The ratio of maximum to minimum enclosed volume gives the compression ratio of the engine. The end plates enclosing the combustion volume accommodate valves, spark plugs or injectors. A representative sketch of the movements of the rotors and the resulting variation of the space they enclose is shown in Figure 1.



At 0 degrees of rotation



At 90 degrees of rotation



At 45 degrees of rotation

Figure 1: Rotor Position at Various Degrees of Rotation

Rotor Shape and Characteristics

The rotors are cylindrical (i.e., their axial sections are constant throughout their length), geared to rotate in the same direction, and supported by bearings at their ends. The rotor cross-sectional shape that assures continuous line contact during rotation is an oval shown in figure 2. It is composed of four circular arcs (a shape commonly identified as the '4-center ellipse'), first described by Hopkins (Patent #2,097,881, dtd 11/2/37). Two identical arcs of a larger radius form the flanks of the oval. The other two circular arcs with a shorter radius form the noses. Each arc subtends an angle of 90 degrees, and the center of the arcs lie on a circle with a radius of r_0, whose center point is at the rotor center, as is also shown in Figure 2.

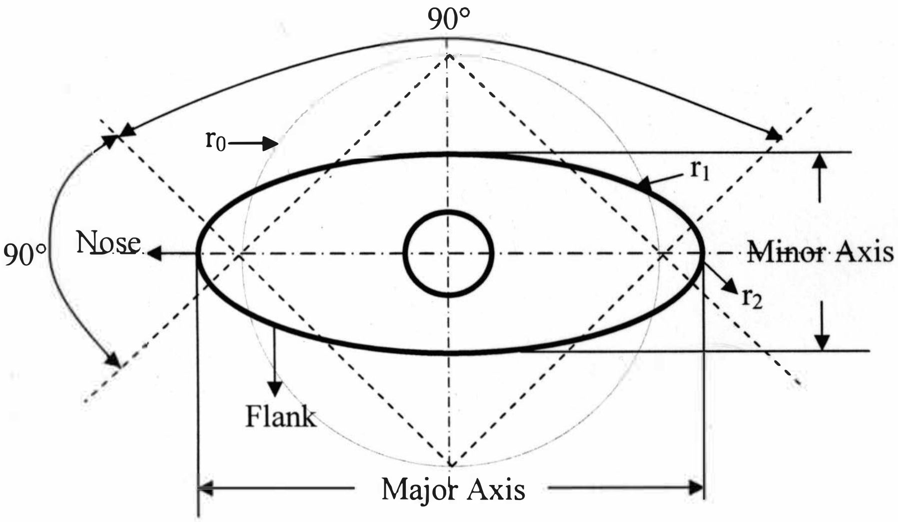


Figure 2: Cross Section of the 4-Center Ellipse Rotor

The selection of the two arc radii, r_1 and r_2, completely defines the geometry of the rotor cross section. When $r_1 = r_2$ the shape of the rotor defines that of a circle and as r_2 approaches '0' a sharp nosed oval shape is created. The nominal ratio (ratio of max to min area enclosed by the rotors) corresponding to these limits will vary between one and infinity, and the aspect ratio of the rotor cross section (ratio of major to minor axes) will range from 1.0 for the circle to 2.4142 ($=2^{1/2} + 1$) for the sharp nosed oval.

Casing of the Rotors

An exterior casing surrounding the rotors is not required, although prior patents dealing with similar mechanisms employ such a casing (or housing). The function of the housing may vary from the containment of the blowby to serving as a compressor for the charge air, or directing the exhaust to the exhaust manifold. End plates held at the requisite axial distance by means of the housing or tie-bolts, serve to seal the expansion space axially. The concept lends itself optionally to a valved or ported admission or discharge of the combustion gases, either through the end plates or through the housing.

Patent History of The Hopkins Rotor Mechanism (HRM)

HRM-type mechanisms have received limited attention and publicity. The relative obscurity probably accounts for the fact that this concept has been repeatedly invented. Infact the HRM-type mechanisms have been described in patents stretching back over nearly a century, for steam engines, internal combustion engines, pumps, compressors and expanders.

Patent searches on the HRM concept revealed that the earliest patent for the HRM-type mechanism was awarded in 1902 to T.S.Colbourne, (#710,756) for a steam engine (i.e., an expander). The earliest patent describing a PRE-type mechanism as a spark ignition (S.I) internal combustion engine pertains to Milton S. Hopkins (#2,097,881, dtd 11/2/37), who describes it with a four rotor concept. More recent patents were issued to Duane P. Snyder (#3,809,026 dtd 5/7/74, #4,934,325 dtd 6/19/90, and #5,341,782 dtd 12/21/93), Dietrich Densch (#4,968,234 dtd 11/6/90) and W. Biswell McCall (#5,341,782). The first Snyder patent describes a six-rotor arrangement with two working volumes. The second Snyder and the McCall patents employ the four-rotor concept, whereas the Densch patent deals with a three-rotor version. Topically the first two Snyder and the Densch patents deal with rotor-to-rotor sealing concepts; the first and the third Snyder patents and the McCall patent deal with internal passages in the rotors for directing the flow of intake charge and exhaust gases.

Advantages of The HRM

The major advantages of the HRM reside in the following aspects:

- The simplicity of the concept; the four identical rotors, and their exterior gearing, revolving about their axes are essentially the only main moving elements of the engine.
- The rotor and the engine are balanced about all three principal axes (radial, circumferential and axial).
- The combustion chamber defined by the rotors expands on all sides as the rotors rotate; only the end faces of the combustion chamber remain fixed. Therefore, the resulting working volume is large relative to the rotors and the entire engine. The result is a compact engine with the potential for high power output.
- The compact combustion volume will moderate heat losses through the engine structure.
- All the strokes of a 4-stroke cycle engine (intake, compression, expansion and exhaust) occur within one revolution of the rotors (as opposed to two crank revolutions for a piston engine).
- A family of HRM engines for different power outputs can be readily designed by expanding the length of the rotors and casing, while retaining all other design dimensions and features of the engine.
- The option of admitting the charge and evacuating the exhaust on the same end of the engine, or using a through-flow design with charge admission on one end, and exhaust discharge on the other is possible.

- A casing surrounding the rotors which captures all blowby, and can be used either for judicious exhaust recirculation schemes or for supercharging in a manner similar to the 'crankcase compression' approach utilized in some 2-stroke cycle engines.

The HRM concept lends itself optionally to the development of homogeneously fueled or stratified charge versions, with pre-compression chambers or direct ignition, and can be designed either as a spark ignition (S.I) or compression ignition (C.I) engine.

CHAPTER II

CHOICE OF HOPKINS ROTOR MECHANISM WORKING CYCLE

Recommended Cycle Choice

The Hopkins rotor mechanism (HRM) has the potential to operate as an Otto cycle, a modified Otto cycle (Atkinson, Miller) or as a Diesel cycle. Within the Otto and Diesel cycles there is potential for either a four-stroke cycle or a two-stroke cycle mode of operation. To examine cycle choice both primary and secondary influences were considered.

The basic HRM design is configured with four rotors, with relationships between each rotor maintained with a gearing system to provide common rotation angles and direction. When the major axes of adjacent rotors are at 90 degrees to each other, maximum volume is contained within. When opposite rotor tips face each other, minimum volume is contained within. The ratio of the maximum volume to the minimum volume establishes the maximum compression and/or expansion ratio for the engine. The HRM experiences two expansions and two compressions for each degree rotation of the rotors. This fundamental characteristic brings the engine from full expansion to full compression in 90 degrees of rotor rotation and from full compression to full expansion in the 2nd 90 degrees of rotor rotation. These could form the compression and power stroke cycles of the engine. From full expansion to minimum volume in the 3rd 90 degree of rotation could be used to expel the inert combustion gases. The 4th 90-degrees of rotation (minimum volume to maximum volume) could be utilized to intake the fresh charge. This 360-degree of rotation would then represent the 720-degrees of rotation of the conventional Otto cycle piston engine. The HRM then establishes, in a single chamber configuration, power output similar to the two-stroke cycle engine. Multiple chambers could also be connected in series to form the equivalent of a "multi-cylinder" engine with greater potential for power density.

In a two-stroke cycle mode of operation the single chamber HRM would provide multiple power pulses per revolution similar to a multi-cylinder two-stroke cycle piston engine. The two-stroke cycle would require "purging" the inert gases from the expanded volume near the end of the power stroke and subsequently "charging" the expanded volume prior to the start of compression. If the rotors were housing-enclosed, the potential for this to occur would be contained in the changing volumes between the rotors and the external housing.

The HRM as a Diesel-cycle could be operated as either a two-stroke Diesel or a four-stroke Diesel. Operation as a two-stroke Diesel would require a "pre-stage" pressurizing mode to assure full expulsion and filling of the cylinder while the rotors are positioned near the full expansion position. Each of these engine configurations will be examined in greater detail in the following discussion.

Diesel Cycle Configuration

The Diesel cycle is characterized by heat addition, which occurs during the expansion stroke. In the pure sense, this cycle is considered a constant pressure heat addition cycle implying that the heat release rate to the cylinder gases is at a rate which is commensurate with the volume expansion resulting in a constant pressure during the heat addition phase of the expansion (power) stroke.

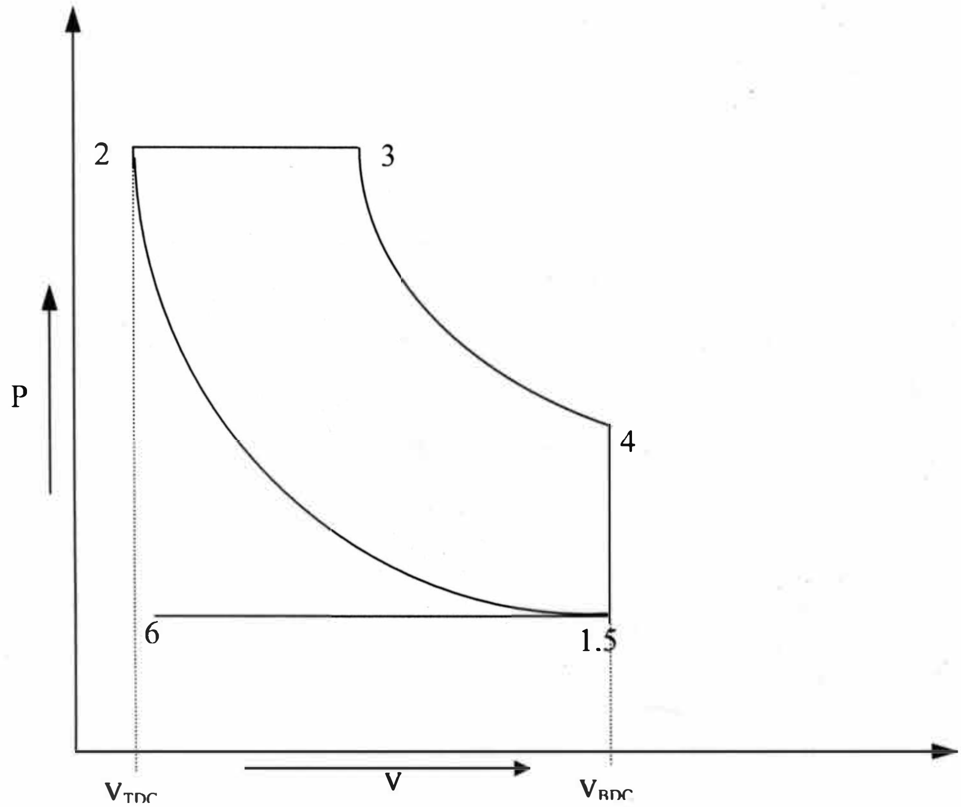


Figure 3: Constant Pressure P-V Diagram of the Diesel Cycle

The Diesel cycle is an unthrottled cycle, as a result power control is accomplished by metering the length of time that fuel is injected and that combustion occurs in the combustion chamber. With an unthrottled, naturally aspirated engine, the mass of air per intake stroke is similarly independent of the work demands from the engine. The mass airflow increases as a function of RPM, but does not change due to increased torque demands, unless the engine is super-charged or turbo-charged. This mode of operation then means that the Diesel cycle has wide variation in mean air/fuel ratios and these vary as the torque and power demands change. The fuel-air mixture must be stratified (non-homogeneous) to assure that combustion is initiated and maintained in a Diesel cycle. An ideal stratified charge would have a stoichiometric mixture at the point of ignition and flame propagation would progress to the leaner regions of the combustion chamber. This type of operation requires either a locally "rich" volume to exist during heat addition or propagation to occur in such a manner that combustion is continuous as fuel is added.

Two types of Diesel engine combustion chamber configurations are used to accomplish this task: the open chamber Diesel which utilizes direct injection and the pre-chamber Diesel which utilizes indirect injection. The open chamber Diesel utilizes controlled injection focused in a region where the charge can begin ignition. This is frequently in a region of the piston where a local stoichiometric mixture can be maintained for a pre-determined angular rotation of the crankshaft (Figure 4).

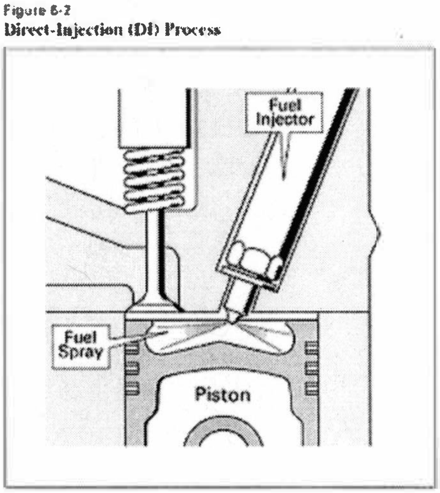


Figure 4: Open Chamber Diesel

(Source: Chevron 1998)

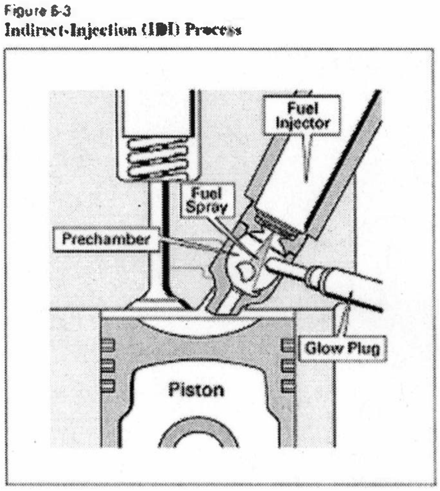


Figure 5: Pre-Chamber Diesel

(Source: Chevron 1998)

The second type of Diesel engine is the pre-chamber Diesel that utilizes a secondary chamber adjacent to the engine cylinder (main chamber). The fuel is injected into the secondary chamber (Figure 5). This pre-chamber experiences the same pressures as the main chamber but allows the fuel-air mixture to be fuel rich (near stoichiometry) to aide in combustion initiation. Due to the complex air motions during compression and expansion of the HRM and other rotary engines, maintaining a stratified charge during combustion would be extremely difficult with an open chamber design. If the Diesel cycle were to be the cycle of choice for the HRM, and the engine were to be designed such that power regulation was important, the pre-chamber Diesel is the logical choice. The pre-chamber Diesel, however, is commonly considered to be a lower efficiency engine due to the transfer efficiency from the pre-chamber to the main chamber and the turbulence that exists in the pre-chamber itself. The pre-chamber engine usually requires a glow plug and higher compression ratios to enable starting and improved efficiency.

Four Stroke Diesel Cycle

The four-stroke Diesel engine operates similarly to a four-stroke gasoline engine. The primary difference between the four-stroke Diesel and the four-stroke gasoline (Otto cycle) is that the Diesel engine operates over a wide range of mean air-fuel ratios usually lean of stoichiometry. The gasoline engine, in contrast, operates at, or near, stoichiometry with power controlled by throttling of the intake charge. The four-stroke Diesel, because of its definite and distinct intake, compression, power and exhaust strokes, typically produces cleaner emissions than the two-stroke cycle and accommodates either naturally aspirated or supercharged intake processes. An intake charge impeller or a compressor could be configured in the HRM to serve the purpose of supercharging the intake gases before they enter the combustion chamber.

Two Stroke Diesel Cycle

The two-stroke Diesel engine operates similarly to a four-stroke cycle Diesel engine except that it does not have separate intake and exhaust strokes. Since the exhaust expulsion and intake introduction occurs during the end of the expansion stroke and the early part of the compression strokes, the intake air is almost always boosted above atmospheric pressure. This is done with a turbocharger or a supercharger. The two-stroke Diesel generates more power (higher power density) but also produces higher emissions.

Otto Cycle Configuration

The Otto cycle engine has its power controlled by throttling the intake charge. Throttling the intake charge reduces the density of the inlet charge on the downstream side of the throttle and, as a result, the mass flow into the engine cylinders is reduced. Fuel is injected into the engine proportional to the mass of air. With modern electronic controls, internal combustion engines operate very close to stoichiometric mixtures for most of the engine operation. Throttling work, done during the intake process, may reduce the efficiency of these engines during part-power operation.

Four Stroke Otto Cycle

This mode of operation for the HRM resembles that of an ideal four-stroke Otto cycle engine. It consists of four strokes: intake, compression, power and exhaust. The cycle is completed once every two revolutions of the crankshaft. The four-stroke cycle, depending on how the mass flow into the engine is controlled, may require a substantial amount of additional work done during the intake process, thereby reducing its efficiency. If conventional throttling is applied as a means of power control on the HRM design, these losses would still be present. If a delayed start of compression is employed to reduce the mass in the chamber, throttling losses could be minimized and part power efficiencies could be improved on the HRM. The delayed start of compression could be accomplished as a late intake closure process.

Two Stroke Otto Cycle

The primary difference between the four-stroke and the two-stroke cycle is that in the two-stroke cycle configuration, there is no separate induction and exhaust stroke. For a given engine size operating at a particular speed, the two-stroke engine is more powerful due to twice as many power strokes per unit time. In the two-stroke engine, the gases at the end of the power stroke have to be expelled and the fresh intake charge introduced without any loss of intake charge out of the exhaust. These two processes have to occur while the chamber is near full expansion position. This is accomplished in reciprocating engines through the use of tuned exhaust, exhaust port control and a transfer design that moves the charge from below the engine piston to the combustion chamber as the piston moves from top dead center (TDC) to bottom dead center (BDC) on the power stroke. During the piston movement from TDC to BDC, the volume in the crankcase is decreasing. The two-stroke operation is possible in the HRM if the outer portion of the rotors are enclosed and the changing volume between the rotors and the case are used as a means to pressurize the intake or a separate intake charge impeller is utilized.

CHAPTER III

TK – SOLVER PROGRAM FOR COMBUSTION CHAMBER ANALYSIS

Introduction to TK – Solver Design Software

TK – Solver is a mathematical as well as a programming tool. It has two characteristics, first TK's declarative allows a user access to all the equations and formulae in any order and without the need to isolate variables and secondly TK solves directly, iteratively or by using a guess value. A single equation or a group of equation can be solved forward or backward. Input and output values can be switched instantly and the equation can be solved again. TK – Solver has a powerful list – solving feature, allowing solution over a range of data'. It calls and solves compiled routine in C and FORTRAN. User can create their own functions for procedural and list-solving operations.

Advantages of TK – Solver Programming

- It is easy to program. No algebraic syntax and assignment statements.
- It solves multidirectionally. Solves for any combination of input and output values without any extra programming.
- It list-solves over a range of inputs to generates data for tables and plots.
- It does table look-ups. It can also interpolate with in data tables.

- TK combines the best of rule-based declarative and procedural programming.

Analysis of HRM Combustion Chamber using TK – Solver

Considering the advantages of TK – Solver, mainly the list – solving features, a program has been written using TK–Solver design software to find the instantaneous surface area of the combustion chamber, volume of the combustion chamber, compression ratio, fraction of fuel burned depending on the start and end timing of combustion, temperature without heat addition, pressure without heat addition and moment for each degree rotation of the rotor. This program requires input values such as Angle of rotation of the rotor, length of major and minor axes, length of the rotor, radius of flank and radius of tip, initial temperature, start and end timing of combustion, Weibe form factor, calorific value of fuel, stoichiometric air-fuel ratio, specific energy and heat ratios for air. This program generates values automatically for each degree of rotor rotation ie., from 0 to 360 degrees. This program also produces graphs that show the variations of the main parameters such as area and volume of combustion chamber, pressure in combustion chamber, temperature change in the combustion chamber and the change in moment in the combustion chamber with respect to the angle of rotation of the rotor.

TK – Solver Program

This program calculates the instantaneous values of all the variables that are involved in the combustion chamber analysis of the Hopkins rotor mechanism. This program generates values for the variables for each degree of rotor rotation. Various steps, procedures and rules have been used in this program to calculate these values. The steps, procedures and rules that are used in this program to calculate the variables are explained below. In this program there are two sets of variables that are used for generating the values for 360° rotation of the rotors. They are input and output variables.

The input variables that are necessary for executing the program are angle of rotation, major axis, minor axis, length of the rotor, major radius, minor radius, start angle of heat release, duration of heat release in terms of angle, closing angle of intake port, opening angle of exhaust port, Weibe form factor, initial temperature, calorific value of the fuel, stoichiometric air fuel ratio and specific heat capacity of air at constant volume.

The output variables that are generated by the program are specific energy of the fuel, heat added during combustion, instantaneous area of combustion chamber, instantaneous volume of combustion chamber, instantaneous surface area of combustion chamber, area of the rotor, compression ratio of the engine, instantaneous compression ratio for each degree rotation of the rotor and instantaneous values of burn fraction, temperature without and with heat addition, pressure without and with heat addition and total moment produced by the four rotors during combustion.

To make this program executable three constants are used as program functionality constants. They are declared by characters ‘a’, ‘b’ and ‘c’. These characters cannot be changed by the user and these are fixed in such a way that they change automatically with respect to the input variables. This program automatically generates graphs for the instantaneous values that are calculated for each degree of rotor rotation. The graphs are generated for 360°, which is one full rotation of the rotor that includes two full compressions and two full expansions.

Engine specification that is used for generating the instantaneous values and to plot the graphs is given in Table 1.

200 cc Engine with Compression Ratio 20			
Rotor Specifications		Combustion Specifications	
Major axis	10.2 cm	Start of Heat release	150 degree
		Duration of Heat release	40 degree
Minor axis	5.948 cm	Closing of intake port	90 degree
		Opening of exhaust port	270 degree
Length	5.612 cm	Weibe form factor ‘n’	6
		Initial Temperature	298 K
Major radius	6.6 cm	Calorific value of fuel	44,000 KJ/Kg
		Stoichiometric Air Fuel ratio	14.8
Minor radius	1.47 cm	Cv of air	0.715
		Case clearance	0 cm

Table 1: 200 cc HRM Engine Specification with CR 20

Cross Sectional Area and Volume of Combustion Chamber

Area of combustion chamber of the HRM is formed by the interaction of the four rotors as shown in Figure 6. The cross sectional area of combustion chamber is as shown in Figure 6. The illustration depicts the rotors approximately between full compression and full expansion ($\approx 45^o$). Figure 6 therefore depicts the areas where both flank and tip radii are included.

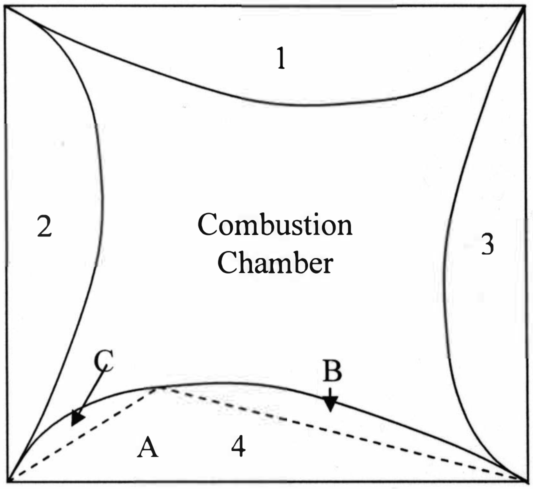


Figure 6: Cross Sectional Area of HRM Combustion Chamber

Combustion chamber area is calculated by assuming a square section that intersects the rotor surface to form the combustion chamber. This is explained in Figure 6 by means of a cross sectional diagram. The rotor surface area that is included in the square section is divided into three regions and these regions are divided to include the flank radius R_f and tip radius R_t. This is represented in the diagram as B and C respectively. The third region represented as A (between the dashed lines and the horizontal axis of the square) is the interaction area that is formed by the areas made by R_f and R_t. From this the combustion chamber area is given by the formula as follows,

Area of Combustion Chamber = Area of Square – 4 ×Area of the Rotor Sector

In Figure 6. 1,2,3 and 4 are the rotor surface areas which contribute to the combustion chamber and A, B and C represent the areas formed by the rotor tip, rotor flank and the area formed by the interaction of tip radius (R_t) and the flank radius (R_f). The graph that is generated for the instantaneous values of area of combustion chamber is shown in Figure 7,

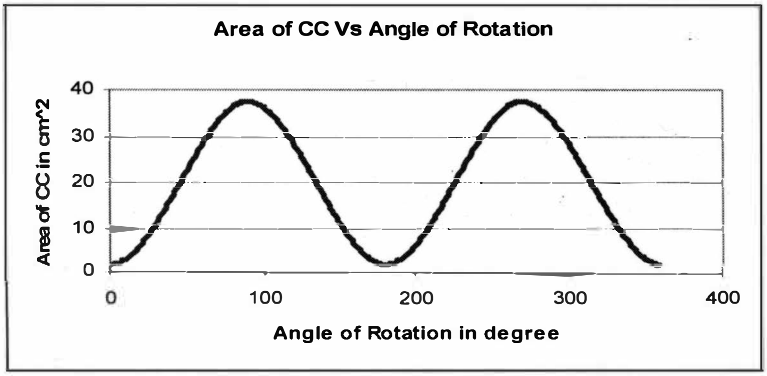


Figure 7: Variation of Cross Sectional Area of CC with Rotational Angle

Volume of combustion chamber is given by the length of the rotor (L) multiplied with the cross sectional area of the combustion chamber.

Volume of Combustion Chamber =Cross sectional area of Combustion Chamber * L

The list-solving procedure is used to generate the instantaneous cross sectional area of the combustion chamber and instantaneous volume of the combustion chamber. When the program is executed, the program automatically generates the values for cross sectional area of combustion chamber and volume of combustion chamber for 360° rotational angle of the rotor. The plot that is generated for the instantaneous values of volume of combustion chamber is shown Figure 8,

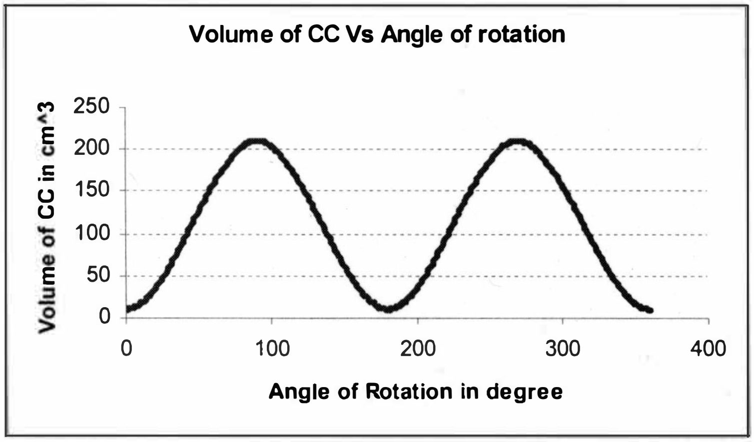


Figure 8: Variation of Volume of CC with Rotational Angle

Surface Area of HRM Combustion Chamber

The surface area of combustion chamber is given by the surface area of each rotor that contributes towards the formation of combustion chamber added with twice the cross sectional area formed by the end plates of the combustion chamber. The surface area calculation of the combustion chamber is shown in the Figure 9,

In Figure 9 the edge of the rotor represented by thick line represents the surface area of the rotor that contributes towards the formation of the combustion chamber. The length of the arc described by the angle BOC represents the surface area of the combustion chamber when the rotor is at 90 degree of rotation (full expansion state) and this includes only the arc made by the major radius R1. The arc described by the angle BO_1A represents the surface area of the combustion chamber when the rotor is at 0 degree rotation (full compression state) and this includes only the arc made by the minor radius R2. In the case of rotor rotation between 0 and 90 degree of rotation both the major and minor arc interact to form the combustion chamber surface area as shown in Figure 6 in which the combustion chamber is formed by the interaction of both the major and minor radius of the rotor. List-function is used for calculating the instantaneous combustion chamber surface area to generate the surface area of the combustion chamber for each degree of rotor rotation. This generates the instantaneous surface area for all 360° rotation of the rotor.

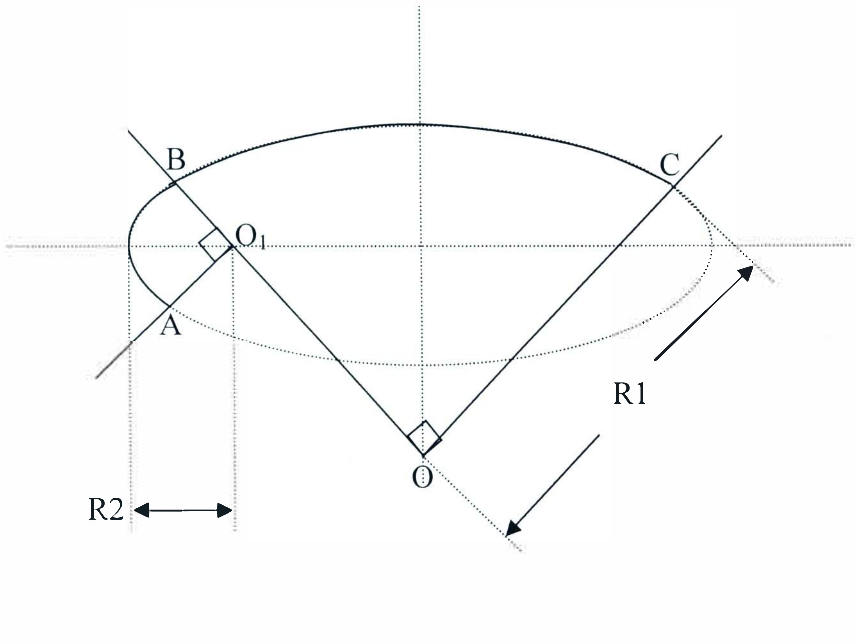


Figure 9: Surface Area of the Rotor made by Major and Minor Radius.

Following formulas are used for calculating the surface area of the combustion chamber.

- L1 = R1 * Angle of rotation of rotor
- L2 = R2 * Angle of rotation of rotor
- Total Length of the Arc (TL) = 4 * (L1+L2)
- Total Surface area of Arc (TSAA) = Length of Rotor * TL
- Surface area of the CC = TSSA + 2 * Cross sectional Area of CC

  where CC is Combustion Chamber.

The plot that is generated for the variation of surface area of combustion chamber with the angle of rotation of the rotor is shown in Figure 10,

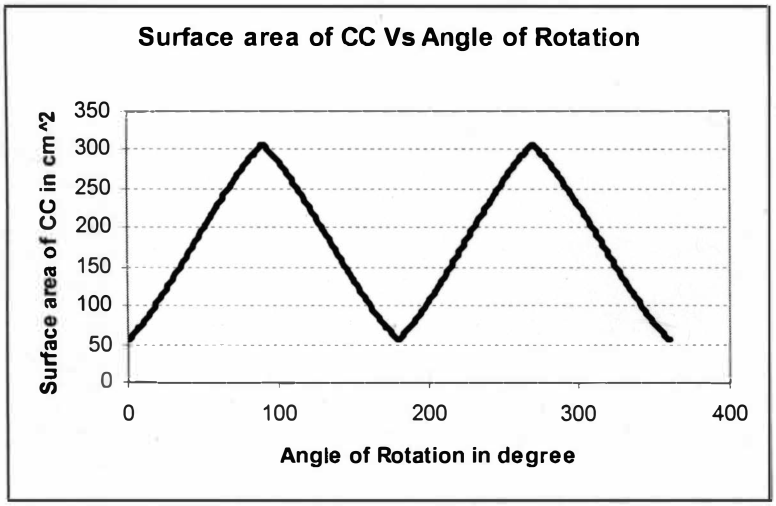


Figure 10: Variation of Surface Area of HRM CC with Rotational Angle

Surface Area of Rotor

Surface area of the rotor is calculated by the summation of the rotor surface area formed by the edges of the four centered ellipse and the cross sectional area of the rotor. The rotor surface area formed by the edges of the four centered ellipse is calculated using the major and minor radius of the ellipse. The cross sectional area of the rotor is calculated by segmenting the rotor cross section into four areas two of each formed by the major and the minor radius of the elliptical rotor. The rotor segmentation and the formulas used for the calculation of the surface area of the rotor are as shown in Figure 11.

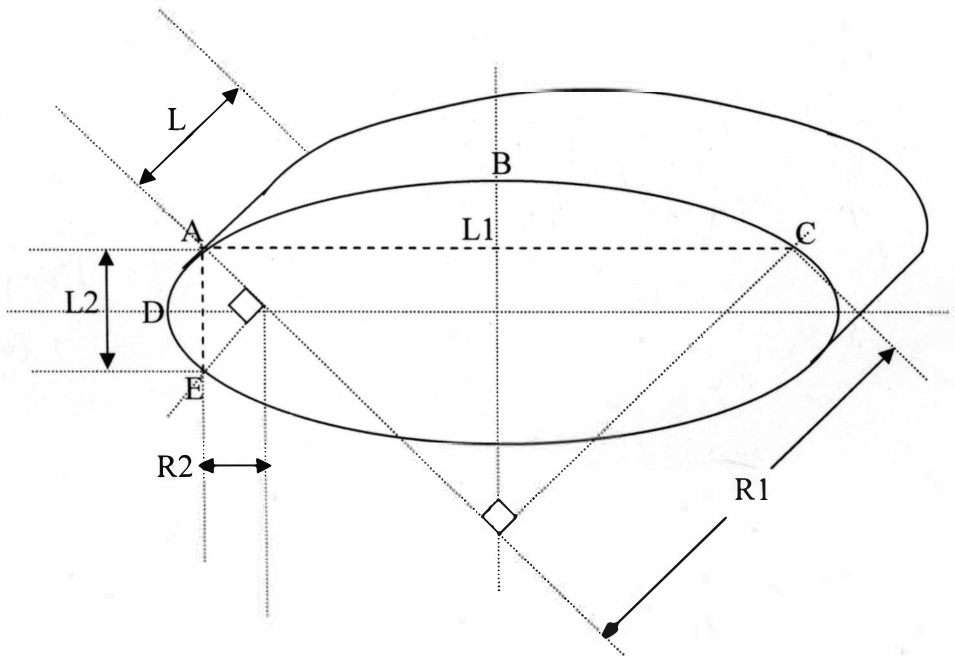


Figure 11: Surface Area of the Rotor

In Figure 11, R1 and R2 represent the major and minor radius of the rotor; L1 and L2 represent the length of the chord formed by the major and minor arcs which are formed by the major and minor radius respectively. L represents the length of the rotor. The following formulas are used for the calculation of the surface area of the rotor.

- Length of major arc AL1 = R1 * Angle of the major arc (where angle = 90°)
- Length of minor arc AL2 = R2 * Angle of the minor arc (where angle = 90°)
- Surface area by elliptical edge (S1) = 2 * (AL1 +AL2) * L
- Area of sector ABC (A1) = $\frac{1}{2}(R1^2 * (\theta - \sin\theta))$
- Area of sector ADE (A2) = $\frac{1}{2}(R2^2 * (\theta - \sin\theta))$
- L1 = 1.414 * R1
- L2 = 1.414 * R2
- Cross sectional Area (S2) = 2 * (A1 +A2) + (L1 *L2)
- Total surface area of rotor = S1 + S2

  where θ is in radians.

Instantaneous Compression Ratio (CR)

Instantaneous compression ratio is the ratio that is calculated for each degree of rotation of rotor. This instantaneous compression ratio is used for the calculation of temperature inside the combustion chamber without heat addition. Instantaneous compression ratio is the ratio of volume of combustion chamber at full expansion to the volume of combustion chamber at the particular degree of rotor rotation. In this calculation list function is used to calculate the instantaneous compression ratio for all 360° of rotor rotation. Main advantage of this list function is that once the program is set to run, the TK – solver automatically generates all the values of the instantaneous compression ratios. The formula used to calculate the instantaneous compression ratio is as follows,

- Instantaneous compression ratio = V_1/V_{ICC}

  where V_1 and V_{ICC} are volume at full expansion and volume of combustion chamber at a particular degree of rotor rotation.

Instantaneous Burn Fraction

Instantaneous burn fraction gives the details of percent fuel burned during the period of heat release. The TK – Solver program includes the input variables like angle of rotation of rotor, start of heat release, duration of heat release and shape factor. For every degree of rotor rotation the burn fraction is calculated and a list solving procedure is adapted. The instantaneous burn fraction can be changed by varying the input variables to determine maximum power output. This burn fraction is used to calculate the temperature inside the combustion chamber when heat is added. The following formula is used in the calculation of burn fraction,

- Burn fraction $=1 - \exp(-((A - A_s)/A_d)$ ^n)

  where A, A_s, A_d, n are angle of rotation of rotor, start angle of combustion, duration of combustion and Weibe form factor respectively. The plot that is generated for the instantaneous variations of burn fraction for each degree of rotor rotation is shown in Figure 12,

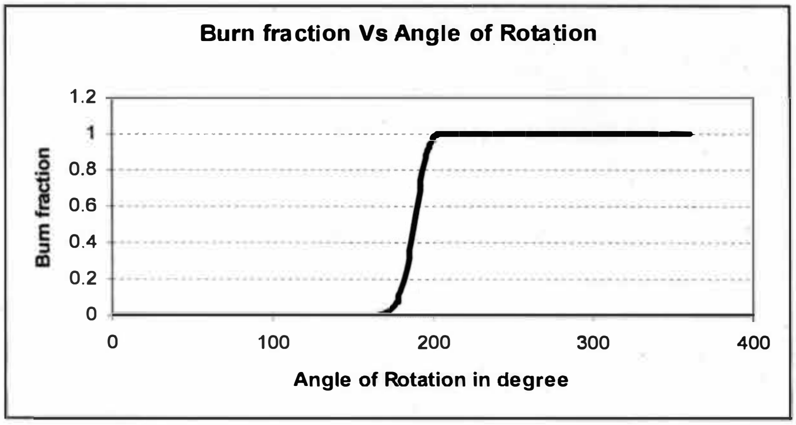


Figure 12: Variation of Burn Fraction with Rotational Angle

Instantaneous Temperature in Combustion Chamber

Temperature variation inside the combustion chamber is divided into two parts ie., with heat and without heat addition. Temperature variation without heat addition is a baseline analysis to determine the amount of heat developed inside the combustion chamber without heat addition depending on the compression ratio. Since the rate of surface area change of the HRM is much more higher than in the reciprocating engine this baseline calculation is done. Temperature without heat depends on the initial temperature, instantaneous compression ratio and ratio of specific heat capacity (γ).

Temperature variation with heat addition depends on temperature without heat, calorific value of the fuel, stoichiometric air fuel ratio and burn fraction. These variables govern the calculation of variation of temperature with heat addition inside the combustion chamber. For both the calculations, list-solving procedure is adapted. In this program, temperature variation with heat addition is divided into two categories, temperature variation with heat addition before the start of combustion and after the start of combustion. Heat addition depends on the input variables, start of combustion and duration of the combustion process.

Following formulas are used to calculate the temperature without heat addition and temperature with heat addition inside the combustion chamber,

- Temperature without heat addition (T_{WOH})= T_1 * (Instantaneous CR) ^(γ)
- Temperature with heat before combustion $T_{WH} = T_{WOH}$
- Temperature at heat release $T_{WH} = T_{WH(n-1)} + (T_n - T_{(n-1)}) + (f_n - f_{(n-1)}) * \Delta T$

  where n and (n-1) are present and previous values and $T_{WH(n-1)}$, T_n, $T_{(n-1)}$, f_n, $f_{(n-1)}$, ΔT are previous temperature with heat, present and previous values of temperature without heat, present and previous values of burn fraction and heat added to the combustion chamber respectively.

Instantaneous Pressure in Combustion Chamber

In this program two types of pressure variations are calculated, without heat addition and with heat addition. Pressure without heat addition describes the pressure that is produced inside the combustion chamber due to its change in compression ratio. Pressure with heat addition depends on three variables, mainly duration of opening and closing of intake port, duration of combustion and expansion process and duration of opening and closing of exhaust port. The values for these three zones are given as input variables before running this program. Depending upon the input values the pressure curve varies. The formulas used for calculation of pressure with and without heat addition are,

- Pressure without heat addition (P_{WOH}) = (V_1 / V_{ICC}) ^(γ)
- Pressure with heat addition (P_{WH}) = (P_{WOH}) * (T_{WH} / T_{WOH})

In this program, the pressure with heat addition, for the period between exhaust valve opening and the intake valve closing, is prefixed to atmospheric pressure.

The plot generated by the program for the variation of pressure without heat addition and with heat addition for each angle rotation of the rotors is shown in Figures 13 and 14 respectively.

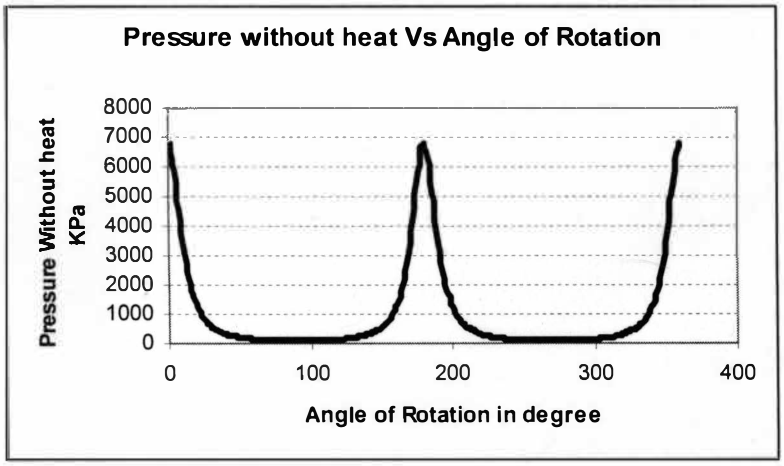


Figure 13: Variation of Pressure without Heat Addition

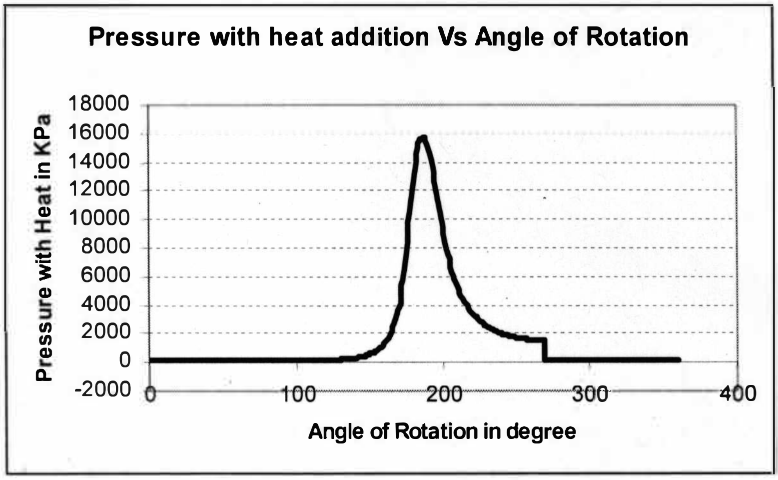


Figure 14: Variation of Pressure with Heat Addition with Rotational Angle

Instantaneous Torque

Since the rotors used in the Hopkins rotor mechanism engine is a four centered ellipse, the torque produced by the rotor is calculated separately for major and minor arcs of the rotor made by the major and minor radius R1 and R2 respectively. For this calculation list-solving procedure is adapted to generate the instantaneous moment for 360° of rotor rotation.

Figure 15 explains the procedure that is used to calculate the moment produced by the major arc when exposed to the combustion chamber. In Figure 10, R1 represents the major radius, O_1E represented by 'L' represents the moment arm, angle COD and AOF represented by 'a' represents the angle of rotation of the rotor and the angle made by the force vector with respect to the neutral axis of the rotor. L1 and L2 are lengths of the chord made by the force vector and major radius 'R1' of the rotor and L2 is the intermediate length that is used in the calculation of the length of the moment arm L. The angles 'a' and 'b' are the angles of the triangle made by the major radius R1, the force vector and the chord length L1. The angles 'c' and'd' are the angles made by the triangle AO_1B described by the intermediate length L2. The triangle BO_1E is a right angled triangle and angle 'e' is the intersection angle made by the force vector 'F' and the intermediate length L2.

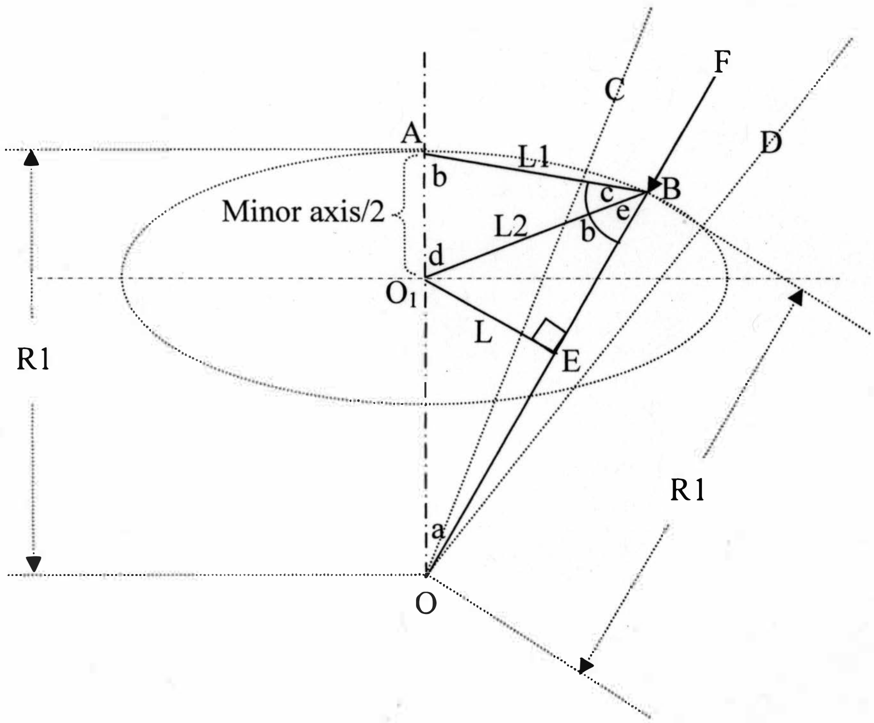


Figure 15: Moment Diagram for Major Arc

The following formulas are used in the calculations for determining the length of the moment arm 'L' produced by the neutral axis of the rotor to the force vector 'F',

- Angle ‘a’ = 45 – (Angle COD / 2)
- Length AO_1 = Minor axis / 2
- Angle ‘b’ = (180 – a) / 2
- Length ‘L1’ = (Sin a / Sin b) * R1
- Length ‘L2’ = sqrt (AO_1^2+L1^2-2* AO_1*L1* cos (b))
- Angle ‘d’ = Sin^{-1} * ((Sin b / L2) * L1)
- Angle ‘c’ = 180 – b – d
- Angle ‘e’ = b – c
- Moment arm length ‘L’ = Sin e * L2
- Force ‘F’ = Pressure with heat * Surface area
- Moment made by Major arc ‘M1’ = Force * Moment arm length

The moment made by the major arc and the procedure that is used calculating the moment produced by the minor arc is shown in the Figure 16 below

In Figure 16, OA represents the minor radius R2, angle ‘a’ made by BOD is the rotational angle of the rotor, ‘b’ and ‘c’ are the angles made by COD and the angle made by the force vector ‘F’ with respect to the major axis respectively. O_1E represent the length of the moment arm made by the neutral axis of the rotor with the force vector ‘F’.

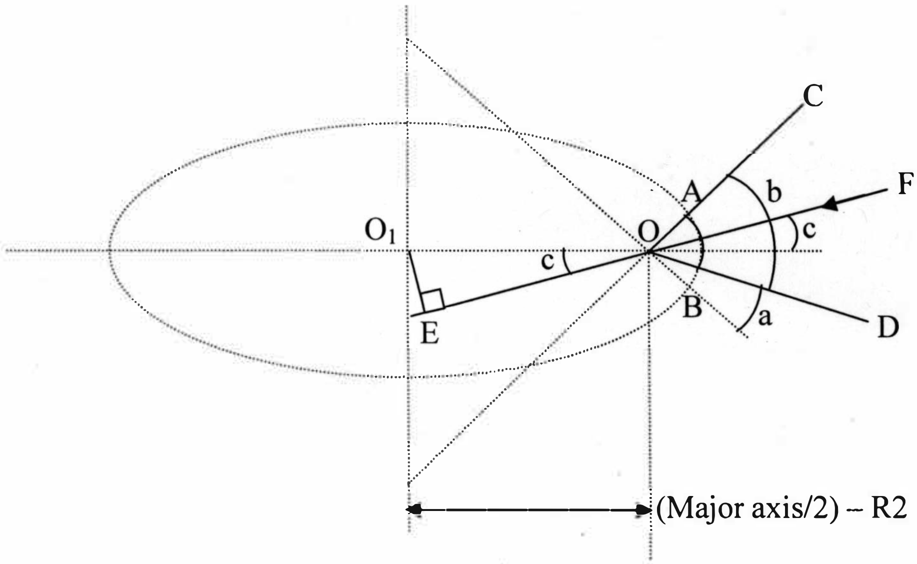


Figure 16: Moment Diagram for Minor Arc

Following formulas are used in the calculation of the length of the moment arm and the moment produced by the minor arc,

- Angle 'a' = Angle of rotation of the rotor
- Angle 'b' = 90 – a
- Angle 'c' = 45 – (b / 2)
- Length (O_1O) = (Major axis / 2) – Minor radius 'R2'
- Length of the Moment arm (O_1E) = Sin c * O_1O
- Force (F) = Pressure with heat addition * Surface area 'AB'
- Moment produced by Minor arc (M2) = Force * O_1O
- Total Moment produced by the single rotor (M) = M1 + M2

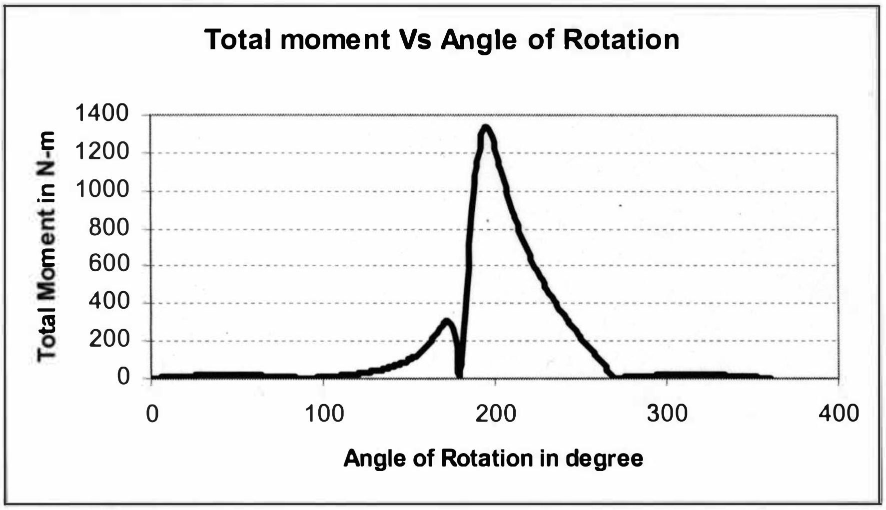


Figure 17: Variation of Total Moment with the Rotational Angle

In this program the moment produced for each degree of rotor rotation is calculated separately for major and minor arcs and are summed together to find the total moment produced by a single rotor. Moment produced by the four rotors is given by multiplying the total moment produced by the single rotor with a factor of 4. For calculating the instantaneous moment list-solving procedure is used to generate values for 360° of rotor rotation. The plot in Figure 17 shows the variation of moment varies with respect to the angle of rotation of the rotor.

CHAPTER IV

HOPKINS ROTOR MECHANISM COMBUSTION CHAMBER

Overview of HRM Combustion Chamber

The Hopkins rotor mechanism consists of an approximately elliptical-shaped cylinder which rotates about its center by means of a central shaft. The HRM engines consist of four rotors that are equally displaced about a common center. The rotor section outline is developed so that the tip of one rotor will engage with the flank of the adjacent rotor with a small clearance, and this clearance is maintained throughout the rotation of the rotors. The four rotors are constrained to rotate synchronously and in the same direction to form the combustion chamber of the HRM. The rotors are indexed so that in the initial or minimum volume position, the major axes of the rotor sections intersects the common center of the rotors, as shown in Figure 1 of chapter I.

Characteristics of Rotor Surface with Combustion Chamber

The rotor surface is characterized by a four-center ellipse and is represented dimensionally by the major axis, minor axis, flank radius (Rf), tip radius (Rt), and aspect ratio. The rotor is a four-center ellipse divided into two major areas, an area made by the flank radius and an area made by the tip radius. The radius of the flank and the tip vary according to the aspect ratio. The aspect ratio determines the compression ratio and volume displaced in the HRM engine. The segmentation of the rotor into areas made by the flank radius and the tip radius is shown in Figure 18.

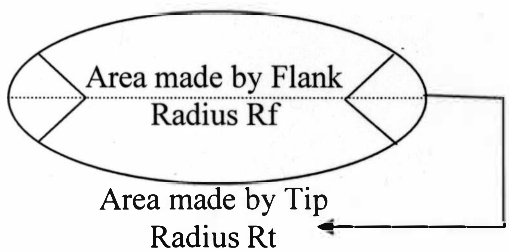


Figure 18: Segmentation of Cross Sectional Area of the Rotor

At full compression state, the area made by the rotor tips, represents the area of the combustion chamber. At full expansion state, the combustion chamber area is represented by the rotor flanks. Considering this, the angle limit between full compression and full expansion is found to be 90 degrees. Variation of the combustion chamber area from full compression to full expansion is shown in Figure 19.

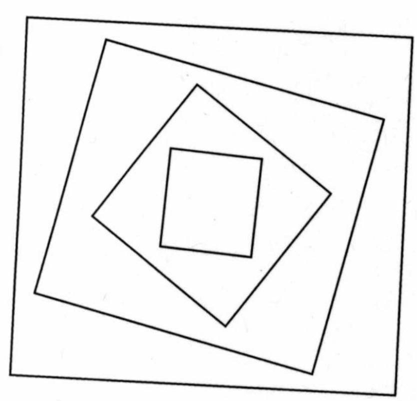


Figure 19: Change in Combustion Chamber Area with Rotor Rotation

Surface Area Diagram of HRM Combustion Chamber

The cross sectional area of combustion chamber is shown in Figure 20. It shows the rotor position approximately between full expansion and full compression state (≈45 degrees) Figure 20 depicts areas where both the flank and tip radii are included.

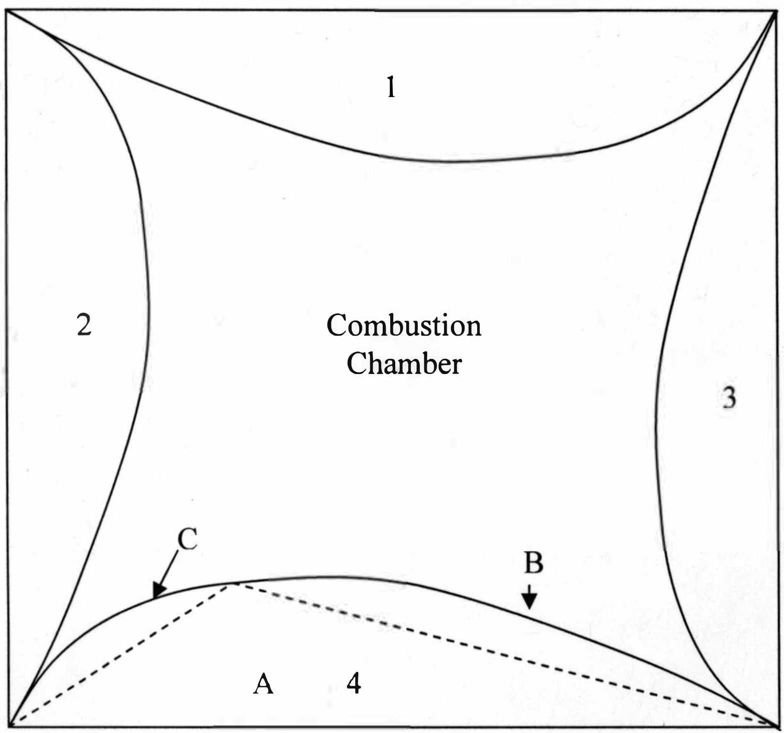


Figure 20: Cross Sectional Area of CC at Rotors Approximately 45 Degrees

The combustion chamber area is calculated by assuming a square section that intersects with the rotor surface to form the combustion chamber. This is explained by means of a cross sectional diagram. The rotor surface area that is included in the square section is divided into three regions and these regions are divided to include the flank radius R_f and tip radius R_t. This is represented in the diagram as B and C respectively. The third region represented as A (between the dashed lines and the horizontal axis of the square) is the interaction area that is formed by the areas made by R_f and R_t. From this the combustion chamber area is given by,

$$\text{Area of Combustion Chamber} = \text{Area of Square} - 4 \times \text{Area of the Rotor Sector}$$

In Figure 20 1,2,3 and 4 are the rotor surface areas which contribute to the combustion chamber and A, B and C represent the areas formed by the rotor tip, rotor flank and the area formed by the interaction of tip radius (R_t) and the flank radius (R_f) respectively.

Volume of HRM Combustion Chamber

When the rotors are positioned at 45 degrees, the major and the minor axes of complementary rotors intersects to define a perfect 'square' as shown in Figure 20. The maximum volume is attained after an additional rotation of 45° when the major axes of the rotors define a 'diamond pattern' about the common center. Further rotation of the rotors results in gradual decrease in the combustion chamber volume. When the rotor rotation reaches 180 degrees, minimum volume is contained within the combustion chamber. Continued rotation defines another identical sequence of expanding and contracting volumes. Thus for a full 360° of rotor rotation, two expanding and two contracting volumes are obtained.

The combustion chamber volume is defined by the surfaces of the four rotors, with fixed end plates containing the ends of the cylindrical chamber. When combustion occurs, the gasses produced as a result of combustion, acts on the rotor surface that are exposed inside the combustion chamber causing the rotors to rotate. If the mechanism is to rotate at high speed, a significant portion of the velocity component of the expanding gas can be converted into mechanical energy as in the case of a turbine engine.

CHAPTER V

ANALYSIS AND COMPARISION OF HRM COMBUSTION CHAMBERS GEOMETRIC VARIABLES

Introduction

The geometric variables of the HRM combustion chamber such as surface area, volume and cross sectional area are analyzed and compared with that of the reciprocating engine. In addition to this, the compression ratio and expansion ratio of the HRM combustion chamber are also analyzed to arrive at methods to increase the thermal efficiency. To compare specific characteristics of the HRM engine with that of a single cylinder reciprocating engine, displacements of 100, 200, 300 cubic centimeter (cc) are taken and compared with that of 200, 400, 600 cubic centimeter reciprocating engine. Compression ratios employed for this comparison are 10,15 and 20.

Geometric Comparison of Baseline Reciprocating Engine with HRM Engine

Due to the uniqueness of the HRM engine combustion chamber, analysis of the combustion chamber as a function of angular position of the rotors was deemed prudent. It was felt that the volume rate of expansion as a function of rotor angle ($dv/d\theta$) and as a function of time (dv/dt) could be the limiting factors in determining the operating RPM range of the HRM engine. To analyze the behavior, a set of dimensions that would represent automotive type engine dimensions were employed. To understand the HRM engines volume and combustion characteristics it was important to compare its characteristics to an established baseline. In a reciprocating engine the displacement commonly is near, or below, 500 cc per engine cylinder. The common automobile engine would have a cylinder bore to piston stroke ratio (b/s) of approximately one (1). The baseline reciprocating engine used for the comparison has displacements of 200, 400 and 600 cc each with compression ratio of 10, 15 and 20. In the four-stroke cycle reciprocating engine, two revolutions of the engine output shaft is required to intake, compress, combust and expel the charge that results in one power pulse.

The HRM engine experiences two volume compressions and two volume expansions per revolution. This power to output shaft relationship is similar to a two-stroke cycle Otto-cycle engine. Since the HRM engine in a comparable four-stroke mode can intake, compress, combust and expel the charge in one full revolution of the rotors, a fair comparison of engine behavior is to use 100, 200, 300 cc HRM configuration each with compression ratio of 10, 15 and 20. To gain a similar comparison to the single cylinder of a multi-cylinder automobile engine the dimensions shown in Table 2 and Table 3 are employed.

Geometric parameters of HRM engine				
Displacement in cc	Geometric Parameters in cm	Compression Ratio (CR)		
		10	15	20
100	Major axis	8.5	8.32	8.1
	Minor axis	5.48	5	4.723
	Rotor length	4.6	4.45	4.486
	Major radius	5.317	5.315	5.244
	Minor radius	1.675	1.37	1.167
200	Major axis	10.8	10.4	10.2
	Minor axis	6.968	6.318	5.948
	Rotor length	5.67	5.66	5.612
	Major radius	6.756	6.644	6.6
	Minor radius	2.128	1.715	1.47
300	Major axis	12.32	11.9	11.7
	Minor axis	7.948	7.23	6.822
	Rotor length	6.532	6.485	6.414
	Major radius	7.707	7.6	7.575
	Minor radius	2.428	1.96	1.686

Table 2: Geometric Parameters of HRM Engine

Geometric Parameters of Reciprocating engine				
Displacement in cc	Geometric Parameters	Compression Ratio (CR)		
		10	15	20
600	Bore diameter in cm	9.143	9.143	9.143
	Stroke length in cm	9.143	9.143	9.143
	Combustion chamber volume in cc	66.67	42.86	31.58
	Crank radius in cm	4.572	4.572	4.572
	Connecting rod length in cm	13.715	13.715	13.715
400	Bore diameter in cm	7.987	7.987	7.987
	Stroke length in cm	7.987	7.987	7.987
	Combustion chamber volume in cc	44.44	28.57	21.05
	Crank radius in cm	3.994	3.994	3.994
	Connecting rod length in cm	11.981	11.981	11.981
200	Bore diameter in cm	6.339	6.339	6.339
	Stroke length in cm	6.339	6.339	6.339
	Combustion chamber volume in cc	22.22	14.29	10.53
	Crank radius in cm	3.170	3.170	3.173
	Connecting rod length in cm	9.509	9.509	9.509

Table 3: Geometric Parameters of Reciprocating Engine

Compression Ratio (CR) and Expansion Ratio (ER) Analysis

Since the ratio of the shaft work output to the energy input is a common comparison, called the specific fuel consumption, it can be desirable to have lower specific fuel consumption. The actual specific fuel consumption is the product of the mechanical efficiency, the cycle efficiency (air standard efficiency) and how closely the air standard cycle can be approached considering heat transfer to the chambers and the heat addition rate. It is anticipated that the specific fuel consumption for the HRM engine would be lower than what is achievable with the reciprocating engine. This is based on calculations indicating higher mechanical friction and slightly greater combustion chamber surface areas (≈6%), which would promote higher combustion heat losses in the HRM engine. Potential methods of offsetting these losses would be to utilizing higher compression ratios and the potential use of different compression ratio to expansion ratios. The usage of higher compression ratios to obtain the maximum torque output has been analyzed and the results are shown in Table 4.

Displacement in cc	Average Torque Produced for 2 revolutions of rotor N-m		
	CR 10	CR 15	CR 20
100	49.5	53.3	54.9
200	98.5	105.0	108.8
300	147.6	157.5	163.8

Table 4: Compression Ratio Analysis

The thermal efficiency of a four-stroke Otto cycle engine was taken and its properties were studied to estimate the range of compression ratio at which the HRM has the greatest potential for gains over the conventional Otto cycle engine. To begin the study, an "ideal" Otto cycle engine analysis was used. This analysis, called an air standard analysis, assumes that there is no heat loss during the compression and expansion process. The air standard analysis also assumes that instantaneous heat addition is possible and occurs.

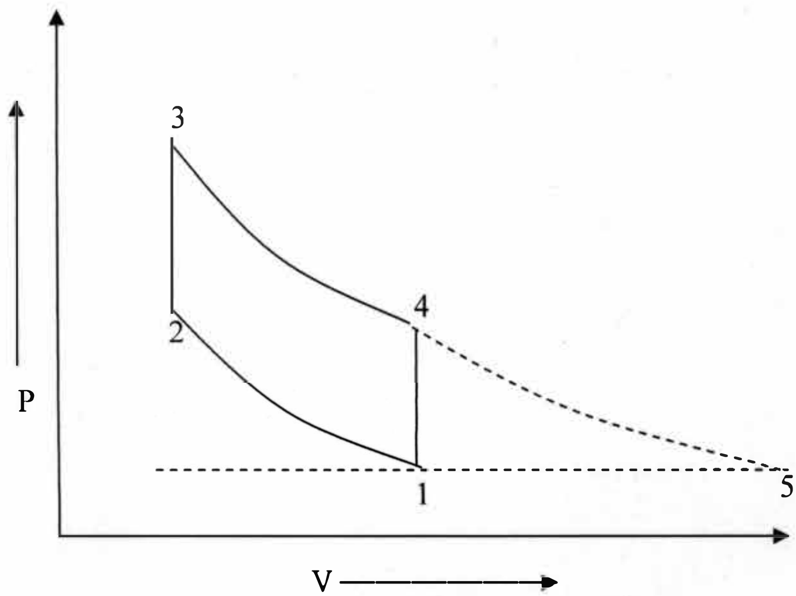


Figure 21: Air Standard Otto Cycle P – V (Work Diagram)

Although neither of these conditions exists in reality these assumptions form the basis for comparison of potential gains and set the bounds for the upper limits of efficiency of the Otto cycle engine and, more importantly, enable the study of altering the process to provide gains, if possible, with the HRM. The single cylinder

Otto-cycle engine is taken and the p-v diagram of a representative engine is drawn. From the p-v diagram, pressure vs. volume at the various points is noted.

From the plot shown in Figure 21, it is obvious that if the area enclosed by 4-5-1 could fully, or partially, be utilized in the HRM, benefits could be accrued. This area represents work wasted due to the constraint that compression ratio (CR) and expansion ratio (ER) are equal in an Otto cycle engine. Modern camshaft and valving technology are being employed and/or studied to benefit from this area in Otto cycle engines for the last decade, with a cost penalty and only as a benefit when the engine is throttled.

The compression ratio of the Otto cycle engine is the ratio of the maximum volume when the piston is at BDC to the minimum volume when the piston at TDC. The volume contained between the piston and the cylinder head when the piston is at TDC is called the clearance volume. The difference between the volume contained when the piston is at BDC to the clearance volume is called the swept volume. The swept volume, to a great extent, determines the power that can be produced from any engine. The compression ratio of the HRM is the ratio of the maximum volume enclosed by the rotors (adjacent rotor major axes are at 90 degrees to each other; adjacent rotor tips are at 90 degrees to each other) to the minimum volume enclosed by the rotors (opposite rotor tips facing each other).

$$CR = \frac{V_{\max}}{V_{\min}} = \frac{V_1}{V_2} \qquad ER = \frac{V_{\max}}{V_{\min}} = \frac{V_4}{V_3}$$

The thermal efficiency of the air standard Otto cycle is a function of the compression ratio only and is depicted in the following equation.

$$\eta_{th} = 1 - \frac{1}{CR^{\gamma-1}}$$

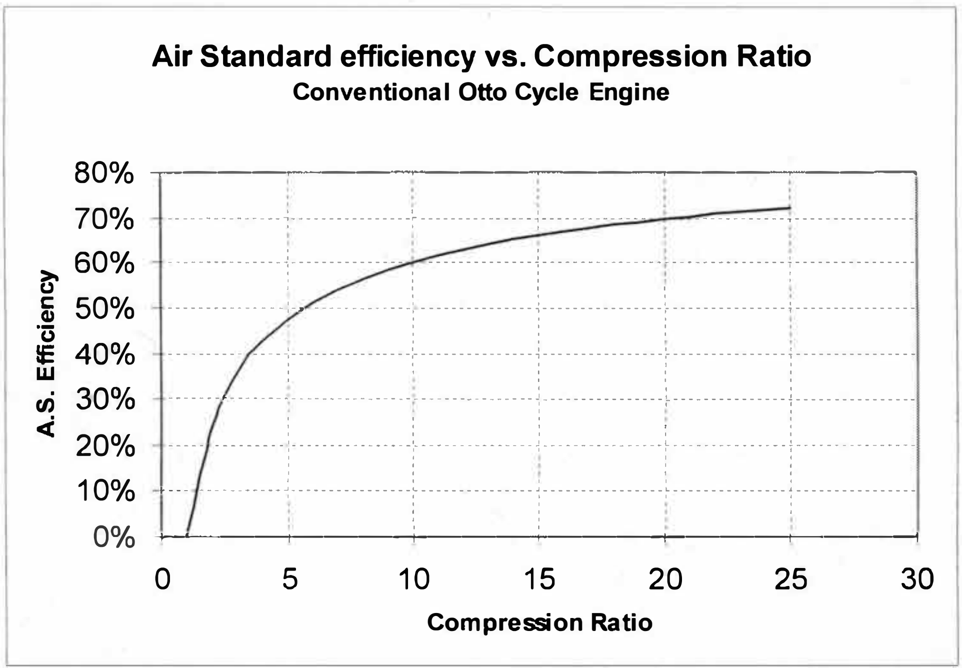


Figure 22: Air Standard Efficiency Limits vs. Compression Ratio

The concept of enabling the HRM to work with the expansion ratio not equal to the compression ratio (ER>CR) was felt to be an important concept to explore. Allowing the combustion gases to expand completely can enhance the work output and hence can improve the overall efficiency of the process. The maximum thermal efficiency that can be achieved with a conventional Otto cycle engine (CR=ER) at a compression ratio of 20:1 is approximately 69.8% and for 15:1 is approximately

66%. If the compression ratio is decreased to 10:1, the efficiency is reduced to approximately 60%. This is the upper limit of efficiency with the assumption that heat losses do not occur during compression, expansion and heat addition. Basically this would represent a true adiabatic process.

Due to the efficient packaging of the HRM, with high ratios of V_{max}/V_{min} achievable, it was felt that ER>CR was in fact possible. Taking the expansion ratio equal to 'n' times the compression ratio, the analysis was performed. The air standard thermal efficiency and the heat rejected were predicted.

Using practical values for the expansion ratio to compression ratio (n), it might be possible to raise the efficiency limits above those possible with conventional reciprocating engines. Table 5 shows the efficiency possible and the potential gains as 'n' is increased from 1 (ER=CR) to n=4, (ER=4xCR). The theoretical value of "n" at which the thermal efficiency becomes 100% for different compression ratios is also included. From a computational analysis it is found that the air standard efficiency for compression ratio of 10, 15 and 20 is 100% when the expansion ratio is 667, 712.5 and 760 with 'n' varying from 66.75, 47.5 and 37.75 respectively. The high expansion ratios (667, 712.5 and 760) are impractical, even for the HRM, as the physical size of the engine would be too great. The lower ratios (2=<4) might be practical with the HRM as its packaging efficiency is high.

Compression Ratio (CR)	n = (ER/CR)					n for 100% A.S. efficiency
	n = 1	n = 2		n = 3		
	A.S. efficiency %	A.S. efficiency %	Gain %	A.S. efficiency %	Gain %	
10	60.2	71.6	11.4	80.2	20.0	66.75
15	66.1	76.1	10.0	83.7	17.6	47.50
20	69.8	78.9	9.1	85.8	16.0	37.75

Table 5: Potential Gain with Expansion Ratio (ER) > Compression Ratio (CR)

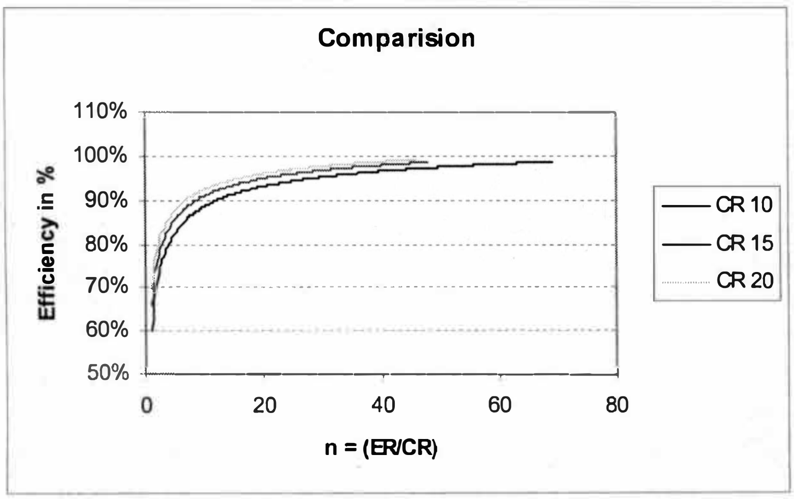


Figure 23: Variation of Air Standard Thermal Efficiency with n

The graph in Figure 23 shows the variation of air standard thermal efficiency for different values of compression ratio's with different values of n (n=ER/CR). From this preliminary analysis employing greater expansion ratios would provide substantial gains in efficiencies that could further promote the HRM benefits. It would be favorable to further explore and exploit these benefits in the HRM design.

Surface Area Comparison Between Reciprocating and HRM Engine

The surface area constantly changes when the rotors changes angular position in the HRM engine. The HRM engine has a greater surface area rate of change than the baseline reciprocating engine.

Due to the complex and constantly changing combustion chamber shape, the HRM engines combustion chamber surface area was compared with that of 200, 400 and 600 cc baseline reciprocating engine. Generally, the surface inside the combustion chamber that is exposed to the gases produced due to combustion is called the combustion chamber surface area. In the reciprocating engine this surface includes the cylinder liner area, which varies as a function of crankshaft position, together with the surface of the piston and the cylinder head that is approximated to have the shape of the piston surface. In the HRM engine surface area was considered to be the rotor surface and the end plate surfaces that are exposed to the combustion gases. Only surface areas exposed to combustion gases were considered in the instantaneous surface evaluation. The surface area, and the temperature of the surface area, is an indicator of the heat transfer from the combustion gases and an indicator of the isentropic losses that would occur.

The instantaneous surface area comparison is done by comparing the HRM engine with a displacement of 100 cc and the reciprocating engine with a displacement of 200 cc. A compression ratio of 10 was employed for this comparison. The resulting plot obtained is shown in Figure 24.

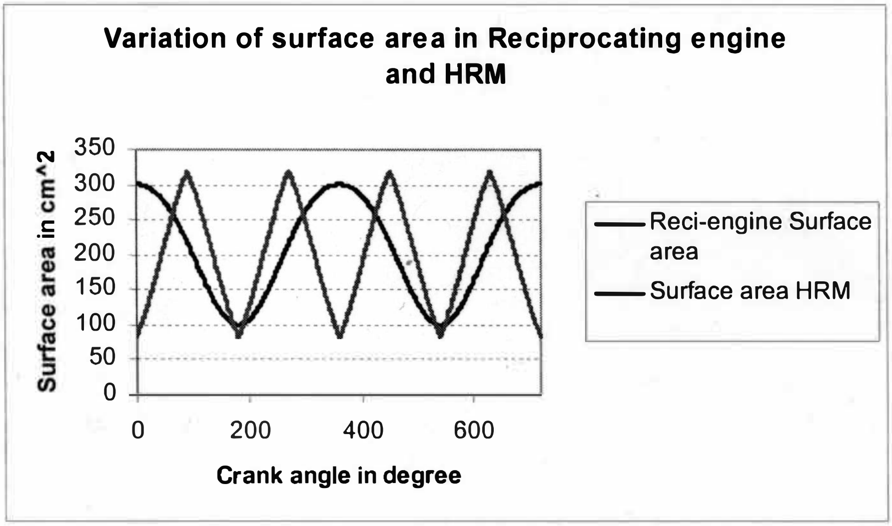


Figure 24: Variation of Surface Area in HRM and Reciprocating Engine

The surface area of the HRM varies, in a periodic fashion, for every 180 degrees of rotor rotation. The surface area of the reciprocating engine varies in a periodic fashion for every 360 degrees of crankshaft rotation. The surface area is important as it serves as quench areas and, at the rotor intersection, as quench volumes. Large surface to volume ratios are usually not emission friendly. Crevice volumes are tight areas where small volumes are supported by large surfaces. The surface to volume ratio, the surface area and the crevice volumes are constantly changing as the rotors of the HRM engine change angular position.

Figure 24 indicates that the average surface area for the reciprocating engine is greater than the average value of surface area for the HRM by almost 20%. This is primarily due to the fact that the HRM has one half the physical dimension of the reciprocating power plant (200 cc vs. 400 cc) for comparable power. However, the

PRE actually has a greater surface area rate of change than the baseline reciprocating engine. This rate of change of surface area depicts the time any surface is exposed to elevated temperature, with the time proportional to the heat loss that is likely to occur. The HRM has a 42.5% greater surface to volume ratio than the conventional reciprocating engine and proved by the Table 6, 7 and 8.

The smaller surface area of the HRM would indicate the potential for lower isentropic losses while the higher surface to volume ratios would lead one to believe higher emissions are possible from the engine exhaust prior to after-treatment. After-treatment would be in the form of catalytic conversion of the harmful exhaust constituents.

Engine Type	Power Stroke per 2 revolution	Displacement in cc	CR	RPM	Surface area in cm^2	
					Minimum	Maximum
HRM	2	100	10	4000	53.2	202.2
Reci	1	200			77.2	203.4
HRM	2	100	15	4000	44.0	197.1
Reci	1	200			72.2	198.4
HRM	2	100	20	4000	37.8	195.0
Reci	1	200			69.8	196.0

Table 6: 100 cc HRM and 200 cc Reciprocating Engine Surface Area Comparison

Engine Type	Power Stroke per 2 revolution	Displacement in cc	CR	RPM	Surface area in cm^2	
					Minimum	Maximum
HRM	2	200	10	4000	87.3	319.0
Reci	1	400			122.5	322.9
HRM	2	200	15	4000	70.0	312.1
Reci	1	400			114.5	314.9
HRM	2	200	20	4000	59.7	307.5
Reci	1	400			110.8	311.2

Table 7: 200 cc HRM and 400 cc Reciprocating Engine Surface Area Comparison

Engine Type	Power Stroke per 2 revolution	Displacement in cc	CR	RPM	Surface area in cm^2	
					Minimum	Maximum
HRM	2	300	10	4000	114.7	418.3
Reci	1	600			160.5	423.1
HRM	2	300	15	4000	91.7	408.8
Reci	1	600			150.1	412.7
HRM	2	300	20	4000	78.2	403.8
Reci	1	600			145.1	407.7

Table 8: 300 cc HRM and 600 cc Reciprocating Engine Surface Area Comparison

Comparison of Volume Variation Between HRM and Reciprocating Engine

It was felt that the volume rate of expansion as a function of rotor angle ($dV/d\theta$) and as a function of time (dV/dt) could be limiting factors in determining the operating RPM range of the HRM engine. The baseline used was the reciprocating engine with displacements of 200, 400, 600 cc. In the four-stroke cycle engine two revolutions of the engine output shaft is required to intake, compress, combust and expel the charge that results in one power pulse. The HRM engine experiences two volume compressions and two volume expansions for each complete revolution of the rotor. This power to output shaft relationship is similar to a two-stroke cycle Otto cycle engine. Since the HRM engine in a comparable four-stroke mode can intake, compress, combust and expel the charge in one full revolution of the rotors, a fair comparison of engine behavior is used, that is one-half the displacement produced by the reciprocating engine.

The variation of volume is shown for a compression ratio of 10 and a displacement of 400cc.The plot shows that the cylinder volume is at maximum when the piston is at the bottom dead center (0 degrees). The maximum volume of the cylinder occurs at BDC and is 444.4 cm^3, which is the swept volume, added with the clearance volume.

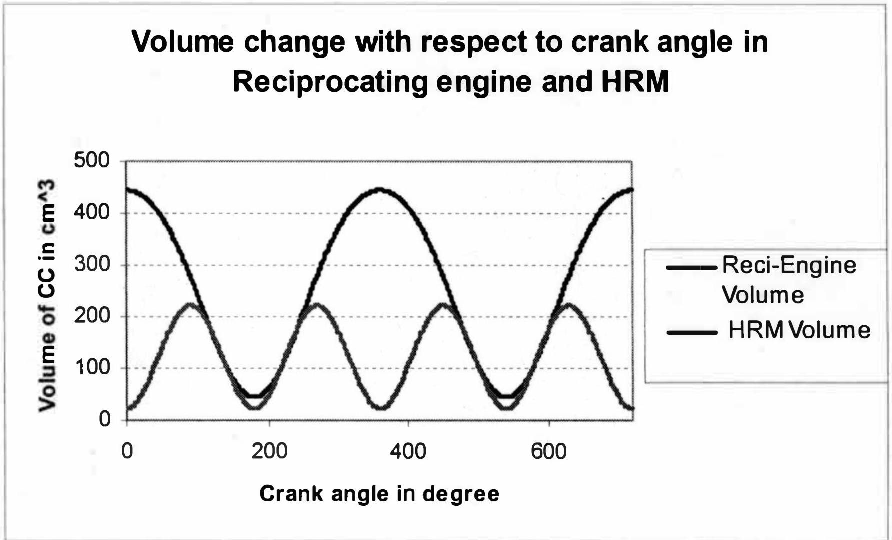


Figure 25: Volume Variation in HRM and Reciprocating Engine

The volume of the combustion chamber has a direct influence on the pressure variation produced. When the volume in the combustion chamber is a minimum, the pressure in the cylinder reaches a peak. This shows that volume and pressure are inversely proportional. Variation of volume of the combustion chamber with the angle of rotation of the rotors in the HRM and in the reciprocating engine is shown in Figure 25. In the HRM engine for every 90 degrees of rotor rotation, the volume of the combustion chamber varies from a maximum to a minimum. This variation in the volume of the combustion chamber influences the pressure variation. The variation of volume is shown for the HRM with a compression ratio of 10 and a displacement of 200 cc. The maximum volume attained is calculated using TK-Solver. The maximum volume of the combustion chamber in this case is 222.15 cc.

Engine Type	Power Stroke per 2 revolution	Displacement in cc	CR	RPM	Rate of Volume change	
					cc/deg	cc/sec
HRM	2	100	10	4000	1.11	26640
Reci	1	200			1.23	29520
HRM	2	100	15	4000	1.11	26640
Reci	1	200			1.19	28560
HRM	2	100	20	4000	1.11	26640
Reci	1	200			1.17	28080

Table 9: Volume Comparison for 100 cc HRM with 200 cc Reciprocating Engine

Engine Type	Power Stroke per 2 revolution	Displacement in cc	CR	RPM	Rate of Volume change	
					cc/deg	cc/sec
HRM	2	200	10	4000	2.22	53280
Reci	1	400			2.47	59280
HRM	2	200	15	4000	2.22	53280
Reci	1	400			2.38	57120
HRM	2	200	20	4000	2.22	53280
Reci	1	400			2.34	56160

Table 10: Volume Comparison for 200 cc HRM with 400 cc Reciprocating Engine

Engine Type	Power Stroke per 2 revolution	Displacement in cc	CR	RPM	Rate of Volume change	
					cc/deg	cc/sec
HRM	2	300	10	4000	3.33	79920
Reci	1	600			3.70	88800
HRM	2	300	15	4000	3.33	79920
Reci	1	600			3.57	85680
HRM	2	300	20	4000	3.33	79920
Reci	1	600			3.51	84240

Table 11: Volume Comparison for 300 cc HRM with 600 cc Reciprocating Engine

To analyze the operating cycle considerations of the HRM, properties of the reciprocating engine with dimensions, compression ratio and volume displaced similar to that of the HRM were taken and analyzed. The HRM engine taken with a displacement half that of the reciprocating engine gives similar rate of volume change as in the case of a reciprocating engine. This fact is substantiated in Tables 9, 10 and 11.

CHAPTER VI

ANALYSIS AND COMPARISION OF HRM COMBUSTION CHAMBERS COMBUSTION VARIABLES

Introduction

The combustion chambers variables of the HRM engines combustion chamber such as burn rate, pressure and torque are analyzed and compared with that of the reciprocating engine. To compare the combustion characteristics of the HRM engine with that of a single cylinder reciprocating engine, displacements of 100, 200, 300 cubic centimeters are taken and compared with that of 200, 400 and 600 cubic centimeter displacement of the reciprocating engines. Compression ratios employed for this comparison are 10, 15 and 20.

Burn Rate Analysis

In the air standard Otto and Diesel cycle reciprocating engines, it is assumed that the heat is released instantaneously. The fuel is assumed to burn at rates which results in constant volume top dead center combustion, or constant pressure combustion respectively. Actual engine pressure and temperature profile data do not match these simple models, and more realistic models, such as a finite heat release model is required. In order to study and analyze the spark timing, duration of combustion, flame speed, pressure and temperature variation in the combustion chamber finite heat release model is used in the HRM engine.

Method Employed for HRM Finite Heat Release Model

The finite heat release model specifies heat release as a function of crank angle. Similar methods are employed for the HRM engine to determine the heat release. In the HRM engine the angle made by the rotor rotation is taken instead, for the crank angle which is the case for reciprocating engine. The model employed for the determination of heat release in the HRM has been explained in chapter III. If the start of heat release begins too late, then heat release will occur in an expanding volume, resulting in lower combustion pressure and lower net work. If the start of heat release begins too early during the compression stroke, the negative compression work will increase, since the work done is against the expanding combustion gases.

The burn rate analysis for the HRM engine was carried out at different starting periods of combustion and for different durations of combustion. This preliminary analysis was done to determine the operating pressure range of the HRM engine. For this preliminary analysis, three different start angles of combustion and two different durations of combustion were considered. Each of these start angles of combustion are examined with the durations of combustion to determine the HRM engines operating pressure range. The start angles of combustion that are considered for this preliminary analysis are 150, 145 and 140 degrees of the rotor rotation. Duration of combustion is taken to be 40 and 35 degrees.

The finite heat release model for various start angle of combustion and duration of combustion is shown in Table 12.

Combustion specifications in degree	Finite heat release model for 100, 200 and 300 cc engine displacement					
Start angle	150	150	145	145	140	140
Duration angle	40	35	40	35	40	35
CR	10	10	10	10	10	10
Start angle	150	150	145	145	140	140
Duration angle	40	35	40	35	40	35
CR	15	15	15	15	15	15
Start angle	150	150	145	145	140	140
Duration angle	40	35	40	35	40	35
CR	20	20	20	20	20	20

Table 12: Finite Heat Release Model for HRM Engine

The above table explains how the start angle of combustion and duration of combustion are used in the determination of heat release in the HRM engine.

Pressure Analysis in the HRM Combustion Chamber

From the preliminary analysis the pressure variations for different heat release models are calculated. It has been found that the peak pressure calculated for the heat release model converges towards air standard cycle. Using these pressure variations the operating pressure range for the HRM engine is determined for maximum torque output. The pressure variations that are obtained for various heat release models shown in Table 12 are given in Table 13.

Start of Combustion in degree		150		145		140	
Duration of Combustion in degree		40	35	40	35	40	35
Displacement in cc	CR	Combustion Pressure in KPa					
100	10	9363	12121	11852	14503	14192	16123
	15	12532	17182	16925	21613	21140	24559
	20	15749	22369	22183	29121	28513	33613
200	10	9365	12054	11855	14507	14197	16132
	15	12505	17138	16881	21552	21080	24484
	20	15727	22334	22149	29072	28464	33553
300	10	9363	12121	11851	14502	14191	16127
	15	12523	17166	16909	21545	21118	24532
	20	15745	22363	22177	29112	28505	33602

Table 13: Variation of Pressure in the HRM for Finite Heat Release Model

Pressure variations in the HRM engines for the air standard Otto cycle is calculated for 100, 200 and 300 cubic centimeter displacements each with compression ratio of 10, 15 and 20. The air standard pressure variation in the HRM engine that produces maximum torque output is given in Table 14.

Combustion Parameter	Engine Displacement	Compression Ratio (CR)		
		10	15	20
Air standard Peak Pressure in the HRM in KPa	100 cc	12,121	17,182	22,369
	200 cc	12,054	17,138	22,334
	300 cc	12,121	17,166	22,363

Table 14: Effective Pressure for the HRM Engine

Operating Pressure Range in the HRM Engine

The heat release parameters are optimized, in such a way that, the peak pressure produced in the HRM engine for 100, 200 and 300 cubic centimeter displacements matches with the peak pressure produced by the reciprocating engine for 200, 300 and 400 cubic centimeter displacements. The compression ratios used for this comparison are 10, 15 and 20. In a reciprocating engine the common heat release parameters such as start angle of combustion is around 40 degrees before TDC and duration of combustion is near or below 120 degrees. The duration of combustion take place for 180 degrees.

In the HRM engine one complete power cycle occurs within 360 degrees. The heat release in HRM engine should occur within 90 degrees of the rotor rotation when compared to that of 180 degrees of crank shaft rotation in the reciprocating engine. Hence the start of combustion has been taken as 20 degrees before full compression and duration of combustion has been taken as 60 degrees compared to reciprocating engine. Using these heat release characteristics, the HRM engine has been analyzed and the peak pressure obtained from this analysis is given in Table 15.

Combustion Parameter	Engine Displacement	Compression Ratio (CR)		
		10	15	20
Operating Peak Pressure in KPa	100 cc	3323	5491	8054
	200 cc	3324	5471	8036
	300 cc	3322	5484	8050

Table 15: Operating Peak Pressure in the HRM Engine

Torque Output for the Peak Pressure

The peak pressure obtained for the optimized heat release characteristics has been used to calculate the average torque output from the HRM engine. The average torque output has been calculated for 100, 200 and 300 cubic centimeter displacements each with compression ratios of 10, 15 and 20. The average torque variation for these displacements and compression ratios are given in Table 16.

Combustion Parameter	Engine Displacement	Compression Ratio (CR)		
		10	15	20
Average Torque in N-m	100 cc	49.5	53.3	54.9
	200 cc	98.5	105.0	108.8
	300 cc	147.6	157.5	163.8

Table 16: Average Torque for Optimized Heat Release Model

Pressure Comparison Between the HRM Engine and Reciprocating Engine

The HRM engine completes one power cycle within 360 degrees of the rotor rotation compared to that of 720 degrees of crank shaft rotation in the reciprocating engine. Hence the HRM engine produces two power cycles for 720 degrees when compared to one power cycle in the reciprocating engine. In any engine, the rate of heat release and rate of volume expansion controls the rate of pressure rise in the combustion chamber. The rate of volume expansion in the HRM engine is greater compared to the reciprocating engine. So the heat release is also greater. Hence it can be assured that the HRM could produce net pressure gain.

To verify this, the peak pressure of the HRM engine with displacements 100, 200 and 300 cubic centimeter are compared with the peak pressure of the reciprocating engine with displacements 200, 400 and 600 cubic centimeter each with compression ratios 10, 15 and 20. This comparison is given in Table 17.

Peak Pressure Comparison in KPa										
Displacement		100 cc HRM and 200 cc Reciprocating Engine			200 cc HRM and 400 cc Reciprocating Engine			300 cc HRM and 600 cc Reciprocating Engine		
Compression Ratio		10	15	20	10	15	20	10	15	20
Engine Type	HRM	3323	5491	8054	3324	5471	8036	3322	5484	8050
	RECI	3142	5280	7716	3142	5282	7721	3142	5282	7720

Table 17: Peak Pressure Comparison

Torque Comparison Between the HRM Engine and Reciprocating Engine

Since the HRM engine has greater volume rate of expansion and rate of heat release, the net pressure gain in the HRM engine is more when compared to the reciprocating engine. This has been compared and given in Table 18. As the HRM engine has net pressure gain when compared to the reciprocating engine, it should also produce correspondingly positive net power output than the reciprocating engine. To verify that the HRM engine produces positive net power output, the average torque produced for the peak pressures of the HRM engine and the reciprocating engine are compared. The average torque comparison is given in Table 18.

Average torque comparison for the optimized heat release characteristics in N-m										
Displacement		100 cc HRM and 200 cc Reciprocating Engine			200 cc HRM and 400 cc Reciprocating Engine			300 cc HRM and 600 cc Reciprocating Engine		
Compression Ratio		10	15	20	10	15	20	10	15	20
Engine Type	HRM	49.5	53.3	54.9	98.5	105.3	108.8	147.6	157.5	166.8
	RECI	41.7	40.3	39.5	83.4	80.5	79.0	125.0	120.7	118.5

Table 18: Average Torque Comparison

Form the Tables 17 and 18 this analysis confirms the basic claim of high power density for the HRM engine.

CHAPTER VII

DISCUSSION AND RECOMMENDATIONS

Discussion

The Hopkins Rotor Mechanism (HRM) has been examined from a perspective of commercial viability. Factors that were studied herein include basic operating characteristics, cycle choice, combustion effects, behaviors unique to the HRM engine and dimensional changes that may impose limits to commercial viability.

In its basic operation, the HRM experiences two volume compressions and two volume expansions for one complete revolution of the rotors. The HRM, in a comparable four-stroke mode, can intake, compress, combust and expel the charge in one full revolution of the rotors. This power to output shaft relationship, in a four-stroke cycle operating mode, is similar to a two-stroke cycle reciprocating Otto cycle engine. To compare specific characteristics of the HRM engine, in a comparable four-stroke mode, hypothetical single cylinder reciprocating engine with displacements of 200, 400 and 600 cc and with compression ratios 10, 15 and 20 was used. For similar outputs this resulted in the use of a 100, 200 and 300 cc HRM configuration each with compression ratios 10, 15 and 20. This analysis confirms the basic claim of high power density for the HRM design.

From the preliminary analysis, employing greater expansion ratios than compression ratios would be possible. This would provide substantial gains in efficiencies that could further promote the HRM benefits. It would be favorable to further explore and exploit these benefits in the HRM design. If a delayed start of compression is employed to reduce the mass in the chamber, throttling losses could be minimized and part-power efficiencies could be further improved in the HRM engine. The delayed start of combustion could be accomplished as a late intake closure process.

Critical analysis was performed to ascertain that, using conventional heat release characteristics, and by comparing the HRM engine with the conventional reciprocating engine, the HRM could match the work output of a reciprocating engine. Since the rate of heat release and the rate of volume expansion control the rate of pressure rise within the chamber of any engine it had to be assured that the HRM could produce a net pressure gain and correspondingly positive net work output.

Since the ratio of the shaft work output to the energy input is a common comparison, called the specific fuel consumption, it would be desirable to have lower specific fuel consumption. The actual specific fuel consumption is the product of the mechanical efficiency, the cycle efficiency (air standard efficiency) and how closely the air standard cycle can be approached considering heat transfer to the chambers and the heat addition rate. It is anticipated that the specific fuel consumption (work out per unit fuel in) for this engine would be lower than what is achievable with the reciprocating engine. This is based on calculations indicating higher mechanical friction and slightly greater combustion chamber surface areas (≈6%), which would promote higher combustion heat losses in the HRM engine. Potential methods of offsetting these losses would be to utilize higher compression ratios and by the potential use of different compression to expansion ratios.

The surface to volume ratios, the surface areas and the crevice volumes are constantly changing as the rotors change angular position in the HRM engine. The HRM engine has a greater surface area rate of change than the baseline reciprocating engine. The HRM engine also has a greater surface to volume ratio than the conventional reciprocating engine. When both factors are taken into account, the smaller surface area of the HRM would indicate the potential for lower isentropic losses while the higher surface to volume ratios would lead one to believe that higher emissions are possible from the engine exhaust prior to after-treatment.

Recommendations

Major areas that are needed to be addressed for the development and to establish commercial viability of the HRM are

- Material selection
- Evaluation of mechanical friction
- Sealing design and analysis
- Intake and exhaust port design and analysis
- Lubrication
- Dynamic analysis of the engine

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APPENDICES

Appendix A

Geometric and Combustion Variations in 100 cc HRM Engine

100 cc with CR 10 HRM Combustion Chamber Analysis

Geometric Parameters

Displacement	100	cc
Major axis	8.5	cm
Minor axis	5.48	cm
Length of Rotor	4.6	cm
Major Radius	5.317	cm
Minor Radius	1.675	cm
Compression Ratio	10	

Angle	CS Area	Volume	SA	BF	ICR
degree	cm^2	cm^3	cm^2	%	
0	2.4	11.1	53.2	0	10.1
45	13.3	61.4	127.7	0	1.8
90	24.3	111.6	202.2	0	1.0
135	13.3	61.4	127.7	0	1.8
180	2.4	11.1	53.2	0.0363596	10.1
225	13.3	61.4	127.7	0.7195646	1.8
270	24.3	111.6	202.2	0.997892	1.0
315	13.3	61.4	127.7	1	1.8
360	2.4	11.1	53.2	1	10.1

Combustion Parameters

Start of Combustion	160	degree
Duration of Combustion	60	degree
Initial Temperature	298	K
Closing of Intake Valve	90	degree
Opening of Exhaust Valve	270	degree
Air / Fuel Ratio	14.8	
Fuel Calorific Value	44,000	KJ/Kg

Angle	T_woh	T_wh	Pres_woh	Pres_wh	Moment
degree	K	K	Kpa	Kpa	N-m
0	750.8	750.8	2572.4	101.3	0.0
45	378.6	378.6	234.2	101.3	10.1
90	298.0	298.0	101.3	101.3	0.0
135	378.6	378.6	234.2	234.2	23.3
180	750.8	924.4	2572.4	3167.0	0.0
225	378.6	3433.9	234.2	2124.4	211.6
270	298.0	298.0	101.3	101.3	0.0
315	378.6	163.8	234.2	101.3	10.1
360	750.8	29.6	2572.4	101.3	0.0

100 cc with CR 15 HRM Combustion Chamber Analysis

Geometric Parameters

Displacement	100	cc
Major axis	8.32	cm
Minor axis	5	cm
Length of Rotor	4.45	cm
Major Radius	5.315	cm
Minor Radius	1.37	cm
Compression Ratio	15	

Angle	CS Area	Volume	SA	BF	ICR
degree	cm^2	cm^3	cm^2	%	
0	1.6	7.2	41.5	0	15.1
45	12.9	57.5	119.3	0	1.9
90	24.2	107.9	197.1	0	1.0
135	12.9	57.5	119.3	0	1.9
180	1.6	7.2	41.5	0.03636	15.1
225	12.9	57.5	119.3	0.719565	1.9
270	24.2	107.9	197.1	0.997892	1.0
315	12.9	57.5	119.3	1	1.9
360	1.6	7.2	41.5	1	15.1

Combustion Parameters

Start of Combustion	160	degree
Duration of Combustion	60	degree
Initial Temperature	298	K
Closing of Intake Valve	90	degree
Opening of Exhaust Valve	270	degree
Air / Fuel Ratio	14.8	
Fuel Calorific Value	44,000	KJ/Kg

Angle	T_woh	T_wh	P_woh	Pres_wh	Ioment
degree	K	K	Kpa	Kpa	N-m
0	881.5	881.5	4511.4	101.3	0.0
45	383.2	383.2	244.4	101.3	10.2
90	298.0	298.0	101.3	101.3	0.0
135	383.2	383.2	244.4	244.4	24.6
180	881.5	1054.3	4511.4	5395.5	0.0
225	383.2	3438.2	244.4	2192.4	220.3
270	298.0	298.0	101.3	101.3	0.0
315	383.2	158.9	244.4	101.3	10.2
360	881.5	19.8	4511.4	101.3	0.0

100 cc with CR 20 HRM Combustion Chamber Analysis

Geometric Parameters

Displacement	100	cc
Major axis	8.1	cm
Minor axis	4.723	cm
Length of Rotor	4.486	cm
Major Radius	5.244	cm
Minor Radius	1.167	cm
Compression Ratio	20	

Angle	CS Area	Volume	SA	BF	ICR
degree	cm^2	cm^3	cm^2	%	
0	1.2	5.2	35.2	0	20.2
45	12.4	55.6	115.1	0	1.9
90	23.6	105.9	195.0	0	1.0
135	12.4	55.6	115.1	0	1.9
180	1.2	5.2	35.2	0.0363596	20.2
225	12.4	55.6	115.1	0.7195646	1.9
270	23.6	105.9	195.0	0.997892	1.0
315	12.4	55.6	115.1	1	1.9
360	1.2	5.2	35.2	1	20.2

Combustion Parameters

Start of Combustion	160	degree
Duration of Combustion	60	degree
Initial Temperature	298	K
Closing of Intake Valve	90	degree
Opening of Exhaust Valve	270	degree
Air / Fuel Ratio	14.8	
Fuel Calorific Value	44,000	KJ/Kg

Angle	T_woh	T_wh	P_woh	P_wh	Moment
degree	K	K	Kpa	Kpa	N-m
0	991.5	991.5	6807.3	101.3	0.0
45	385.7	385.7	249.9	101.3	10.1
90	298.0	298.0	101.3	101.3	0.0
135	385.7	385.7	249.9	249.9	24.9
180	991.5	1163.3	6807.3	7987.0	0.0
225	385.7	3440.5	249.9	2229.2	222.2
270	298.0	298.0	101.3	101.3	0.0
315	385.7	156.4	249.9	101.3	10.1
360	991.5	14.8	6807.3	101.3	0.0

Appendix B

Geometric and Combustion Variations in 200 cc HRM Engine

200 cc with CR 10 HRM Combustion Chamber Analysis

Geometric Parameters

Displacement	200	cc
Major axis	10.8	cm
Minor axis	6.968	cm
Length of Rotor	5.67	cm
Major Radius	6.756	cm
Minor Radius	2.128	cm
Compression Ratio	10	

Angle	CS Area	Volume	SA	BF	ICR
degree	cm^2	cm^3	cm^2	%	
0	3.9	22.0	83.6	0	10.1
45	21.5	122.1	201.3	0	1.8
90	39.2	222.2	319.0	0	1.0
135	21.5	122.1	201.3	0	1.8
180	3.9	22.0	83.6	0.0363596	10.1
225	21.5	122.1	201.3	0.7195646	1.8
270	39.2	222.2	319.0	0.997892	1.0
315	21.5	122.1	201.3	1	1.8
360	3.9	22.0	83.6	1	10.1

Combustion Parameters

Start of Combustion	160	degree
Duration of Combustion	60	degree
Initial Temperature	298	K
Closing of Intake Valve	90	degree
Opening of Exhaust Valve	270	degree
Air / Fuel Ratio	14.8	
Fuel Calorific Value	44,000	KJ/Kg

Angle	T_woh	T_wh	P_woh	P_wh	Moment
degree	K	K	Kpa	Kpa	N-m
0	750.9	750.9	2573.5	101.3	0.0
45	378.6	378.6	234.2	101.3	20.1
90	298.0	298.0	101.3	101.3	0.0
135	378.6	378.6	234.2	234.2	46.4
180	750.9	924.5	2573.5	3168.3	0.0
225	378.6	3433.9	234.2	2124.4	421.0
270	298.0	298.0	101.3	101.3	0.0
315	378.6	163.8	234.2	101.3	20.1
360	750.9	29.6	2573.5	101.3	0.0

200 cc with CR 15 HRM Combustion Chamber Analysis

Geometric Parameters

Displacement	200	cc
Major axis	10.4	cm
Minor axis	6.318	cm
Length of Rotor	5.66	cm
Major Radius	6.644	cm
Minor Radius	1.715	cm
Compression Ratio	15	

Angle	CS Area	Volume	SA	BF	ICR
degree	cm^2	cm^3	cm^2	%	
0	2.5	14.3	66.0	0	15.0
45	20.2	114.4	189.1	0	1.9
90	37.9	214.5	312.1	0	1.0
135	20.2	114.4	189.1	0	1.9
180	2.5	14.3	66.0	0.0363596	15.0
225	20.2	114.4	189.1	0.7195646	1.9
270	37.9	214.5	312.1	0.997892	1.0
315	20.2	114.4	189.1	1	1.9
360	2.5	14.3	66.0	1	15.0

Combustion Parameters

Start of Combustion	160	degree
Duration of Combustion	60	degree
Initial Temperature	298	K
Closing of Intake Valve	90	degree
Opening of Exhaust Valve	270	degree
Air / Fuel Ratio	14.8	
Fuel Calorific Value	44,000	KJ/Kg

Angle	T_woh	T_wh	P_woh	P_wh	Ioment
degree	K	K	Kpa	Kpa	N-m
0	880.5	880.5	4493.5	101.3	0.0
45	383.2	383.2	244.3	101.3	20.1
90	298.0	298.0	101.3	101.3	0.0
135	383.2	383.2	244.3	244.3	48.4
180	880.5	1053.3	4493.5	5375.1	0.0
225	383.2	3438.2	244.3	2192.0	434.5
270	298.0	298.0	101.3	101.3	0.0
315	383.2	158.9	244.3	101.3	20.1
360	880.5	19.9	4493.5	101.3	0.0

200 cc with CR 20 HRM Combustion Chamber Analysis

Geometric Parameters

Displacement	200	cc
Major axis	10.2	cm
Minor axis	5.948	cm
Length of Rotor	5.612	cm
Major Radius	6.6	cm
Minor Radius	1.47	cm
Compression Ratio	20	

Angle	CS Area	Volume	SA	BF	ICR
degree	cm^2	cm^3	cm^2	%	
0	1.9	10.4	55.5	0	20.2
45	19.6	110.1	181.5	0	1.9
90	37.4	209.8	307.5	0	1.0
135	19.6	110.1	181.5	0	1.9
180	1.9	10.4	55.5	0.03636	20.2
225	19.6	110.1	181.5	0.719565	1.9
270	37.4	209.8	307.5	0.997892	1.0
315	19.6	110.1	181.5	1	1.9
360	1.9	10.4	55.5	1	20.2

Combustion Parameters

Start of Combustion	160	degree
Duration of Combustion	60	degree
Initial Temperature	298	K
Closing of Intake Valve	90	degree
Opening of Exhaust Valve	270	degree
Air / Fuel Ratio	14.8	
Fuel Calorific Value	44,000	KJ/Kg

Angle	T_woh	T_wh	P_woh	P_wh	Moment
degree	K	K	Kpa	Kpa	N-m
0	990.8	990.8	6791.3	101.3	0.0
45	385.7	385.7	249.9	101.3	20.0
90	298.0	298.0	101.3	101.3	0.0
135	385.7	385.7	249.9	249.9	49.3
180	990.8	1162.7	6791.3	7969.1	0.0
225	385.7	3440.5	249.9	2229.1	440.2
270	298.0	298.0	101.3	101.3	0.0
315	385.7	156.4	249.9	101.3	20.0
360	990.8	14.8	6791.3	101.3	0.0

Appendix C

Geometric and Combustion Variations in 300 cc HRM Engine

300 cc with CR 10 HRM Combustion Chamber Analysis

Geometric Parameters

Displacement	300	cc
Major axis	12.32	cm
Minor axis	7.948	cm
Length of Rotor	6.532	cm
Major Radius	7.707	cm
Minor Radius	2.428	cm
Compression Ratio	10	

Angle	CS Area	Volume	SA	BF	ICR
degree	cm^2	cm^3	cm^2	%	
0	5.1	33.1	109.8	0	10.1
45	28.0	183.1	264.0	0	1.8
90	51.0	333.1	418.3	0	1.0
135	28.0	183.1	264.0	0	1.8
180	5.1	33.1	109.8	0.03636	10.1
225	28.0	183.1	264.0	0.719565	1.8
270	51.0	333.1	418.3	0.997892	1.0
315	28.0	183.1	264.0	1	1.8
360	5.1	33.1	109.8	1	10.1

Combustion Parameters

Start of Combustion	160	degree
Duration of Combustion	60	degree
Initial Temperature	298	K
Closing of Intake Valve	90	degree
Opening of Exhaust Valve	270	degree
Air / Fuel Ratio	14.8	
Fuel Calorific Value	44,000	KJ/Kg

Angle	T_woh	T_wh	P_woh	P_wh	Moment
degree	K	K	Kpa	Kpa	N-m
0	750.8	750.8	2572.2	101.3	0.0
45	378.6	378.6	234.2	101.3	30.1
90	298.0	298.0	101.3	101.3	0.0
135	378.6	378.6	234.2	234.2	69.6
180	750.8	924.3	2572.2	3166.7	0.0
225	378.6	3433.9	234.2	2124.3	631.1
270	298.0	298.0	101.3	101.3	0.0
315	378.6	163.8	234.2	101.3	30.1
360	750.8	29.6	2572.2	101.3	0.0

300 cc with CR 15 HRM Combustion Chamber Analysis

Geometric Parameters

Displacement	300	cc
Major axis	11.9	cm
Minor axis	7.23	cm
Length of Rotor	6.485	cm
Major Radius	7.6	cm
Minor Radius	1.96	cm
Compression Ratio	15	

Angle	CS Area	Volume	SA	BF	ICR
degree	cm^2	cm^3	cm^2	%	
0	3.3	21.4	86.5	0	15.0
45	26.4	171.5	247.6	0	1.9
90	49.6	321.5	408.8	0	1.0
135	26.4	171.5	247.6	0	1.9
180	3.3	21.4	86.5	0.0363596	15.0
225	26.4	171.5	247.6	0.7195646	1.9
270	49.6	321.5	408.8	0.997892	1.0
315	26.4	171.5	247.6	1	1.9
360	3.3	21.4	86.5	1	15.0

Combustion Parameters

Start of Combustion	160	degree
Duration of Combustion	60	degree
Initial Temperature	298	K
Closing of Intake Valve	90	degree
Opening of Exhaust Valve	270	degree
Air / Fuel Ratio	14.8	
Fuel Calorific Value	44,000	KJ/Kg

Angle	T_woh	T_wh	P_woh	P_wh	Moment
degree	K	K	Kpa	Kpa	N-m
0	881.2	881.2	4504.8	101.3	0.0
45	383.2	383.2	244.3	101.3	30.1
90	298.0	298.0	101.3	101.3	0.0
135	383.2	383.2	244.3	244.3	72.6
180	881.2	1053.9	4504.8	5388.1	0.0
225	383.2	3438.2	244.3	2192.3	651.3
270	298.0	298.0	101.3	101.3	0.0
315	383.2	158.9	244.3	101.3	30.1
360	881.2	19.8	4504.8	101.3	0.0

300 cc with CR 20 HRM Combustion Chamber Analysis

Geometric Parameters

Displacement	300	cc
Major axis	11.7	cm
Minor axis	6.822	cm
Length of Rotor	6.414	cm
Major Radius	7.575	cm
Minor Radius	1.686	cm
Compression Ratio	20	

Angle	CS Area	Volume	SA	BF	ICR
degree	cm^2	cm^3	cm^2	%	
0	2.4	15.7	72.8	0	20.2
45	25.8	165.8	238.3	0	1.9
90	49.3	315.9	403.8	0	1.0
135	25.8	165.8	238.3	0	1.9
180	2.4	15.7	72.8	0.0363596	20.2
225	25.8	165.8	238.3	0.7195646	1.9
270	49.3	315.9	403.8	0.997892	1.0
315	25.8	165.8	238.3	1	1.9
360	2.4	15.7	72.8	1	20.2

Combustion Parameters

Start of Combustion	160	degree
Duration of Combustion	60	degree
Initial Temperature	298	K
Closing of Intake Valve	90	degree
Opening of Exhaust Valve	270	degree
Air / Fuel Ratio	14.8	
Fuel Calorific Value	44,000	KJ/Kg

Angle	T_woh	T_wh	P_woh	P_wh	Moment
degree	K	K	Kpa	Kpa	N-m
0	991.4	991.4	6804.4	101.3	0.0
45	385.7	385.7	249.9	101.3	30.1
90	298.0	298.0	101.3	101.3	0.0
135	385.7	385.7	249.9	249.9	74.3
180	991.4	1163.2	6804.4	7983.8	0.0
225	385.7	3440.5	249.9	2229.2	662.9
270	298.0	298.0	101.3	101.3	0.0
315	385.7	156.4	249.9	101.3	30.1
360	991.4	14.8	6804.4	101.3	0.0

Appendix D

Geometric Variations in 200, 400 and 600 cc Reciprocating Engine

Geometric Cheracteristics of 200 cc Reciprocating Engine

Displacement	200	cc
Bore	6.339	cm
Stroke	6.339	cm
CC-Volume	22.22	cm^3
Crank-Radius	3.170	cm
Conn rod - L	9.509	cm
CC-Length	0.704	cm

Combustion Chamber Volume in cm^3

CR 10	22.22

Angle in degree	Stroke L	CR 10	
		Volume	Sur-Area
0	6.34	22.22	77.14
45	5.68	59.97	100.96
90	3.71	139.41	151.09
135	1.20	201.43	190.23
180	0.00	222.28	203.38
225	1.20	201.43	190.23
270	3.71	139.41	151.09
315	5.68	59.97	100.96
360	6.34	22.22	77.14
405	5.68	59.97	100.96
450	3.71	139.41	151.09
495	1.20	201.43	190.23
540	0.00	222.28	203.38
585	1.20	201.43	190.23
630	3.71	139.41	151.09
675	5.68	59.97	100.96
720	6.34	22.22	77.14

Combustion Chamber Volume in cm^3

CR 15	14.35

Angle in degree	Stroke L	CR 15	
		Volume	Sur-Area
0	6.34	14.29	72.14
45	5.68	52.04	95.96
90	3.71	131.48	146.09
135	1.20	193.50	185.22
180	0.00	214.35	198.37
225	1.20	193.50	185.22
270	3.71	131.48	146.09
315	5.68	52.04	95.96
360	6.34	14.29	72.14
405	5.68	52.04	95.96
450	3.71	131.48	146.09
495	1.20	193.50	185.22
540	0.00	214.35	198.37
585	1.20	193.50	185.22
630	3.71	131.48	146.09
675	5.68	52.04	95.96
720	6.34	14.29	72.14

Combustion Chamber Volume in cm^3

CR 20	10.59

Angle in degree	Stroke L	CR 20	
		Volume	Sur-Area
0	6.34	10.53	69.76
45	5.68	48.28	93.59
90	3.71	127.72	143.71
135	1.20	189.74	182.85
180	0.00	210.59	196.00
225	1.20	189.74	182.85
270	3.71	127.72	143.71
315	5.68	48.28	93.59
360	6.34	10.53	69.76
405	5.68	48.28	93.59
450	3.71	127.72	143.71
495	1.20	189.74	182.85
540	0.00	210.59	196.00
585	1.20	189.74	182.85
630	3.71	127.72	143.71
675	5.68	48.28	93.59
720	6.34	10.53	69.76

Geometric Cheracteristics of 400 cc Reciprocating Engine

Displacement	400	cc
Bore	7.987	cm
Stroke	7.987	cm
CC-Volume	44.44	cm^3
Crank-Radius	3.994	cm
Conn rod - L	11.981	cm
CC-Length	0.887	cm

Combustion Chamber Volume in cm^3

CR 10	44.61

Angle in degree	Stroke L	CR 10	
		Volume	Sur-Area
0	7.99	44.44	122.46
45	7.15	119.95	160.28
90	4.68	278.85	239.86
135	1.51	402.92	301.99
180	0.00	444.61	322.87
225	1.51	402.92	301.99
270	4.68	278.85	239.86
315	7.15	119.95	160.28
360	7.99	44.44	122.46
405	7.15	119.95	160.28
450	4.68	278.85	239.86
495	1.51	402.92	301.99
540	0.00	444.61	322.87
585	1.51	402.92	301.99
630	4.68	278.85	239.86
675	7.15	119.95	160.28
720	7.99	44.44	122.46

Combustion Chamber Volume in cm^3

CR 15	28.74

Angle in degree	Stroke L	CR 15	
		Volume	Sur-Area
0	7.99	28.57	114.51
45	7.15	104.08	152.33
90	4.68	262.98	231.91
135	1.51	387.05	294.04
180	0.00	428.74	314.92
225	1.51	387.05	294.04
270	4.68	262.98	231.91
315	7.15	104.08	152.33
360	7.99	28.57	114.51
405	7.15	104.08	152.33
450	4.68	262.98	231.91
495	1.51	387.05	294.04
540	0.00	428.74	314.92
585	1.51	387.05	294.04
630	4.68	262.98	231.91
675	7.15	104.08	152.33
720	7.99	28.57	114.51

Combustion Chamber Volume in cm^3

CR 20	21.22

Angle in degree	Stroke L	CR 20	
		Volume	Sur-Area
0	7.99	21.05	110.75
45	7.15	96.56	148.57
90	4.68	255.46	228.14
135	1.51	379.53	290.28
180	0.00	421.22	311.16
225	1.51	379.53	290.28
270	4.68	255.46	228.14
315	7.15	96.56	148.57
360	7.99	21.05	110.75
405	7.15	96.56	148.57
450	4.68	255.46	228.14
495	1.51	379.53	290.28
540	0.00	421.22	311.16
585	1.51	379.53	290.28
630	4.68	255.46	228.14
675	7.15	96.56	148.57
720	7.99	21.05	110.75

Geometric Cheracteristics of 600 cc Reciprocating Engine

Displacement	600	cc
Bore	9.143	cm
Stroke	9.143	cm
CC-Volume	66.66	cm^3
Crank-Radius	4.572	cm
Conn rod - L	13.715	cm
CC-Length	1.015	cm

Combustion Chamber Volume in cm^3

CR 10	66.94

Angle in degree	Stroke L	CR 10	
		Volume	Sur-Area
0	9.14	66.66	160.47
45	8.19	179.94	210.03
90	5.36	418.30	314.31
135	1.73	604.40	395.73
180	0.00	666.94	423.09
225	1.73	604.40	395.73
270	5.36	418.30	314.31
315	8.19	179.94	210.03
360	9.14	66.66	160.47
405	8.19	179.94	210.03
450	5.36	418.30	314.31
495	1.73	604.40	395.73
540	0.00	666.94	423.09
585	1.73	604.40	395.73
630	5.36	418.30	314.31
675	8.19	179.94	210.03
720	9.14	66.66	160.47

Combustion Chamber Volume in cm^3

CR 15	43.14

Angle in degree	Stroke L	CR 15	
		Volume	Sur-Area
0	9.14	42.86	150.06
45	8.19	156.14	199.62
90	5.36	394.50	303.90
135	1.73	580.60	385.32
180	0.00	643.14	412.68
225	1.73	580.60	385.32
270	5.36	394.50	303.90
315	8.19	156.14	199.62
360	9.14	42.86	150.06
405	8.19	156.14	199.62
450	5.36	394.50	303.90
495	1.73	580.60	385.32
540	0.00	643.14	412.68
585	1.73	580.60	385.32
630	5.36	394.50	303.90
675	8.19	156.14	199.62
720	9.14	42.86	150.06

Combustion Chamber Volume in cm^3

CR 20	31.86

Angle in degree	Stroke L	CR 20	
		Volume	Sur-Area
0	9.14	31.58	145.13
45	8.19	144.86	194.68
90	5.36	383.22	298.96
135	1.73	569.32	380.38
180	0.00	631.86	407.75
225	1.73	569.32	380.38
270	5.36	383.22	298.96
315	8.19	144.86	194.68
360	9.14	31.58	145.13
405	8.19	144.86	194.68
450	5.36	383.22	298.96
495	1.73	569.32	380.38
540	0.00	631.86	407.75
585	1.73	569.32	380.38
630	5.36	383.22	298.96
675	8.19	144.86	194.68
720	9.14	31.58	145.13

Appendix E

Pressure Variations in 200, 400 and 600 cc Reciprocating Engine

Pressure Variations in Reciprocating Engine

Combustion Parameters

Start Angle of Combustion	320	degree
Duration of Combustion	120	degree
Start Temperature of Combustion	298	K
Opening of Exhaust Valve	540	degree
Air Fuel Ratio	14.8	
Fuel Calorific Value	44,000	KJ/Kg

Pressure in Kpa			
Crank Angle	200 cc Engine		
degree	CR 10	CR 15	CR 20
0	100.00	100.00	100.00
45	100.00	100.00	100.00
90	100.00	100.00	100.00
135	100.00	100.00	100.00
180	99.97	99.97	99.97
225	115.50	116.15	116.48
270	195.51	201.91	205.24
315	648.64	756.07	821.91
360	3054.10	5237.68	7696.63
405	2155.94	2414.41	2567.42
450	1787.38	1827.86	1848.81
495	1580.93	1586.86	1589.84
540	1492.71	1492.72	1492.72
585	100.00	100.00	100.00
630	100.00	100.00	100.00
675	100.00	100.00	100.00
720	100.00	100.00	100.00

Pressure in Kpa			
Crank Angle	400 cc Engine		
degree	CR 10	CR 15	CR 20
0	100.00	100.00	100.00
45	100.00	100.00	100.00
90	100.00	100.00	100.00
135	100.00	100.00	100.00
180	99.95	99.95	99.95
225	115.48	116.13	116.46
270	195.48	201.88	205.21
315	648.56	756.04	821.88
360	3054.10	5239.97	7701.29
405	2155.73	2414.33	2567.33
450	1787.17	1827.66	1848.59
495	1580.73	1586.66	1589.64
540	1492.78	1492.79	1492.79
585	100.00	100.00	100.00
630	100.00	100.00	100.00
675	100.00	100.00	100.00
720	100.00	100.00	100.00

Pressure in Kpa			
Crank Angle	600 cc Engine		
degree	CR 10	CR 15	CR 20
0	100.00	100.00	100.00
45	100.00	100.00	100.00
90	100.00	100.00	100.00
135	100.00	100.00	100.00
180	99.95	99.95	99.94
225	115.47	116.12	116.45
270	195.47	201.86	205.20
315	648.53	755.98	821.80
360	3054.10	5239.20	7699.73
405	2155.65	2414.16	2567.13
450	1787.08	1827.55	1848.49
495	1580.65	1586.57	1589.55
540	1492.81	1492.82	1492.82
585	100.00	100.00	100.00
630	100.00	100.00	100.00
675	100.00	100.00	100.00
720	100.00	100.00	100.00

Appendix F

Torque Variations in 200, 400 and 600 cc Reciprocating Engine

Torque Variations in Reciprocating Engine

Combustion Parameters

Start Angle of Combustion	320	degree
Duratioı	120	degree
Combustion Start Temperature	298	K
Opening of Exhaust Valve	540	degree
Air Fuel Ratio	14.8	
Fuel Calorific Value	44,000	KJ/Kg

Crank	Torque in N-m		
Angle	200 cc Engine		
degree	CR 10	CR 15	CR 20
0	-2	-1	-1
45	-6	-5	-5
90	-14	-13	-13
135	-20	-19	-19
180	-22	-21	-21
225	-23	-22	-22
270	-27	-27	-26
315	-39	-39	-40
360	68	75	81
405	129	126	124
450	249	240	236
495	318	307	302
540	332	320	314
585	-20	-19	-19
630	-14	-13	-13
675	-6	-5	-5
720	-2	-1	-1

Crank	Torque in N-m					
Angle	400 cc Engine			600 cc Engine		
degree	CR 10	CR 15	CR 20	CR 10	CR 15	CR 20
0	-4	-3	-2	-7	-4	-3
45	-12	-10	-10	-18	-16	-14
90	-28	-26	-26	-42	-39	-38
135	-40	-39	-38	-60	-58	-57
180	-44	-43	-42	-67	-64	-63
225	-47	-45	-44	-70	-67	-66
270	-55	-53	-52	-82	-80	-79
315	-78	-79	-79	-117	-118	-119
360	136	150	162	204	225	243
405	259	251	248	388	377	372
450	498	481	472	748	721	708
495	637	614	603	955	921	905
540	664	640	629	996	960	943
585	-40	-39	-38	-60	-58	-57
630	-28	-26	-26	-42	-39	-38
675	-12	-10	-10	-18	-16	-14
720	-4	-3	-2	-7	-4	-3