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UNCOOLED TWO-STROKE GAS ENGINE FOR HEAT PUMP DRIVE

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ABSTRACT

This paper describes the design and analysis of a family of natural gas fueled, uncooled, two-stroke, lean burn, thermal-ignition engines. The engines were designed specifically to meet the requirements dictated by the commercial heat pump application.* The engines have a power output ranging from 15 to 100 kW; a thermal efficiency of 36 percent; a mean time between failure greater than 3 years; and a life expectancy of 45,000 hours. To meet these specifications a family of very simple, uncooled, two-stroke cycle engines were designed which have no belts, gears or pumps. The engines utilize crankcase scavenging, lubrication, stratified fuel introduction to prevent raw fuel from escaping with the exhaust gas, use of and ceramic rolling contact bearings. The Thermal Ignition Combustion System (TICS) [1] is used for ignition to enable the engines to operate with a lean mixture and eliminate spark plug erosion.

INTRODUCTION

Thermally activated heat pumps have a number of potential benefits. Nearly half of all the natural gas used during a typical heating season could be conserved with efficient gas-fired heat pumps. During the cooling season, consumption of natural gas for space conditioning can reduce electrical demand, which helps both the gas and the electric utilities by eliminating large seasonal peaks and leveling the demand for both utilities.

One obstacle to the widespread use of engine-driven gas-fired heat pumps has been the extreme durability requirements for the prime mover. In order to compete effectively with electrically driven equipment, the durability of present natural gas-fired engines must be increased to on the order of 45,000 hours." [2]

PARAMETRIC OPTIMIZATION

Given the requirement to design a new engine for a power output of 15 to 100 kW, a parametric optimization was carried out to determine the following key engine design parameters:

Number of Cylinders
Brake Mean Effective Pressure (BMEP)
Mean Piston Speed (PS)
Bore to Stroke Ratio (B/S)

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Once these parameters are specified, the piston bore, piston stroke and engine rotational speed of each engine are fixed.

To perform the parametric optimization a series of design goals were determined and ranked as follows:

1. LIFE
2. COST
3. WEIGHT
4. TECHNICAL RISK

The first parameter which was fixed was to make the 15 kW engine with only one cylinder. A historical data base of successful two-stroke cycle engines was then compiled and studied to determine, on a historical basis, practical limits for BMEP, piston speed, and bore/stroke ratio. A simplified design formula for engine weight as a function of these design parameters was also used in determining the sensitivity of engine weight (and by deduction cost) to these same parameters. The result of this optimization was an engine with a rotational speed between 1500 and 2000 RPM. In order to make the engine applicable to co-generation and other generator set applications, the rated engine speed was set at 1800 RPM. The following is a listing of the optimized engine parameters and the resulting engine design dimensions:

POWER RATING	15 kW
RATED SPEED	1800 RPM
BMEP	379 kPA
BORE/STROKE	0.9
PISTON SPEED	7.65 M/S
DISPLACEMENT	1.32 L
BORE	115 mm
STROKE	127 mm

SCAVENGE GEOMETRY

In order to make the engine have the fewest number of parts the "loop scavenged" symmetrical porting configuration was chosen instead of others which utilize intake and/or exhaust valves or which have multiple pistons per cylinder. This type of engine requires that the intake air be provided to the intake port at a pressure higher than the exhaust manifold pressure. This compressed air can be provided by an engine driven compressor or by using the bottom of the piston as a compressor in either the crankcase or in a separate volume (by using a cross-head type piston). For simplicity, and hence long life, the crankcase compression scavenging approach was selected.

An existing engine cycle simulation [3] was modified to model the crankcase compression, loop scavenged, symmetrically ported engine. Using this cycle simulation the scavenge geometry was optimized. Using assumptions based on historical data for an initial design, each of the key scavenge geometries was iterated to determine its sensitivity and optimal value. Figure 1 is a plot of BMEP versus Crankcase Compression Ratio (CCR) which shows that the CCR optimizes between 1.40 and 1.45.

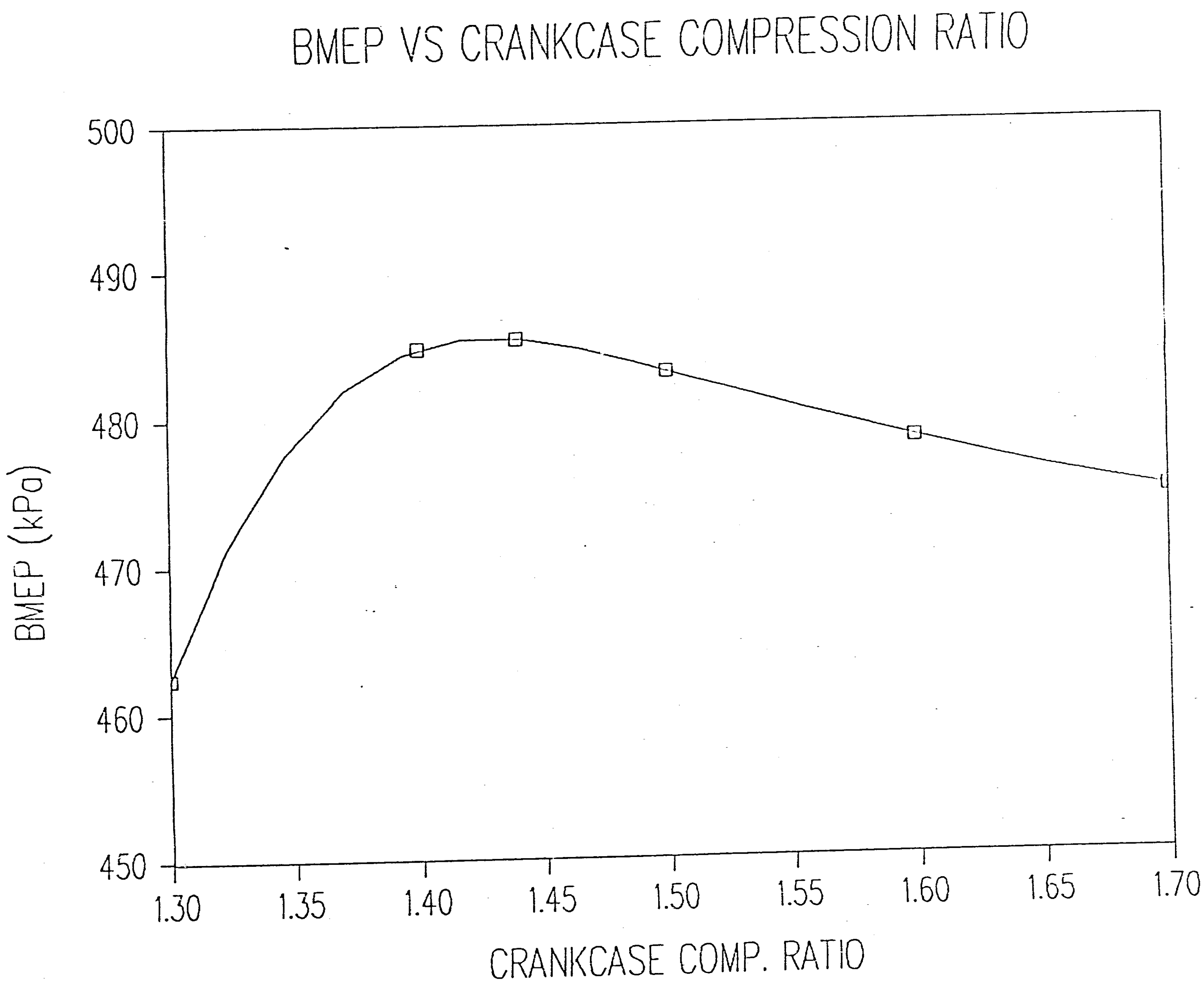

FIGURE 1 BMEP vs. Crankcase Compression Ratio

The other two key geometric variables which required optimization were the inlet port open duration and the exhaust lead. Figure 2 shows that the inlet opening duration optimized at 92 degrees which means that the inlet port opens at 46 degrees before bottom dead center (BDC) and closes at 46 degrees after BDC. Figure 3 illustrates the sensitivity of power output to exhaust lead (exhaust lead is the number of crankangle degrees between the opening of the exhaust port and the opening of the inlet port). This plot shows that a lead of 14 degrees is optimal, but that increased leads have a very minor effect on performance.

PERFORMANCE ANALYSIS

The cycle simulation program was further modified to incorporate the Thermal Ignition Combustion System (TICS) and Thin Thermal Barrier Coatings (TTBC) [3] to predict the performance of the engine. Figure 4 is a plot of the predicted cylinder pressure history for the rated load condition which shows a peak pressure of 5.9 MPa (850 psi). Figure 5 is a plot of the absolute pressure in the cylinder, crankcase and exhaust pipe versus crankangle which shows the pressure potentials which drive exhaust gases into the exhaust pipe and prevent reverse flow of exhaust gas back into the cylinder. It also shows that at the instant which the intake port opens that the cylinder pressure is higher than the crankcase pressure which causes some burnt in-cylinder gases to backflow through the intake port into the transfer passage leading to the crankcase. The temperatures in the crankcase, cylinder and exhaust pipe are shown in Figure 6. At the point when intake air starts flowing into the cylinder the cylinder is divided into two zones, a fresh air zone and a combustion gas zone until an arbitrary point is reached in the cycle (75% cylinder volume). At this arbitrary point, the gases are perfectly mixed and the two zone temperatures become one mixed temperature.

Figure 7 shows a summary of the predicted energy balance of the engine at its rated conditions. The thermal efficiency of the engine is 36.1% and an additional 33.9% of the heat input is available in the exhaust heat for space heat augmentation or for hot water or steam generation. Table 1 lists all of the performance related parameters of the engine.

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MECHANICAL DESIGN

CRANKSHAFT AND BEARINGS

Crankcase scavenged two-stroke cycle engines cannot use plain type bearings but must use rolling contact bearings as the amount of lubricant in the crankcase is only a slight mist. Due to the need to assemble the connecting rod and big end bearing on the crankchaft it is required that either the rolling bearing and connecting rod big end be split or the crankshaft must be made of several pieces. It was decided that the method which will result in the longest engine life is to use a built-up crankshaft, solid race bearings and a one piece connecting rod. The crankshaft is built with integral counterweights to minimize engine vibration, and the engine is fitted with a flywheel which is sized to limit instantaneous torsional speed variations to 2% of rated speed.

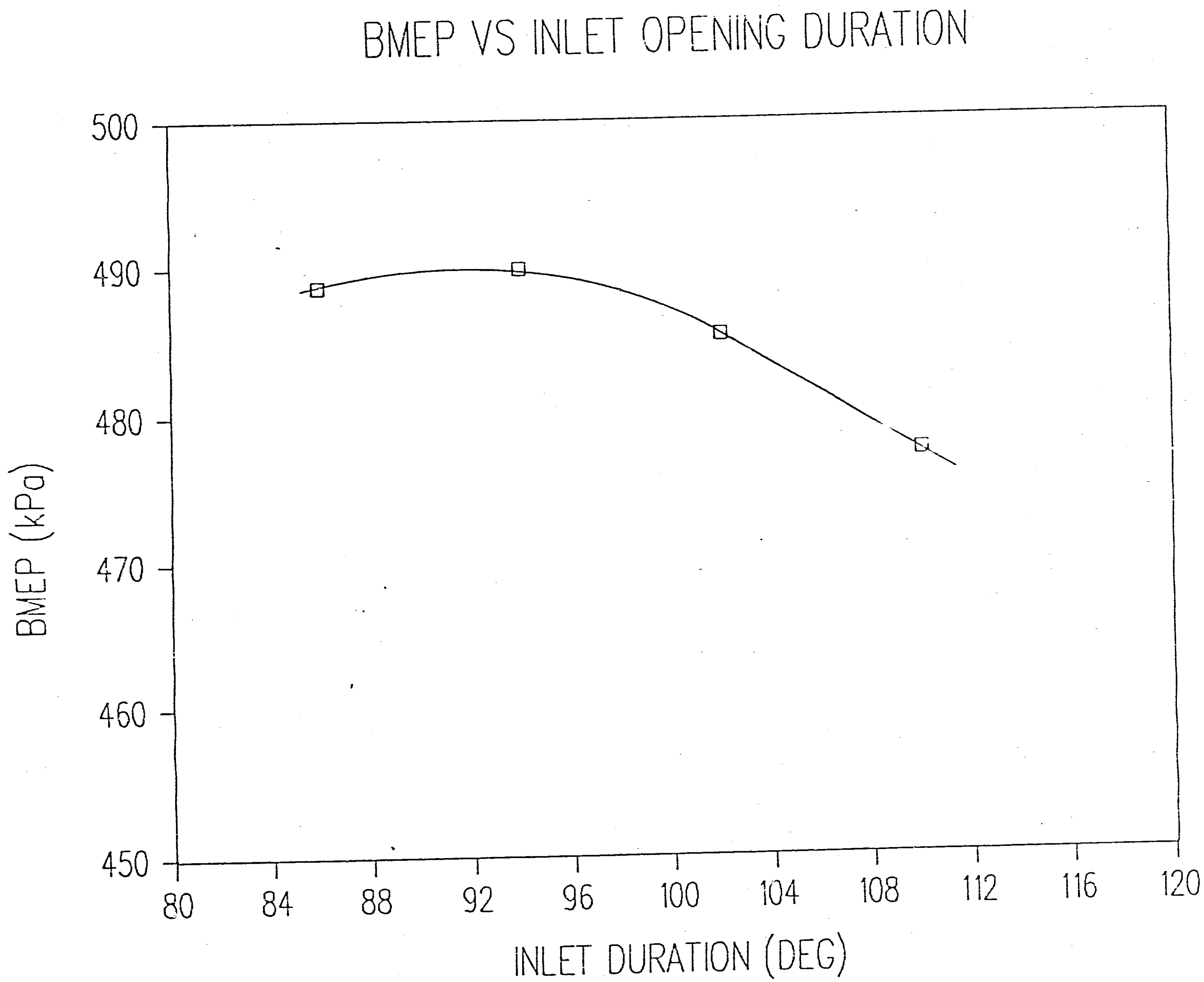

FIGURE 2 BMEP vs. Inlet Opening Duration

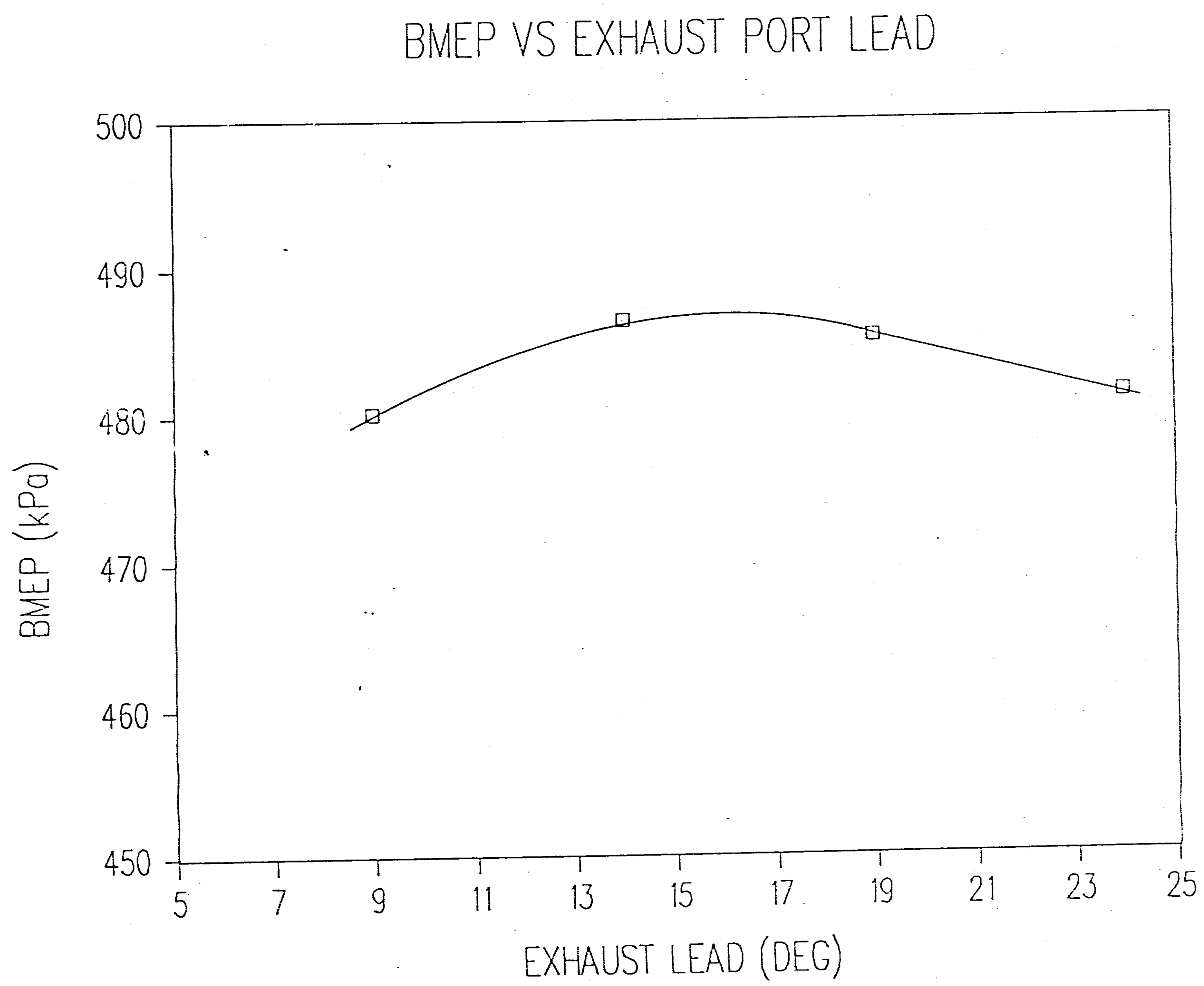

FIGURE 3 BMEP vs. Exhaust Port Lead

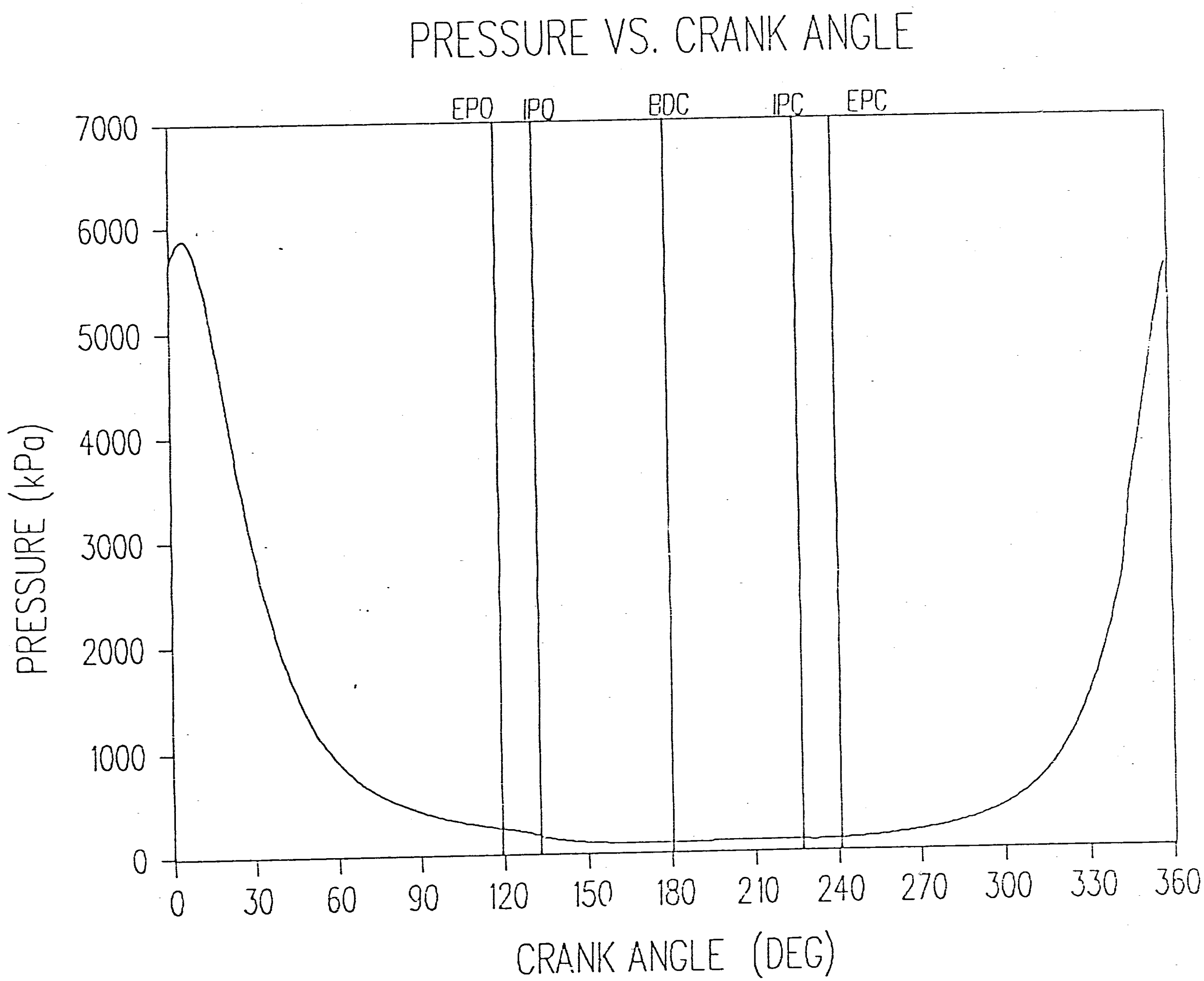

FIGURE 4 Cylinder Pressure vs. Crank Angle

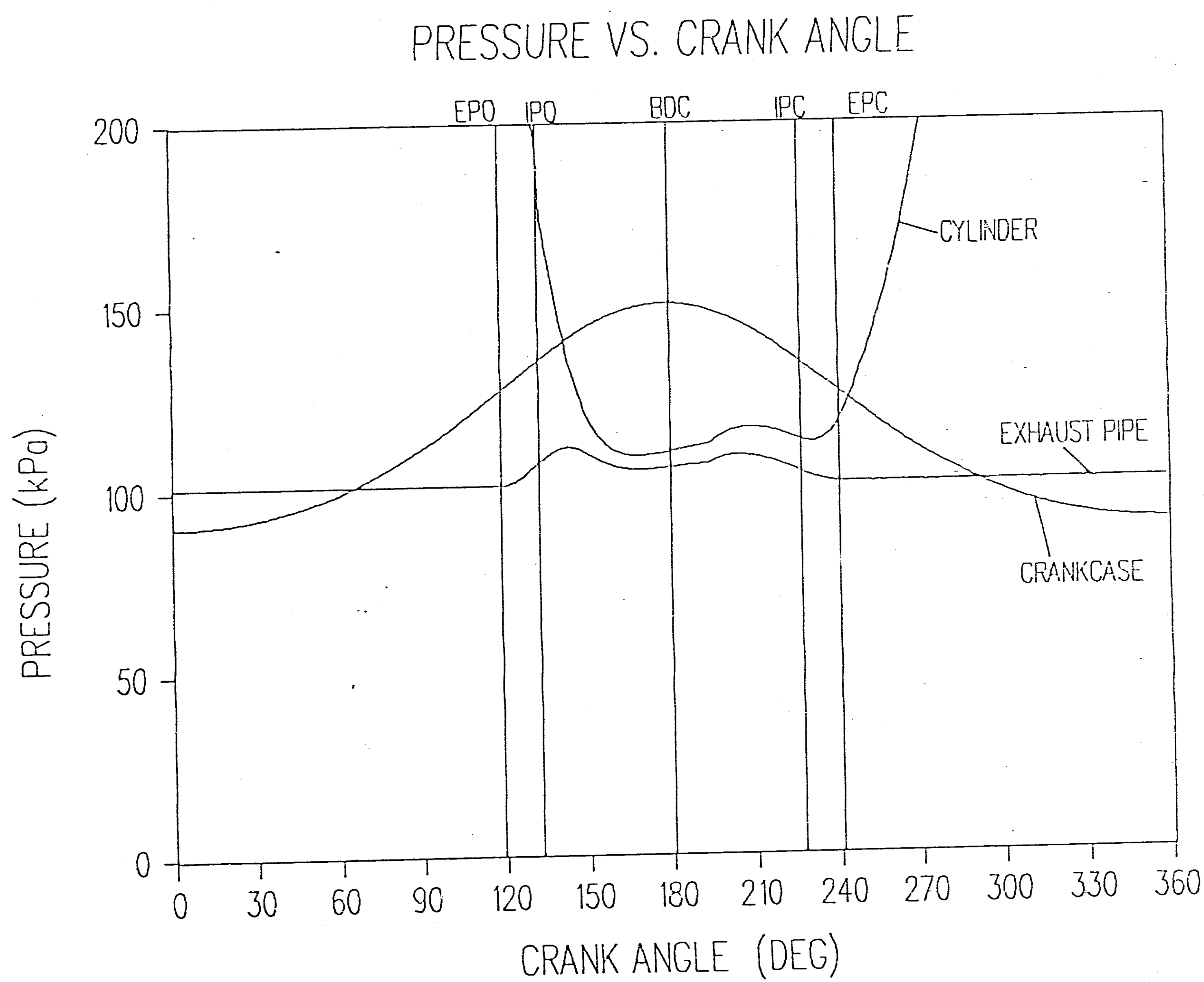

FIGURE 5 Pressure vs. Crank Angle

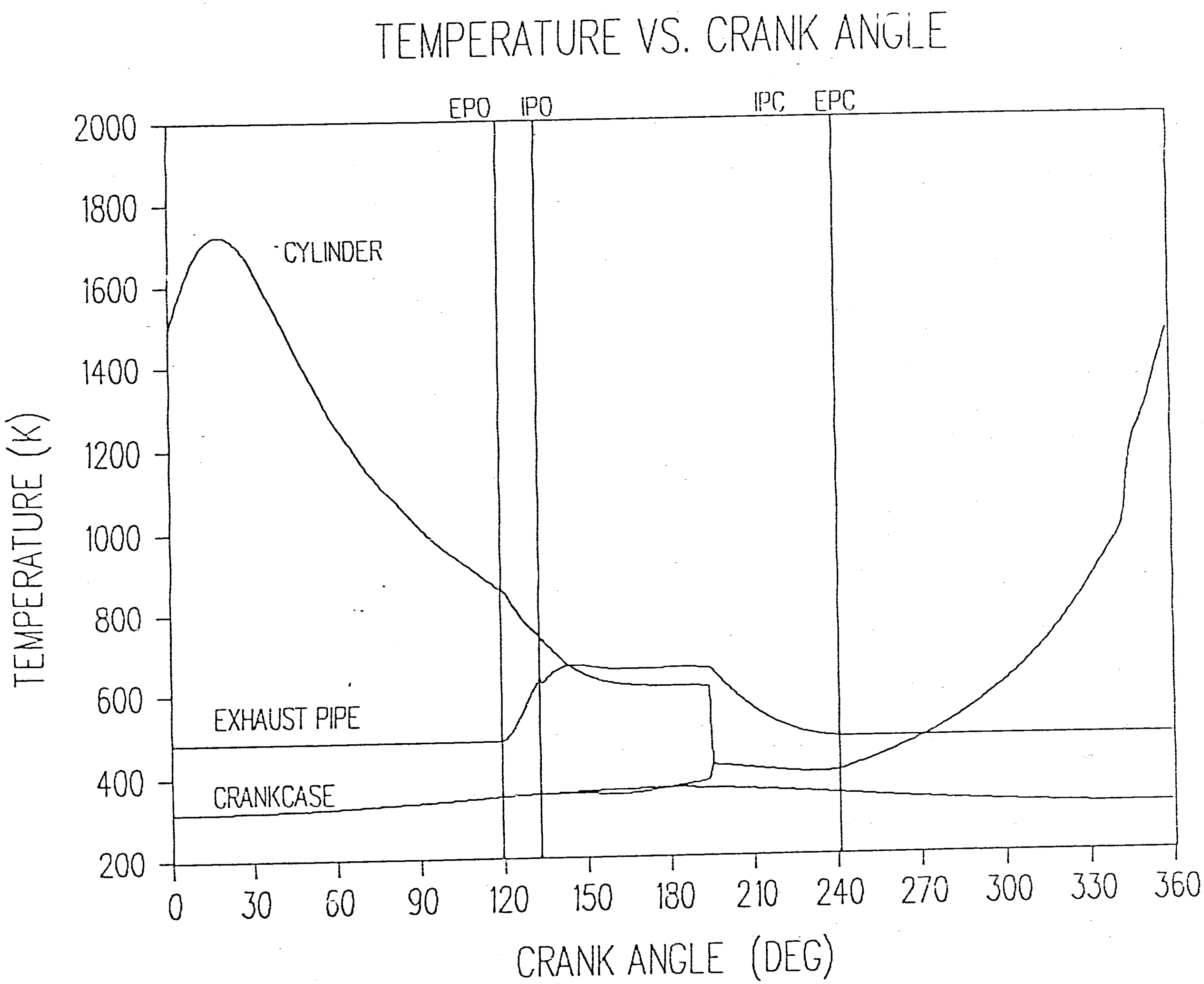

FIGURE 6 Temperature vs. Crank Angle

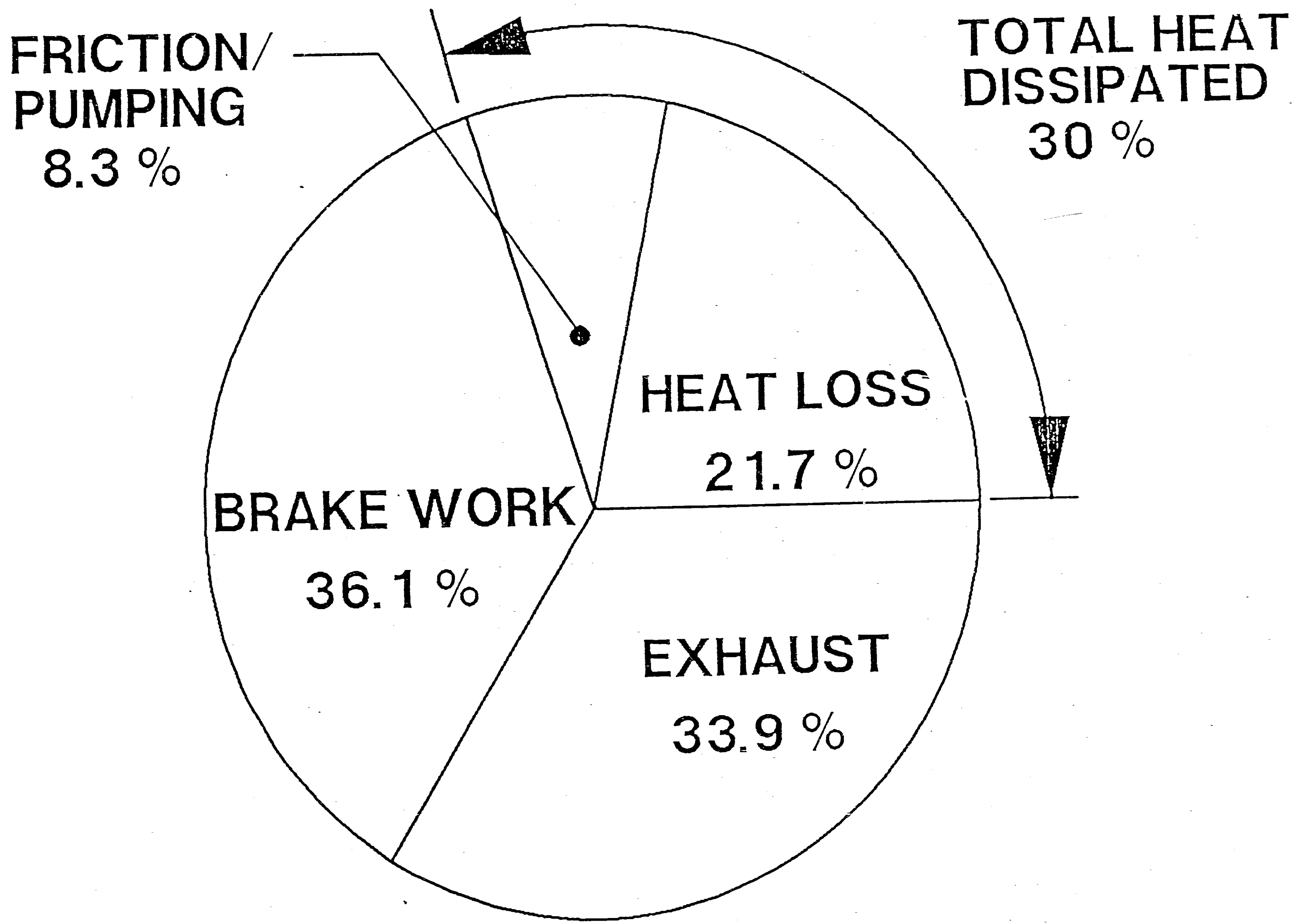
FRICTION/
PUMPING
8.3 %
TOTAL HEAT
DISSIPATED
30 %
HEAT LOSS
21.7 %
BRAKE WORK
36.1 %
EXHAUST
33.9 %

FIGURE 7

TABLE 1

ENGINE RATED POINT PERFORMANCE

BORE	4.52 IN.	(115 mm)
STROKE	5.02 IN.	(127 mm)
DISPLACEMENT	80 CU.IN.	(1303 cc)
TOTAL TICS VOLUME	.53 CU.IN.	(8.74 cc)
GEO. CR.	15.9	
TRAPPED CR	12.7	
CRANKCASE CR	1.44	
CRANKCASE VOLUME	180.7 CU.IN.	(2961 cc)
ENGINE RPM	1800 RPM	
POWER	20 HP	15 KW
FUEL FLOW	.110 LB/MIN, 2357BTU/MIN	49.9 g/MIN, 2487 kJ/MIN
FUEL RATE	6.16 x 10 LB/STROKE, 1.32 BTU/STROKE	.028 g/STROKE, 1.39 kJ/STROKE
AIR FLOW	5.60 LB/MIN, 76.1 CFM @ 80 F	2.54 kg/min, 2154 l/min
TRAPPED AIR FLOW	4.30 LB/MIN	1.95 kg/MIN
BSFC	7057 BTU/HP-HR	9980 KJ/KW-HR
BMEP	55.0 PSI	379.2 kPa
FMEP	12.41 PSI	85.6 kPa
PMEP	.13 PSI	.90 kPa
THERMAL EFFICIENCY	36.1	
TRAPPED A/F RATIO	40/1	
OVERALL A/F RATIO	52.3/1	
DELIVERY RATIO	.929	
TRAPPING EFFICIENCY	75.5%	
CHARGING EFFICIENCY	70.1%	
SCAVENGING EFFICIENCY	92.6%	
SCAVENGING RATIO	.871	
IGNITION TIMING	18 BTDC	

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Selection of rolling element bearings were based on life calculations considering different bearing types (roller, needle and ball), size, loads, speed and bearing material (conventional metal construction and ceramic-metal combinations). The selected bearings all utilize silicon nitride ceramic rolling elements as this material has shown up to a hundredfold increase in life as compared to the best metal bearings, has better capability to operate with marginal lubrication, and does not suffer from problems caused by corrosion. Needle bearings were selected for the wrist pin and crank pin locations and ball bearings are being used for the crankshaft main bearings.

PISTON

In order to retain an acceptable volumetric efficiency for the engine and avoid having an active cooling system, it is necessary to minimize the amount of heat which can transfer through the piston into the crankcase. Also, to minimize the leakage of air from the intake port to the exhaust port around the piston, it is desirable that the piston pin bore not penetrate the piston skirt. To accomplish these objectives the piston is designed with a separate pin carrier which also forms a very effective air gap insulation to reduce heat transfer.

CERAMIC COATINGS

The combustion chamber surfaces of the cylinder head and piston are to be coated with a "Thin Thermal Barrier Coating" [4] tc reduce the transient portion of the heat transfer to raise the temperature of the gases in the cylinder; to increase the rate of combustion and also raise the exhaust temperature.

The cylinder bore will be coated with a ceramic coating system designed for low friction, low wear and elevated temperature capabilities. The coating will be selected from a group of materials including chrome oxide, chrome carbide, etc. which has been shown to be capable of maintaining low friction levels and acceptable wear for 45,000 hours of operation.

ENGINE LAYOUT, SIZE, WEIGHT

The engine layout drawings are shown as Figures 8 thru 13 including an isometric view of the front, intake side corner. The drawings show the exterior of the engine to be heavily ribbed in a waffle pattern. This feature is included to add structural rigidity to the engine and to increase the exterior surface area to enhance the natural convection cooling. The engine has a total parts count (including all anxiliaries and fasteners) of 164 with a total of 87 different parts. The engine has a length of 43 cm (17 inches), a width of 47 cm (18.5 inches) and a height of 69 cm (27 inches). The total weight of the engine (including all auxiliaries such as filters, mufflers, controls, etc. is 130 kg (287 lbs).

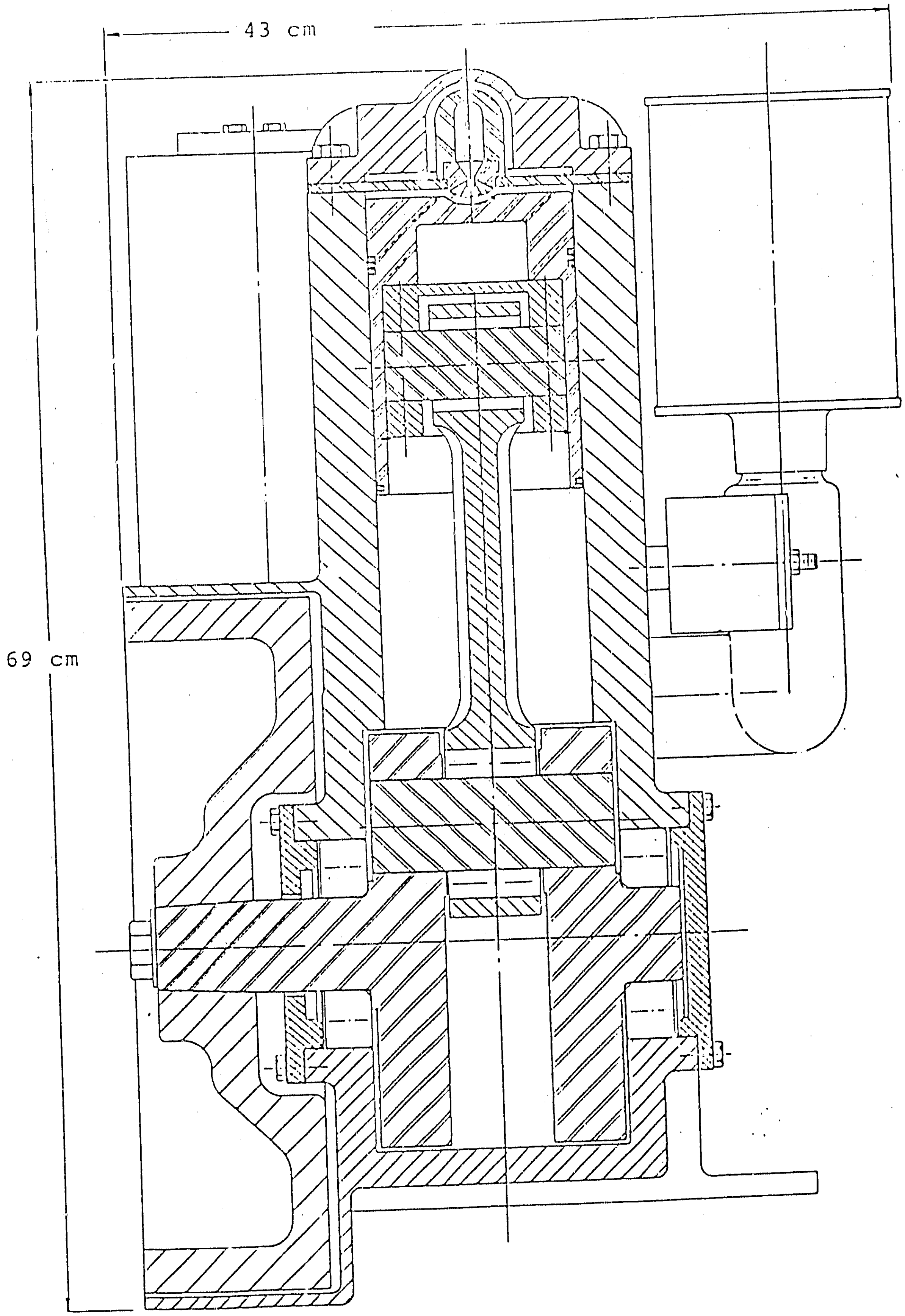

Figure 8 Engine Cross Section

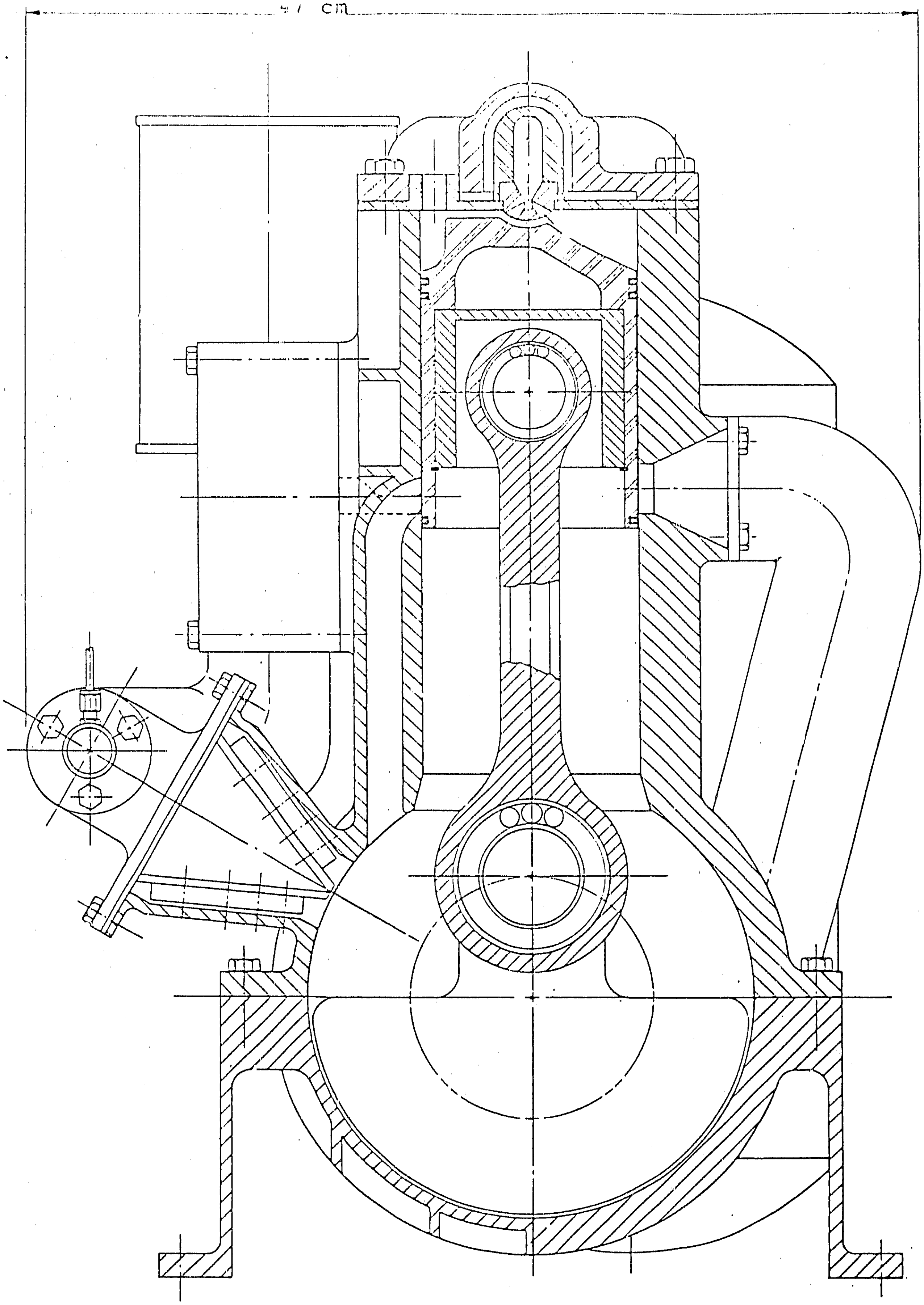

Figure 9 Engine Cross Section

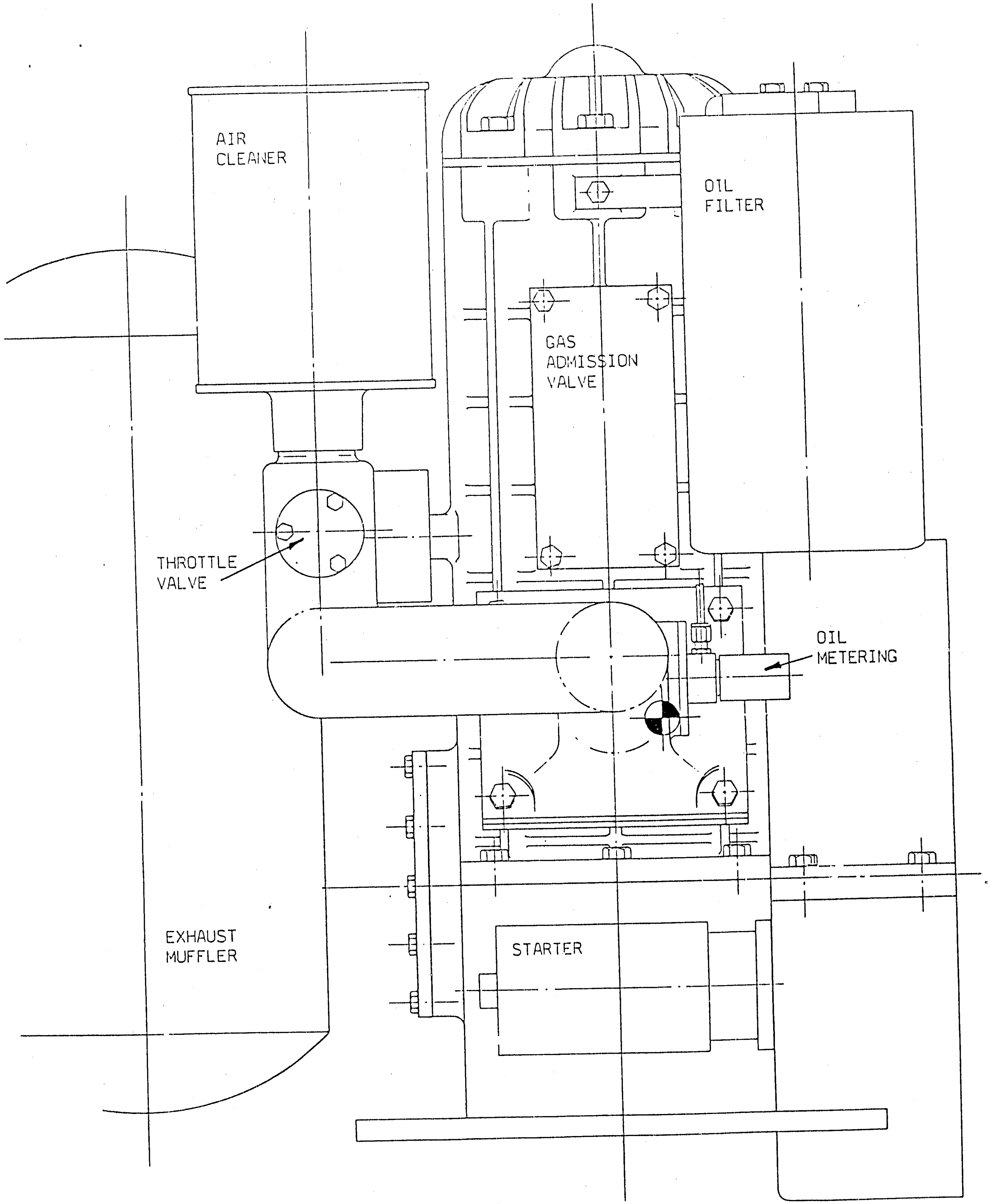

Figure 10 Engine Side View

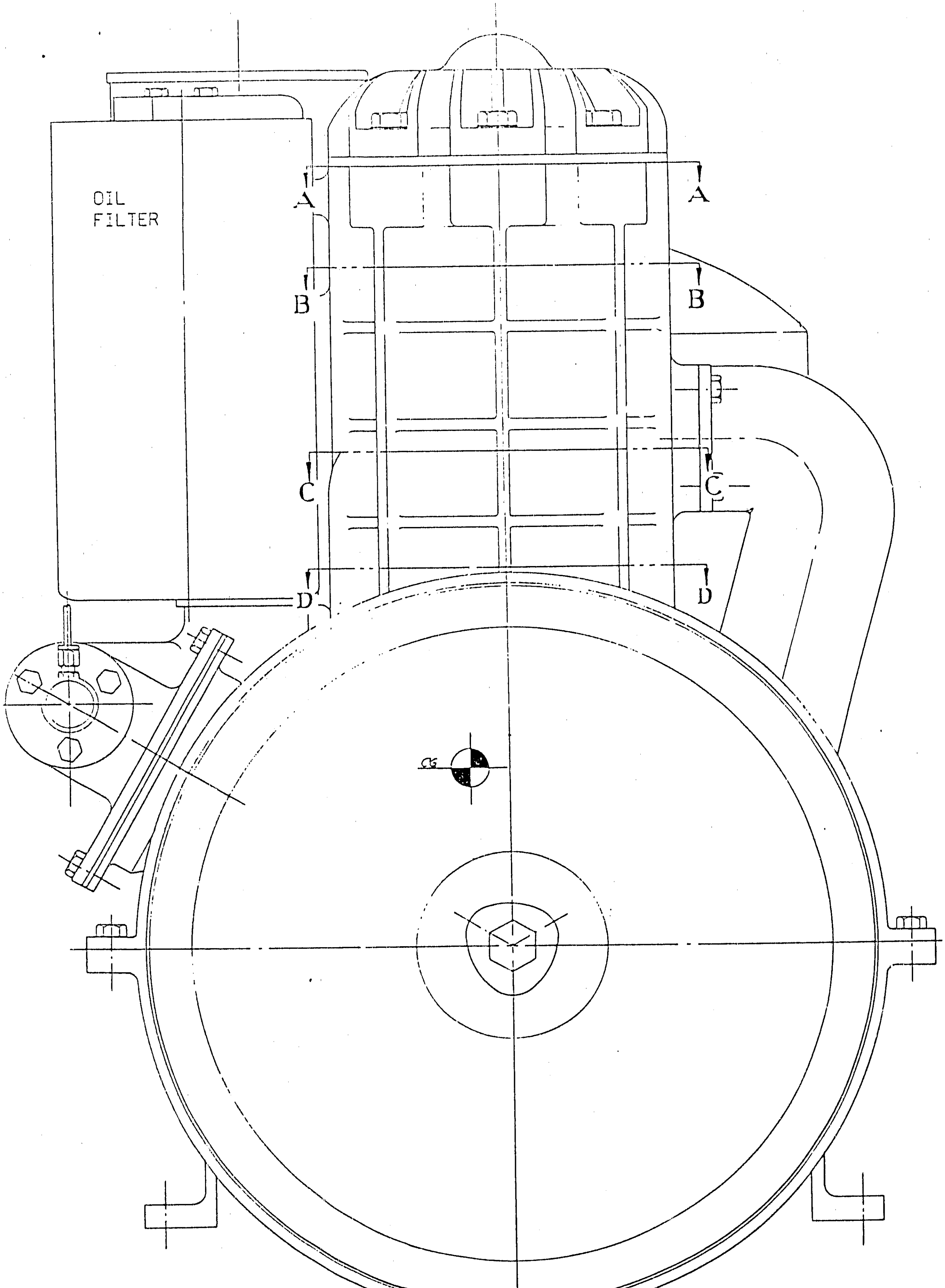

Figure 11 Flywheel End View

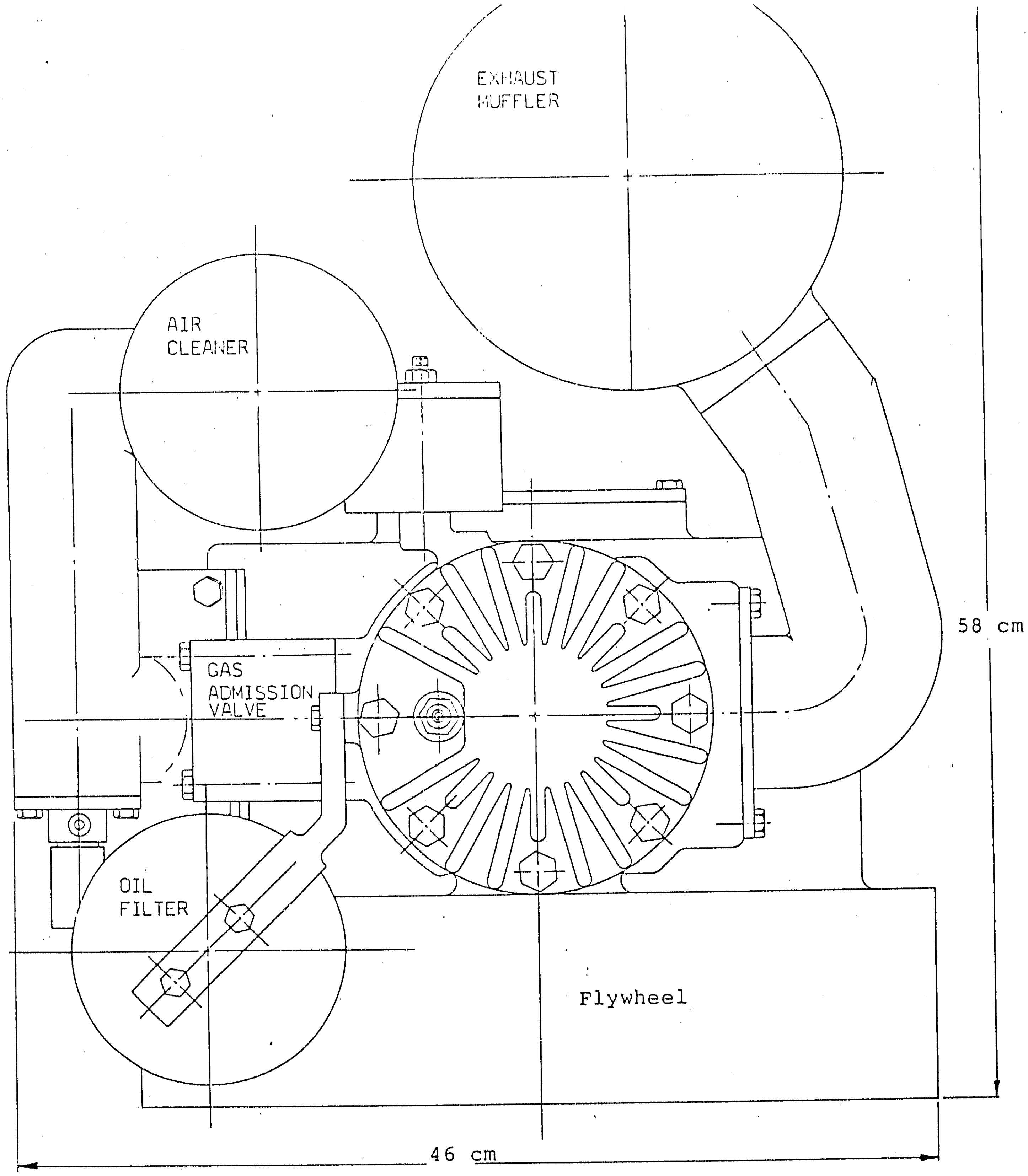

Figure 12 Top View

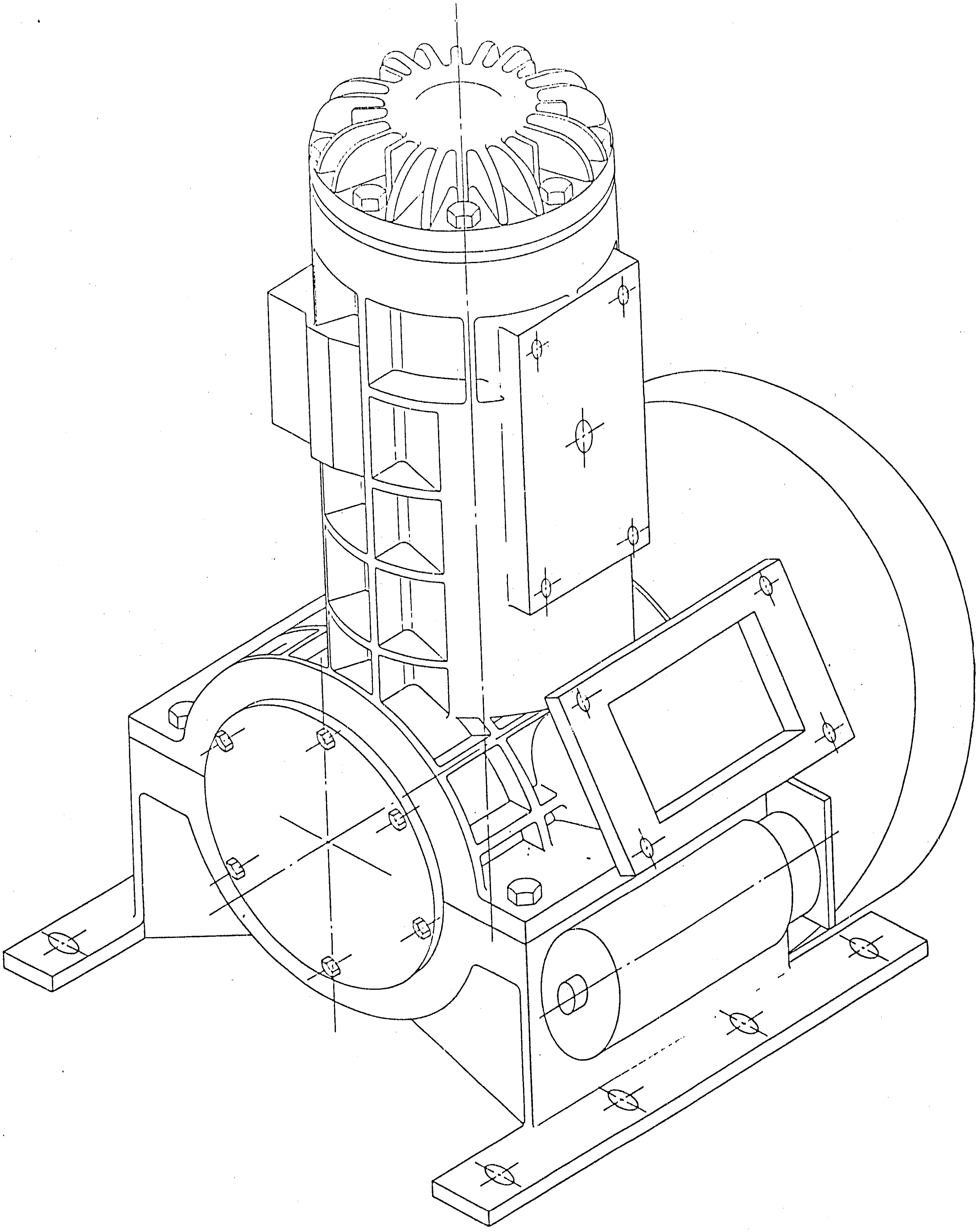

Figure 13 Heat Pump Engine - Isometric

RELIABILITY AND MAINTAINABILITY

As part of the design procedure, a reliability analysis, a failure modes and effects analysis and a maintainabilty analysis were performed. The combined result of these analyses is a mean time of failure estimate for the complete engine including all accessories of 10,253 hours. The vast majority of the failures which determine this failure rate occur with accessory items which can be easily replaced. By eliminating failures which are repairable without removing the cylinder head, crankcase, or otherwise opening up the core of the engine, the basic engine is predicted to have a mean time between failure of 60,060 hours.

are these significant figures

significance ?

Scheduled maintenance for the engine consists of filling the lubricating oil reservoir once yearly (assuming that the engine runs at full power for one half of the time in an average year) and changing the sparkplug (which is used only during starting). Based upon the predicted component reliability, an unscheduled maintenance will be required once every three years to replace a failed auxillary component.

CHARGE STRATIFICATION & THERMAL IGNITION COMBUSTION SYSTEM

In order to achieve high efficiency and low emissions from a two-stroke cycle engine it is important to have a fuel system which prevents raw fuel from escaping with the exhaust gas. To accomplish this the natural gas is being admitted through a separate inlet port which is opened by the piston after the exhaust blowdown; and after and initial blowdown of crankcase air into the cylinder. By delaying the gas introduction until after these events have taken place, the gas fuel is not allowed sufficient time to traverse the cylinder to the exhaust port, prior to exhaust port closure. Prior to producing a running engine this feature will be verified by running a full scale flow visualization model of the engine, photographing the air and fuel flows, and modifying the geometry of the ports and piston top contour to assure that no raw fuel escapes.

In order to ignite lean mixtures of natural gas and air it is necessary to have an extremely high source of ignition energy. To provide this energy the engine has been designed to use the Thermal Ignition Combustion System (TICS). TICS is a proven concept [1] using an uncooled precombustion chamber, which operates at a very high temperature to provide a source of energy to heat the air and fuel charge in the chamber to its ignition temperature. A jet of hot gases then sprays out of the prechamber and ignites the remainder of the fuel charge in the main chamber.

COMPONENT TEMPERATURE

As an aid to determining the estimated operating temperature of the major engine components, a finite element model of the complete engine was generated. The components modeled were the cylinder head and Thermal Ignition Combustion System antechamber which were modeled using a three dimensional model, and a cross section thru the engine using a two dimensional model. Additional two dimensional models were made for cross sections of the cylinder block in the area below the cylinder head and thru the intake and exhaust ports. Boundary conditions for the models were determined by using outputs from the cycle simulation program for cylinder pressure, cylinder temperature and in-cylinder heat transfer coefficients. Heat transfer conditions for the exterior of the engine were based upon published convective heat transfer coefficients for quiescent air. Figure 14 is a view of the combustion side of a section of the cylinder head which shows the interior surface of the TICS chamber to be at a temperature of 960 C (1760 F) while the temperature of the cylinder head surface in the main combustion chamber is less than 538 C (1000 F) and the exterior surface is less than 315 C (600 F).

Figure 15 shows the thermal results for the complete engine and Figure 16 is an enlargement showing details in the head/piston area. These figures dramatically illustrate the insulation effect of the air gap between the TICS chamber and cylinder head, and between the piston and pin carrier.

SUMMARY AND CONCLUSIONS

The heat pump application demands an efficient internal combustion engine which has the life and reliability of an electric motor, is as maintenance free as a gas furnace, which has acceptable exhaust emissions and is extremely quiet. These demands require that a new approach to engine design be used. A reciprocating engine has been designed which meets these goals by using a very simple uncooled, two-stroke cycle engine approach with an absolute minimal number of moving parts and advanced materials, tribology and combustion technology made possible by research in the adiabatic engine field.

ACKNOWLEDGEMENTS

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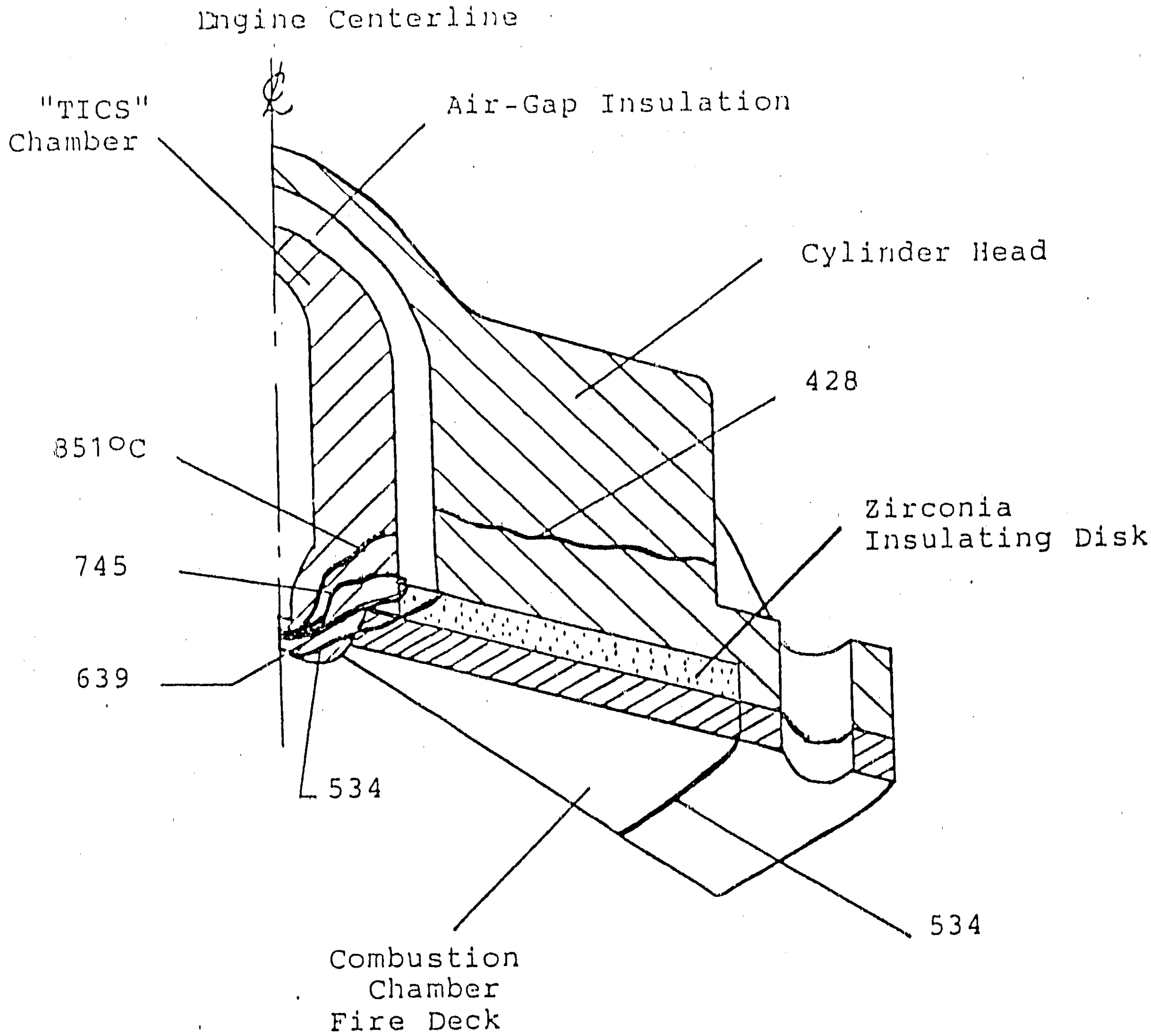

Figure 14 , Temperature Distribution of Cylinder Head (Isometric View of a Cross-Section Viewed From the Cylinder Side).

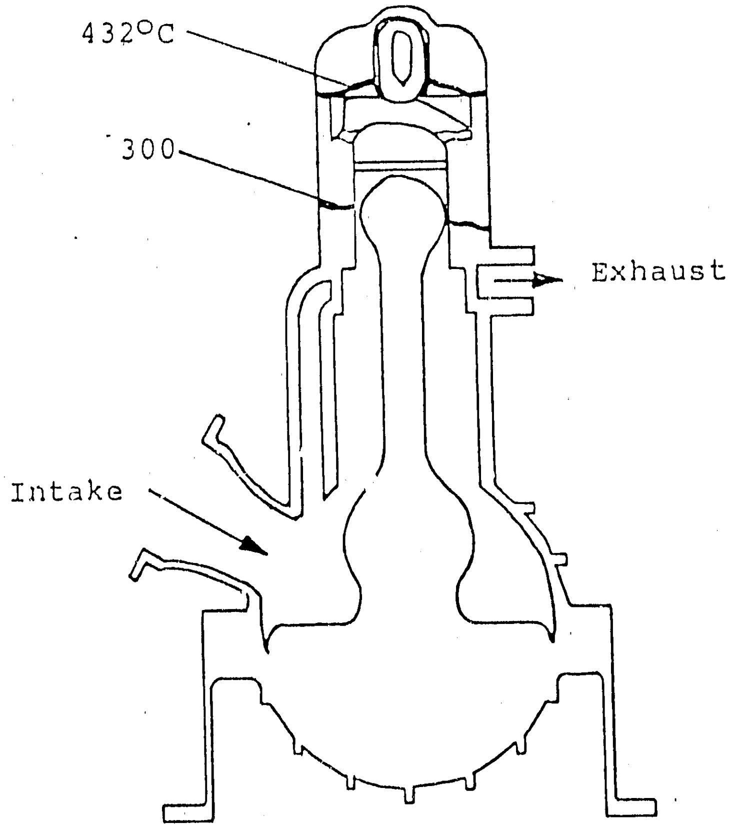

Figure 15 Thermal Model of Engine

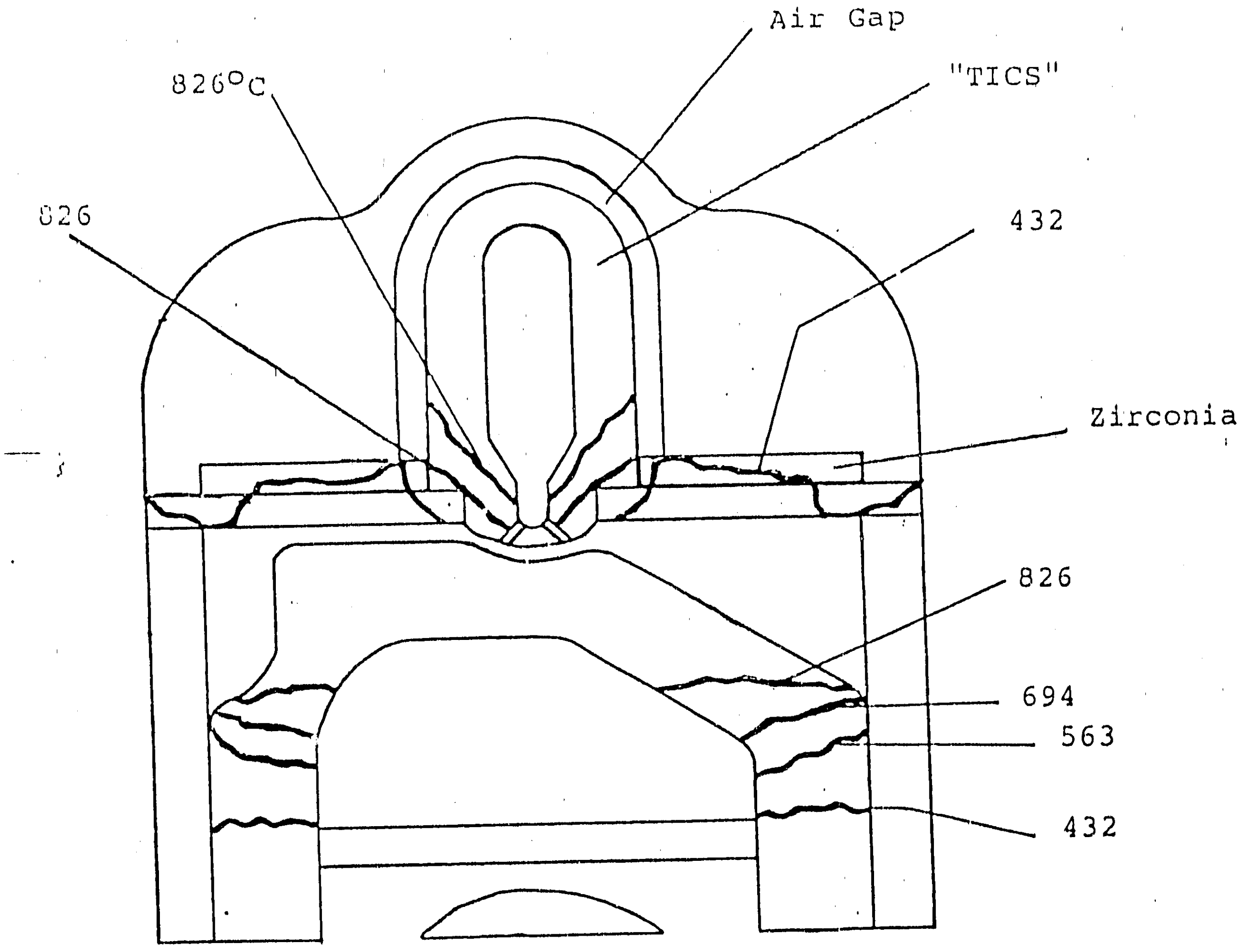

Figure 16 Thermal Model of Head/Piston Area

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