ROTARY ENGINE

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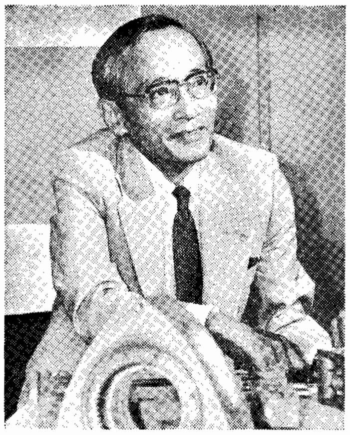

PREFACE

Nikolaus August Otto of Germany succeeded in making a practical internal combustion engine in 1876, and Gottlieb Daimler made an automobile utilizing an internal combustion engine in 1886. These were both the reciprocating type piston engines. Since then, the development of the internal combustion engine for automotive purposes has been remarkable. As a consequence, however, the history of the internal combustion engine mainly consisted of engines that possessed the reciprocating mechanism.

It is not appropriate to conclude, however, that the mechanism of the internal combustion engine must be of the reciprocating type solely because the early history of the internal combustion engine was primarily that of the reciprocating one.

As the matter of fact, there have been numerous challenges made, in the course of development of the internal combustion engine, to create practical rotary engines.

However, satisfying all the requirements for a practical internal combustion engine is not so simple as to be achieved by a mere casual idea and automotive applications demand still more rigorous requirements. This explains the reason why many efforts in the past for perfecting a practical rotary engine did not succeed.

There is solid background and good reason why the NSU-Wankel type rotary engine invented by Felix Wankel have become the only practical rotary engine. First, its principle is superior. Secondly, enormous and untiring efforts have been made to make it work.

No matter how innovative an invention may be, there is no invention that is absolutely perfect from the beginning. The case of the NSU-Wankel type rotary engine is one good example of untiring enthusiasm and effort having prevented an excellent invention from falling into oblivion.

The development history of the NSU-Wankel type rotary engine includes the improvement of the engine itself and the progress of its application. The number of years and the scale of research work devoted toward the rotary engine compared with that of the reciprocating engine, however, is still small. Accordingly, this book which introduces the actual circumstances of development of the rotary engine up to the present stage may touch merely part of the full potential of the NSU-Wankel type rotary engine.

We are very pleased that the rotary engine has by now established a firm position in the history of internal combustion engine. We will further continue to pursue the path of improvements and new discoveries.

KENICHI YAMAMOTO

CHAPTER 1
OUTLINE OF THE ROTARY ENGINE

In 1954, Dr. Felix Wankel (West Germany) became the first in the world to successfully develop a rotary engine with the cooperation of NSU (West Germany).

The attention of those in the field of mechanical engineering, not to mention those related to the automotive industry, has gradually increased blending high expectations for the new engine with a cognizance of past failures.

While being watched by engineers throughout the world, this rotary engine was made a practical engine as a result of tenacious effort. In spite of the many trials and difficulties it encountered, it gradually came to be acknowledged as an engine applicable to automobile as well as industrial purposes. Commercial production of the rotary engine began ten odd years ago.

The NSU-Wankel type rotary engine will be described from the viewpoint of its historical background and the requirements for an internal combustion engine.

1.1 HISTORY OF THE ROTARY ENGINE

To this day, many inventors have designed various types of machines to obtain motive power. Most of them were of rotary piston structure composed of rotary motion parts, because the idea was natural and the structure was simple. However, most of such ideas were those that could not provide sufficient airtightness and durability, and were of complicated configuration. Hence, engines with reciprocating pistons were developed first. Untiring research and development of the reciprocating engine as an automotive engine took place since then and made the engine what it is today.

Incidentally, reciprocating engines have the following shortcomings:

(1) Because of reciprocating parts, vibration, noise and power loss become greater as engine revolutions increase.

(2) Because of its cranking mechanism, it is heavy and large compared to power output.

(3) Because of its intake and exhaust valve mechanism, it generates mechanical noise and this mechanism requires many parts.

The rotary engine, which does not require an intake-exhaust valve mechanism and draws power directly from a rotating motion, has been the subject of research and development for many inventors and engineers as an ideal engine that overcomes the above-mentioned shortcomings of reciprocating engines.

Various types of rotary engines which grew from the rotary mechanisms of pumps and compressors have been developed to date, but it took many years before a rotary engine could be made practical. The following is a history of development of rotary engines in chronological order:

1588 Ramelli invented the first rotary piston type water pump.
Fig. 1.1 shows its rotary piston mechanism in which a seal that slides radially is incorporated in the rotor.

1636 Pappenheim invented a gear type pump. This gear type pump is the first pump composed of moving parts that only rotate. It is still used for oil pumps on automobiles, etc.

1759 James Watt invented the first rotary steam engine.
Although he designed an engine which has a rotor that derives its rotating power directly from steam pressure, it could not be put to practical use due to an airtightness problem.

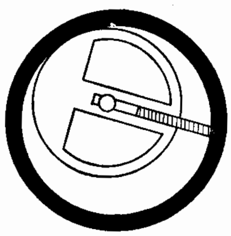

Fig. 1.1 Ramelli's pump

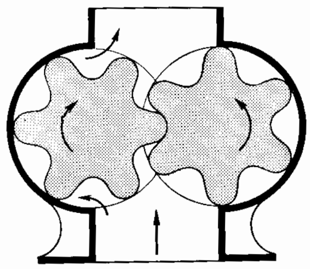

Fig. 1.2 Pappenheim's gear pump

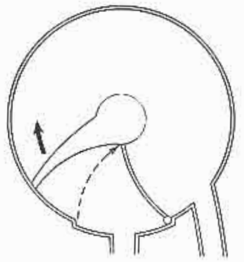

Fig. 1.3 Rotary steam engine by James Watt

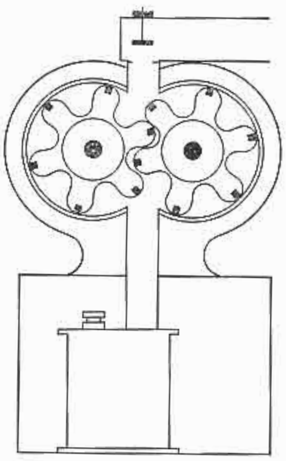

Fig. 1.4 Murdock's rotary steam engine

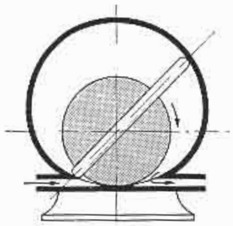

Fig. 1.5 Franchot's rotary compressor

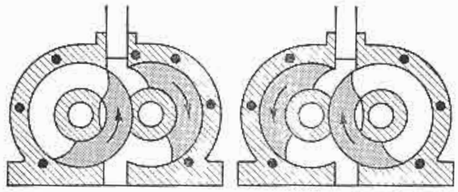

Fig. 1.6 Behrens's rotary steam engine

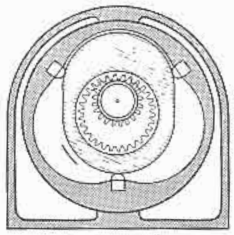

Fig. 1.7 Cooley's rotary steam engine

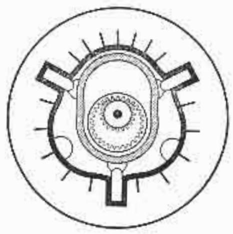

Fig. 1.8 Umpleby's rotary engine

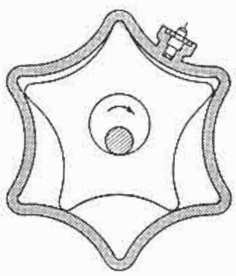

Fig. 1.9 Rotary engine by Wallinder & Skoog

1799 Murdock also invented a rotary steam engine.
He used wooden seals to improve airtightness, and succeeded in generating power for machine work and water pumps, but the engine did not provide sufficient airtightness and durability.

1860 Oldham and Franchot manufactured a rotary type compressor.
As a curve to form the working chamber, the peritrochoid curve was applied for the first time.

1867 Behrens manufactured a rotary steam engine with improved airtightness.
A recess that corresponds to the circular arc of the housing's inner wall is provided on part of the fixed shaft. The rotor rotates by coming in contact with this part. Since the sealing method was changed to a face contact seal, water pumps and steam engines with this structure were able to operate under a high pressure that could not be used in those days.

Beginning with the 20th century, the theories on mathematics, geometry, thermodynamics, materials, processing, metrology, etc. had progressed along with the development of the reciprocating type internal combustion engine, and also research on the rotary type internal combustion engine further advanced.

1901 Cooley manufactured a rotary steam engine in which both inner and outer rotors rotate. The inner rotor uses the peritrochoid curve and the outer rotor uses its outer envelope.

1908 Umpleby advanced Cooley's steam engine into a rotary type internal combustion engine.

1923 Wallinder and Skoog announced their joint research on the rotary engine.

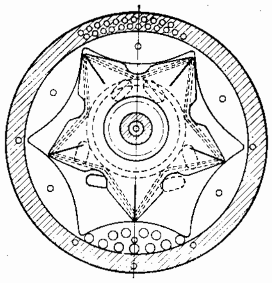

Fig. 1.10 Rotary engine by Sensaud de Lavou

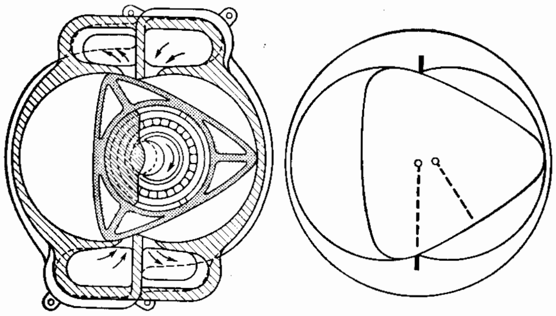

Fig. 1.11 Maillard's rotary compressor

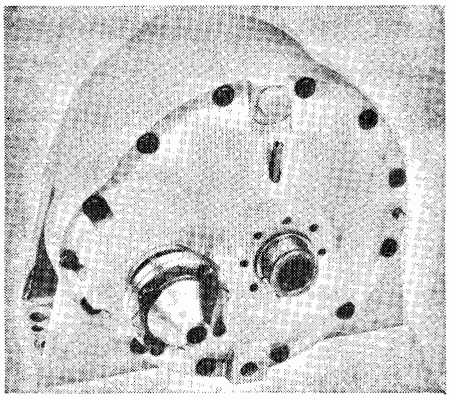

Fig. 1.12 Wankel's rotary compressor

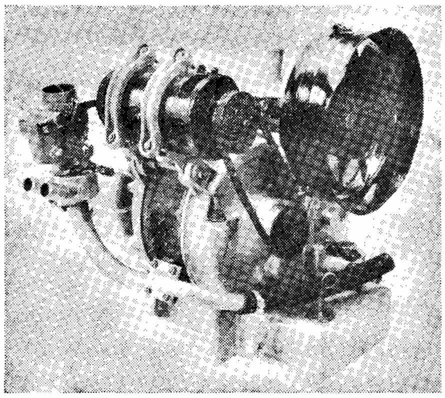

Fig. 1.13 250cc×1 rotary engine (NSU, West Germany)

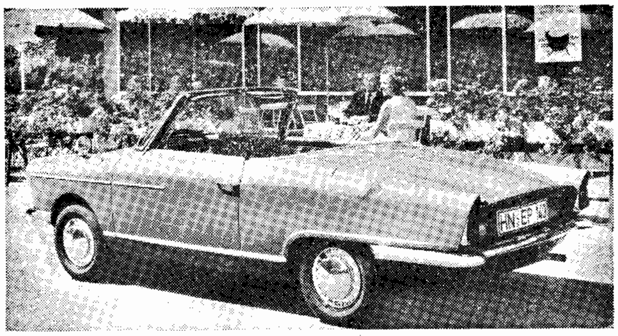

Fig. 1.14 Spider (NSU, West Gormany)

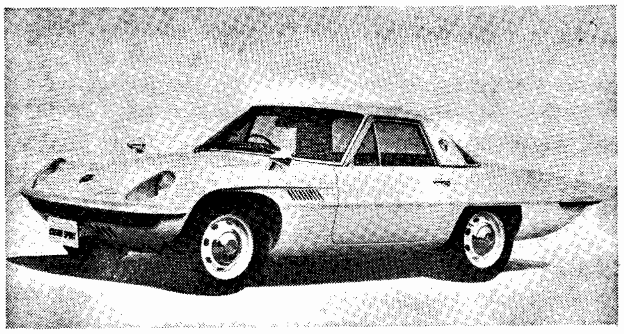

Fig. 1.15 Cosmo Sports (Toyo Kogyo)

This rotary engine uses the hypotrochoid curve for the inner wall of the housing and uses its inner envelope for the external shape of the rotor.

1938 Sensaud de Lavou further advanced the rotary theory in those days.
He devised an internal combustion engine that goes through the four strokes in one housing and does not require additional components such as intake and exhaust valves, but this engine could not be put to practical use because airtightness, lubrication, cooling, etc. were insufficient.

1943 Maillard devised a compressor by applying the rotary theory.
This compressor uses the hypotrochoid curve for the rotor and its outer envelope for the inner surface of the housing. He further advanced his research to apply this theory to engines, and contributed a great deal to the geometrical analysis of the rotary engine.

1951 Wankel manufactured a rotary compressor. This compressor was installed on NSU's motorcycle as a supercharger and established a speed record in the United States.

1959 The NSU-Wankel type rotary engine succeeded in a durability test.
A 250cc×1 rotor rotary engine of NSU completed a durability test of one hundred hours, and the report of "The Rotary Engine Achieves Success" was announced throughout the world.

1962 NSU announced a water-cooled 150cc×1 rotor rotary engine mounted on a boat for water-skiing.

1963 Toyo Kogyo made a prototype car, Cosmo

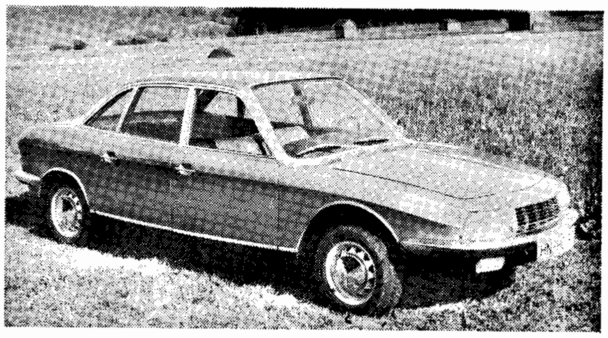

Fig. 1.16 Ro 80 (NSU, West Germany)

Sports, installing a 400cc×2 rotor rotary engine, and displayed the engine and main parts at the Tokyo Motor Show for the first time.

1964 NSU announced its sports car Spider installing a 497.5cc×1 rotor rotary engine, and this became the first rotary engine mounted car in the market.

1967 Toyo Kogyo introduced its Cosmo Sports (491cc×2 rotor) to the market. NSU also introduced its Ro80 (497.5cc×2 rotor).

Since then, rotary engine mounted cars proliferated and have been improved constantly to suit market requirements and to improve their marketability.

1.2 TYPES OF ROTARY ENGINE

Currently, the term rotary engine is familiar to those in the automotive industry and to the public as well. However, this term is interpreted in a number of ways, so it cannot be said that the interpretation is unified.

In this book, the rotary engine is defined as ; "An internal combustion engine that performs the four strokes of intake, compression, expansion and exhaust while the working chamber changes its volume and the moving parts always rotate in the same direction."

From this meaning, there are cases that the terms rotary piston engine, or rotary combustion engine are used in order to more accurately describe the rotary engine.

In the case of gas turbine, although it consists of rotating parts only, it is not included in the term, rotary engine, used here, because a gas turbine gets its power by the flow of gas. The volume of the working chamber does not change.

Since James Watt devised the first rotary engine in 1759, many rotary engines have been studied in the world. It is not easy to classify the ideas of the many rotary engines systematically, but it is possible to group them into the three types—single rotating engine, oscillatory rotating engine and planetary rotating engine.

1.2.1 SINGLE ROTATING ENGINE

A single rotating engine is an engine in which the rotor rotates at a certain angular velocity and the center

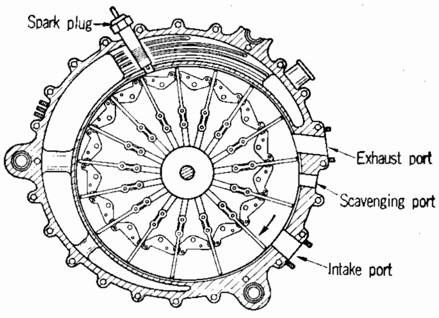

Fig. 1.17 Rotary engine by Mallory Co. (mechanism)

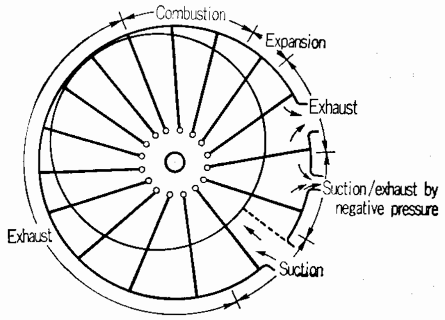

Fig. 1.18 Rotary engine by Mallory Co. (function)

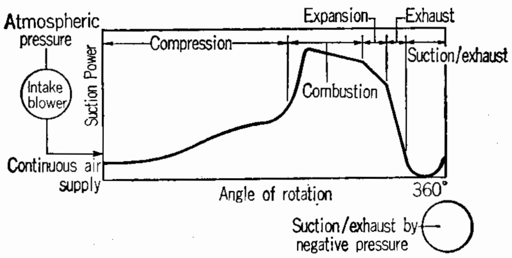

Fig. 1.19 Rotary engine by Mallory Co. (pressure change)

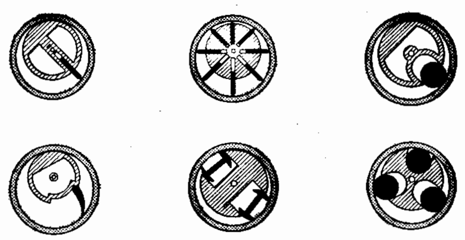

Fig. 1.20 Single rotating rotary engine

of its rotation does not move. An example is the Mallory engine (United States) which can be explained as follows: The rotor, which is divided by a number of vanes, rotates eccentrically in the housing. **Fig. 1.18** shows how the engine operates. The air-fuel mixture intake is supercharged by a blower and is compressed as the rotor rotates. The mixture, which is ignited after a compression stroke of approximately 180 degrees in rotation angle of the output shaft, successively completes the strokes of expansion and exhaust. And, in order to prevent the exhaust gas from mixing with fresh mixture, a scavenging port is provided which sucks out

the exhaust gas by negative pressure.

As for other single rotating engines, there are those of various structures as shown in **Fig. 1.20.**

1.2.2 OSCILLATORY ROTATING ENGINE

An oscillatory rotating engine is an engine in which a plural number of rotors rotate around the center of rotation by changing their angular velocity, and the chamber volume changes as the rotors come close to each other or separate from each other. As a typical example, this may be explained by introducing the structure and operation of Kauertz's engine.

In this engine, a single rotating rotor and an oscillatorily rotating rotor rotate concentrically and perform a relative motion inside a round housing. The relative motion of the two rotors are controlled by gears and linkages, and, if appropriate intake and exhaust ports are provided, each stroke of intake, compression, expansion and exhaust can be performed by the rotation f rotors which open and shut the ports.

As this type of engine requires a complicated oscillatory motion mechanism, problems concerning strength and noise are apt to occur.

Fig. 1.23 shows another example of an oscillatory rotating engine.

1.2.3 PLANETARY ROTATING ENGINE

A planetary rotating engine is an engine in which a rotor rotates by making a planetary motion. A typical example of this is the NSU-Wankel type engine which will be explained in detail in 1.4 and from Chapter 2.

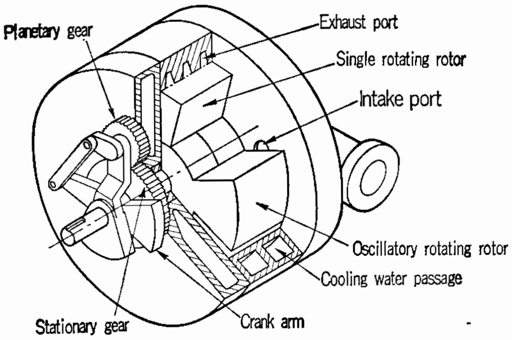

Fig. 1.21 Kauertz engine (construction)

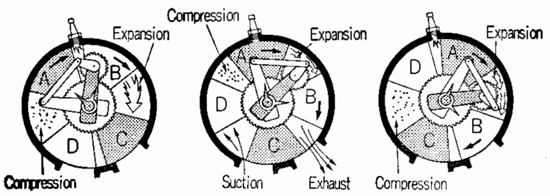

Fig. 1.22 Kauertz engine (function)

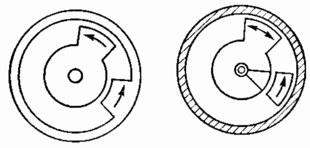

Fig. 1.23 Oscillatory rotating rotary engine

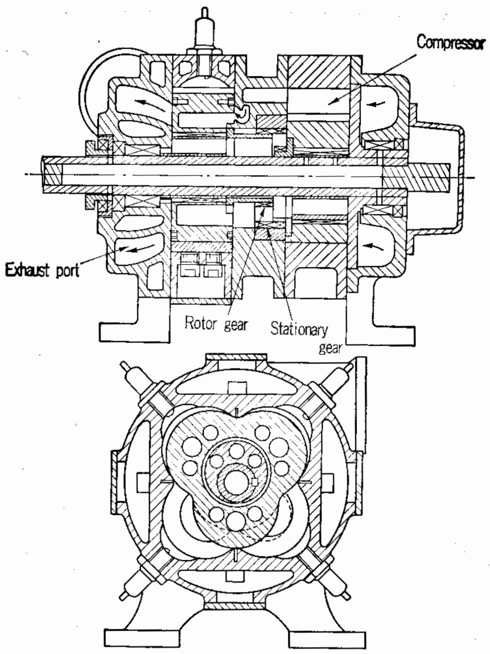

Fig. 1.24 Turbo-radial engine (construction)

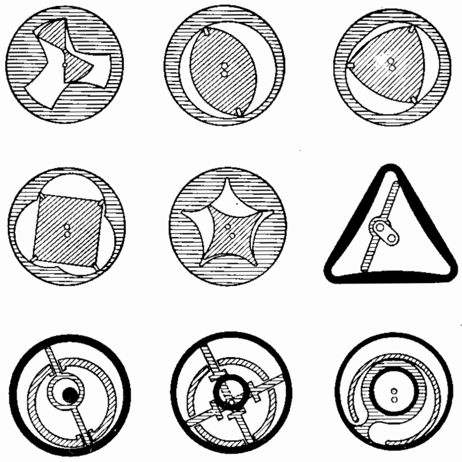

Fig. 1.25 Planetary rotating rotary engine

Therefore, in this section, a turbo-radial engine and its mechanism will be introduced.

In this engine, a peritrochoid curve is used on the rotor and its outer envelope is used on the housing. A rotor gear is projected from the rotor, and it meshes with a stationary gear which is concentric with the output shaft. Since the rotor center and the output shaft center are eccentric to each other, the rotating power of the output shaft can be obtained by the gas pressure which the rotor receives. In this engine, the working chamber does not make a rotating motion. Therefore, the gas seals to secure gas tightness are arranged on the housing side.

There are other planetary rotating engines which are of different structure as shown in **Fig. 1.25.**

1.3 REQUIREMENTS FOR A PRACTICAL ROTARY ENGINE

Although various types of rotary engines have been developed to date, there are few that do not qualify as internal combustion engines or do not qualify as a practical engine even if they satisfy the qualifications of an internal combustion engine.

The requirements for a rotary engine to be qualified as a practical internal combustion engine can be summed up in the following five items. The practicality of various ideas on rotary engines can be judged by evaluating them by these criteria:

(1) Every moving part, including the timing mechanism, should make a rotating motion.

A mechanism having a reciprocating inertia will increase mechanical noise and vibration, and will work against high speed and high revolution. Accordingly, structures that require intake and exhaust valves and mechanisms using the rotors oscillatory motion are not desirable.

(2) Gas seals of the working chamber should be three-dimensionally reliable.

The gas seal mechanism of a rotary engine should be constituted by connecting the individual seals three-dimensionally. Among the many ideas we come across are those that do not show such three-dimensional thoughts.

(3) Appropriate gas exchange of intake and exhaust should take place.

Together with having a mechanism that can correctly open and close the ports, sufficient time for intake and exhaust should be provided especially for high speed and high revolution. Among the ideas on rotary engines, there are some that ignore this point.

(4) Every component part should have the strength to endure high speed and high pressure.

As the component parts are exposed to high pressure, high sliding velocity and high heat load, every part should have sufficient allowance in size and in shape.

(5) Sufficient cooling and lubrication should be provided.

In order to qualify as an nternal combustion engine, although this is related to above (4), durability against high heat load, high sliding velocity, etc. are required. Therefore, the structure should be simple, and the ideas on rotor cooling, lubrication of seal parts, and oil seal structure should be those that are given adequate thought.

Also, it is desirable for a practical rotary engine to have a simple and compact structure. When we evaluate the various types of rotary engines devised to date in view of the above-mentioned requirements, the NSU-Wankel type rotary engine is the engine that best satisfies the above-mentioned requirements.

The NSU-Wankel type rotary engine will simply be called a rotary engine, and the reciprocating piston engine will be called a reciprocating engine hereinafter.

1.4 OUTLINE OF THE NSU-WANKEL TYPE ROTARY ENGINE

1.4.1 BASIC STRUCTURE

The basic structure of the NSU-Wankel type rotary engine is shown in **Fig. 1.26**.

The inner surface of the rotor housing is cocoon shaped, and the rotor performs a rotating motion inside the housing. By arranging side housings to both sides of the rotor housing, three working chambers are formed. The rotor housing and side housings correspond to the cylinder and cylinder block of the reciprocating engine, and the rotor corresponds to the piston.

As phasing gears to control the rotating motion of the rotor, a rotor gear and a stationary gear is fitted to the rotor and side housing with a gear ratio of 3:2. As shown in **Fig. 1.27**, by having the rotor gear rotate while being meshed with the stationary gear, the apex of the rotor will rotate by drawing a peritrochoid which is the basic curve of the rotor housing.

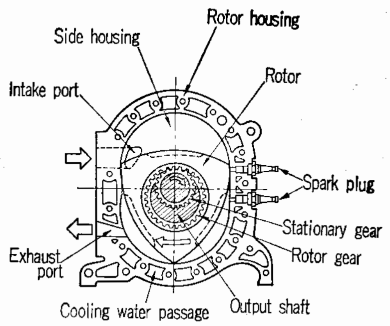

Fig. 1.26 Basic construction of NSU-Wankel type rotary engine

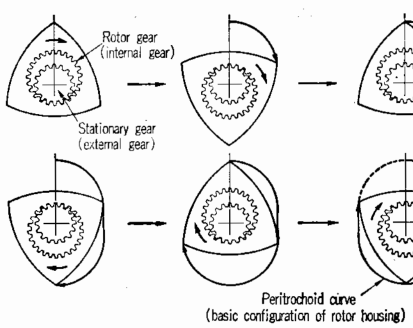

Fig. 1.27 Generation of peritrochoid curve

1.4.2 PRINCIPLE OF OPERATION

Fig. 1.28 shows the operation of the rotary engine.

The intake port opens at ①, where the intake stroke begins. The volume of the working chamber gradually expands as the rotor rotates as illustrated by ② and ③, and reaches the maximum volume at ④. The intake port automatically closes at ⑤.

The air-fuel mixture is compressed by ⑥, ⑦ and ⑧, and goes into the expansion stroke after being ignited near the compression top dead center ⑨. After going through the expansion stroke ⑩, ⑪ and ⑫, the exhaust port opens at ⑬. The exhaust stroke goes through ⑭, ⑮, ⑯ and ⑰, and completes at ⑱. From this point the stroke returns to ①, where the intake stroke begins again.

As described in the above, the rotary engine is a four stroke one cycle engine (hereinafter referred to as a four stroke engine) which continuously performs clearly distinguishable four strokes of intake, compression, expansion and exhaust. Until one of the working chamber completes the four strokes starting from ①, the strokes illustrated from i) to vi) of **Fig.** 1.28 are repeated

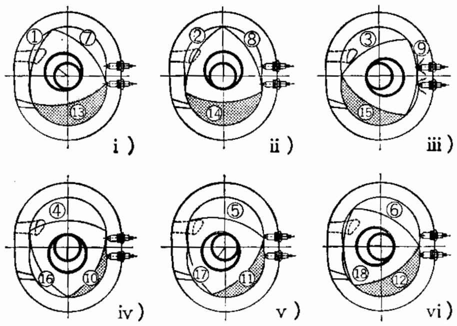

Fig. 1.28 Function principle of rotary engine

three times, which in turn means that the rotor rotates once and the output shaft three times. Also, the other two working chambers will respectively complete the four strokes once during this period, which in turn means that a total of three explosions take place in the same period. In other words, the rotary engine undergoes one explosion while the output shaft rotates once, as in the case of a two-stroke reciprocating engine.

Furthermore, since the output shaft rotates three times while one working chamber completes the four strokes once, the time required of one stroke will be 270 degrees in terms of the rotating angle of the output shaft. When compared with a four stroke reciprocating engine (180 degrees for one stroke), this is 1.5 times longer.

1.4.3 FEATURES OF THE ROTARY ENGINE

As compared with the reciprocating engine, the rotary engine has the following basic features:

(1) There are no reciprocating parts.

Since reciprocating engines have reciprocating parts, there are problems of unbalance caused by the inertia of reciprocating parts and complicated engine vibration. Whereas, in a rotary engine is composed of only rotating parts, engine

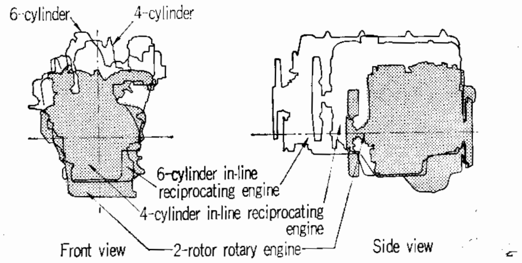

Fig. 1.29 Comparison of engine size

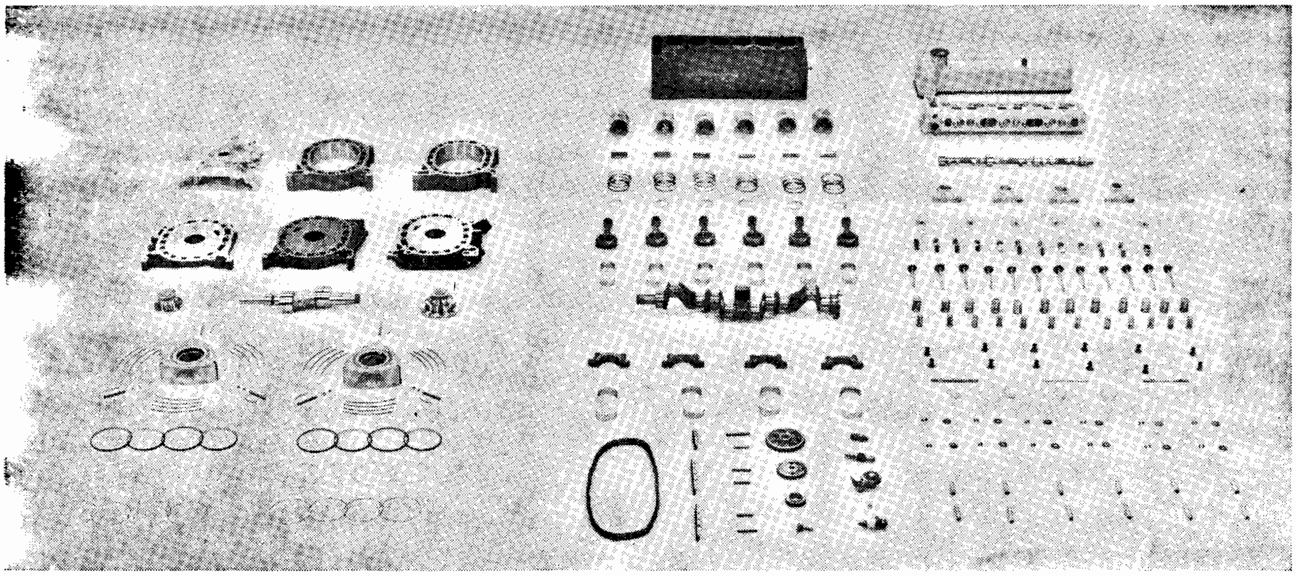

Fig. 1.30 Comparison of main parts of engine

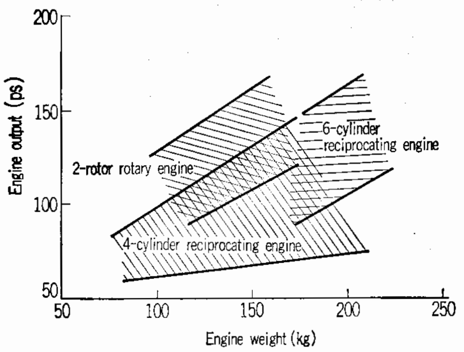

Fig. 1.31 Output and weight distribution of engine

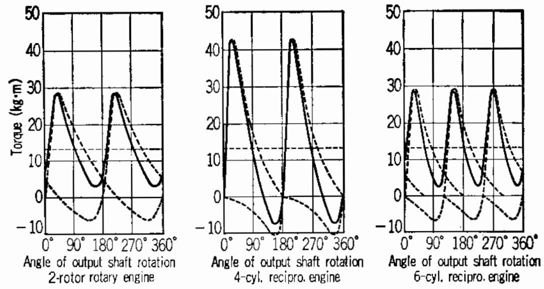

Fig. 1.32 Comparison of torque curve

vibration is very low because it is possible to perfectly balance the engine by using balancing weights.

Also, since the rotary engine does not require a cranking mechanism, it has the advantages of smooth motion, less mechanical loss, simple construction, and compactness.

(2) There are no intake-exhaust valve mechanism.

For those engines which require an intake-exhaust valve mechanism like a reciprocating engine, mechanical noise is generated by the opening and closing of the valves, and the valves themselves obstruct air flow. There are additional problems such as the valves being unable to follow the motion of the cam at high revolutions.

Since the rotary engine has a rotor that directly opens and closes the intake and exhaust ports, it does not require an intake-exhaust valve mechanism ; hence, the correct timing for opening and closing can be maintained even at high speeds.

(3) The time for one stroke is 270 degrees in terms of the rotating angle of the output shaft, and there is one explosion for one rotation of the output shaft.

The long intake stroke means that the volumetric efficiency becomes higher even at the high speed range, and reduces the torque drop. Also, the long expansion stroke means that it is advantageous from the point of torque fluctuation. In a rotary engine that has two rotors, the expansion stroke overlaps so the torque fluctuation is as low as in the case of a six cylinder reciprocating engine.

As described in the foregoing lines, the rotary engine has the superb features of light weight, compactness, less vibration and noise, flat torque characteristics, etc. Vehicles installing a rotary engine provide more freedom in designing the body for style and comfort. And, the driver will experience quieter, more comfortable and more responsive driving.

1.4.4 LICENSERS AND LICENSEES

Twenty odd companies of six countries are respectively in collaboration with Audi NSU Auto Union AG and Wankel GmbH, and are advancing their development of the NSU-Wankel type rotary engine. Although there are licensees who have different fields of industrialization of the engine in terms of engine output, kinds of fuel, etc., they are conducting their development work in their respective fields while mutually exchanging technical information under their common aim of research and development of the rotary engine.

Table 1.1 Licensers and Licensees (as of 1977)

Licensers

Country	Company	Field of Application
West Germany	Audi NSU Auto Union AG	Motor vehicles
	Wankel GmbH	......

Licensees

Country	Company	Field of Application
West Germany	Fichtel & Sachs AG	Boats, Land vehicles and Industrial purposes
	Klöckner-Humboldt-Deutz AG	All technical fields
	Daimler-Benz AG	All technical fields
	MAN (Maschinenfabrik Augsburg-Nürnberg AG)	All technical fields
	Friedrich Krupp GmbH	All technical fields
	Dr. Ing. h.c.F. Porsche AG	Motor vehicles
	Johannes Graupner	Model engines

Country	Company	Field of Application
Luxemburg	Comotor S.A. (Peugeot/Citroen)	Motor vehicles
Great Britain	Rolls-Royce Motors Limited NVT Motorcycles Ltd.	All technical fields Motorcycles
Switzerland	CROCO Engines GmbH	Off-road vehicles
United States	Cumtiss-Wright Corporation Outboard Marine Corporation Ingersoll-Rand Company American Motors Corporation General Motors Corporation	All technical fields Boats and Industrial purposes Industrial purposes Motor vehicles All fields excepting airplanes
Japan	Yanmar Diesel Co., Ltd. Toyo Kogyo Co., Ltd. Nissan Motor Co., Ltd. Toyota Motor Co., Ltd.	All technical fields excepting motor vehicles All technical fields Passenger cars Passenger cars

Note: NSU became Audi NSU in August 1969 by merger.

References:

1) F. Huf: Zur Geschichte der Rotationskolbenmaschinen, Automobil Revue No. 49 (1961)
2) F. Wankel: ROTARY PISTON MACHINES, London Illiffe Books Ltd. (1965)
3) D. Korp: Protokoll einer Erfindung DER WANKEL-MOTOR, MOTORBUCH VERLAG STUTTGART (1975)
4) Kenichi Yamamoto: Rotary Engine, Nikkan Kogyo Shimbunsha (1969)
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CHAPTER 2

BASIC DIMENSIONS

The NSU-Wankel type rotary engine is a simple mechanism consisting of a rotor housing in which a rotor and an eccentric shaft rotate. In principle, the mechanism is composed of a peritrochoid curve and its inner envelope. Their elements are called the basic dimensions.

Being closely related with the shape of each component of the rotary engine, the basic dimensions of this trochoid greatly affect all the engine characteristics including displacement, performance, durability, productivity, etc.

Various organizations in the world are actively participating in the competition for research and development of the rotary engine. However, in comparison with the many years of systematic studies on the reciprocating engine, few basic studies on the rotary engine have been conducted at universities and institutes. The relation between these basic dimensions and the characteristics of the rotary engine remain to be pursued further.

2.1 PERITROCHOID

Peritrochoid is a basic curve forming the inner surface of the rotor housing. It is the actual base for the geometric construction of the rotary engine.

2.1.1 EQUATION OF PERITROCHOID

Peritrochoid is the locus of the tip point P of an arm fixed on the revolving circle B in Radius q when it rolls along the periphery of the base circle A in Radius p as inscribed.

Therefore, the equation of peritrochoid can be expressed by the coordinates of Point P (x, y) in the coordinates x, y of the center of Base Circle A, as initial point, as

$$\left.\begin{aligned} x &= e\cos\alpha + R\cos\beta \\ y &= e\sin\alpha + R\sin\beta \end{aligned}\right\} \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (2.1)$$

where e : center distance between Base Circle A and Rolling Circle B
R : length of arm fixed on Rolling Circle B
α : angle of rotation of Revolving Circle B around Base Circle A
β : angle of rotation of Revolving Circle B on its axis

In **Fig. 2.1**, Point F on Base Circle A and Point G on Revolving Circle B are the point of contact between the circles when $\alpha=0$ and $\beta=0$. From the relation $\overset{\frown}{FH}=\overset{\frown}{GH}$,

$$q(\alpha-\beta)=p\alpha$$

$$\therefore \quad \beta=(1-p/q)\alpha$$

This shows that the angle of rotation α of revolving Circle B around Base Circle A and that of β on its axis are in a proportional relation.

This curve will be closed if

$$1-p/q=1/m \quad (m \text{ being an integer})$$

Therefore, the relation between p and q will be

$$p/q=(m-1)/m \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots \quad (2.2)$$

The equation of peritrochoid becomes

$$\left.\begin{aligned} x &= e\cos\alpha + R\cos\alpha/m \\ y &= e\sin\alpha + R\sin\alpha/m \end{aligned}\right\} \quad \cdots\cdots\cdots\cdots\cdots \quad (2.3)$$

Fig. 2.2 shows the configurations of the curves when $m=2, 3$ and 4, respectively.

The rotary engine in practical application presently uses the trochoid of $m=3$, the equation of which is expressed, as

$$\left.\begin{aligned} x &= e\cos\alpha + R\cos\alpha/3 \\ y &= e\sin\alpha + R\sin\alpha/3 \end{aligned}\right\} \quad \cdots\cdots\cdots\cdots\cdots \quad (2.4)$$

Since Radius p of the base circle and Radius q of the revolving circle used to draw this trochoid satisfies $p/q=2/3$ from Eq. (2.2) and $q-p=e$ from the center distance e, $p=2e$ and $q=3e$ will result.

As shown in Eq. (2.4), the center distance e between the revolving and base circles and the length R of the arm fixed on the revolving circle are the basic values to determine the trochoid. In general, e is called

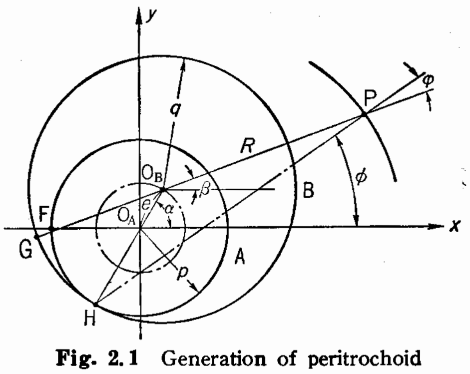

Fig. 2.1 Generation of peritrochoid

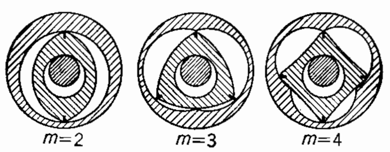

Fig. 2.2 Configuration of peritrochoid and inner envelope

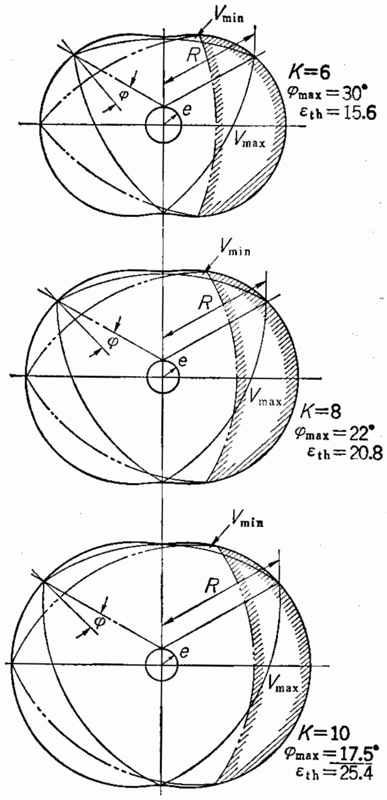

Fig. 2.3 Trochoid constant and trochoid configuration

the amount of eccentricity and R, the generating radius.

The ratio of the generating radius R to the eccentricity e is called the trochoid constant K, which will provide a typical index for a trochoid to indicate its geometric configuration.

$$K=R/e \quad \cdots\cdots \quad (2.5)$$

Fig. 2.3 shows the trochoid constant K and the configuration of the trochoid when the working volume is taken as constant. The trochoid constant is an important index to determine the basic dimension of the engine, such as maximum angle of oscillation, theoretical compression ratio, circumferential velocity of the apex seal, outside dimensions of engine, etc. Generally, it is taken between $K=6\sim8$.

2.1.2 PARALLEL TROCHOID CURVE

In the actual rotary engine, the curve used for the inner surface of its rotor housing is the trochoid given by **Eq.** (2.4) that has been moved outward in parallel by a constant amount $\boldsymbol{a}$.

The purpose of this is to form the top of the apex seal in a circular section in the radius $\boldsymbol{a}$ and to change its line of contact with the inner surface of the rotor housing as the rotor rotates.

The parallel trochoid curve is expressed by

$$\left.\begin{aligned} x&=e\cos\alpha+R\cos\alpha/3+a\cos(\alpha/3+\varphi) \\ y&=e\sin\alpha+R\sin\alpha/3+a\sin(\alpha/3+\varphi) \end{aligned}\right\} \quad \cdots\cdots \quad (2.6)$$

where φ is the angle of oscillation of the apex seal, as described in 2.3.

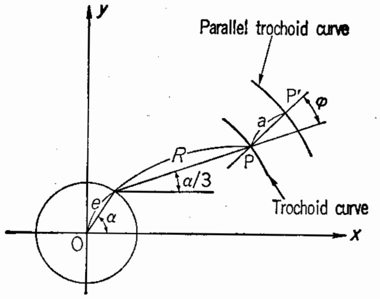

Fig. 2.4 Parallel trochoid curve

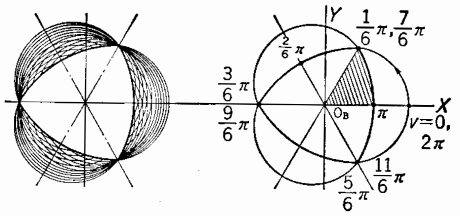

Fig. 2.5 Envelopes of two-lobe peritrochoid

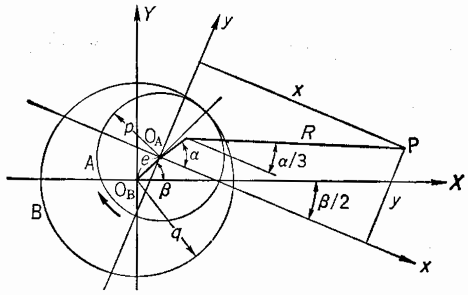

Fig. 2.6 Generation of inner envelope of peritrochoid

This curve is closely approximate to the trochoid drawn with the generating radius R increased by the amount of the parallel transfer $\boldsymbol{a}$. In ordinary calculation, a sufficient accuracy for practical use can be obtained by substituting $R+a$ for the generating radius R in Eq. (2.4).

2.2 INNER ENVELOPE OF PERITROCHOID

As shown in **Fig. 2.6,** the envelope of a peritrochoid is a pattern drawn by the peritrochoid fixed on Circle A when it rolls along the inner surface of the stationary Circle B as inscribed. The inner line of this pattern is called the inner envelope.

The inner envelope of the trochoid is a basic curve forming the contour of the rotor. As shown in **Fig. 2.5**, its three apexes are always in contact with the peritrochoid.

2.2.1 EQUATION OF INNER ENVELOPE

As shown in **Fig. 2.6**, when the $x-y$ coordinate system rotates as fixed to the revolving circle A, the following relation between the $x-y$ and $X-Y$ coordinate systems will be established:

$$\left.\begin{array}{l} X=e\cos\beta+x\cos\beta/2+y\sin\beta/2 \\ Y=e\sin\beta-x\sin\beta/2+y\cos\beta/2 \end{array}\right\} \cdots (2.7)$$

Since Point P (x, y) draws a peritrochoid, substituting Eq. (2.4) in Eq. (2.7), using

$$\beta=u-3v$$
$$\alpha-\beta/2=u+3v$$

the group of curves will be given by

$$\left.\begin{array}{l} X=2e\cos 3v\cdot\cos u+R\cos 2v \\ Y=2e\cos 3v\cdot\sin u+R\sin 2v \end{array}\right\} \cdots\cdots (2.8)$$

The envelope is given by the following additional conditions:

$$\frac{dX}{dv}=\frac{\partial X}{\partial v}+\frac{\partial X}{\partial u}\cdot\frac{du}{dv}=0$$
$$\frac{dY}{dv}=\frac{\partial Y}{\partial v}+\frac{\partial Y}{\partial u}\cdot\frac{du}{dv}=0$$

Therefore, the equation of the envelope can be expressed, as

$$\left.\begin{array}{l} X=R\cos 2v+\dfrac{3e^2}{2R}(\cos 8v-\cos 4v) \\ \quad\pm e\left(1-\dfrac{9e^2}{R^2}\sin^2 3v\right)^{1/2}\cdot \\ \quad(\cos 5v+\cos v) \\ Y=R\sin 2v+\dfrac{3e^2}{2R}(\sin 8v+\sin 4v) \\ \quad\pm e\left(1-\dfrac{9e^2}{R^2}\sin^2 3v\right)^{1/2}\cdot \\ \quad(\sin 5v-\sin v) \end{array}\right\} \cdots\cdots\cdots\cdots (2.9)$$

These functions are cyclic functions with the period of 2π. The inner envelope corresponds to

$$v=\frac{1}{6}\pi\sim\frac{1}{2}\pi,\ \frac{5}{6}\pi\sim\frac{7}{6}\pi,\ \frac{3}{2}\pi\sim\frac{11}{6}\pi$$

in Eq. (2.9).

2.2.2 PARALLEL INNER ENVELOPE

In the same manner as using the parallel trochoid curve for the configuration of the inner surface of the rotor housing, the parallel transferred inner envelope obtained in 2.2.1 is used for the configuration of the rotor of the actual engine.

Eq. (2.9) being parallel transferred outward by a', the equation of the parallel inner envelope will be:

$$\left.\begin{array}{l} X=R\cdot\cos 2v+\dfrac{3}{2}\cdot\dfrac{e^2}{R}(\cos 8v-\cos 4v) \\ \quad\pm e(\cos 5v+\cos v)\cdot \\ \quad\left(1-\dfrac{9e^2}{R^2}\sin^2 3v\right)^{1/2}\pm\dfrac{3}{2}\cdot\dfrac{ea'}{R} \\ \quad(\cos 5v-\cos v)+a'\cos 2v \\ \quad\left(1-\dfrac{9e^2}{R^2}\sin^2 3v\right)^{1/2} \\ Y=R\cdot\sin 2v+\dfrac{3}{2}\cdot\dfrac{e^2}{R}(\sin 8v+\sin 4v) \\ \quad\pm e(\sin 5v-\sin v)\cdot \\ \quad\left(1-\dfrac{9e^2}{R^2}\sin^2 3v\right)^{1/2}\pm\dfrac{3}{2}\cdot\dfrac{ea'}{R} \\ \quad(\sin 5v+\sin v)+a'\sin 2v \\ \quad\left(1-\dfrac{9e^2}{R^2}\sin^2 3v\right)^{1/2} \end{array}\right\} \cdots\cdots\cdots\cdots (2.10)$$

Generally, a' can be determined, as follows:

$$a'=a-S_p$$

where a: amount of parallel transfer of trochoid

S_p: minimum clearance between rotor and rotor housing

In this case, S_p is determined considering error in manufacture, thermal deformation and bearing clearance, etc. of the rotor and rotor housing.

The use of the parallel inner envelope for the rotor configuration makes it possible to always keep constant the minimum clearance between the rotor and the rotor housing.

2.3 ANGLE OF OSCILLATION

The angle of oscillation is that formed by the generating radius of the peritrochoid and the normal of the trochoid. This angle of oscillation shows the inclination of the apex seal to the inner periphery of the rotor housing. It is a factor greatly affecting the various performances of the engine.

2.3.1 CALCULATION OF ANGLE OF OSCILLATION

In **Fig. 2.1**, taking ϕ as an angle formed by the normal of the trochoid and the x–axis, the angle of oscillation φ can be expressed as $\varphi=\phi-\alpha/3$

$$\cos\varphi=\cos(\phi-\alpha/3)$$
$$=\cos\phi\cos\alpha/3+\sin\phi\sin\alpha/3 \quad (2.11)$$

Thus ϕ satisfies the following equation:

$$\tan\phi=\frac{dx}{dy}=\frac{3e\sin\alpha+R\sin\alpha/3}{3e\sin\alpha+R\cos\alpha/3} \quad\cdots(2.12)$$

Substituting $\sin\phi$ and $\cos\phi$ obtained from Eq. (2.12) for those in Eq. (2.11),

$$\cos\varphi=\frac{3e\cos\frac{2}{3}\alpha+R}{\left(9e^2+R^2+6eR\cos\frac{2}{3}\alpha\right)^{1/2}} \cdots\cdots\cdots\cdots (2.13)$$

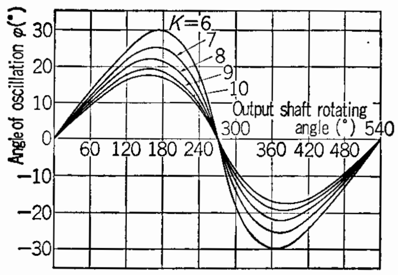

Fig. 2.7 Angle of oscillation and rotating angle of main axis

The angle of oscillation can be obtained by this equation. **Fig.** 2.7 shows the relation between φ and α.

2.3.2 MAXIMUM ANGLE OF OSCILLATION

The maximum value of the angle of oscillation is called the maximum angle of oscillation. The maximum angle of oscillation φ max is given by α, the value of which can be obtained by applying the condition $d\varphi/d\alpha=0$ to Eq. (2.13), as

$$\cos\frac{2}{3}\alpha=-\frac{3e}{R} \qquad \cdots\cdots(2.14)$$

Substituting this in Eq. (2.13), φ_{max} becomes

$$\left.\begin{aligned}\cos\varphi_{max}&=\left\{1-\left(\frac{3e}{R}\right)^2\right\}^{1/2}\\ \sin\varphi_{max}&=\frac{3e}{R}\end{aligned}\right\} \qquad \cdots\cdots(2.15)$$

Using the trochoid constant K in Eq. (2.15),

$$\sin\varphi_{max}=3/K \qquad \cdots\cdots(2.16)$$

From this, it is clear that the maximum angle of oscillation will become smaller as the trochoid constant becomes greater and vice versa.

The maximum angle of oscillation is itself an important factor greatly affecting the design and performance of the apex seal, while it is also a convenient constant in calculating the volume of th? working chamber, compression ratio, etc., as later described.

2.4 VOLUME OF WORKING CHAMBER

The change in the volume of the working chamber is one of the fundamental requirements for the rotary engine to be regarded as an internal combustion engine.

The volume of the working chamber of the rotary engine is defined as the product of the area (side area) enclosed by a side of the rotor contour and the inner surface of the peritrochoid and the width of the rotor housing.

2.4.1 SIDE AREA OF WORKING CHAMBER

The side area F of the working chamber shown in **Fig.** 2.8 is expressed as

$$F=F_1-F_2-F_3-F_4 \qquad \cdots\cdots(2.17)$$

where F_1: area enclosed by the peritrochoid and Lines OP_1 and OP_2

F_2: area of $\triangle OP_1O'$

F_3: area of $\triangle OP_2O'$

F_4: area enclosed by $O'P_1P_2$ (1/3 of the rotor side area)

The coordinates (x, y) of any point P between P_1 and P_2 on the trochoid are

$$\left.\begin{aligned}x&=e\cos(\alpha+\theta)+R\cos\left(\frac{\alpha+\theta}{3}\right)\\ y&=e\sin(\alpha+\theta)+R\sin\left(\frac{\alpha+\theta}{3}\right)\end{aligned}\right\} \qquad \cdots\cdots(2.18)$$

From this,

$$\begin{aligned}F_1&=\frac{1}{2}\int_0^{2\pi}\left(x\frac{dy}{d\theta}-y\frac{dx}{d\theta}\right)d\theta\\ &=\left(e^2+\frac{R^2}{3}\right)\pi-\sqrt{3}eR\sin\left(\frac{2}{3}\alpha+\frac{\pi}{6}\right)\end{aligned} \qquad \cdots\cdots(2.19)$$

From the formula of the triangle area,

$$F_2=\frac{1}{2}eR\sin\frac{2}{3}\alpha \qquad \cdots\cdots(2.20)$$

$$F_3=eR\sin\left(\frac{2}{3}\alpha+\frac{\pi}{3}\right) \qquad \cdots\cdots(2.21)$$

From Eq. (2.9) of the inner envelope,

$$\begin{aligned}F_4&=\frac{1}{2}\int_{5/6\pi}^{7/6\pi}\left(X\frac{dX}{dv}-Y\frac{dX}{dv}\right)dv\\ &=\int_{\pi}^{7/6\pi}\left(X\frac{dX}{dv}-Y\frac{dX}{dv}\right)dv\\ &=\frac{\pi}{3}(R^2+2e^2)-2eR\cos\varphi_{max}\\ &\quad-\left(\frac{2}{9}R^2+4e^2\right)\varphi_{max}\end{aligned} \qquad \cdots\cdots(2.22)$$

The side area of the working chamber is

$$\begin{aligned}F&=\frac{\pi}{3}e^2+eR\\ &\quad\left\{2\cos\varphi_{max}-\frac{3\sqrt{3}}{2}\sin\left(\frac{2}{3}\alpha+\frac{\pi}{6}\right)\right\}\\ &\quad+\left(\frac{2}{9}R^2+4e^2\right)\varphi_{max}\end{aligned} \qquad \cdots\cdots(2.23)$$

where φ_{max} is the maximum angle of oscillation obtained in 2.3.2.

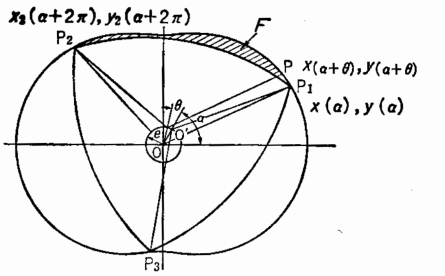

Fig. 2.8 Side area of the working chamber

2.4.2 CHANGE IN VOLUME OF WORKING CHAMBER

With the width of the rotor housing taken as ***b***, the volume of the working chamber is expressed, as

$$\begin{aligned}V&=b\cdot F=V_{min}+\frac{3\sqrt{3}}{3}eRb\\ &\quad\left\{1-\sin\left(\frac{2}{3}\alpha+\frac{\pi}{6}\right)\right\}\end{aligned} \qquad \cdots\cdots(2.24)$$

where

$$\begin{aligned}V_{min}&=\left\{\frac{\pi}{3}e^2+2eR\cos\varphi_{max}\right.\\ &\quad\left.+\left(\frac{2}{9}R^2+4e^2\right)\varphi_{max}-\frac{3\sqrt{3}}{2}eR\right\}b\end{aligned} \qquad \cdots\cdots(2.25)$$

This indicates that the volume of the working chamber of the rotary engine changes in sine curve.

On the other hand, the volumetric change of the reciprocating engine is expressed, as

$$V=V_{\min}+\gamma\cdot s\left\{(1-\sin\alpha)+\frac{1}{4\lambda}(1+\cos 2\alpha)\right\}\quad\cdots(2.26)$$

where γ : radius of crank
s : area of cylinder
α : angle of rotation of crank
λ : length of connecting rod/radius of crank

The volumetric change of the working chamber of the rotary engine corresponds to that of the reciprocating engine taken as $\lambda\to\infty$. Also, the period of the volumetric change is 1.5 times that of the reciprocating engine, showing a smoother change.

Fig. 2.9 shows the curves of volumetric change for the rotary and reciprocating engines. For easy comparison of the volumetric change, different readings for the angle of rotation of their main axes are used, respectively.

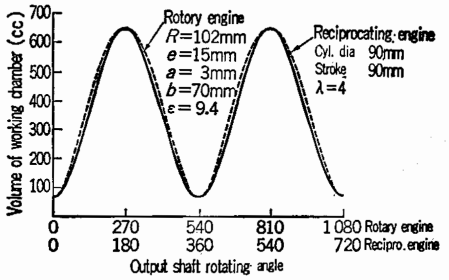

Fig. 2.9 Comparison of change in volume of working chamber between rotary and reciprocating engines

2.4.3 STROKE VOLUME

The stroke volume is given by the difference between the maximum and minimum values of the volume of the working chamber.

From Eq. (2.24), the stroke volume V_H is

$$V_H=3\sqrt{3}\,eRb\quad\cdots\cdots\cdots\cdots(2.27)$$

With the major and minor axes of the peritrochoid taken as A and B, respectively, the following relations will be established:

$$A=2(R+e)$$
$$B=2(R-e)$$

Using these relations, also

$$V_H=\frac{3\sqrt{3}}{16}(A^2-B^2)b\quad\cdots\cdots\cdots(2.28)$$

In the above caluculation, the parallel transfers a and a' of the trochoid and the inner envelope, respectively, have been disregarded for convenience. In practice, they need to be considered. In such a case, the volume of the working chamber can be approximately calculated in practically sufficient accuracy by taking $R_1=R+a$ and $R_2=R+a'$, as follows:

$$V'=V'_{\min}+\frac{\sqrt{3}}{2}e(2R_1+R_2)b\left\{1-\sin\left(\frac{2}{3}\alpha+\frac{\pi}{6}\right)\right\}$$

When

$$V'_{\min}=\left\{\frac{\pi}{3}e^2+\frac{R_1^2-R_2^2}{3}\pi+2eR_2\cos\varphi_{\max}+\left(\frac{2}{9}R_2^2+4e^2\right)\varphi_{\max}-\frac{\sqrt{3}}{2}e(2R_1+R_2)\right\}b\quad\cdots\cdots(2.29)$$

The stroke volume V_H becomes

$$V_H'=\sqrt{3}\,e(2R_1+R_2)b\quad\cdots\cdots\cdots\cdots(2.30)$$

2.5 COMPRESSION RATIO

It is impossible for the rotary engine due to its mechanism to obtain a higher compression ratio than the theoretical compression ratio that is expressed by the ratio of the maximum to minimum values of the volume of the working chamber as obtained in 2.4. This is one of the major problems in designing a diesel rotary engine

2.5.1 THEORETICAL COMPRESSION RATIO

The theoretical compression ratio ε_{th} is

$$\varepsilon_{th}=\frac{V_{\max}}{V_{\min}}$$

$$=\frac{2eR\cos\varphi_{\max}+\left(\frac{2}{9}R^2+4e^2\right)\varphi_{\max}+\frac{\pi}{3}e^2+\frac{3\sqrt{3}}{2}eR}{2eR\cos\varphi_{\max}+\left(\frac{2}{9}R^2+4e^2\right)\varphi_{\max}+\frac{\pi}{3}e^2-\frac{3\sqrt{3}}{2}eR}$$

$$=\frac{2K\cos\varphi_{\max}+\left(\frac{2}{9}K^2+4\right)\varphi_{\max}+\frac{\pi}{3}K+\frac{3\sqrt{3}}{2}K}{2K\cos\varphi_{\max}+\left(\frac{2}{9}K^2+4\right)\varphi_{\max}+\frac{\pi}{3}K-\frac{3\sqrt{3}}{2}K}\quad\cdots\cdots\cdots(2.31)$$

As being clear from this, the theoretical compression ratio ε_{th} is determined only by the trochoid constant K. **Fig.** 2.10 shows the relation between K and the maximum angle of oscillation and the theoretical

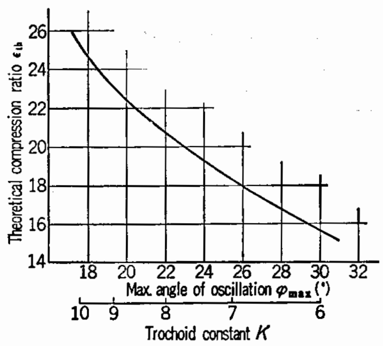

Fig. 2.10 Relation between trochoid constant and max. angle of oscillation and theoretical compression ratio

compression ratio. This indicates that, as the trochoid constant increases, the theoretical compression ratio also increases. If the trochoid constant is taken between $K=6\sim10$, the theoretical compression ratio will become 15 $\sim$25.

2.5.2 COMPRESSION RATIO OF ROTARY ENGINE

The compression ratio of the actual rotary engine is lower than the theoretical compression ratio due to the effects of the parallel movements of the peritrochoid and its inner envelope, the volume of the recess for the combustion chamber provided in the rotor surface, the volume of the spark plug hole, etc.

Taking the theoretical minimum volume of the working chamber and the stroke volume as V'_{min} and V_H', respectively, when the parallel movement is considered, and the sum of the volume of the combustion chamber recess and the volume of the spark plug hole as V_r, the compression ratio ε of the actual rotary engine can be obtained by

$$\varepsilon=\frac{V'_{min}+V_r+V_H'}{V'_{min}+V_r} \quad \cdots\cdots\cdots\cdots\cdots\cdots(2.32)$$

2.6 CIRCUMFERENTIAL VELOCITY AND ACCELERATION OF ROTOR VERTEX

The speed of the rotor vertex, i.e. the sliding speed of the apex seal over the peritrochoid curve, corresponds to that of the piston ring of the reciprocating engine. It is one of the important factors to determine the durability of the apex seal and the inner surface of the rotor housing. The acceleration of the rotor vertex is closely related with inertia force acting on the apex seal, greatly affecting the apex seal functions.

2.6.1 CIRCUMFERENTIAL VELOCITY OF ROTOR VERTEX

The circumferential velocity v is obtained from Eq. (2.4), as

$$v=\left\{\left(\frac{dx}{dt}\right)^2+\left(\frac{dy}{dt}\right)^2\right\}^{1/2}$$
$$=\frac{\omega}{3}\left(9e^2+R^2+6eR\cos\frac{2}{3}\alpha\right)^{1/2} \quad \cdots\cdots(2.33)$$

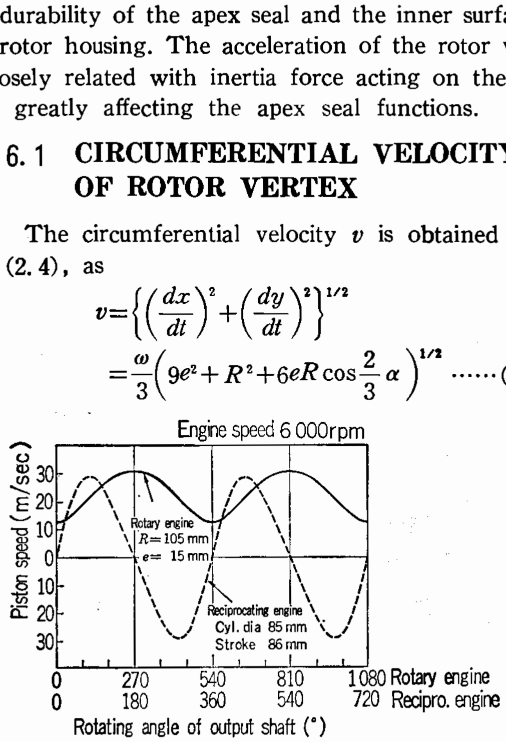

Fig. 2.11 Rotor vertex speed and piston speed

Where, $\omega=d\alpha/dt$ (rad/sec) is the rotating angular velocity of the output shaft.

Fig. 2.11 shows a comparison between the circumferential velocity of the rotor vertex and the piston speed of the reciprocating engine.

In the rotary engine, its rotor always rotates in the same direction eliminating the speed-zero point, the presence of which is inherent in the reciprocating engine.

The maximum and minimum values of the peripheral speed are

$$\left.\begin{aligned} &v_{max}=\frac{\omega}{3}(R+3e) \\ &\text{when } \alpha=0,\ 3\pi \text{ (on major axis)} \\ &v_{min}=\frac{\omega}{3}(R-3e) \\ &\text{when } \alpha=\frac{3}{2}\pi,\ \frac{9}{2}\pi \text{ (on minor axis)} \end{aligned}\right\} \cdots\cdots(2.34)$$

If the distance from the rotor center to any point on the rotor is taken as r, the point will draw a peritrochoid curve with the eccentricity e and the generating radius r. That is, the circumferential velocity of this point can be obtained by substituting r for R in Eq. (2.33).

2.6.2 ACCELERATION OF ROTOR VERTEX

The x- and y- elements of the vertex acceleration, a_x and a_y, respectively, are

$$\left.\begin{aligned} a_x&=\frac{d^2x}{dt^2}=-\omega^2\left(e\cos\alpha+\frac{R}{9}\cos\frac{\alpha}{3}\right) \\ a_y&=\frac{d^2y}{dt^2}=-\omega^2\left(e\sin\alpha+\frac{R}{9}\sin\frac{\alpha}{3}\right) \end{aligned}\right\}$$
$$\cdots\cdots\cdots\cdots\cdots\cdots(2.35)$$

The magnitude of acceleration, a_c, is

$$a_c=(a_x^2+a_y^2)^{1/2}$$
$$=\frac{\omega^2}{9}\left\{\left(81e^2+R^2+18eR\cos\frac{2}{3}\alpha\right)\right\}^{1/2}$$
$$\cdots\cdots\cdots\cdots\cdots\cdots(2.36)$$

The acceleration is divided into the parallel element a_r and the vertical element a_α to the generating radius, as

$$\left.\begin{aligned} a_r&=a_x\cdot\cos\frac{\alpha}{3}+a_y\cdot\sin\frac{\alpha}{3} \\ &=-\frac{\omega^2}{9}\left(R+9e\cos\frac{2}{3}\alpha\right) \\ a_\alpha&=a_x\cdot\sin\frac{\alpha}{3}-a_y\cdot\cos\frac{\alpha}{3} \\ &=\omega^2 e\sin\frac{2}{3}\alpha \end{aligned}\right\} \cdots\cdots\cdots(2.37)$$

Generally, the inertia force is given by the product of acceleration and mass. The inertia force acting on the apex seal can be obtained by multiplying Eq. (2.37) by the mass of the apex seal.

2.7 PRACTICAL APPLICATIONS OF THE BASIC DIMENSIONS

The basic dimensions must be carefully determined depending on intended use of the engine and considering its size, range of practical revolutions, etc.

The displacement of the rotary engine can be increased or decreased by changing the width of the rotor housing obtained by the same trochoid curve. Therefore, sufficient prospective view on the development plan for the future engine displacement will also be necessary to determine the basic dimensions.

Table 2.1 shows the practical applications of the basic dimensions.

Table 2.1 Practical applications of the basic dimensions.

V_H(cc)	e(mm)	R(mm)	a(mm)	b(mm)	K	φ_{max}(degree)
60	7.6	47	0.5	32	6.18	29.0
129	9.5	65	0.5	40	6.85	26.0
150	10.5	66	1	41	6.28	28.5
221	10.5	72	3	54	6.86	25.9
253	11.0	84	1	52	7.63	23.2
452	14.0	94	3	64	6.71	26.5
491	15.0	101	4	60	6.73	26.5
573	15.0	102	3	70	6.80	26.2
654	15.0	102	3	80	6.80	26.2
655	17.5	116	4	60	6.63	26.9
747	17.0	118.5	4	69	6.97	25.5

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2) W.D. Bensinger: Rotationskolben-Verbrennungsmotoren, Springer-Verlag (1973)
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CHAPTER 3
BASIC CONSTRUCTION OF THE ROTARY ENGINE

The rotary engine is composed of the housing system, rotation system, intake-exhaust system, cooling system and lubricating system. It does not have a valve mechanism and can transmit its output power only by means of rotary motion, allowing it to be more compact and more simply constructed than the reciprocating engine.

A considerable number of its component parts are of the same construction and have the same function as those of the reciprocating engine. Their construction and shapes are relatively fixed. Other parts have functions and construction unique to the rotary engine. There is the possibility of newer construction and flexible factors in the manufacturing process and materials, as research and development proceed further.

This Chapter describes the basic construction of the rotary engine, mainly the construction and characteris ics of the component parts unique to the rotary engine.

3.1 ROTOR

The rotor has a function corresponding to the piston and connecting rod of a reciprocating engine, directly transmitting the pressure of combustion gas to the output shaft as turning force. Also, the rotor does the work of the intake-exhaust valves. The rotor automatically opens and closes the intake-exhaust ports as it rotates.

The rotor is equipped with an apex seal on each apex, and side seals, corner seals and oil seals on both sides. The rotor gear and bearing are incorporated in its center. **Fig. 3.1** shows the names of the parts of the rotor.

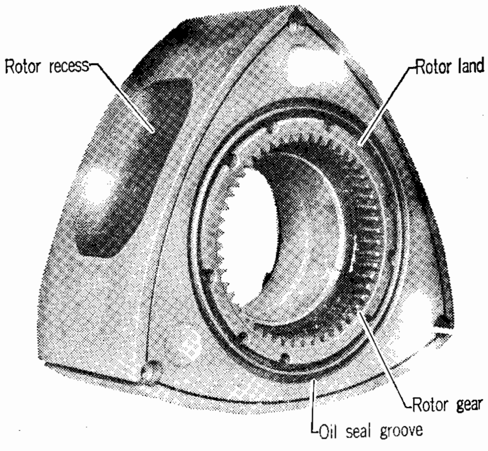


Fig. 3.1 Names of parts of the rotor

3.1.1 BASIC CONSTRUCTION

The contour of the rotor is based on the inner envelope of a trochoid, as described in 2.2.

In a practical engine, the rotor housing inner surface is formed by the curve of the basis trochoid parallel transferred outward by a fixed distance. The outer surface of the rotor is also formed by the inner envelope parallel transferred outward in the same manner, accordingly.

A clearance between the rotor and the rotor housing is secured by giving slightly smaller parallel transfer for the rotor than that for the rotor housing. From the viewpoint of the engine performance, a minimum possible amount of this clearance is desirable. Generally, it is set at approximately 0.5 mm considering deflection of the output shaft, thermal deformation, bearing clearance, tolerances for manufacture, etc.

The three sides of the rotor periphery are called the rotor flanks, which are machined by numerically controlled machines or copying grinders.

The rotor is of hollow construction for cooling and lighter weight, provided with ribs for rigidity and higher cooling effect.

(1) ROTOR RECESS

A recess provided in each rotor flank is called the rotor recess. Its volume determines the compression ratio of the engine. Moreover, its configuration and location have a great effect on the combustion characteristics of the engine, as described in 4.5.

Therefore, an optimum rotor recess has been selected to meet the particular requirements for output performance, fuel consumption performance, driveability exhaust gas, etc. of the engine.

The rotor recess may be machined by numerically controlled machines or copying grinders. In general, it is precision cast eliminating the need of machining.

(2) ROTOR LAND

A projection provided on the side face of the rotor is called the rotor land that defines the position of contact between the rotor and the side housing for most favorable lubrication.

The land is generally located on the inner side of the oil seal where sufficient lubricating oil can be supplied and the sliding speed over the side housing is small.

The top face of the land is tapered for better lubricating effect.

The projection of the rotor land is generally set at 0.1~0.15 mm. A smaller projection will cause the part other than the rotor land to contact with the side housing when the rotor leans, while a greater projection will increase the area of the side seal directly exposed to combustion gas causing gas leak, seal temperature rise, etc.

The clearance between the rotor land and the side housing is generally set at 0.1~0.2 mm to prevent the two parts from sticking together as well as to keep to a minimum noise due to inclination and axial movement of the rotor.

(3) BLOW-BY GAS RECOVERY GROOVE

A difference in pressure of blow-by gas leaked into both side spaces of the rotor from the side seals and corner seals will press the rotor to one side in the side housing.

To eliminate such a pressure difference, a balancing hole connecting the two side spaces may be cut in the rotor. However, the result will be a complicated structure and increased weight of the rotor. Instead of this balance hole, a blow-by gas recovery groove has been provided, as shown in **Fig. 3.2.**

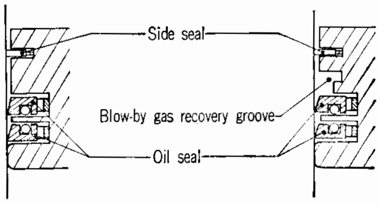


Fig. 3.2 Blow-by gas recovery groove

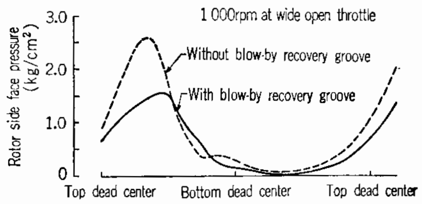


Fig. 3.3 Comparison of rotor side face pressure

The blow-by gas leaked into both side spaces will be recovered into the side intake port through this groove.

Such a blow-by gas recovery groove will also reduce the variation in gas pressure in the rotor side spaces and stabilize the oil sealing function.

3.1.2 ROTOR MATERIAL

The requirements for the rotor material are: (1) higher fatigue strength at high temperature ; (2) smaller thermal expansion coefficient ; (3) higher wear resistance ; and (4) better castability and workability ; etc. In general, nodular graphite cast iron is used.

However, from the viewpoint of the rotor weight, this nodular graphite cast iron has the disadvantage of its greater specific gravity.

Less weight of the rotor will reduce the load on its bearing, which is effective to obtain a wider range of allowable engine speed.

For that purpose, the development of a rotor using light weight material such as aluminum alloys, etc. is under way.

3.2 CONSTRUCTION OF HOUSING

The housing of the rotary engine, corresponding to the cylinder block and cylinder head of the reciprocating engine, is composed of a rotor housing and its side housings forming the side walls.

Fig. 3.4 shows the construction of the housing of the rotary engine.

3.2.1 ROTOR HOUSING

In the reciprocating engine, each stroke for intake, compression, expansion and exhaust takes place in the same space. The heat load on its cylinder head and

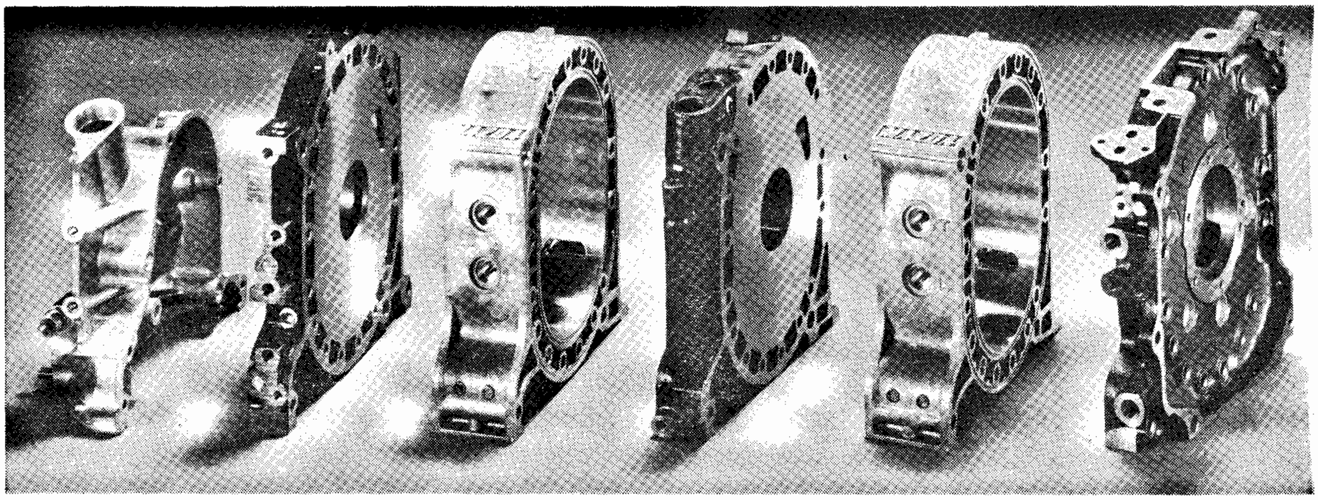


Fig. 3.4 Construction of housing

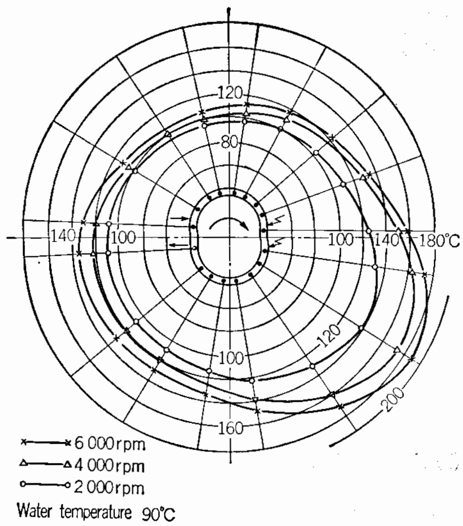


Fig. 3.5 Temperature distribution of inner surface of rotor housing

cylinder block is relatively evenly distributed.

In the rotary engine, on the other hand, the working chamber moves as the stroke proceeds. The space for the intake stroke in the housing is always cooled by fresh air, while that for the expansion stroke is always exposed to high temperature and high pressure.

Furthermore, pressure due to centrifugal force of the apex seal and gas pressure acts on the inner surface of the rotor housing, always showing a greater value on a particular position.

Fig. 3.5 shows an example of the temperature distribution over the inner surface of the rotor housing.

As shown, the housing is subject to different temperature and pressure with a great difference on different areas over its surface. Therefore, the following items have been taken into consideration for material, structure, surface treatment, etc. of the rotor housing:

(1) To provide a strength sufficient to withstand mechanical stresses to be caused by combustion pressure, fastening bolts, etc.

(2) To minimize the temperature difference as well as to withstand the thermal stress to be caused due to uneven temperature distribution. Especially, to secure sufficient means for cooling and lubrication in the vicinity of the spark plug hole to be exposed to the highest temperature.

(3) To minimize deformation of the inner surface of the rotor housing, thereby preventing gas leak at the apex seal leading to a reduction in engine performance.

(1) BASIC STRUCTURE

The apex seal slides over the inner surface of the rotor housing. To reduce wear in the apex seal, the inner surface of the rotor housing is formed by a peritrochoid curve, geometrically, as determined, that has been parallel transferred outward by a fixed distance.

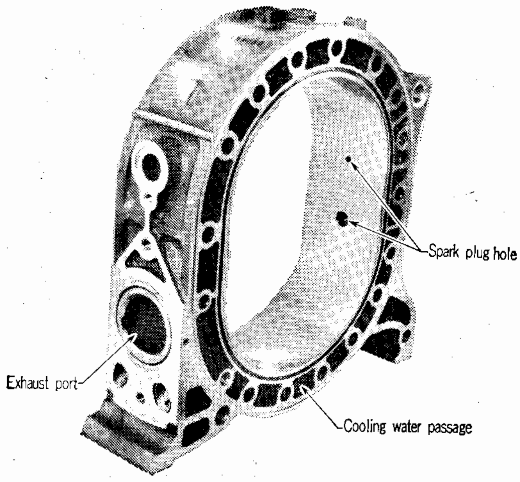


Fig. 3.6 Construction of rotor housing

Passages for cooling air or water are provided around the outside of the inner surface of the rotor housing. Ribs or fins are also provided for higher rigidity and better cooling effect. Further, in the rotor housing, spark plug holes, an exhaust port, etc. are provided at their respective optimum positions meeting the requirements for engine characteristics.

Fig. 3.6 shows an example of the structure of the rotor housing.

The cooling method for the rotor housing can be classified into the circumferential flow cooling system and the axial flow cooling system depending on the direction of flow of the cooling medium.

In the circumferential flow cooling system, air or water flows around the outer side of the trochoid. **Fig. 3.7** shows an example of the rotor housing for circumferential flow cooling.

In the axial cooling system, air or water flows through the outer side of the trochoid in the axial direction.

Fig. 3.8 shows examples of the rotor housing structure for water cooling (a) and air cooling (b).

For automobiles, the axial flow water cooling

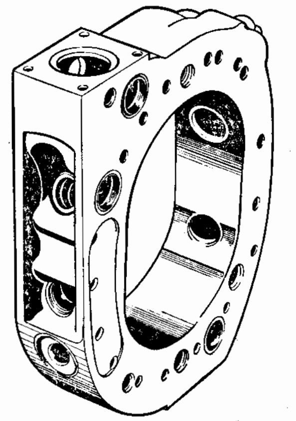
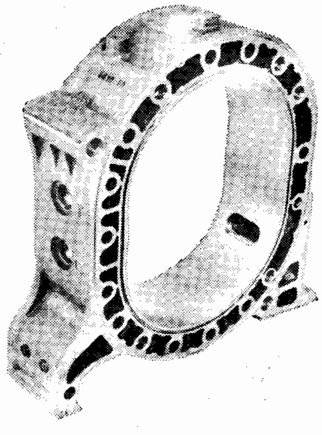


Fig. 3.7 Construction of circumferential flow-type water cooled rotor housing


(a) Water cooling type

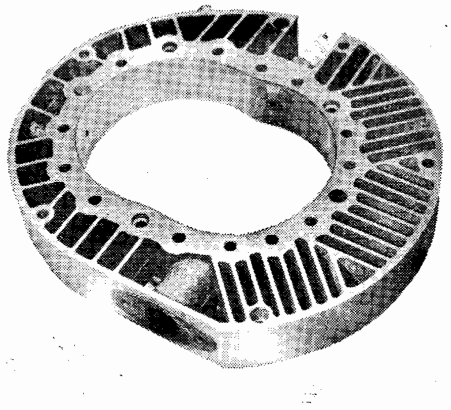

(b) air cooling type

Fig. 3.8 Construction of axial flow-type cooling of rotor housing

system is generally applied because of its relatively easy manufacturing.

(2) MATERIAL

Since the rotor housing is subjected to a much higher heat load than the side housing, its material must be of high strength, small thermal expansion coeficient and high heat conductivity.

In general, gamma silmin aluminum alloy is used for the rotor housing.

Fig. 3.9 shows a comparison of temperature distribution of cast iron and aluminum alloy rotor housings.

The cast iron rotor housing of low heat conductivity shows its highest temperature approximately 150℃ higher than that of the aluminum alloy rotor housing, exceeding the critical temperature for maintaining lubricating oil film.

However, cast iron has a smaller thermal expansion coefficient than aluminum alloy causing less thermal deformation and is lower in cost.

Therefore, further studies are being conducted for improving the housing structure.

For the aluminum alloy housing, its inner surface

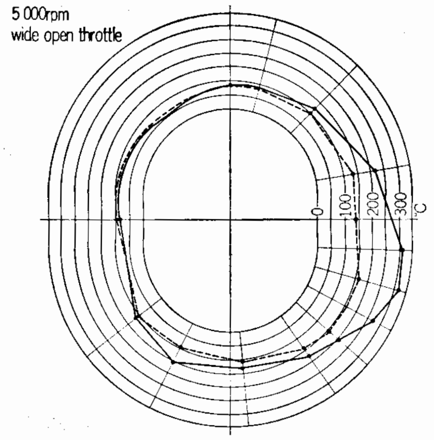


Fig. 3.9 Comparison of temperature distribution

over which the apex seal slides has been chrome plated for higher wear resistance. To improve the adhesive property of chrome plating, a new manufacturing method, called SIP (Sheet-metal Insert Process), has been developed.

In this method, a sheet of steel formed to the trochoid is inserted inside the rotor housing and its inner surface is hard-chrome plated. The outer surface of the sheet is provided with a fine-saw-toothlike cut for a better bond to the aluminum alloy.

Fig. 3.10 shows its cross sectional structure.

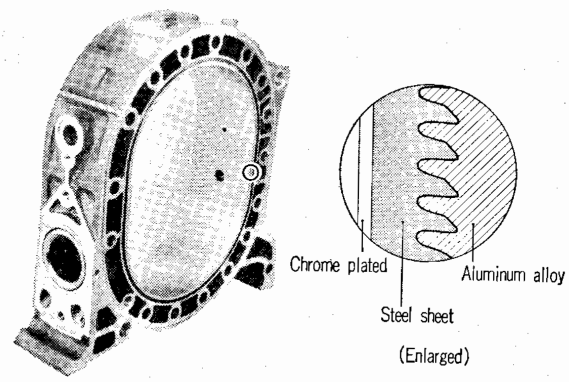


Fig. 3.10 Cross-sectional structure of SIP rotor housing

(3) INNER SURFACE TREATMENT OF HOUSING

In the early stage of development of the rotary engine, the greatest obstruction encountered was chatter marks caused on the inner surface of the rotor housing. The chatter marks were caused by frictional vibration of the apex seal and greatly affected the durability of the engine.

To solve this problem, a long period of time was spent on the study of the apex seal material and the configuration to be used, and on the inner surface treatment of the rotor housing.

As a result, it has proved effective to use hard chrome plating, nickel plating containing silicon carbide

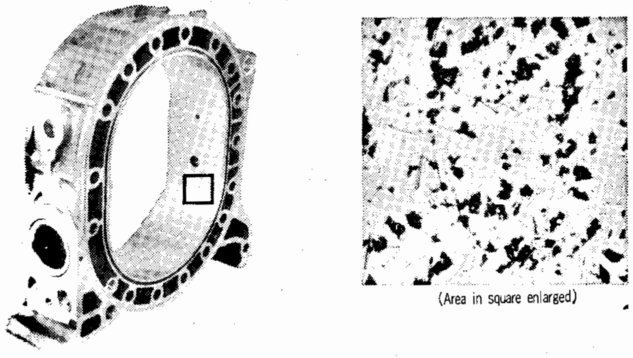


Fig. 3.11 Structure of porous chrome-plated surface

particle (Nickasil, Elnisil), spraying of combined carbide or molybdenum alloy, etc.

The hard chrome plating is usually performed over a steel sheet layer in the SIP method, as abovementioned, since the direct chrome plating on aluminum alloy results in poor adhesive property and strength.

Further, to improve the maintenance property of lubricating oil film, small holes or grooves may be mechanically cut in the chrome plated surface, or the surface may be porous-chrome plated electrically.

Fig. 3.11 shows the porous chrome plated surface.

(4) SPARK PLUG THROUGH HOLE

The spark plug of the rotary engine must be located on the outer side of the inner surface of the rotor housing so as to avoid contact with the apex seal and rotor. Therefore, a through hole is cut in the rotor housing connecting the spark plug to the working chamber.

Since the location of the spark plug through hole has a great effect on combustion efficiency, ignitability, etc. its optimum location is to be selected also in relation to the rotor recess.

For automobile engines, a two-spark plug system, one spark plug installed on the trailing side (retarded side) and the other on the leading side (advanced side) of the minor axis, is generally used to obtain better combustion efficiency over a wider range of operating conditions.

Although a large diameter plug through hole is advantageous to ignitability, gas may leak due to a pressure difference between the two adjacent working chambers caused when the apex seal passes over the hole. For this reason, a greater diameter is used for the leading side hole of the minor axis where the pressure in both working chambers nearly balances and a smaller diameter for the trailing side hole where a greater pressure difference develops.

The area in the vicinity of the spark plug through hole is heated to the highest temperature in the rotor housing and is subject to severe thermal stress. The lubricating oil film is liable to be eroded away when the apex seal is pressed against the inner surface of the rotor housing due to the high pressure of the combustion gas.

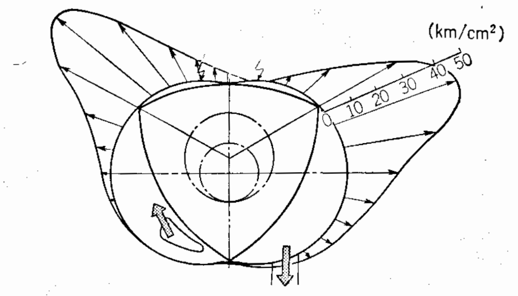


Fig. 3.12 Pressure difference between adjacent working chambers

Therefore, special considerations have been given to mechanical strength, cooling and lubrication in the vicinity of the spark plug through hole.

3.2.2 SIDE HOUSING

The side housing has sliding surfaces for the corner seals, side seals and oil seals. It is subject to gas pressure and temperature difference in its areas, as well as the rotor housing. However, the lubricating condition of the side housing is not so severe as that of the rotor housing because of its smaller heat load, circumferential velocity and contact pressure of the sliding seals.

(1) BASIC STRUCTURE

Fig. 3.13 shows the cross-sectional structure of an axial-flow water-cooled side housing.

The side housing is provided with an intake port. Its interior is hollow forming a cooling water passage. Ribs are arranged therein for higher rigidity and better cooling effect.

Also, in the central part of the side housing, a lubricating oil passage is provided for the rotor interior cooling oil returning to the oil pan.

Further, a secondary air or EGR gas passage may be provided for cleaner exhaust gas.

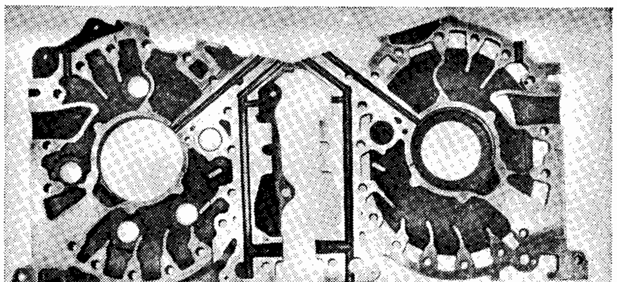


Fig. 3.13 Cross-sectional structure of axial flow type water-cooled side housing

(2) MATERIAL AND SLIDING SURFACE TREATMENT

For the side housing, cast iron or aluminum alloy may be used. Generally, cast iron of low cost is used because of its smaller heat load than that of the rotor

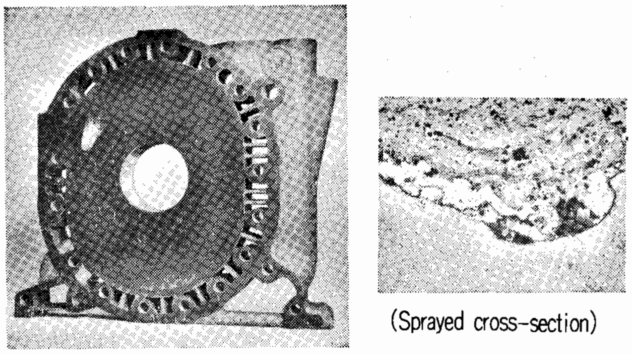


Fig. 3.14 Carbon steel sprayed side housing

housing.

For a cast iron side housing used in an engine operating under relatively light load, no special surface treatment is required ; provided the sliding surface has been sufficiently finished and the material for gas and oil seals properly selected.

On the other hand, for a high power engine subject to high heat loads, the sliding surface is hardened by nitrization, induction hardening, etc.

Aluminum alloy is generally advantageous to the phases of cooling and lighter weight, but poor in wear resistance thus requiring treatment by metal spraying, etc. on the sliding surface. In such a case, an effective method to improve the adhesive property on the aluminum alloy is to first spray on molybdenum over which carbon steel is then sprayed.

Fig. 3.14 shows the sliding surface of a side housing thus treated.

However, metal spraying causes problems in the working environment. Therefore, methods of bonding a steel sheet over aluminum alloy and of forming a hardened layer over the surface by chemical treatment, etc. are being studied.

3.3 PHASING GEAR MECHANISM

The rotary engine is equipped with a phasing gear mechanism for accurate control of the rotary motion of the rotor.

The phasing gear consists of an external gear (stationary gear) fixed on the side housing and an internal gear (rotor gear) installed in the rotor, corresponding respectively to the peritrochoid generating base circle and rolling circle, as described in 2.1.

These phasing gears are meshed at the rotating ratio 1 : 3 of the rotor to the output shaft. The rotor apex draws a trochoid curve that is the basic curve for the inner surface of the rotor housing.

3.3.1 BASIC STRUCTURE OF PHASING GEAR

For phasing gears, the profile shifted spur gearing is generally used.

The stationary and rotor gears are designed for a gear ratio 2 : 3 and for their pitch diameters $4e$ and $6e$, respectively, so that the center distance will be equal to the eccentricity e of the trochoid.

The normal backlash for the gearing is determined considering the clearances between the journals and the rotor shaft and main shaft bearings.

The stationary gear is solid with the main shaft bearing housing, as shown in **Fig. 3.15**, which is generally press fit and bolted to the side housing.

The spring constant of the stationary gear itself is closely related with the magnitude of gear load and the point of resonance. Therefore, considerations have been given to the thickness and shape of the stationary gear for sufficient fatigue strength and proper rigidity.

The rotor gear is usually in the shape of a ring gear, as shown in **Fig. 3.1**.

The rotor gear may be bolted to the rotor, but, in general, spring pins are used for its flexibility to absorb gear load, as shown in **Fig. 3.16**.

The side faces of the rotor gear also often play the role of the rotor land that governs the axial movement of the rotor. Provisions have been made for face shape and lubricating to prevent seizure with the

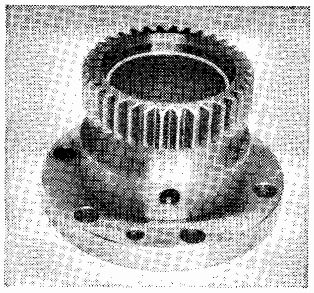


Fig. 3.15 Stationary gear

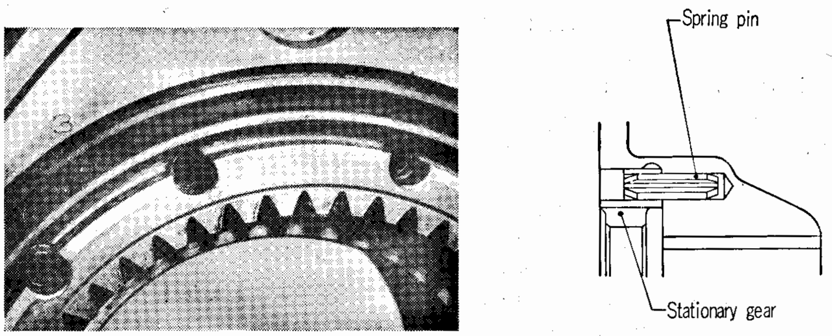


Fig. 3.16 Rotor gear installed with spring pins

side housing.

The gear material is normally S 45 C or equivalent of carbon steel conditioned for mechanical structure. For particular specifications for a racing car engine, alloy steel or surface hardened material are used to increase fatigue strength.

3.3.2 TOOTH LOAD

Various concepts have been proposed on the generating mechanism of tooth load and some of them theoretically analyzed. In general, the following factors are thought to be the possible causes of generating a tangential load on the tooth:

(1) Variation in the rotor speed.
(2) Unbalanced inertia force in the revolution system.
(3) Clearance between bearings and journals.
(4) Deflection in the output shaft.
(5) Errors in gear cutting and in installation of gears.

Fig. 3.17 shows an example of measurement of tooth load on the stationary gear.

The tooth load tends to increase as the engine speed and load increases, and has the point of resonance showing the maximum value to the engine speeds.

To reduce the tooth load, the effective method is to relieve the above factors generating the load. Further effective methods may be to give flexibility to the phasing gears allowing them to absorb the tooth load, or to move the point of resonance out of the range of normal engine speed by changing the spring constant of the gear itself.

Moreover, the tooth load will be affected depending on the condition of combustion and an the characteristics of the power transmission system.

Fig. 3.18 shows a comparison of tooth load between the single-and dual-spark plugged engines.

Also, a similar effect is apparent due to change in the spark plug position and the rotor recess configuration. Abnormal combustion such as knocking, preignition, etc. will further greatly increase the tooth load.

Fig. 3.19 shows a comparison of tooth load between the manual and automatic transmissions.

When an automatic transmission is installed, the tooth load is greatly reduced. This indicates that the tooth load will be also greatly affected by damping capacity and inertia force of the power plant to be installed.

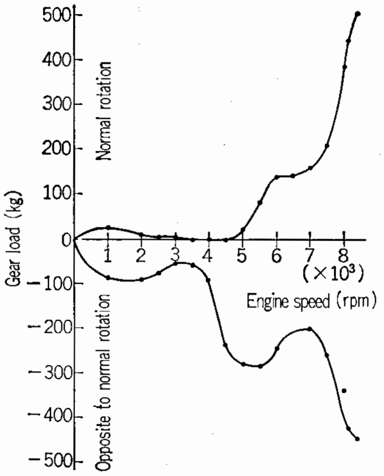


Fig. 3.17 Gear load (at W.O.T.)

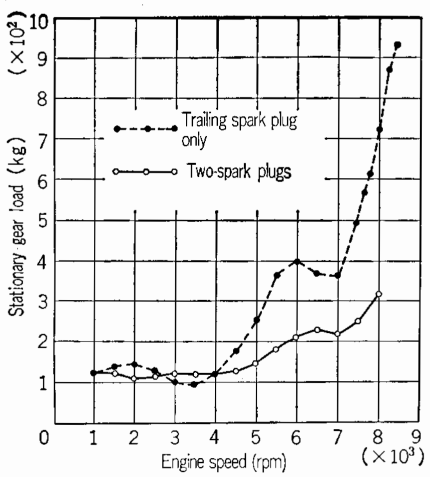


Fig. 3.18 Comparison of gear load

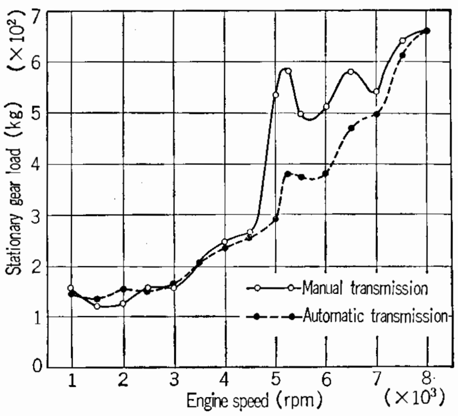


Fig. 3.19 Comparison of gear load

3.4 OUTPUT SHAFT SYSTEM AND BEARINGS

The output shaft of the rotary engine corresponds

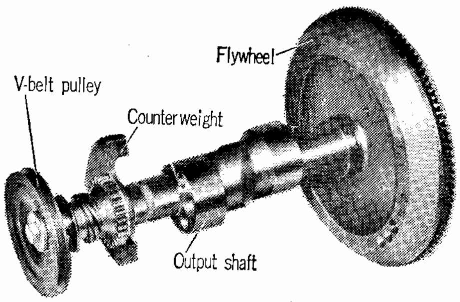


Fig. 3.20 Construction of output shaft

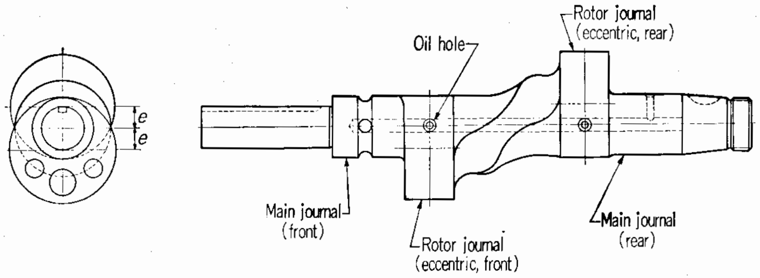


Fig. 3.21 Construction of output shaft

to the crank shaft of the reciprocating engine. It functions to generate a turning force by receiving the combustion pressure applied on the rotor journal eccentric to the center of rotation.

Fig. 3.20 shows the output shaft system for a 2-rotor engine.

The output shaft is supported by the main bearings incorporated in the front and rear stationary gears. A balance weight, an auxiliary equipment driving gear, V-belt pulley, etc., are installed on the front part ; and a flywheel on the rear.

3.4.1 STRUCTURE OF OUTPUT SHAFT

Fig. 3.21 shows the structure of the output shaft of a 2-rotor engine.

The rotor journal is eccentric to the center of rotation of the output shaft by the eccentricity e of the trochoid.

A lubricating oil passage running through the output shaft is provided to supply oil to the bearings and the rotor interior.

The diameter of each journal is determined considering the design of the phasing gears, arrangement of seals, bending rigidity of the output shaft, etc.

The output shaft has considerably higher rigidity than that of the crank shaft of the reciprocating engine, requiring no consideration about torsional vibration, in general.

The output shaft material is normally a forged chrome steel, chromemolybdenum steel, etc. of high bending rigidity, and each journal is hardened.

3.4.2 BALANCE OF INERTIA FORCE

To balance the inertia force caused by the moving parts, in general, the balance of both rotating mass and reciprocating mass must be fundamentally considered.

Of the two, the rotating mass can be completely balanced by a counterweight.

On the other hand, the reciprocating mass leaves an unbalanced inertia force and an unbalanced couple of inertia which cannot be removed by a counterweight. They increase in proportion to the square of the engine speed.

If such unbalanced forces remain, a reaction force or a reaction moment resisting them will consequentially act on the main bearing. Its magnitude and direction will be regularly changed to become a cause of vibration that will restrict the allowable engine speed.

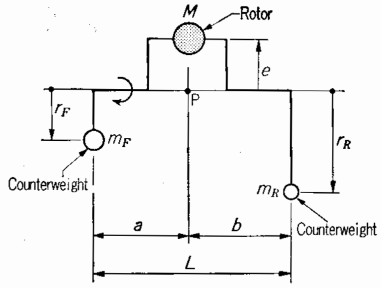


Fig. 3.22 Arrangement of counterweight

Having no reciprocating mass, the rotary engine can be completely balanced only by considering the balance of the rotating mass.

The capacity of the counterweight required for a complete balance of a single-rotor engine can be determined, as follows:

As shown in **Fig. 3.22**, in which

M: mass of the rotor (including seals and rotor cooling oil) andeccentric mass of rotor journal

e : eccentricity of rotor journal

ω : angular velocity of output shaft

m_F, r_F and m_R, r_R: capacity of each counterweight to be installed on front and rear ends of output shaft

The total sum of centrifugal forces ΣF is

$$\Sigma F = Me\omega^2 - (m_F \cdot r_F + m_R \cdot r_R)\omega^2$$

From the condition of static balance $\Sigma F = 0$,

$$Me = m_F \cdot r_F + m_R \cdot r_R \quad \cdots\cdots (3.1)$$

The total sum of couples ΣM_P with respect to Point P on the output shaft is

$$\Sigma M_P = m_R \cdot r_R \cdot b\omega^2 - m_F \cdot r_F \cdot a\omega^2$$

From the condition of dynamic balance $\Sigma M_P = 0$,

$$m_R \cdot r_R \cdot b = m_F \cdot r_F \cdot a \quad \cdots\cdots (3.2)$$

From Eqs. (3.1) and (3.2), the capacity of each counterweight can be obtained, as

$$\left.\begin{aligned} m_F \cdot r_F &= \frac{b}{a+b} Me \\ m_R \cdot r_R &= \frac{a}{a+b} Me \end{aligned}\right\} \quad \cdots\cdots (3.3)$$

M: rotor mass, e: eccentricity, m_Fr_F, m_Rr_R: capacity of counterweight for perfect balance

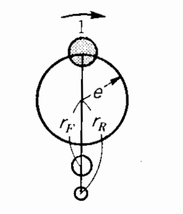
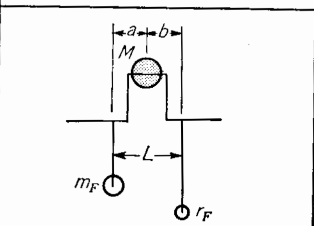
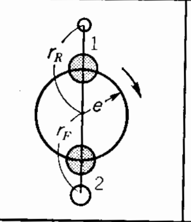
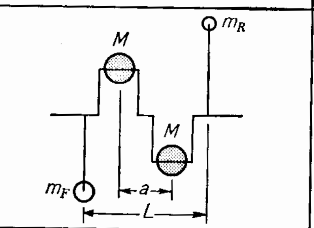
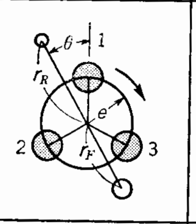
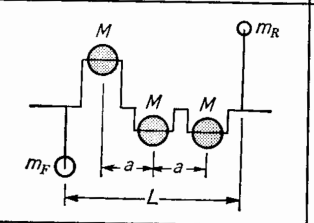
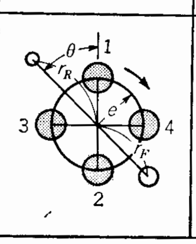
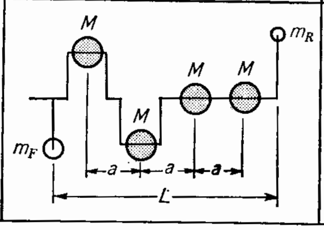
No. of rotor	Rotor & counterweight (clockwise rotation)	Output shaft	Balancing condition
1-rotor			$m_Fr_F=\frac{b}{L}M_e$ $m_Rr_R=\frac{a}{L}M_e$
2-rotor			$m_Fr_F=m_Rr_R=\frac{a}{L}M_e$
3-rotor			$m_Fr_F=m_Rr_R=\sqrt{3}\frac{a}{L}M_e$ (θ=30°)
4-rotor			$m_Fr_F=m_Rr_R=\sqrt{2}\frac{a}{L}M_e$ (θ=45°)

Fig. 3.23 Balancing condition of rotary engine

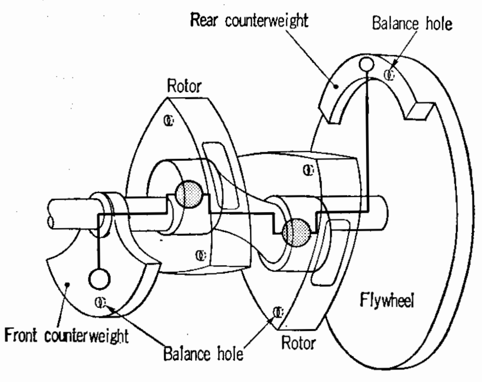


Fig. 3.24 Arrangement of counterweight

Fig. 3.23 shows the typical balancing condition and arrangement of counterweights for single to four rotors.

For the actual engine, the rear counterweight is generally made solid with the flywheel, as shown in **Fig. 3.24**

3.4.3 BEARINGS AND BEARING LOADS

For the main and rotor bearings in small sized engines, ball and roller bearings are mostly used.

For automobile engines to be operated over a wide range of revolutions under severe load conditions, plain bearings of babbit metal or aluminum alloy are generally used, as shown in **Fig. 3.1**.

Fig. 3.25 shows the characteristics of load on the main bearing of a rotary engine.

In the rotary engine, the inertia force of its moving parts can be completely balanced. The load acting on the main bearing is fundamentally only the element in the center direction of the burned gas pressure applied on the rotor.

On the rotor bearing, the burned gas pressure applied on the rotor and centrifugal force caused by the planetary motion of the rotor will be applied.

Since these two forces act nearly in opposition to each other, the load on the rotor bearing to the engine speed has characteristics as shown in **Fig. 3.26**.

This graph has been prepared by plotting the

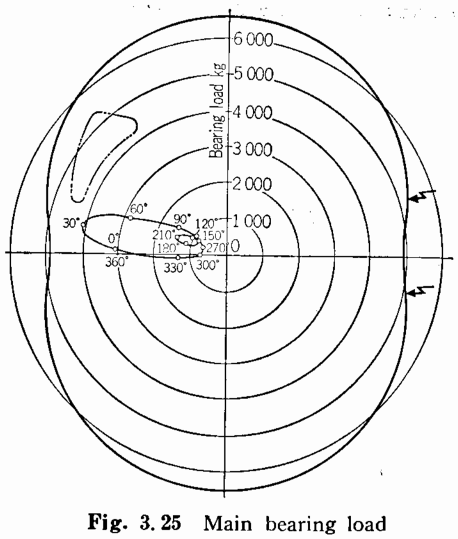


Fig. 3.25 Main bearing load

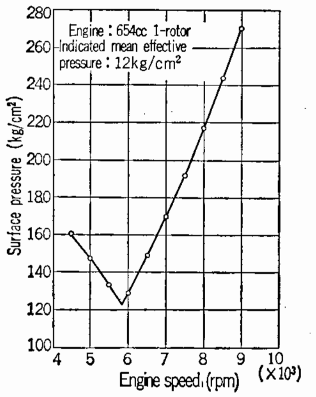


Fig. 3.26 Main bearing load

maximum value of bearing loads at each engine speed specified, indicating that the greater effect of gas pressure in the low speed range and that of centrifugal force in the high speed range are apparent.

3.5 GAS SEALING MECHANISM

The gas sealing mechanism for the rotary engine corresponds to the piston and compression rings of the reciprocating engine. It is composed of three-dimensionally combined seals with specially devised connections to cover a relatively larger area to be sealed.

The gas seals are subject to greater loads including heat load. Sufficient consideration must be given to their material, configuration, and the material and surface treatment of the sliding surfaces (**Fig. 3.27**).

3.5.1 CONSTRUCTION OF GAS SEALING MECHANISM

The gas sealing mechanism for the rotary engine consists of a side seal corresponding to the compression ring of the reciprocating engine, an apex seal to seal each working chamber from its adjacent ones, and a corner seal used at the junction of the above two seals.

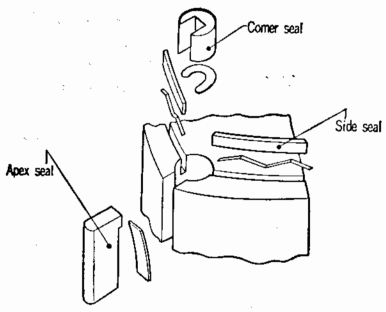


Fig. 3.27 Construction of gas seal

Each seal is provided with a spring on its back side which will allow the seal to keep close contact with the surface to be sealed even when it is worn.

In such a manner, the gas sealing mechanism for the NSU-Wankel type rotary engine can make each working chamber completely independent from the others with continuous seals to keep it gas-tight. This type of gas sealing mechanism is also called "Wankel-Grid."

3.5.2 APEX SEAL

The apex seal is provided on each apex of the rotor to keep each working chamber gas-tight. For the peripheral port, it also plays the roles of both intake and exhaust valves, which is an important part unique to the rotary engine. Since it is directly exposed to high pressure combustion gas and subject to various kinds of restraint involved in the planetary motion, most efforts have been concentrated on the study of its performance and durability.

(1) SEALING FUNCTION OF APEX SEAL

As shown in **Fig. 3.28**, the apex seal functions with its top and side to achieve gastightness. The top of the apex seal is pressed against the inner surface of the trochoid by the gas pressure and the spring tension acting on its bottom, and its side against one side of its groove by the gas pressure acting on the other side.

When thermal deformation and errors in manufacture of the seal and the housing are considered, a geometrical clearance to some extent between the seal end face and the side housing will be required. The use of a split-type apex seal can reduce such clearance to a minimum.

The apex seal keeps completely independent and

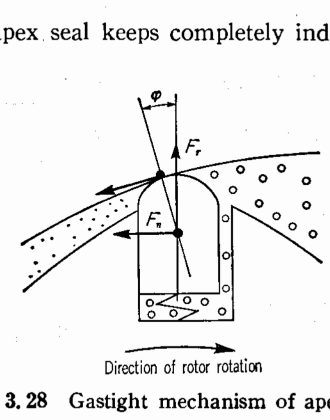


Fig. 3.28 Gastight mechanism of apex seal

gastight each of the three working chambers in which the heat cycle is continuously repeated. Each working chamber functions with a phase difference of 360°, causing a pressure difference between the two adjacent working chambers.

Fig. 3.12 shows the distribution of the pressure difference between the two adjacent chambers.

The pressure in a working chamber in the combustion stroke is higher than that in the compression stroke till the apex seal slightly passes the minor axis. Such a difference will soon disappear bringing a higher pressure to the working chamber in the compression stroke.

This means that the variation in gas pressure acting on the side of the apex seal will cause it to move back and forth in its groove.

In addition to this gas pressure, various kinds of inertia force to be caused by the planetary motion of the apex seal with the rotor must be taken into consideration with respect to the motion of the apex seal in its groove.

(2) FORCES ACTING ON APEX SEAL

The sliding velocity of the apex seal over the inner surface of the rotor housing is expressed, as described in 2.6.1, by

$$v=\frac{\omega}{3}\left(9e^2+R^2+6eR\cos\frac{2}{3}\alpha\right)^{1/2}$$

where v : sliding velocity
ω : angular speed of output shaft
α : angle of output shaft

Fig. 3.29 shows the distribution of sliding velocity of the apex seal in comparison with that of the compression ring of the reciprocating engine.

The rotary engine shows a smaller difference between the maximum and minimum values, that is, a smaller change in velocity.

In case of the reciprocating engine, the piston in reciprocating motion reverses in its sliding direction each time it reaches its top and bottom dead center. On the other hand, the rotary engine provides the sliding speed

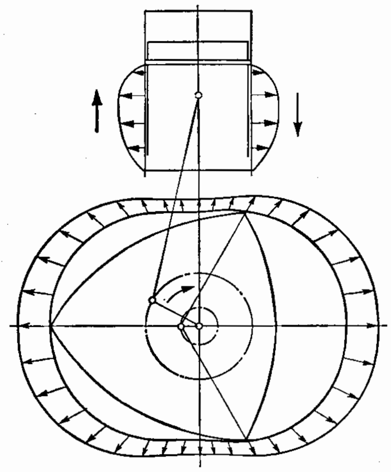


Fig. 3.29 Sliding velocity of apex seal

continuously changing in the same direction of rotation, which is also advantageous to the formation of oil film.

Such speed change of the apex seal indicates that centrifugal and inertia forces are acting on it.

The radial centrifugal force F_r of the rotor and the normal inertia force to its radius F_n can be expressed, as

$$F_r=\frac{W}{g}\cdot\omega^2\cdot\left(\frac{r}{9}+e\cdot\cos\frac{2}{3}\alpha\right)$$

$$F_n=\frac{W}{g}\cdot\omega^2\cdot e\cdot\sin\frac{2}{3}\alpha$$

where W: weight of apex seal
r : distance between the center of gravity of apex seal and the center of rotor

F_r becomes maximum and minimum on the major and minor axes of the trochoid, respectively.

$$F_{r\,max}=\frac{W}{g}\cdot\omega^2\cdot\left(\frac{r}{9}+e\right)$$

$$F_{r\,min}=\frac{W}{g}\cdot\omega^2\cdot\left(\frac{r}{9}-e\right)$$

Since generally $\frac{r}{9}<e$, $F_{r\,min}$ will become a centripetal force from the centrifugal force between the major and minor axes.

The apex seal spring is designed to overcome such centripetal force and to provide the seal with positive contact pressure.

(3) CONFIGURATION OF APEX SEAL

Various configurations for the apex seal have been devised and studied for improving its gastightness. However, from the viewpoint of durability, productivity and installation, excessively complicated configuration or construction is not practical. **Fig. 3.30** shows the three typical types of the apex seal, of which (a) is the simplest solid type.

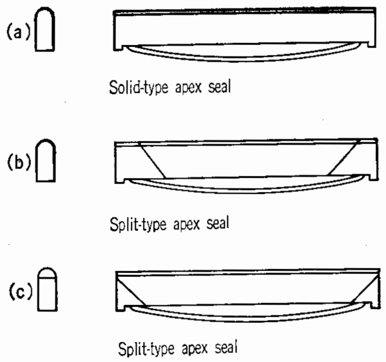


Fig. 3.30 Configurations of apex seal

(4) MATERIAL OF APEX SEAL

Durability of the apex seal is closely related to the inner surface of the rotor housing. To prevent wavy wear on the inner surface of the rotor housing, called chatter marks, many kinds of material have been studied.

Self-lubricating special carbon material has been often used against the chrome-plated inner surface of

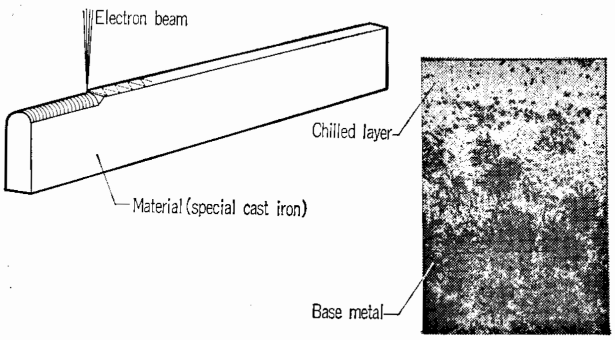


Fig. 3.31 Metal apex seal

the rotor housing. As the chrome-plating and metal surface treatment techniques progressed, a metal seal of special cast iron as base metal, chilled by electron beam, began to be mostly used against the porous chrome-plated inner surface of the rotor housing (**Fig. 3.31**). Besides, there were cases in which compound Ni-SiC plating performed on the inner surface of the rotor housing and the apex seal of special sintered alloy were used.

3.5.3 SIDE SEAL AND CORNER SEAL

The side seal of special cast iron, sintered alloy, etc. is installed on the rotor side to prevent the high pressure gas in the working chamber from leaking into the side space of the rotor. It contacts the sliding face of the side housing and may pass inside the locus of the oil seal, which is more advantageous to lubrication than that of other gas seals. A clearance of approximately 0.05～0.15 mm is provided at its junction with the corner seal, allowing for possible thermal expansion.

The corner seal keeps gastight the junction of the apex seal and the side seal. To keep gastight with the side of the rotor seal hole, it is necessary to give as small a clearance as possible between the seal hole and the seal diameter. Too small a clearance will seize the corner seal in the hole. Therefore, the corner seal may be made flexible, as shown in **Fig. 3.32**.

Provision has been made for the corner seal to be free inside the seal hole even with a small clearance by reducing its radial rigidity.

The corner seal is generally of special cast iron with its outer surface chrome-plated to improve wear resistance of the seal diameter and the rotor seal hole.

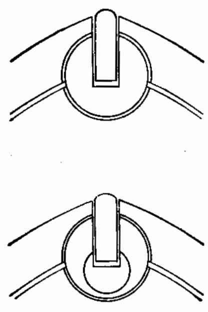


Fig. 3.32 Configuration of corner seal

3.6 INTAKE AND EXHAUST MECHANISM

The working chamber of the rotary engine moves in the housing as its volume is varied. The intake and exhaust ports provided at appropriate locations in the housing assure proper exchange of gases.

These intake and exhaust ports are automatically opened and closed by the rotor rotation, eliminating the need of the valve system as in the reciprocating engine. This has resulted in the characteristics of the rotary engine, such as simple construction, less mechanical noise and excellent high speed performance, etc.

3.6.1 INTAKE AND EXHAUST PORT SYSTEM

The intake and exhaust ports can be provided either in the rotor housing or the side housing (**Fig. 3.33**). Providing ports in the rotor housing is called the peripheral port system (**Fig. 3.33**), and this system has the following features: ① The ports are opened and closed by the apex seal. Either port is always open to either of the working chambers. The port-open time to each working chamber in terms of the angle of rotation of the output shaft is greater than 360°, ② The two adjacent chambers are intercommunicated through the port while the apex seal passes over it. ③ The direction of the flow of gas is the same as that of the rotor revolution, causing little resistance to the flow of gas.

On the other hand, the intake ports provided in the side housing is called the side port system. This system has the following features: ① The contour of the intake port is subject to geometrical restrictions (**Fig. 3.34**). That is, the side seal and corner seal must clear the inner surface of the rotor housing over the specified distance to prevent them from dropping into the

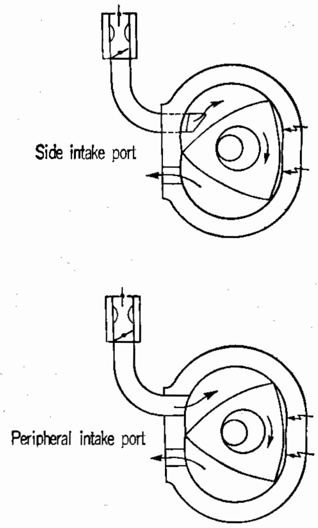


Fig. 3.33 Intake port system

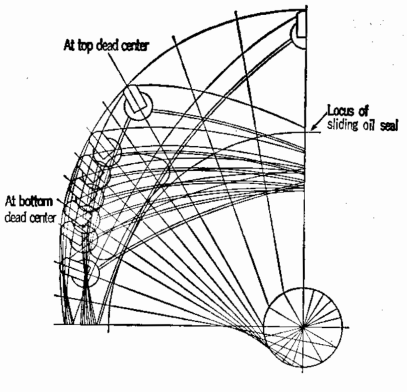


Fig. 3.34 Profile of side intake port

ports. Furthermore, the intake ports must be located outside the locus of the oil seal to prevent lubricating oil inside the oil seal from flowing into the port. ② The length of open-time of the port is shorter than that of the peripheral port by the time while the rotor side closes the former port. ③ The direction of the flow of gas differing from that of the rotor causes a greater resistance to the flow of gas.

Each location for the intake and exhaust ports are to be selected making full use of the advantages of the above side port and peripheral port systems.

For the exhaust port, the peripheral port system is normally used.

The reason for this is that the exhaust port of the side port system will cause a great amount of leak of high temperature exhaust gas into the rotor side space causing deterioration of the side and oil seals.

For the intake port, the peripheral and side port systems are selected for the characteristics required for each engine.

In the peripheral intake system, the longer open-time of the intake port and the same direction of the intake flow as that of the rotor revolution can contribute to a higher charging efficiency and output during high-speed or heavy loading operations. On the other hand, during low-speed or light loading operations, there arises a problem that the combustion will become unstable due to inclusion of a great amount of burned gas in intake gas caused by a longer overlap time of the opened intake and exhaust ports and by intercommunication between the adjacent working chambers in exhaust- and intake-strokes when the apex seal passes over the port.

In the side intake system, although its high-speed performance is lower than that of the peripheral intake system, the direction of intake normal to that of the rotor revolution easily generates swirl and the shorter overlap time between the intake and exhaust ports results in stable combustion even during low-speed or light loading operations.

Therefore, for racing engines, etc. which put great emphasis on high-speed performance, the peripheral intake system is used. On the other hand, the side intake system is used for automobile engines for general use which require balanced performance over a wide range of operation from low to high speeds.

3.6.2 PORT TIMING

The closing time and opening area of the intake port has a great effect on the performance characteristics of the engine. For higher full-open output performance a longer open-time and a greater open-area is required ; while, for better operating performance at low-speed and light loading, a minimum possible overlap of the intake and exhaust ports is required.

Fig. 3.35 shows the relation between the closing time and opening area in each port system.

The peripheral system is generally used for the exhaust port. Its opening time cannot be excessively advanced in view of the thermal efficiency. Therefore, an appropriate closing time is to be determined so as to secure a sufficient opening area.

The intake port is also required to have a maximum possible opening time and area. An excessively retarded closing time will, on the contrary, result in a lower charging efficiency because the mixture drawn in will be forced back into the intake port.

Thus, in the peripheral intake system, the opening time will have to be advanced also to secure a sufficient opening area, leading to a greater overlap with the exhaust port.

In the side intake system, the intake port will be opened after the top dead center of the exhaust-stroke, causing a smaller overlap with the exhaust port but a shorter opening time and a smaller freedom of the opening area. Therefore, it is often the case with the side intake system that the intake port is provided in both

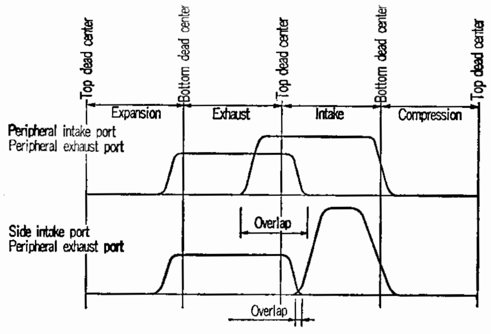


Fig. 3.35 Port timing

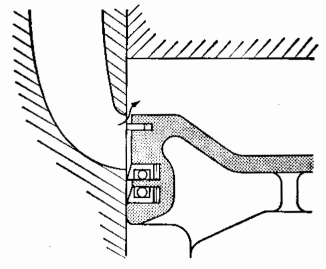


Fig. 3.36 Beginning of intake

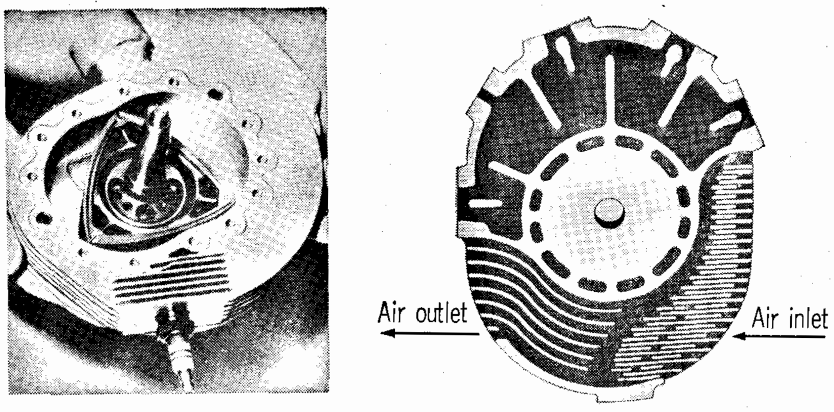


Fig. 3.37 Fins arranged for circumferential flow air cooling system

side housings.

Further, the side intake port timing is usually discussed on the basis of the profile of the rotor. In practice, the port is timed by the side seal because of a clearance lying between the rotor side and the side housing, as shown in **Fig. 3.36**. To minimize the overlap with the exhaust port, both the distance between the side seal and the rotor profile and the clearance between the rotor side and the side housing must be minimized.

3.7 COOLING SYSTEM

In the rotary engine, the strokes of intake, compression, expansion and exhaust are performed in their respective fixed positions causing a great temperature difference between the stroke positions of the housing. The temperature on the inner surface of the housing greatly affects the formation of lubricating oil film, and gastightness by the seal will be reduced due to thermal deformation.

For the housing cooling system it is important to minimize the temperature difference over the housing to prevent local thermal stresses and deformations as well as reducing its highest temperature.

The rotor is also exposed to burned gas over a wider area than that of the piston of the reciprocating engine. Its temperature greatly affects the durability of the seals installed on the rotor and knocking of the engine. For this reason, the rotor also requires a cooling system.

3.7.1 COOLING SYSTEM FOR THE HOUSING

(1) COOLING METHOD

The cooling methods for the housing are generally classified into air and water cooling methods by the type of cooling medium used, and also into circumferential and axial flow types by the direction of flow of the cooling medium.

Figs. 3.37 and **3.38** show the arrangements of fins for the circumferential and axial-flow air-cooling

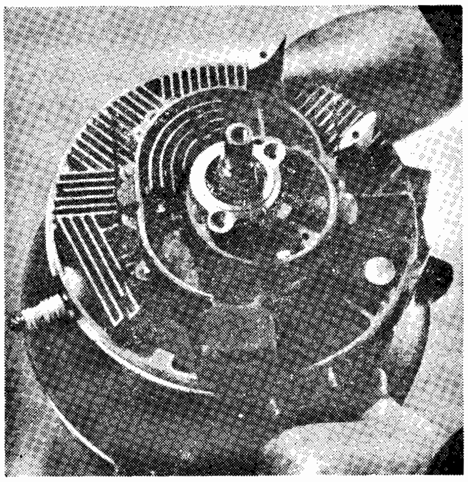


Fig. 3.38 Fins arranged for axial flow type air cooling system

systems for the housing, respectively.

In the high temperature zone, a maximum number of the thinnest possible fins are provided to increase the radiating surface and wind speed for better cooling efficiency.

Such air-cooling systems are used for small engines. For automobile engines of heavy heat load, the water cooling system is generally used.

Figs. 3.39 and **3.40** show the cooling water passages within the housing of the circumferential and axial flow water cooling systems for the housing, respectively.

In the circumferential flow type, the cooling water runs independently of each housing, being capable of

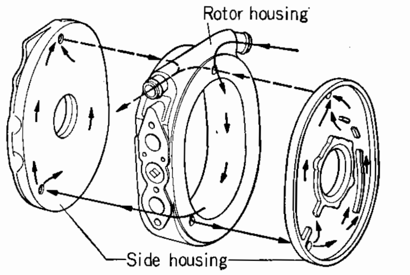


Fig. 3.39 Cooling water passage for circumferential flow type water cooling system

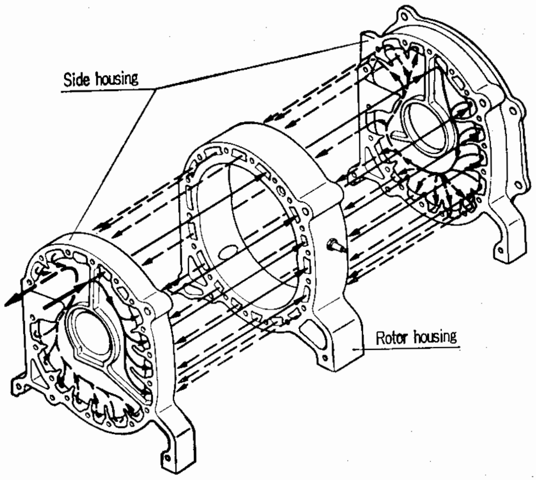


Fig. 3.40 Cooling water passage for axial flow type water cooling system

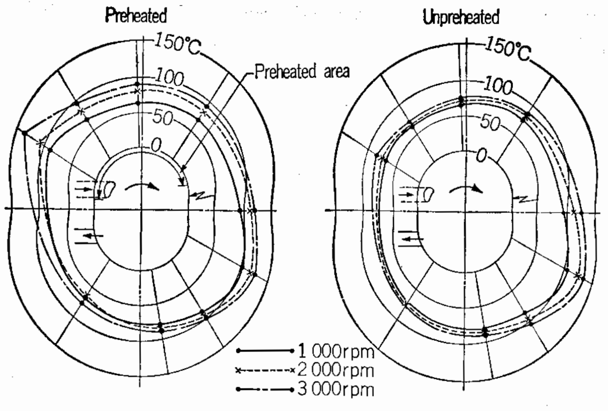


Fig. 3.42 Comparison of wall temperature of rotor housing

reducing the temperature difference between the housings and suitable for multi-rotor engines such as 4-rotors, etc. However, manufacture of the rotor housing is made difficult due to its closed passage of cooling water. Therefore, in general, the axial type is used for automobile engines.

The vicinity of the spark plug hole exposed to heavy heat load is provided with ribs and fins effectively arranged for increasing its radiating area. Higher flow speed of the cooling water also raises cooling efficiency. The low temperature area in the vicinity of the intake port is rather heated by the cooling water, instead.

Thus, the wall temperature can be averaged over the housing and, at the same time, the mixture is preheated to increase vaporization and atomization.

To raise such effects, high temperature exhaust gas may be further led to the low temperature area of the housing.

Fig. 3.41 shows its construction. **Fig. 3.42** shows a comparison of the wall temperature distribution of the rotor housing preheated by exhaust gas and that is not preheated.

(2) SEALING MECHANISM FOR COOLING WATER

A sealing system is required to prevent the cooling water leaking from the fitted surfaces between the rotor housing and the side housing.

For the circumferential flow type, the cooling water hole connecting the housing can be provided in areas avoiding the high temperature area. For the axial flow type, sealing is required along the entire trochoid periphery.

The rotor housing and side housing show a difference in deformation due to different material, temperature, pressure, etc. To accommodate any deviation, a rubber seal ring is installed in the groove provided in the rotor housing.

Since the burned gas from the working chamber enters into the sealed area, high heat resistant rubber is used for the seal. A protective layer of Teflon may be glued on the working chamber side.

3.7.2 COOLING OF ROTOR

The rotor is cooled by mixture in the intake-

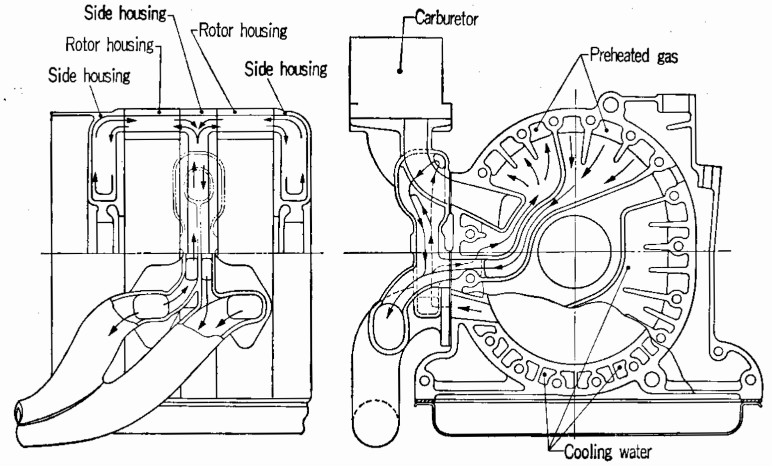


Fig. 3.41 Intake mixture heated by exhaust gas

stroke and heated by burned gas in the combustion-stroke.

The rotor is required to be cooled in view of durability of seals and knocking of the engine, but, desirably, to be kept at an appropriate high temperature in view of thermal efficiency.

Water, being a cooling medium of highest cooling efficiency, cannot be used for cooling the rotor in planetary motion due to the difficulty of providing a perfect sealing mechanism on the rotor. Therefore, intake mixture or lubricating oil is used as a cooling medium.

Fig. 3.43 shows the rotor cooling system by means of intake mixture. The mixture enters into the rotor from the side housing through a through-hole provided in the rotor side face and sucked into the working chamber through the side housing on the opposite side. During this process, the mixture cools the rotor and is, at the same time, preheated by the heated rotor which in turn pronotes vaporization and atomization of the mixture.

This system requires no special heat exchanger, such as oil cooler, etc., and is suitable for small-sized light weight engines. However, the mixture cooling system requires a longer intake passage for the intake mixture, bringing about a problem of reduction in output due to increased suction resistance. For automobile engines, in general, the rotor cooling system by means of lubricating oil is used. **Fig. 3.44** shows the behavior of lubricating oil inside the rotor.

Lubricating oil is injected into the rotor through the output shaft. The oil runs eddying along the inner surface of the rotor to take heat from it according to the change in inertia force caused by the planetary motion of the rotor. Then, it flows out from the rotor by the centripetal force toward the rotor center and is recovered in the oil pan through the side housing.

When the oil temperature becomes too high it will have insufficient cooling effects on the rotor as well as reducing its viscosity and film strength.

To maintain sufficient lubrication and cooling effect, a heat exchanger (oil cooler) is generally used for cooling the oil.

The rotor surface temperature is higher on the leading side due to difference in burning condition. The trailing side temperature has a great influence on causing knocking. Therefore, consideration has been given to the arrangement for ribs provided inside the rotor so that the lubricating oil will be properly circulated through both leading and trailing sides.

Fig. 3.45 shows the temperature distribution on the rotor surface.

Fig. 3.46 shows a comparison of the rotor surface temperatures with and without cooling by means of lubricating oil.

The temperature greatly changes according to the engine revolution. A more positive cooling is required over the range of higher revolutions, while no cooling is rather desirable in some range of low revolutions.

For this purpose, in some engines, the supply of cooling lubricating oil into the rotor is controlled according to operating conditions.

Fig. 3.47 shows a valve mechanism to control injection of lubricating oil according to the engine revolution. The control valve is installed on the output shaft. In slow revolutions, the injection passage of lubricating oil is closed by a spring force. As the revolutions increase, the centrifugal force and oil pressure applied on the ball will open the oil passage to inject oil into the rotor.

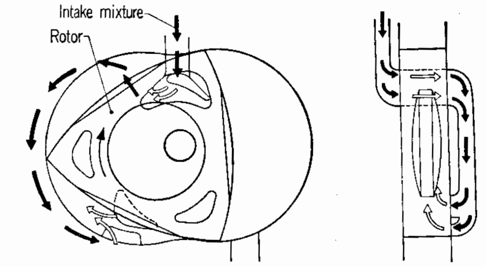


Fig. 3.43 Rotor cooling by intake mixture

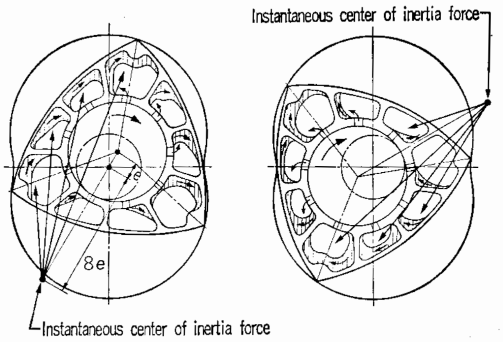


Fig. 3.44 Behavior of lubricating oil inside the rotor

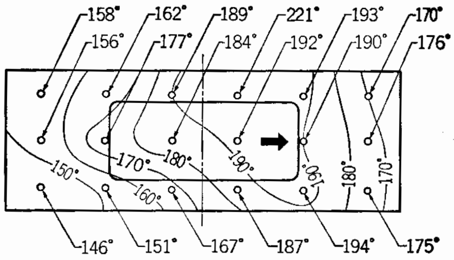


Fig. 3.45 Temperature distribution of rotor surface

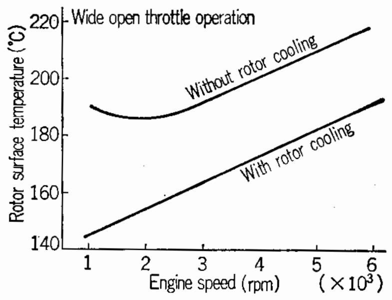


Fig. 3.46 Comparison of temperature of rotor housing

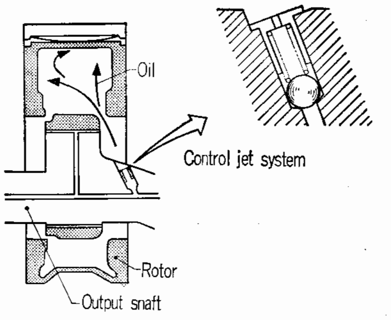


Fig. 3.47 Control of lubricating oil injection

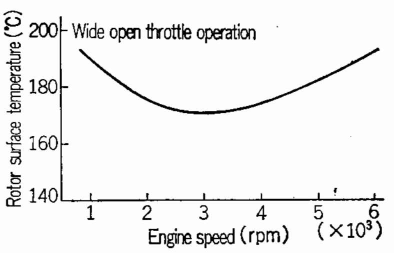


Fig. 3.48 Change in rotor surface temperature

As shown in **Fig. 3.48,** the system is provided to control the rotor surface temperature at approximately the same for that in the low revolution region as for that in the high revolution region.

Further, to maintain the optimum rotor temperature at all times, methods of changing the valve opening revolution and changing the quantity of injection according to oil temperature are also being studied.

3.8 LUBRICATION SYSTEM

The lubrication system for the rotary engine includes lubrication of gas seals and their sliding surfaces as well as main and rotor bearings, phase gears, etc. of the output shafting.

Generally, in small-sized engines, the lubricating oil is pre-mixed with the fuel ("fueloil" lubrication) and supplied into the engine for simpler construction.

In automobile engines, however, a combination of a forced-lubrication system for feeding the proper amount of oil only to the required points of the output shafting and a separate lubrication system for the gas seal sliding surfaces is used for covering the wide range of operation.

3.8.1 LUBRICATION OF OUTPUT SHAFTING

Fig. 3.49 shows an example of flow passage of lubricating oil in the forced lubrication system for the rotary engine.

The automobile rotary engines are generally equipped with an oil cooler to prevent the lubricating oil from being overheated.

In some rotary engines, an oil cooler is provided with a thermo-valve mechanism, as shown **in Fig. 3.50.**

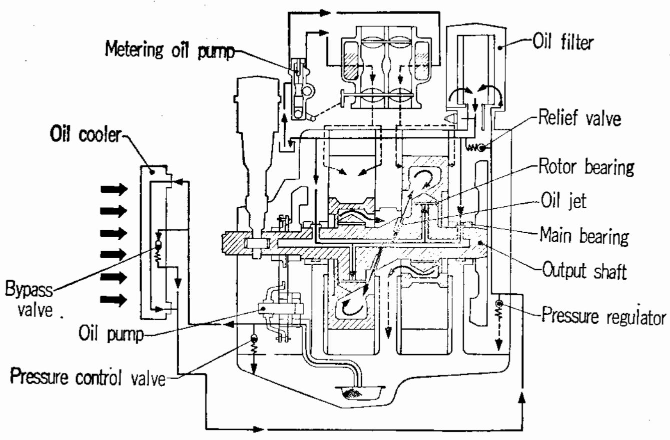


Fig. 3.49 Lubricating oil flow passage

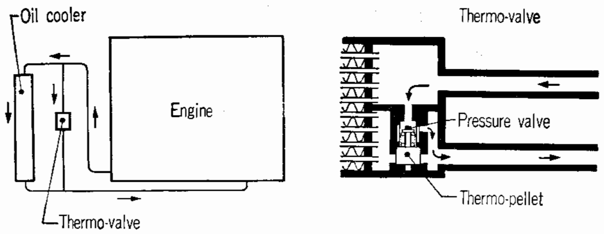


Fig. 3.50 Thermo-valve mechanism

It is actuated to open the bypass in the oil cooler while the oil temperature is low, thereby preventing heat loss due to overcooling and shortening warming-up time of the engine.

3.8.2 LUBRICATION OF GAS SEAL SLIDING SURFACE

The lubricating oil supplied to gas seals and their sliding surfaces functions to improve gastightness by the gas seal as well as to lubricate them.

There are two lubricating methods used: the pre-mixed oil-fuel ("fueloil") lubrication system, and the separate lubrication system using a small oil pump.

The "fueloil" lubrication system, of simple construction, is required to keep a high oil mixing ratio, which is liable to cause seizure of the seals by the products of combustion, fouling of spark plugs, generation of smoky exhaust, etc.

For automobile engines that are subjected to a wide range of operating condtion, the separate lubrication system using a metering oil pump is generally used. It discharges a proper quantity of lubricating oil according to engine speeds and load engine.

Figs. 3.51 and **3.52** show the construction of a variable stroke plunger type metering pump and its function, respectively.

The discharge of the pump is proportional to engine speeds. The throttle lever of the carburetor is linked with the control lever of the pump. The height of the control cam of the pump is changed according to the load applied to adjust the stroke of the plunger,

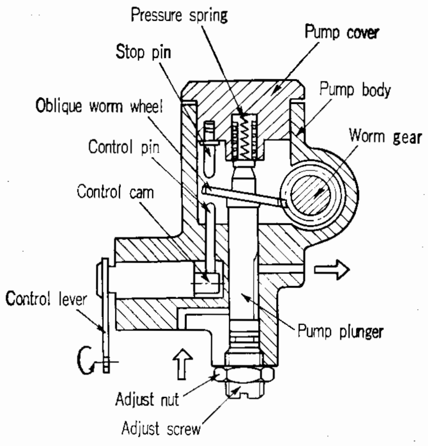


Fig. 3.51 Construction of metering oil pump

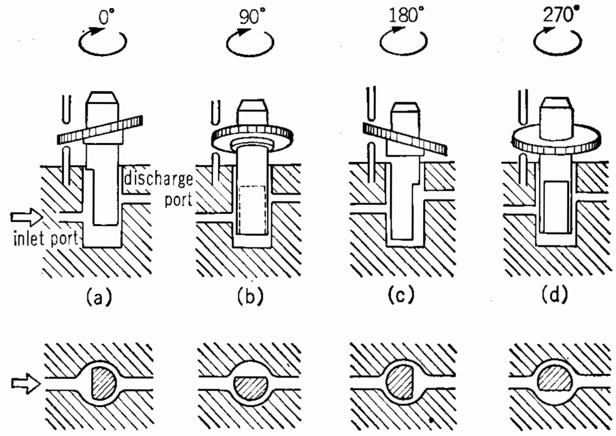


Fig. 3.52 Operation of metering oil pump

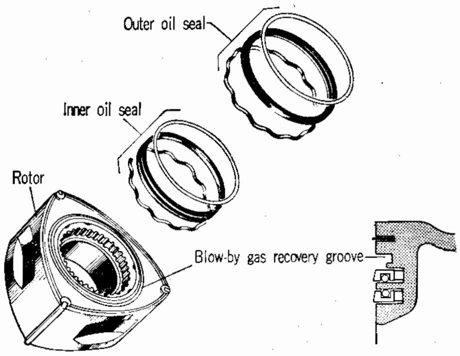


Fig. 3.53 Construction of oil seal

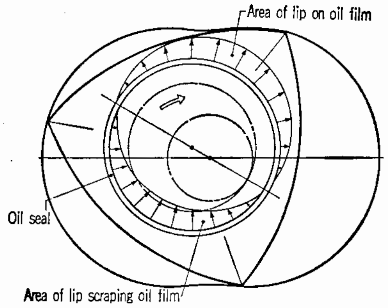


Fig. 3.54 Oil seal in motion

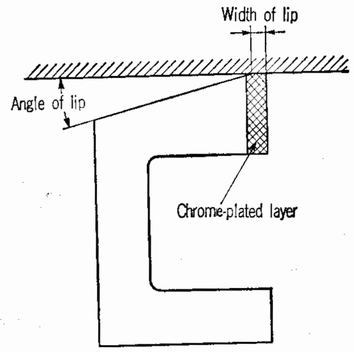


Fig. 3.55 Cross-section of oil seal

that is, the quantity of discharge per revolution.

3.8.3 OIL SEAL

The forced lubrication system for the rotary engine is provided with oil seals to prevent the lubricating oil for its output shafting and for rotor cooling from leaking into the working chamber through the gap between the rotor side face and the side housing.

Fig. 3.53 shows a typical construction of oil seals.

The oil seals are installed in grooves provided in the rotor side face and pressed against the side housing by springs. The contacting part of the oil seal with the side housing is called the oil seal lip.

"O"-rings are also incorporated with the oil seals to prevent the lubricating oil from leaking through the bottom of the oil seal grooves.

Fig. 3.54 shows the motion of the oil seal. The oil seal lip in motion scrapes oil film when it moves inward and rides on the oil film when it moves outward.

For this purpose, the lip is tapered as shown in **Fig. 3.55**.

The surface pressure of the lip greatly affects the oil seal performance. Its lower surface pressure will reduce the scraping function of the oil film, while its higher surface pressure will reduce the riding function on the oil film. Therefore, the oil seal is required to be composed of a lip with a proper angle and width and an oil seal spring of a proper load.

The worn oil seal lip shows a wider lip width, which may reduce its surface pressure leading to deterioration of the oil seal performance. To prevent this by improving the wear resistance of the oil seal, high chrome cast iron is used for the seal material and the inner surface of the rotor is hard chrome plated. Furthermore, the wear resistance of the oil seal is greatly affected by the material and surface treatment of the sliding surface. In many cases, the surface of the side housing is metal sprayed or soft nitrided.

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1) Kenichi Yamamoto: Rotary Engine, Nikkan Kogyo Shimbunsha (1969)
2) W.D. Bensinger: Rotationskolben-Verbrennungsmotoren, Springel-Verlag (1973)
3) R.F. Ansdale: The Wankel RC Engine, London Iliffe Books Ltd. (1968)
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CHAPTER 4
PERFORMANCE AND COMBUSTION

The rotary engine can be treated and evaluated with respect to performance characteristics as a displacement type internal combustion engine of four-stroke, one-cycle similar to the reciprocating engine.

However, the rotary engine has unique performance characteristics inherited from the features of its construction such as no reciprocating parts, a flat type working chamber with a high ratio of surface to volume, the rotation of the working chamber as the rotor rotates, and working time of each stroke 1.5 times longer than that of the reciprocating engine, etc.

Furthermore, the rotary engine also has different characteristics from those of the reciprocating engine in conjunction with the freedom of selection of principal components such as the intake-exhaust system, rotor recess, gas seals, ignition system, etc., and their effects on engine performance.

This chapter describes various devices and considerations being given to promote the practicability of the rotary engine according to its application as well as its fundamental performance characteristics and the effects of its principal components on engine performance.

4.1 PERFORMANCE CHARACTERISTICS

4.1.1 CALCULATION OF OUTPUT

In the rotary engine, one revolution of the rotor (three revolutions of the output shaft) completes one cycle of the four strokes of intake, compression, combustion and exhaust of each of the three working chambers. Therefore, since one revolution of the output shaft provides one combustion stroke, the output performance of the rotary engine can be calculated by the formulas given in **Table 4.1**.

4.1.2 FRICTION LOSS

Fig. 4.1 shows a comparison of friction loss between the rotary engine and the reciprocating engine.

The rotary engine shows lower values of friction loss even in the high-speed range because of the absence of a mass of reciprocating motion.

Fig. 4.2 shows the analysis of friction loss.

To reduce the sliding friction of the gas and

Table 4.1 Formulas for calculating rotary engine output

Total displacement $V(\ell)$	$V=Z\cdot V_H$ Z: number of rotors V_H: Volume of single working chamber (ℓ)
Indicated mean effective pressure P_{mi}(kg/cm²)	$P_{mi}=\dfrac{1}{V_H}\cdot\int_{\theta=0°}^{\theta=1080°}PdV_\theta$ P: gas pressure in working chamber (kg/cm²) θ: output shaft angle of rotation V_θ: Volume of single working chamber when output shaft angle of rotation is $\theta°$ (ℓ)
Indicated horsepower N_i(PS)	$N_i=P_{mi}\cdot\dfrac{V\cdot n}{450}$ n: number of revolutions of output shaft (rpm)
Shaft horsepower N_e(PS)	$N_e=\eta_m\cdot N_i=\eta_m\cdot P_{mi}\dfrac{V\cdot n}{450}=P_{me}\dfrac{V\cdot n}{450}$ η: mechanical efficiency
Net mean effective pressure P_{me}(kg/cm²)	$P_{me}=\eta_m\cdot P_{mi}=\dfrac{450\cdot N_e}{V\cdot n}$
Shaft torque T_e(kg·m)	$T_e=\dfrac{10V\cdot P_{me}}{2\pi}=\dfrac{4500\cdot N_e}{2\pi\cdot n}$

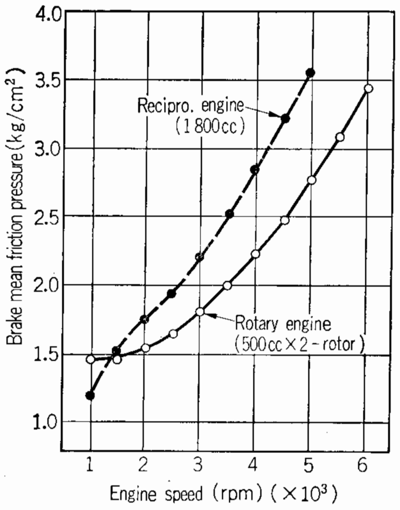

Fig. 4.1 Comparison of friction loss

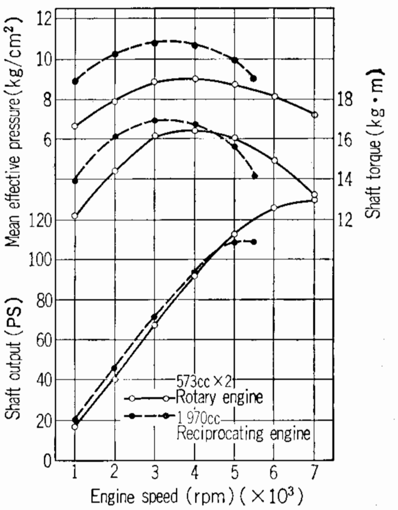

Fig. 4.3 Comparison of performance at W.O.T.

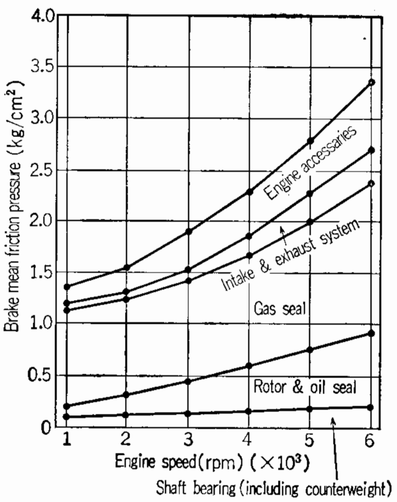

Fig. 4.2 Analysis of friction loss

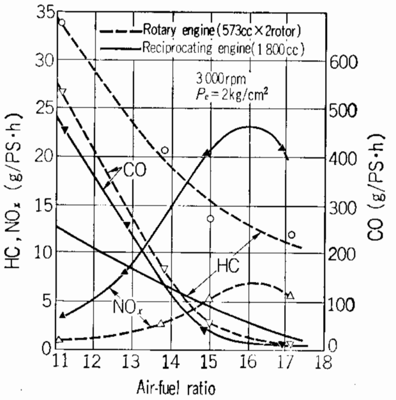

Fig. 4.4 Comparison of exhaust emission characteristics

oil seals, a reduction in spring force and seal weight, a smaller area under pressure, improvement of surface treatment and lubrication, etc. are effective measures that may be taken.

To reduce the gas leak loss, it is also important to prevent thermal deformation of the gas seals and their sliding surfaces, in addition to the improving gas seals as described in 3.5.

To reduce the friction loss of bearings, a reduction in weight of the rotating inertia system including the rotor and flywheel, etc. is effective. The use of thinner and lighter components of equal strength is being studied.

4.1.3 PERFORMANCE AT WIDE OPEN THROTTLE

Fig. 4.3 shows a comparison of the performance at wide open throttle (W.O.T.) between rotary and reciprocating engines.

As shown, the rotary engine, in general, has a slightly lower output at low-speeds but develops high performance at high-speeds.

The lower output at low speeds is attributable mainly to the greater time required for the compression and combustion strokes (1.5 times longer than that of the reciprocating engine) which causes some gas leakage loss.

Higher output at high-speeds is attributable to the small friction loss of the engine even at high speeds because of the absence of a reciprocating mass, and a high charging efficiency due to longer intake and exhaust strokes, and the absence of intake and exhaust valves causing gas flow resistance, etc.

4.1.4 CHARACTERISTICS OF EXHAUST EMISSIONS

Fig. 4.4 shows a comparison of the exhaust emission characteristics between rotary and reciprocating engines.

The rotary engine generally emits a similar amount of CO, more HC, and less NO_x, than the reciprocating engine.

The causes of the high rate of HC emission are:

deteriorated combustion efficiency due to burned gas carried into the intake-stroke, quenching effect along the wall surface of the working chamber, unburned gas leaked from the working chamber in combustion-stroke into that in exhaust-stroke, emission of fuel attached on the wall surface of the working chamber scraped by the gas seals, etc.

Effective remedies for minimizing the rate of HC emission are: reduction of the overlap time between the intake and exhaust ports, heat insulation of the wall surface of the working chamber, improvement of gas sealing performance of the apex seal, etc., improvement of atomization and vaporization of mixture, etc.

Further, the low NO_x emission rate is caused by the low maximum combustion temperature due to a high rate of burned gas in the fresh mixture, a high ratio of surface to volume of the working chamber, etc.

4.1.5 FUEL CONSUMPTION PERFORMANCE

The thermal efficiency of an automobile engine is an important factor in determining the fuel consumption performance of an automobile.

Thermal efficiency is the rate of the heat energy of fuel supplied, to the actual work done thereby.

The theoretical thermal efficiency of the Otto cycle is determined by the compression ratio ε and the specific heat ratio κ of the working gas, as expressed by

$$\eta_{th}=1-\frac{1}{\varepsilon^{\kappa-1}}$$

This shows that a higher compression ratio will improve thermal efficiency.

In practice, however, various heat losses will prevent the thermal efficiency of an engine from reaching theoretical thermal efficiency.

Fig. 4.5 shows the measurements of heat balance of a rotary engine.

Lean combustion by improving the ignition performance of the engine will reduce the rate of emission of unburned elements of CO, HC, etc., which is effective for higher thermal efficiency of the engine.

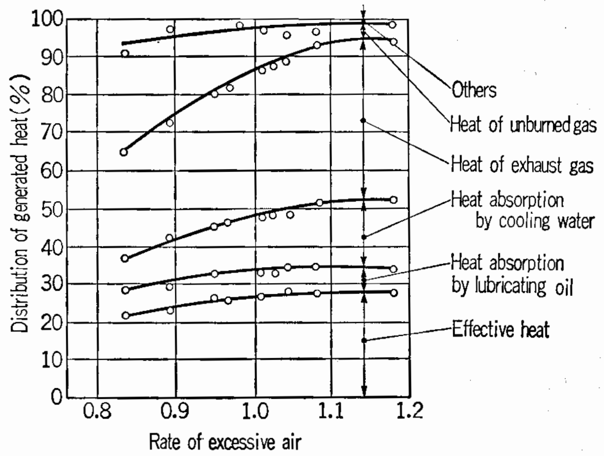

Fig. 4.5 Heat balance of rotary engine

In the rotary engine, the surface to volume ratio of the working chamber is greater and the time required for one-cycle is 1.5 times longer than those of the reciprocating engine, causing a greater cooling loss. The greater surface area/volume ratio of the working chamber also causes a greater loss of unburned gas.

Therefore, insulating the housing and the rotor to reduce cooling loss and loss of unburned gas is very effective for higher thermal efficiency of the rotary engine.

For higher thermal efficiency, it is also effective to raise the burning speed close to the isovolumetric combustion and to reduce the friction loss.

To raise the burning speed, the shape of the rotor recess is selected to obtain stronger squish flow and a 2-spark plug system is used.

To reduce friction loss, reduction in sliding resistance of the seal, improvement of gastightness, reduction in weight of the rotating inertia system, etc. are effective measures, as described in 4.1.2.

The thermal efficiency of an engine is generally expressed by the rate of fuel consumption. The rate of fuel consumption is indicated by the weight of fuel required to generate a unit output for a unit period of time. And, this will indicate the thermal efficiency by heat conversion.

Fig. 4.6 shows the relation between mean effective pressure and specific fuel consumption. The higher the mean effective pressure is, the better the specific fuel consumption will be.

The main reason for this is that the higher mean effective pressure relatively reduce the rate of friction loss to the former.

The mean effective pressure is in proportion to the torque, as shown in Table 4.1. Consequently, the operation of an engine at high torque is advantageous to low fuel consumption in an automobile, and the use of a smaller reduction ratio in the output transmission system is desirable.

However, such a transmission system with a smaller reduction ratio will lower the acceleration performance of the automobile, accordingly.

Therefore, it is effective for improving the fuel

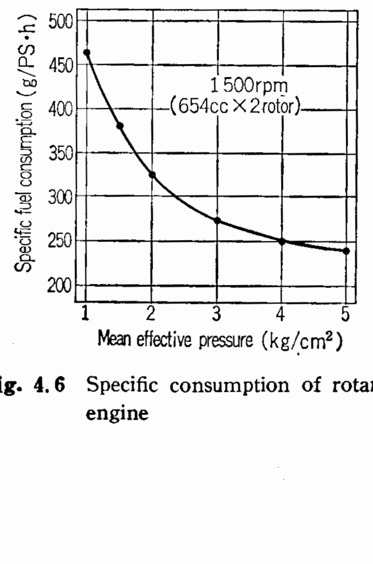

Fig. 4.6 Specific consumption of rotary engine

economy to raise the torque generated at the **time of** W.O.T. operation by taking measures for better gastight performance, reduction of intake-exhaust resistance and higher burning speed, etc. which will make it possible to reduce the reduction ratio for securing the accelaration performance as required.

Further, the rotary engine generates less vibration and noise in addition to its light weight, and requires less material for anti-vibration and sound insulation on the body side. Thus, the weight of the complete automobile is reduced.

Moreover, the rotary engine is more compact in size than a reciprocating engine of equal output. It provides greater freedom in designing bodies of smaller shape with less air resistance.

Taking full advantage of such fundamental features of the rotary engine is also very important in planning better fuel economy in rotary engine automobiles.

4.2 VIBRATION AND NOISE

Recently, car noise has been taken up as one of the environmental problems accompanying the centralization of population in urban areas and the increase in registered number of motor vehicles. Engine noise is the biggest factor causing car noise, and must be reduced to a minimum.

Engine vibration and noise have a great effect on the comfort and fatigue of the driver, and on durability of various parts of both engine and body, etc. noise reduction is a very important problem for car engine designers.

4.2.1 ENGINE VIBRATION

The main causes of vibration in an engine are **imbalances** of inertia force in the moving parts of the nertia couple and variations in torque.

(1) VIBRATION DUE TO INERTIA FORCE AND INERTIA COUPLE

Since the rotary engine has no reciprocating mass, as described in 3.4, the inertia force and inertia couple of the moving parts can be completely balanced by means of counterweights.

To investigate the vibration caused by imbalances in inertia force, a rotary engine and a reciprocating engine were operated at no-load with no variation in torque and their vibration characteristics were compared.

Figs. 4.7 and **4.8** show comparisons of variation in rotating angular velocity and of vertical vibration of engines, respectively.

The rotary engine, having **no reciprocating mass,** shows small variation in its rotating angular velocity and less increase in its vertical vibration even at increased engine speeds.

Fig. 4.9 shows the results of frequency analysis of vertical vibration of both rotary and reciprocating engines.

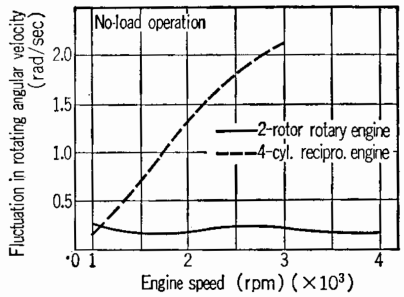

Fig. 4.7 Comparison of fluctuation in rotating angular velocity

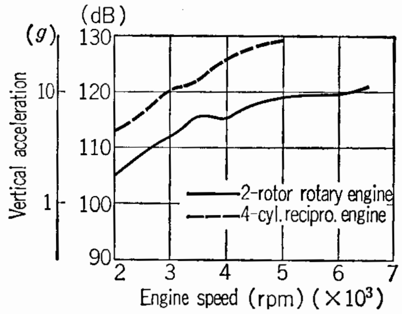

Fig. 4.8 Comparison of vertical vibration (no-load)

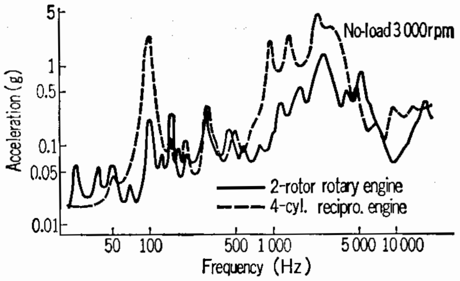

Fig. 4.9 Analysis of vertical vibration frequency

The rotary engine shows less vibration over the frequencies of 500~5,000Hz, greatly affecting the generation of noise, which is also advantageous in suppressing noise due to vibration.

(2) VIBRATION DUE TO TORQUE FLUCTUATION

The torque generating mechanism for **the reciprocating** engine is that the burned gas pressure applied on the piston is transmitted to the crank of the crankshaft through the connecting rod to generate torque.

In the rotary engine, requiring no connecting **rod,** the gas pressure applied on the rotor is directly transmitted to the output shaft to generate torque.

Fig. 4.10 shows the conceptual torque generating **mechanisms** of both engines.

The gas pressure in the working chamber acts on the rotor to generate Force F_g that runs through the rotor center. The eccentricity of the rotor to the output shaft divide Force F_g into its two components, Force F_t to act in the tangential direction with respect

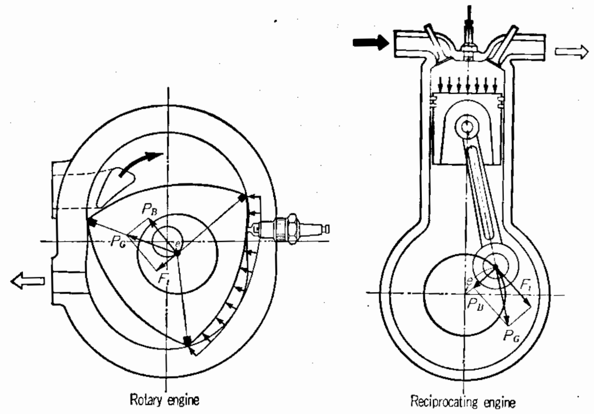

Fig. 4.10 Torque generating mechanism

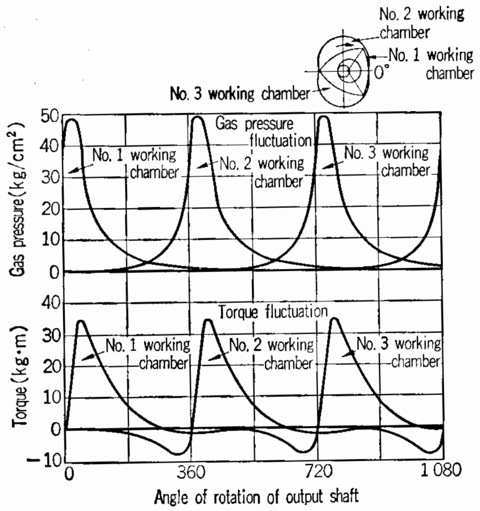

Fig. 4.11 Fluctuation of gas pressure and torque in working chamber

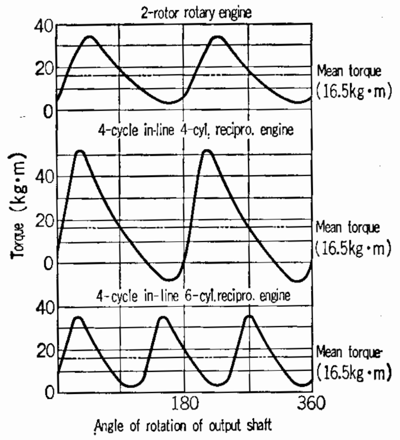

Fig. 4.12 Comparison of torque fluctuation

to the output shaft and Force F_b to act in the center direction.

The eccentricity of the rotor to the output shaft is equal to the eccentricity e of the trochoid. Thus, Torque $F_{t.e}$ is generated in the output shaft.

Fig. 4.11 shows the fluctuations in gas pressure in the working chamber of the rotary engine and in

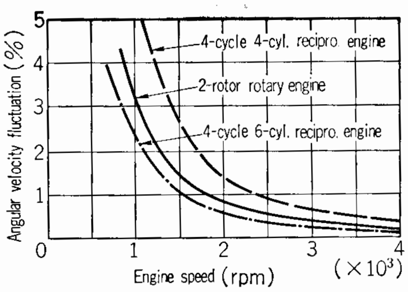

Fig. 4.13 Comparison of angular velocity fluctuation rate

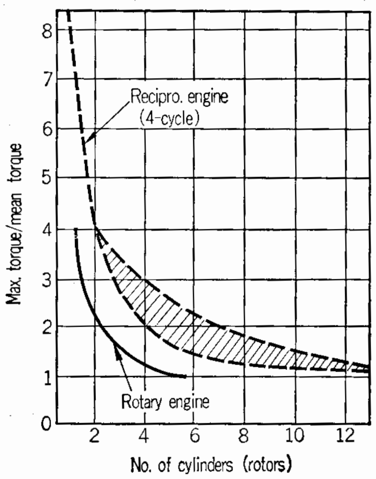

Fig. 4.14 Comparison of torque fluctuation rate

the generated torque.

Fig. 4.12 shows a comparison of fluctuation in torque between the 2-rotor rotary engine and the 4-and 6-cylinder reciprocating engines.

As described in 1.4.3, the 2-rotor rotary engine shows its fluctuations in torque similar to that of the 6-cylinder reciprocating engine.

If a flywheel of the equivalent inertia moment is used, the coefficient of fluctuation in angular velocity of he rotary engine at W.O.T. operation is equal to that of the 6-cylinder reciprocating engine, as shown in **Fig. 4.13.**

Fig. 4.14 shows the relation between the number of cylinders (or rotors) of the reciprocating and rotary engines and the coefficient of fluctuation in torque (maximum to mean torque ratio).

The shadowed area in Fig. 4.14 shows the range of fluctuation due to different types of arrangement of cylinders such as in-line, horizontal, V type, etc.

Since the rotary engine has a small coefficient of fluctuation in torque, multi-rotor type is less required.

The body of a motor vehicle is generally designed to provide higher rigidity in the horizontal direction and lower in the vertical direction while considering comfort and maneuverability.

Therefore, excessive vertical vibration of the output transmission system will cause the body to resonate

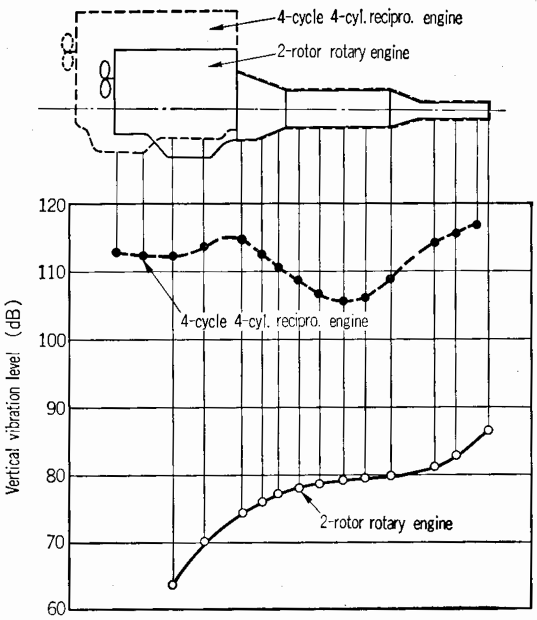

Fig. 4.15 Comparison of vertical vibration in output transmitting system

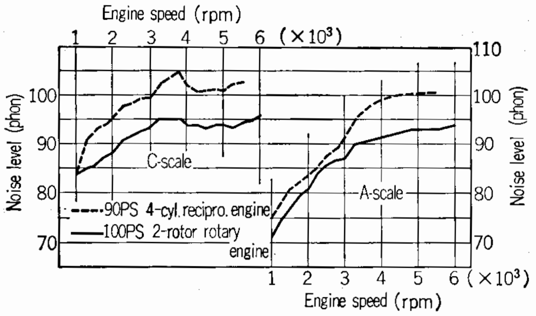

Fig. 4.16 Comparison of engine noise

giving the driver uncomfortable vibrations as well as causing a booming resonance.

Fig. 4.15 shows a comparison of vertical vibration of output transmission systems between rotary and reciprocating engines.

The rotary engine, having fewer fluctuations in torque and better balance than the reciprocating engine will cause little vibromotive force in its output transmission system. Furthermore, the output shaft located in the center of the engine allows the engine to be connected with the transmission around the entire periphery providing higher rigidity at the connection.

Such effects as above account for the small vertical vibration in the output shaft system of the rotary engine.

4.2.2 ENGINE NOISE

Fig. 4.16 shows a comparison of noise between the rotary and reciprocating engines.

The very low noise of the rotary engine as compared with the reciprocating engine is one of the significant features of the former.

Engine noise is roughly divided by its source into combustion noise, mechanical noise, intake and exhaust noise and that of auxiliary equipment.

(1) COMBUSTION NOISE

Combustion noise is caused by pressure fluctuations inside the working chamber. The rotary engine shows a slower pressure change than the reciprocating engine, resulting in lower combustion noise.

Further, for the rotary engine, there are advantages in combustion noise because of the high rigidity of its outside wall. The surface area where noise may radiate outward is small and on the water-cooled engines, a water jacket covers the engine.

(2) MECHANICAL NOISE

Because the rotary engine requires no intake-exhaust valve mechanism, mechanical noise is remarkably low.

In the reciprocating engine, the piston mechanism is a source of piston slap noise. The determination of the clearance between the piston and the cylinder is also a key point in noise reduction.

On the other hand, in the rotary engine, there is almost no variation in thrust load on the rotor. So, no such slapping will occur.

The main mechanical noise of the rotary engine are: meshing noise of the phase gears that control the planetary motion of the rotor, sliding noise of the apex seal, and noise of chains and gears driving the auxiliary equipment. However, the magnitude of these noises propagated outward is much smaller than those of the auxiliary equipment installed outside the engine.

(3) INTAKE-EXHAUST NOISE

Intake-exhaust noise is caused by pulsations and turbulence in the flows of intake mixture and exhaust gas.

As to intake noise, the rotary engine with an ordinary side intake port system has no distinct difference from the reciprocating engine.

Exhaust noise shows a high level at the instant when the exhaust port is opened, being affected by its opening characteristics.

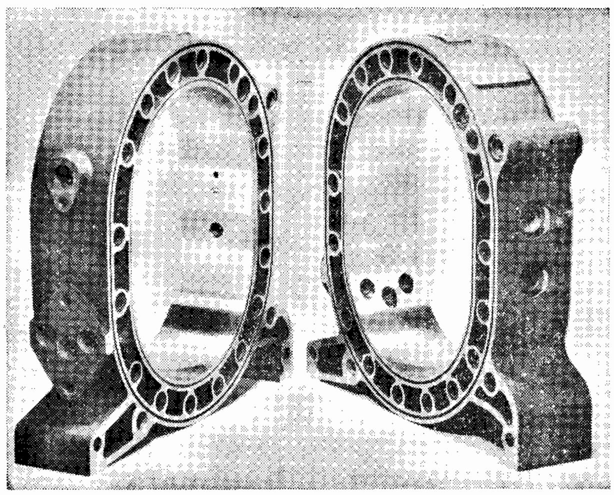

Fig. 4.17 Honey-comb port in rotor housing

In the rotary engine with a peripheral exhaust port system, the exhaust sound at port is high due to rapid opening of the exhaust port. Some rotary engines are provided with an exhaust port made in a honey-comb shape, as shown in **Fig. 4.17**, to lower the exhaust sound at the port.

4.3 COMBUSTION CHARACTERISTICS

In the rotary engine, the working chamber is divided into leading and trailing sides by the minor axis in the vicinity of the top dead center in the compression-stroke. The trailing side is compressed, while the leading side is expanded. This will cause squish flow from the trailing side to the leading side for a long time. This strong squish flow will raise the burning speed. While, the working chamber of flat and high surface/volume ratio will result in lower burning speed. These two factors will compete with each other to bring about the combustion characteristics unique to the rotary engine.

4.3.1 FLAME PROPAGATION

Fig. 4.18 shows the results of investigation on flame propagation using several ion-gaps buried in the housing.

The squish flow carries the flame toward the leading side, causing much faster flame propagation to the leading side than to the trailing side. In the trailing side, the working chamber is further flattened after the top dead center in the combustion-stroke to strengthen the quenching action due to cooling of the wall, which further retards the flame propagation.

The flame propagation speed in the width direction of the rotor is also faster on the leading side than on the trailing side.

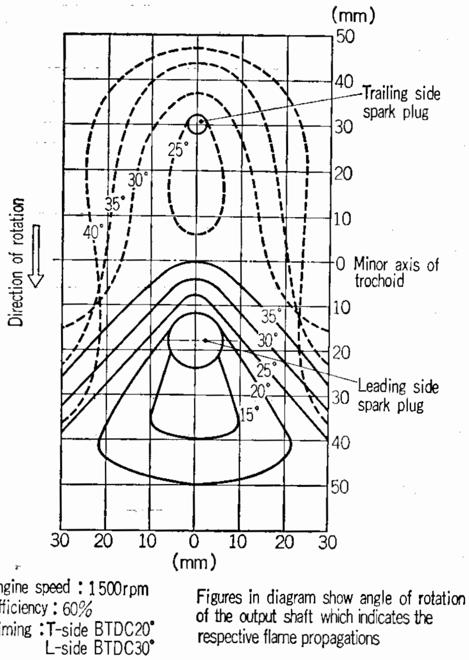

Fig. 4.18 Flame propagation of rotary engine

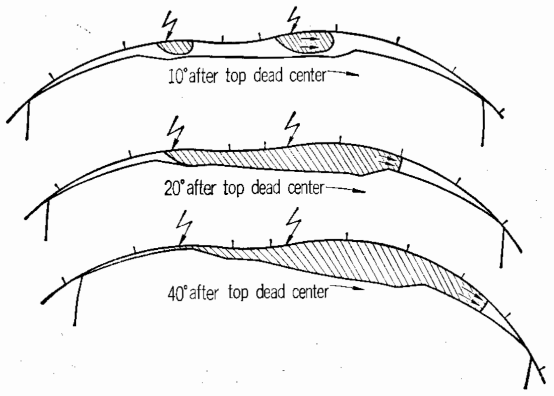

Fig. 4.19 Mode of flame propagation

In the rotary engine, the two-spark plug system is generally used, with the minor axis in between the two plugs, to raise the burning speed.

Fig. 4.19 shows the relative positions of the flame front, the spark plugs and the working chamber estimated from the results of measurement by ion-gaps and flame photographs.

4.3.2 IGNITABILITY IN LIGHT-LOAD OPERATION

The charging efficiency, if lowered, will raise the rate of inclusion of burned gas in the fresh mixture sucked, resulting in poor ignitability.

Fig. 4.20 shows an analysis of causes and the degree of effects of burned gas included in the fresh mixture.

Especially, in decelerating operation, the engine speed will be high and the throttle valve opening small, which will lower the charging efficiency and increase the rate of inclusion of burned gas in the fresh mixture causing a higher rate of misfiring.

During decelerating operation at high engine speed continuous misfiring will occur due to excessively low charging efficiency. During decelerating operation at low engine speed, an ignition-misfire cycle unique to the rotary engine, called the 3:3 ignition-misfire cycle may occur.

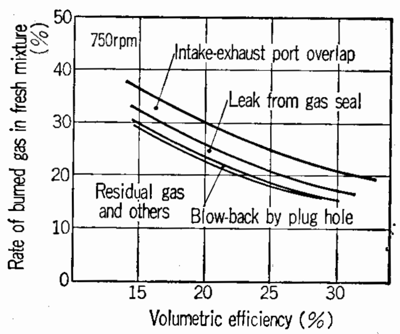

Fig. 4.20 Main analysis of inclusion of burned gas

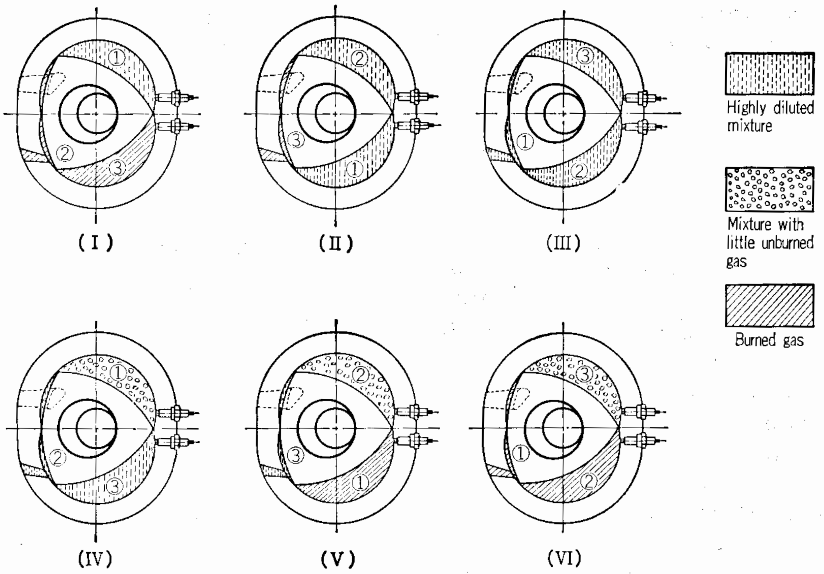

Fig. 4.21 Mechanism of generating 3 : 3 ignition-misfire cycle

Fig. 4.21 shows the mechanism of the 3:3 ignition-misfire cycle.

Misfire will be caused when the rate of burned gas in the working chamber exceeds the limit for ignition. Once misfired, no burned gas is generated. Unburned mixture will be carried, as it is, into the next intake-stroke to be ignited in the combustion-stroke. Then, the ignited working chamber will be misfired at the next combustion stroke due to high rate of burned gas in the mixture sucked in at the intake-stroke.

Such a process of misfiring and ignition occurs in each of the three working chamber. Thus, three consecutive misfires and ignitions will result, alternately. In this case, a torque fluctuation occurring in very low frequency will cause the body to shake back and forth (car bucking) together with the resonance of the driving system.

To prevent this phenomena, the effective measures taken are : a throttle opener or a dash pot used to raise the charging efficiency at the time of deceleration, or a shutter valve used to concentrate the mixture into one side of the working chamber, etc.

4.3.3 ABNORMAL COMBUSTION

(1) KNOCKING

At the time of high-load low-speed operation, the mixture in the end of the working chamber is compressed and self-ignites before the normal flame reaches, causing abnormal noise. This phenomenon is called "knocking".

The rotary engine has disadvantages to "knocking", such as a longer distance of flame propagation because of its flat working chamber, much delay in flame propagation to the trailing side, the trailing side of the working chamber being compressed in the course from the compression-stroke to the combustion-stroke.

On the other hand, advantages to "knocking" are: the mixture is apt to be cooled because of the large

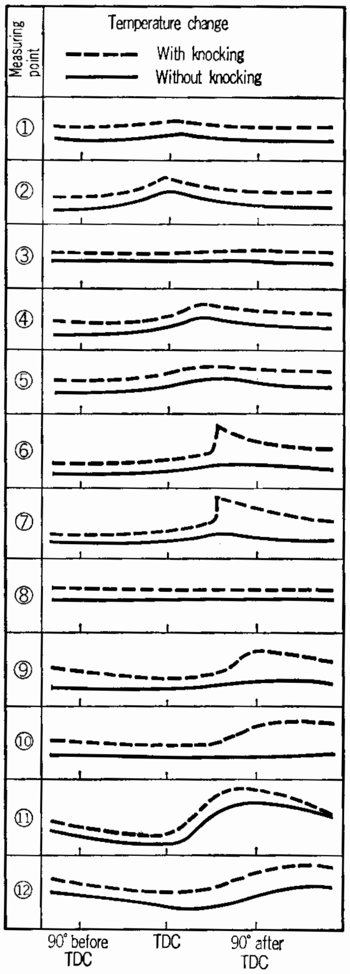

Fig. 4.22 Temperature change at each measuring point

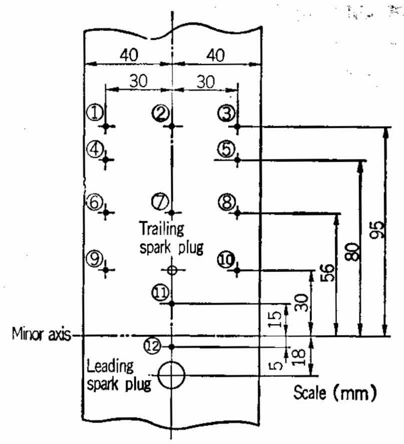

Fig. 4.23 Locations of instantaneous thermometer

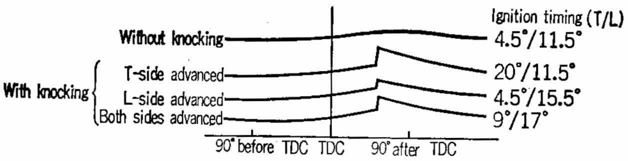

Fig. 4.24 Temperature change at measuring point

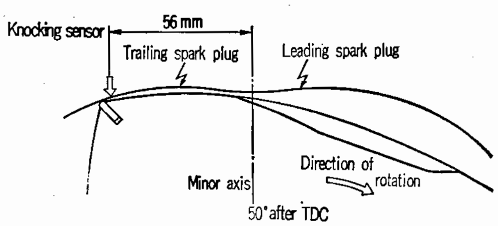

Fig. 4.25 Positions of rotor and knocking sensor

surface to volume ratio of the working chamber, the fuel particles concentrated in the trailing side as the rotor turns to form rich mixture, etc.

Using several instantaneous thermometers buried in the rotor housing, the temperature changes when "knocking" occurred were measured, as shown in **Fig.** 4. 22.

Fig. 4. 23 shows the locations of the instantaneous thermometers buried in the housing.

"Knocking" will not occur in each cycle. Its position will also vary for each cycle.

Fig. 4. 24 shows the temperature change at the measuring point ⑥ by changing the ignition timing to force "knocking" to occur.

The angle of rotation of the output shaft at the time of "knocking" is ATDC $45°\sim50°$ independent of the ignition timing.

From this, it is clear that the relative position of the rotor to the rotor housing is directly related to the occurrence of "knocking".

Fig. 4. 25 shows the position of the rotor at the angle of the output shaft of ATDC 50° and the position of the knocking sensor.

"Knocking" occurs at the end of the trailing side of the rotor.

"Knocking" occurring in the reciprocating engine may melt the exhaust valves leading to serious damage to the engine. In the rotary engine, on the contrary, no damage to the engine will be caused because there is no local temperature rise.

(2) PREIGNITION

In high-load high-speed operation, much heat is generated in the working chamber. Any part not cooled in the working chamber will be subject to a local high temperature rise. The mixture will be ignited before the normal ignition time is reached. This phenomenon is called "preignition".

In the rotary engine, the only source of "preignition" is the spark plug insulator that will be most heated.

The trailing spark plug, if preignited, will cause a rapid pressure rise in the working chamber. The apex seal will be loosened from the inner surface of the rotor housing causing a great amount of gas leak, and output will be greatly reduced.

The reason for this is that the early contact of the trailing spark plug with the fresh mixture makes "preignition" occur much earlier than normal ignition time.

The delayed contact of the leading spark plug with the fresh mixture will not greatly affect engine performance because of delayed "preignition".

Once "preignition" occurs, the spark plug will be abnormally heated damaging its insulator. The broken pieces may enter the engine and damage the apex seal. Therefore, in no case, should even the leading spark plug preignite.

The spark plug of the rotary engine is exposed to high temperatures for a longer time than that of the reciprocating engine. Generally, a high heat value (cold type) spark plug is used to prevent "preignition".

4.4 EFFECT OF INTAKE-EXHAUST SYSTEM

The effect of the specification for the intake-exhaust system on the engine performance is basically the same for both rotary and reciprocating engines. The specification requires raising the charging efficiency from the viewpoint of output performance at W. O. T. and minimizing the rate of burned gas in the fresh mixture from the viewpoint of light-load ignitability.

In the rotary engine, the intake-exhaust ports are automatically opened by the rotor in rotation. The port timing is determined by the location and configuration of the ports in the housing. The limiting conditions for the port timing and the opening area vary with the different port systems. Therefore, the port system and its optimum timing are selected according to the characteristics required for the engine.

4.4.1 INTAKE-EXHAUST PORT SYSTEM

There are two intake-exhaust port systems for

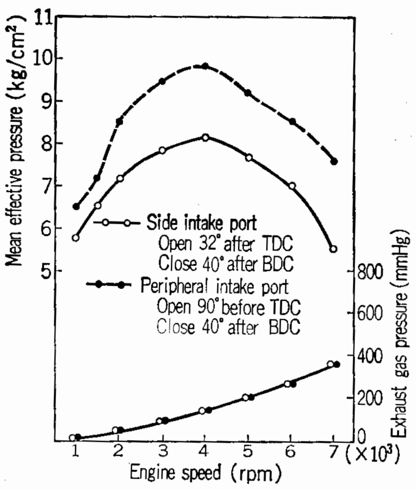

Fig. 4.26 Comparison of W.O.T. performance

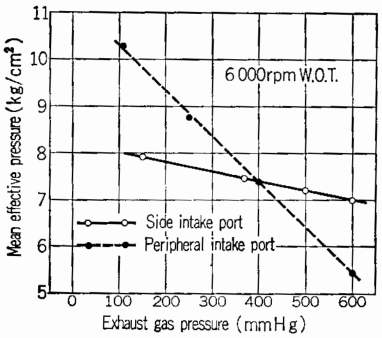

Fig. 4.27 Effect of exhaust pressure on W.O.T. performance

the rotary engine, the peripheral port system with ports in the rotor housing and the side port system with ports in the side housing, as described in 3.6.1.

For the exhaust port, the peripheral port system is generally used. The side port system has a detrimental effect on the side seal and oil seal when a great amount of high temperature exhaust gas enters the rotor side space.

For the intake port, both peripheral and side intake systems are used taking advantage of their respective characteristics.

Fig. 4.26 shows a comparison of performance at W.O.T. between the peripheral and side intake systems.

The peripheral intake system shows a higher charging efficiency and output because of the longer open-time of the port and smaller resistance in intake in the same direction as the rotor revolution.

However, in the peripheral intake system, the longer overlap-time between the intake and exhaust ports will increase the amount of burned gas being carried into the intake-stroke, if the exhaust pressure becomes high, greatly reducing the charging efficiency. To make use of its characteristic of obtaining a high output, it is especially important to combine it with an exhaust system having small gas flow resistance.

Fig. 4.27 shows a comparison between the peripheral and side intake systems as to the effect of exhaust pressure on the performance at W.O.T.

The peripheral intake system develops an excellent output performance in heavy load operation. In light-load operation, it is liable to be unstable due to a high rate of burned gas in the fresh mixture. This is due mostly to the longer overlap-time between the intake and exhaust ports which has been a factor in preventing the peripheral intake system from being put into practical use.

On the other hand, the side intake system has a shorter intake-exhaust port overlap-time, reducing the rate of burned gas in the fresh mixture and securing excellent stable ignitability in light-load operation. Further, the direction of the mixture intake different to that of the rotor revolution facilitates the generation of swirl in the intake-stroke for better vaporization and atomization of fuel.

As described above, the side intake system can provide a balanced output performance and excellent driveability over a wide range of operating condition and is generally used for automobile engines.

4.4.2 PORT TIMING

Ideal port timing is required to secure sufficient open-time for high performance at W.O.T., and to minimize the overlap-time between the intake and exhaust ports for driveability at light-load.

The side intake port system, if used, has little freedom in determining the position of the outside profile that governs the opening time, because it is necessary to prevent the side seal and corner seal from dropping into the port. Therefore, it is especially important to note that the output characteristics of the engine depends on the closing time selected.

Fig. 4.28 shows the effect on performance at W. O. T. of the closing time of the side intake port.

In the high-speed region, the retarded closing time of the intake port will increase the charging efficiency and improve output because of the geometrical effect of longer intake time and greater opening area of the port.

While, in the low-speed region, the intake-time is sufficiently long. Excessively retarded timing for closing the intake-port will force the sucked fresh mixture back into the intake port reducing charging efficiency.

The timing for opening the exhaust-port cannot be excessively advanced due to exhaust loss. Its closing time is determined to secure sufficient open-time and-area.

Fig. 4.29 shows a comparison of performance at W.O.T. of the peripheral exhaust port at different closing times.

The retarded exhaust port closing time can improve the performance at W.O.T., especially in the high-speed region, as shown.

However, the retarded exhaust port closing time

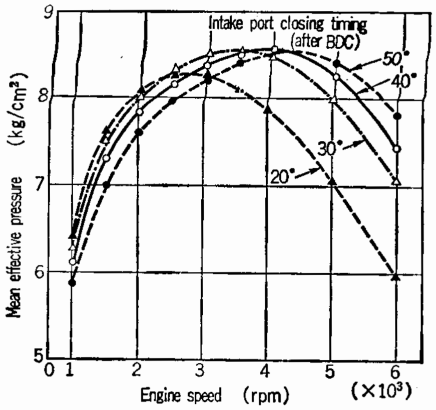

Fig. 4.28 Comparison of W.O.T. performance

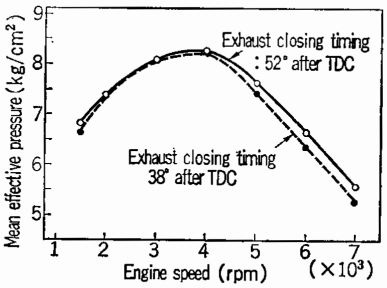

Fig. 4.29 Comparison of W.O.T. performance

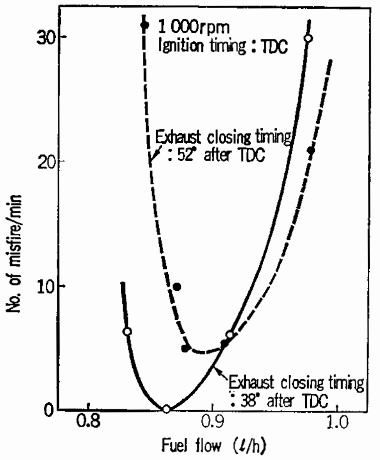

Fig. 4.30 Comparison of stability of idling

will result in a longer overlapping time with the intake port leading to instability of combustion at the time of light-load operation. Therefore, the exhaust port closing time must be selected to satisfy light-load driveability.

Fig. 4.30 shows the frequency of misfires in no-load operation with different exhaust port closing times.

In order to improve the stable ignitability at the time of light-load operation, it is a general practice to replace the burned gas to be carried into the intake-stroke with air by jetting air at the exhaust port.

This improved stable ignitability at the time of light-load operation, as shown in **Fig. 4.31**, further allows the exhaust port closing timing to be retarded for higher performance at W.O.T.

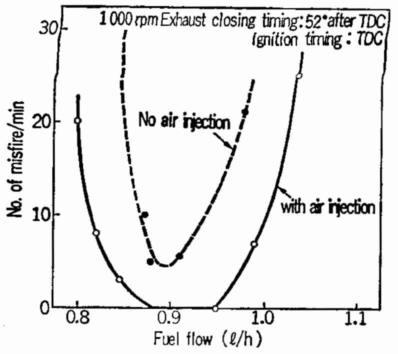

Fig. 4.31 Effect of air jet into exhaust port

4.5 EFFECT OF ROTOR RECESS

The rotor recess provided in the rotor surface has a great effect on the rotary engine. Its location, configuration and volume are important factors, which greatly affect the combustion speed of the rotary engine, govern the intensity and generating time of squish flow in the vicinity of top dead center in the compression stroke, and affect the quantity of mixture existing in the direction of flame propagation.

The higher combustion speed is preferable, in principle, from the viewpoints of output performance and thermal efficiency. On the other hand, however, if it is made excessively higher, problems such as misfiring in light-load operation, and knocking in heavy-load operation, etc. will occur.

For automobile engines to be operated over a wide range of loads and revolutions, the rotor recess is selected for a maximum possible combustion speed without disturbing driveability.

4.5.1 LOCATION AND CONFIGURATION OF ROTOR RECESS

(1) LOCATION OF ROTOR RECESS

The location of the rotor recess closer to the leading side can provide a higher burning speed when the configuration and volume of the rotor recess are the same.

This is because of the geometrical effect of a greater amount of mixture existing in the leading side where flame propagation is faster and the squish flow more intensive.

The squish flow will become more intensive when a greater pressure difference is caused between the trailing side in higher pressure under high compression and the leading side in relatively reduced pressure.

(2) CONFIGURATION OF ROTOR RECESS

When the volume and location of the rotor recess are the same, the squish flow will be more intensive for the rotor with a shallower recess near the minor axis

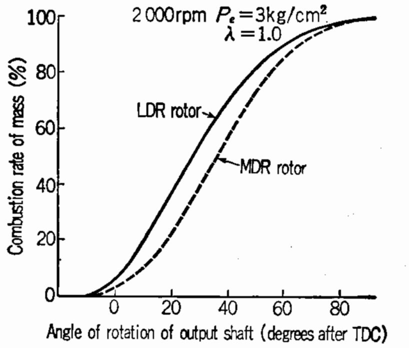

Fig. 4.32 Comparison of combustion speed

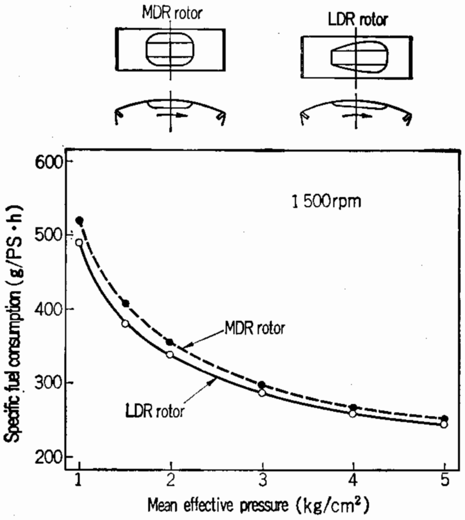

Fig. 4.33 Comparison of specific fuel consumption

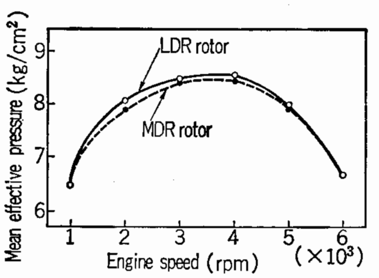

Fig. 4.34 Comparison of W.O.T. performance

when the rotor is in the vicinity of top dead center.

This is because the passage area becomes smaller and the pressure difference between the trailing and leading sides becomes greater.

The cross-section of the rotor recess in an adequate depth to the flame surface profile is preferred to that in smaller depth and wider width to provide a greater flame surface area.

Further, for efficient burning inside the rotor recess, a configuration that provides a smooth squish flow is required.

The above concept is called the LDR (Leading Deep Recess) rotor which provides a rotor recess that is deeper on the leading side.

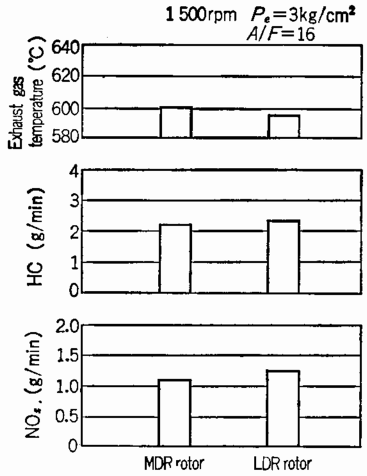

Fig. 4.35 Comparison of emission of exhaust gas

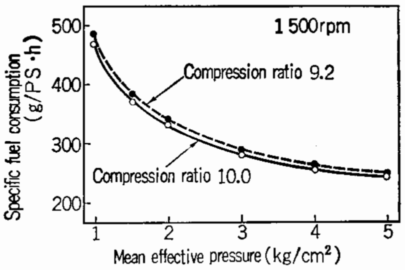

Fig. 4.36 Comparison of specific fuel consumption rate

Fig. 4.32 shows a comparison of burning speeds between the LDR rotor and the MDR (Medium Deep Recess) rotor that has a volumetric center on its center.

Fig. 4.33 through **4.35** show comparisons of specific fuel consumption, performance at W.O.T. and emission of exhaust gas.

The LDR rotor increases the amount of NO_x emission due to higher burning speed, while the amount of HC emission is not related to the location and configuration of the rotor recess as NO_x because of its dependence on a relatively later stage of combustion after the working chamber volume has been expanded.

4.5.2 VOLUME OF ROTOR RECESS

The compression ratio depends mainly on the volume of the rotor recess. A smaller volume of the rotor recess increases the compression ratio improving the theoretical thermal efficiency. An excessively small volume will make the surface/volume ratio greater, which could disturb flame development. In general, the volume of the rotor recess is selected within a range of the compression ratio of 8～10.

Fig. 4.36 through **38** show the effect of the compression ratio on fuel consumption performance, performance at W.O.T. and exhaust emission.

As shown, the smaller volume of the rotor recess improves fuel consumption and performance at W.O.T., but, at the same time, increases the amount of emission of NO_x and HC.

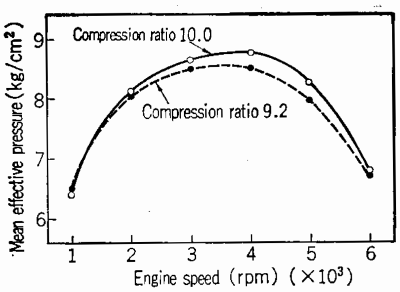

Fig. 4. 37 Comparison of W.O.T. performance

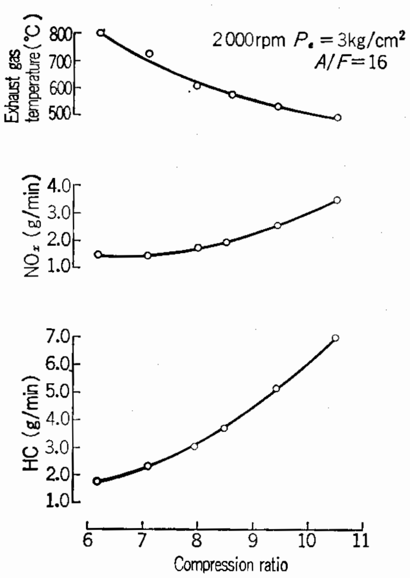

Fig. 4. 38 Compression ratio and exhaust emission

4.5.3 PROBLEMS INVOLVED IN INCREASED BURNING SPEED

(1) KNOCKING

The rotor recess of high compression ratio or of squish flow intensified type will raise the working gas pressure in the trailing side of the combustion chamber at heavy-load. As the burning speed is increased, "knocking" is liable to occur.

Therefore, the rotor recess must be selected so that the working gas pressure in the trailing side will not be raised more than necessary.

Fig. 4. 39 shows the research octane number with the varied compression ratios.

(2) DETERIORATION OF LIGHT-LOAD DRIVEABILITY

A higher combustion speed improves the thermal efficiency, but reduces the charging efficiency in light-load operation such as idling. The rate of burned gas in the fresh mixture, that is, the rate of inactive gas in the mixture, will increase thus decreasing ignitability.

Fig. 4. 40 shows a comparison of the range of fuel flow for stable idling between the LDR and MDR rotors.

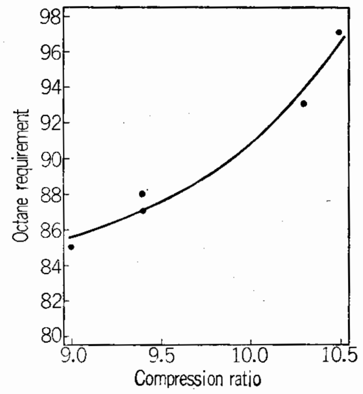

Fig. 4. 39 Compression ratio and research octane number

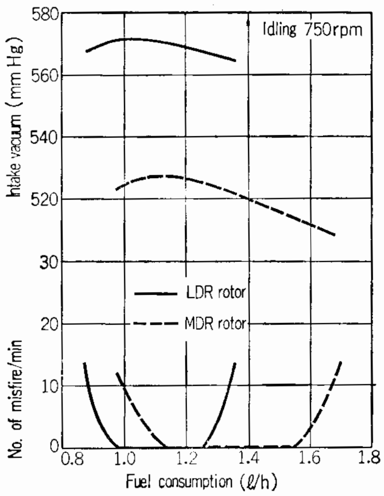

Fig. 4. 40 Comparison of idling stability

4.6 EFFECT OF GAS SEAL

The rotary engine is constructed to maintain gastightness of each working chamber by means of gas seals. Therefore, the gas seals have a great effect on fuel consumption performance, performance at W.O.T. and on exhaust emissions of the engine.

Many efforts have been made for their improvement. The gastightness of the seals has been tremendously improved in comparison with those in the early stage of development of the engine.

Such improvement has been brought about mainly by development of sealing material, improvement of seal configuration, use of smaller tolerances in manufacture and development of surface treatment techniques.

4.6.1 IMPROVEMENT OF GAS SEAL

(1) APEX SEAL

The apex seal functions as described in 3.5. For further improvement of sealing performance of the apex seal, it is required to reduce the clearance at the split

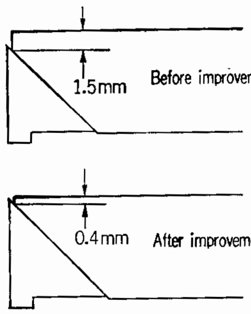

Fig. 4.41 Improved apex seal

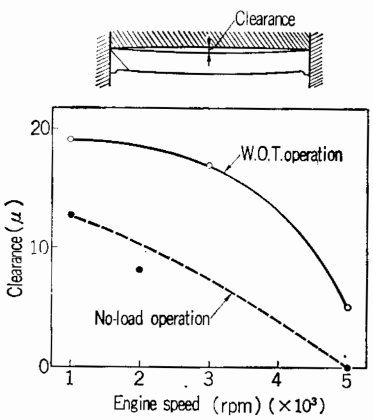

Fig. 4.42 Clearance between apex seal and rotor housing

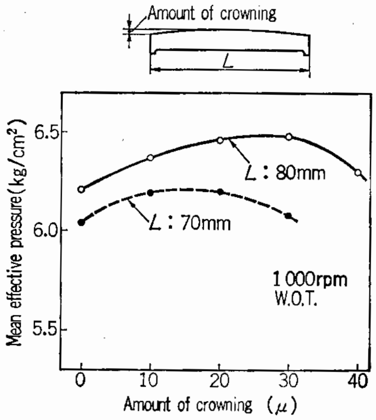

Fig. 4.43 Effect of crowning

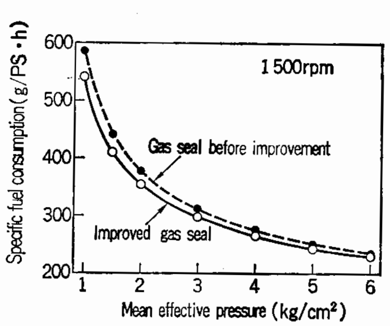

Fig. 4.44 Comparison of fuel consumption rate

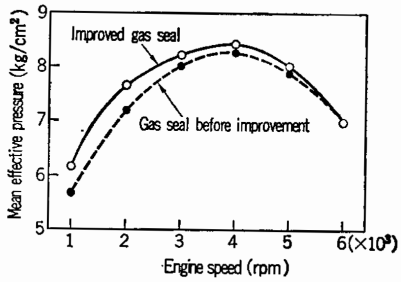

Fig. 4.45 Comparison of W.O.T. performance

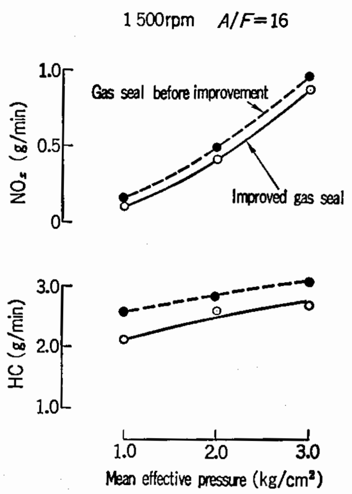

Fig. 4.46 Comparison of exhaust emissions

of the apex seal and that between the apex seal and the housing due to thermal deformation in running condition.

Fig. 4.41 shows the reduced clearance at the apex seal split.

Fig. 4.42 shows the measured clearance caused due to thermal deformation during operation.

This clearance can be reduced by crowning the apex seal.

Fig. 4.43 shows the effect of crowning on the mean effective pressure at W.O.T. 1,000 rpm.

It is clear that the optimum selected value for the crowning will improve the gastightness function.

(2) SIDE SEAL

The side seal raises few problems of gastightness, as described in 3.5.

Its clearance with the corner seal cannot be reduced by the spring pressure or gas pressure, but by adjusting the length of the side seal to that of its groove.

(3) CORNER SEAL

As described in 3.5, the corner seal can be improved for gastightness by cutting its center to make it an expansion seal.

4.6.2 EFFECT OF IMPROVED GAS SEAL

Improved gastightness will reduce the amount of mixture leaking from the working chamber and increase the rate of effective fuel to be burned as well as reducing the loss of working gas by leaking. Also, the charging efficiency in operation at W.O.T. will be increased as the exhaust gas blowing into the working chamber in the intake-stroke is reduced. Such effects will then improve the fuel consumption and the performance at W.O.T.

The amount of HC emission in exhaust gas will be reduced by the reduced unburned gas leaking into the working chamber in the exhaust-stroke and by reduced burned gas leaking into the working chamber in intake-stroke which will improve combustion.

Fig. 4.44 through **4.46** show the effects of improved gastightness performance on fuel consumption, performance at W.O.T. and exhaust emission.

The amount of NO_x emission is decreased. This is because of the reduced amount of mixture to develop the same output along with the development of gastightness.

4.7 EFFECT OF IGNITION SYSTEM

The rotary engine is characterized by the squish flow that forces the flame to propagate faster in the leading direction and slower in the trailing direction.

Therefore, the combustion speed in the entire working chamber depends greatly on the starting position of the flame propagation, hence the spark plug arrangement becomes important.

As described in 4.5.3, however, the higher combustion speed will reduce ignitability in light-load operation.

To obtain higher ignitability is also an important matter.

4.7.1 IGNITION SPARK CHARACTERISTICS

The mixture is ignited when the mixture in the spark gap, electrically energized sufficiently to resist the heat loss in its vicinity, is heated and ionized exceeding its ignition point.

Figs. 4.47 and **4.48** show the relation between the duration of secondary current and arc and the range of fuel flow capable of stable idling.

Ignitability is improved as the secondary current is increased up to approximately 120mA.

The duration of arc is not related to ignitability,

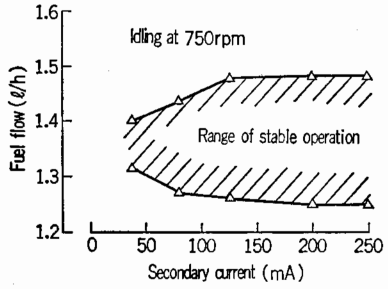

Fig. 4.47 Secondary current and idling stability

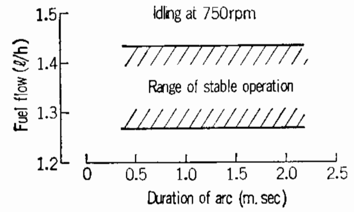

Fig. 4.48 Duration of arc and idling stability

as shown in **Fig. 4.48**. This is because, in low speed operation such as idling, the mixture in the vicinity of the spark plug is not exchanged in the duration of arc given in **Fig. 4.48** due to low flow speed of gas in the plug hole.

4.7.2 SPARK PLUG

As shown in **Fig. 4.49**, the greater spark gap of a spark plug can improve the ignitability because of the longer distance of spark to contact with the mixture and the smaller cooling effect of the electrodes on the flame core.

The configuration of electrodes greatly affect the ignitability due to their cooling effect on the flame core formed in the early stage of ignition and to the remaining burned gas in their vicinity to be scavenged.

Fig. 4.50 shows the effect of the configuration of the spark plug on the ignition limit at light-load operation.

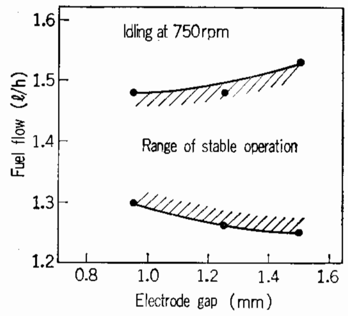

Fig. 4.49 Electrode gap and idling stability

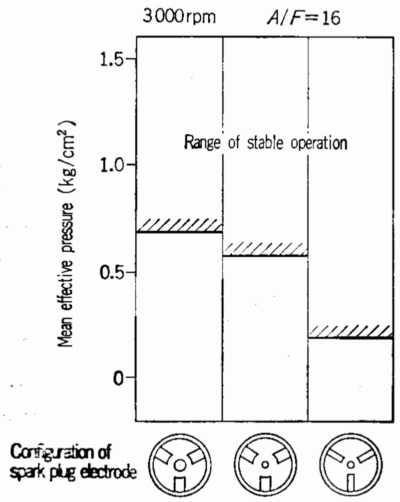

Fig. 4.50 Comparison of light-load ignitability

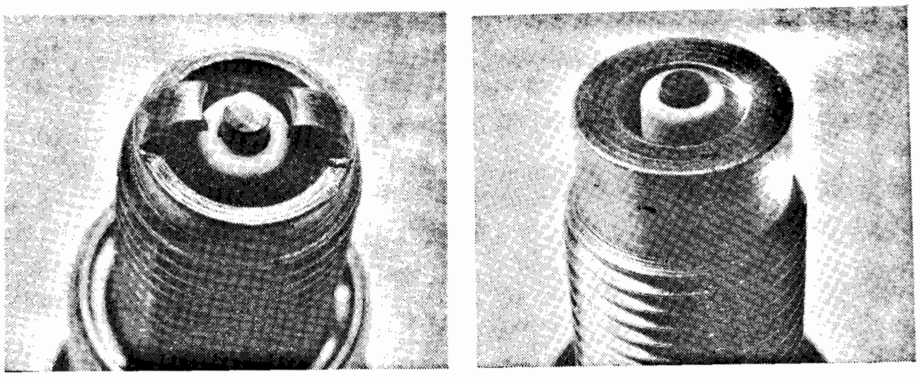

Fig. 4.51 Spark plug

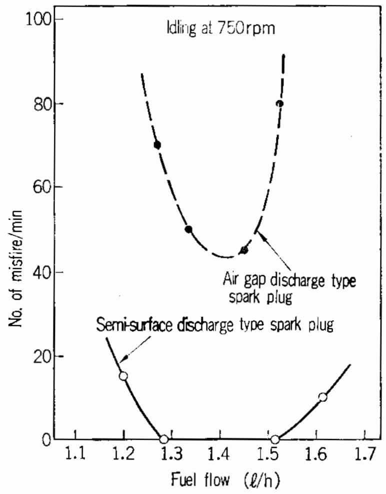

Fig. 4.52 Comparison of idling stability

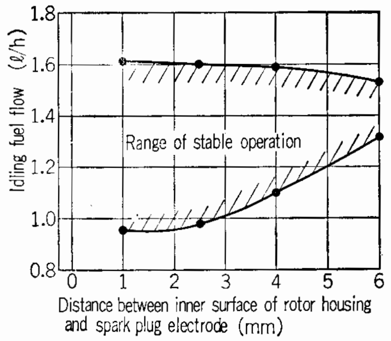

Fig. 4.53 Distance between inner surface of rotor housing and electrode and ignitability

The smaller diameter of the center and ground electrodes will improve ignitability due to reduction in their cooling effect.

Fig. 4.51 shows a space discharge spark plug and a semi-surface spark plug partially incorporating surface discharge.

Fig. 4.52 shows a comparison of ignitability between the space discharge and semi-surface discharge spark plugs during idling in a condition of high rate of burned gas in fresh mixture.

The improved ignitability of the semi-surface discharge spark plug may be considered due to its greater spark gap and to stable sparking because of its smaller equivalent reactance of surface discharge than that of space discharge.

4.7.3 DISTANCE FROM INNER SURFACE OF ROTOR HOUSING TO ELECTRODE

The spark plug electrode brought closer to the inner surface of the rotor housing improves the ignitability due to easier scavenging in the vicinity of the electrode and to reduced blow-back of burned gas from the working chamber in expansion-stroke to that in compression-stroke when the apex seal passes over the plug hole.

Fig. 4.53 shows the relation between the distance from the inner surface of the rotor housing to the electrode and the ignitability in idling operation.

4.7.4 LOCATION AND NUMBER OF SPARK PLUGS

A spark plug installed on the leading side with reference to the minor axis is called the leading spark plug and the other on the trailing side, the trailing spark plug.

Fig. 4.54 shows a comparison of the partial loading performance between the 2-spark plug system provided with the leading and trailing spark plugs and the single spark plug system provided with the leading spark plug only.

In the case of the single spark plug system, less emission of NO_x and HC is observed. On the other hand, deterioration of fuel consumption performance is noted.

The reason for this is that the use of the leading spark plug only causes much delayed combustion of mixture on the trailing side and reduces the general combustion speed.

In the rotary engine, the 2-spark plug system is generally used.

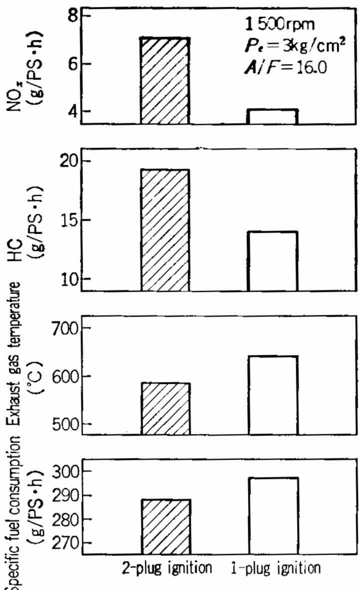

Fig. 4.54 Comparison of partial load performance

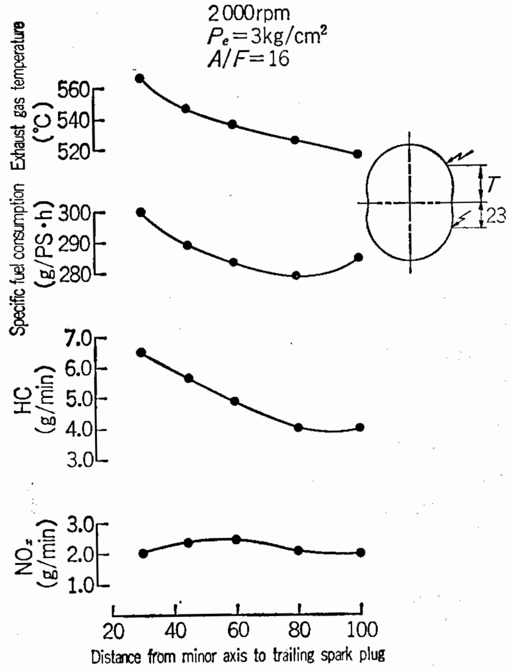

Fig. 4.55 Location of trailing spark plug and partial load performance

The effect of the locations of plugs in the 2-spark plug system on the engine performance was investigated and is shown in **Figs. 4.55** and **4.56.**

Fig. 4.55 shows the relation between the location of the trailing spark plug and the partial loading performance.

The spark plug installed away from the minor axis in the trailing direction will improve the fuel consumption and reduce HC emission.

The main cause for this is that combustion starts

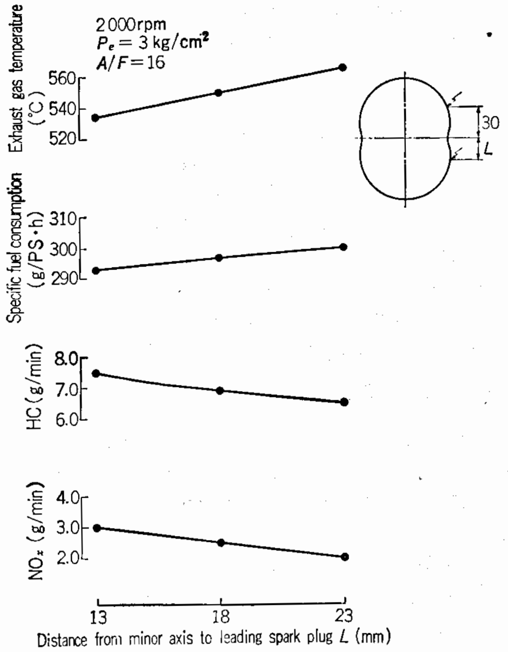

Fig. 4.56 Location of leading spark plug and partial load performance

in the trailing side from the point closer to its end to contribute to combustion in the trailing space where extinction of flame is liable to take place.

In operation at W.O.T., the trailing spark plug installed further away from the minor axis will increase blow-back from the working chamber in expansion-stroke into that in intake-stroke when the apex seal passes over the plug hole. The charging efficiency will be lowered to reduce the performance at W.O.T.

Thus, the trailing spark plug is installed in a location where an improved fuel consumption can be obtained with a minimum reduction in performance at W.O.T.

Fig. 4.56 shows the relation between the location of the leading spark plug and the partial loading performance.

The spark plug installed away from the minor axis in the leading direction will reduce emission of HC and NO_x, but deteriorate the fuel consumption. This is because the initiation of combustion in the less squishing area causes a slow combustion speed.

4.8 EXHAUST EMISSION CONTROL SYSTEM

The social requirement to protect the environment made no exception to motor vehicles, and, since the exhaust emission standards were enforced in 1968 in the United States ; the standards have been made severer each year. The same trend as such can be observed in various parts of the world including Japan, European countries, Australia, etc.

The fundamental concept of the exhaust emission control system for automobile engines meeting the exhaust emission standards lies in how to determine the principal specifications for a desired engine without sacrificing its performance, fuel economy, durability etc.

First, the characteristics of the engine must be clarified. The effect of the properties of exhaust emission, etc. and the characteristics of various parameters of the rotary engine have been described in the foregoing lines. For controlling the exhaust emission, there are two methods being used at present, the thermal reactor system and the catalyst system.

4.8.1 THERMAL REACTOR SYSTEM

The thermal reactor system has been most generally used for exhaust emission control of the rotary engine. Being effective against leaded fuel, this system is used on engines exported to Europe, Australia, etc., where leaded fuel is used.

Fig. 4.57 shows the thermal reactor system used in the rotary engine installed on the ECE RX-7.

The characteristics of the combination of the thermal reactor system with the rotary engine are as follows:

① Based on the exhaust emission properties of less NO_x and high HC, supply of the secondary air to the thermal reactor will put the reactor into a stable purification reaction of exhaust emission by the burning heat of HC.

② Continuous emission of exhaust gas from each exhaust port and relatively high temperature of the exhaust gas are advantageous to the oxidation reaction in the thermal reactor.

③ The thermal reactor can be installed in direct contact with the engine of such favorable construction leading to less heat loss and better purifying efficiency.

④ The fewer number of exhaust ports enables the thermal reactor to be designed compactly.

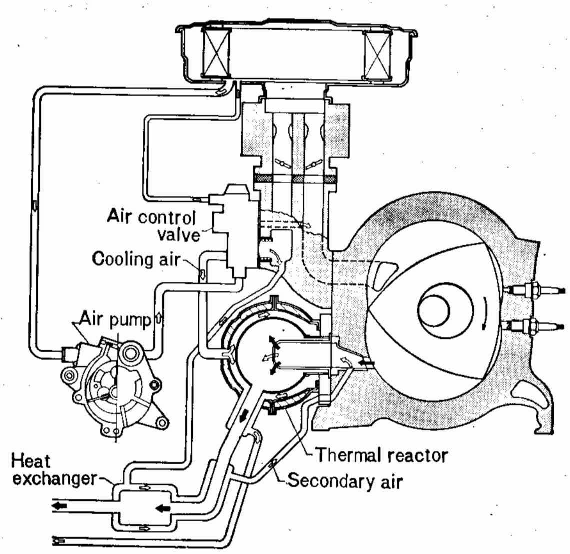

Fig. 4.57 Thermal reactor system

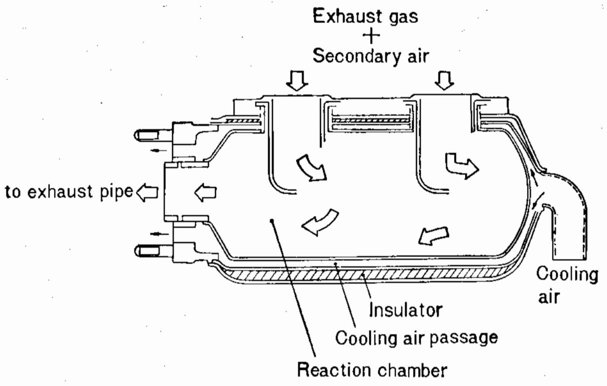

Fig. 4.58 Thermal reactor

⑤ The exhaust port of simple construction can be easily provided with a port insert effective for reducing heat loss of exhaust gas flowing into the thermal reactor.

⑥ The thermal reactor is installed on the bottom of the engine which is advantageous in preventing heat deterioration inside the engine compartment.

Thus, the thermal reactor system can be considered to be a really compatible emission control system with the rotary engine.

(1) THERMAL REACTOR

As well known, conditions for sufficient oxidation reaction of exhaust gas in the thermal reactor are: ① to keep the exhaust gas in the reactor as long as possible, ② to keep the exhaust gas temperature as high as possible, and ③ to mix the exhaust gas with the secondary air as much as possible.

Fig. 4.58 shows an example an the above conditions.

This thermal reactor is an exhaust gas reverse flow type consisting of three components, a reaction chamber, a cooling air passage and an insulator layer covering the outside.

An inlet pipe with a baffle plate is inserted at its inlet.

For durability, the following are to be noted: ① Corrosion by oxidation and creep from high temperature exhaust gas, ② Cracks and damage due to thermal fatigue and ③ Deformation of inner wall due to thermal and mechanical restraints and exhaust gas pressure.

To prevent the above, the thermal reactor is made of well selected material of high thermal strength and supported for free sliding allowing release of thermal stress.

Further, to prevent the thermal reactor wall from becoming extremely hot during high speed and heavy load conditions, provisions have been made to control its temperature, for example, by positively cooling the inner wall with an excessive amount of secondary air, as cooling air, flowing inside the thermal reactor.

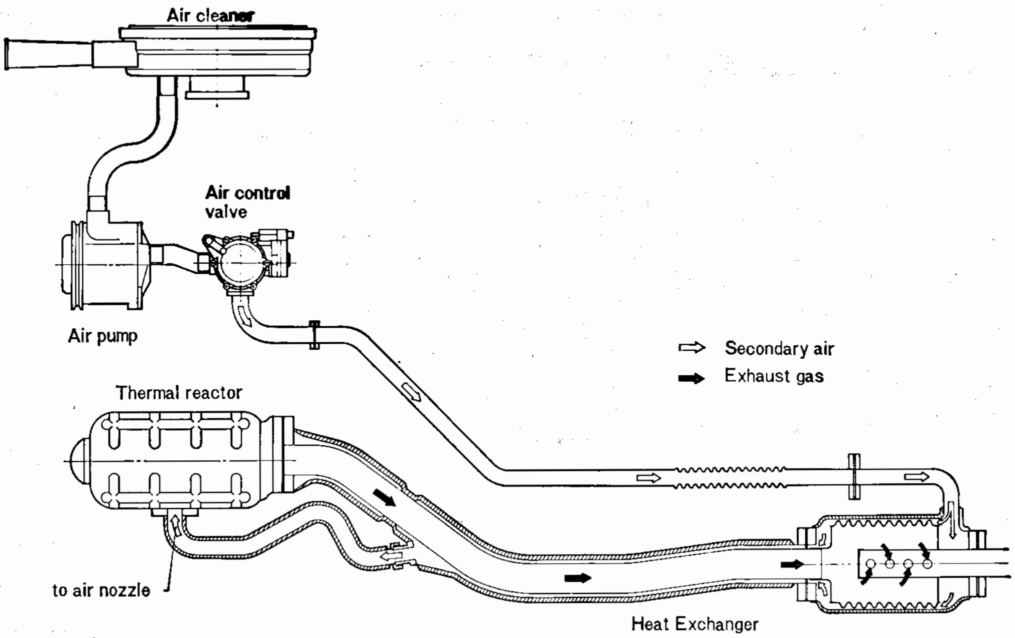

Fig. 4.59 Secondary air pre-heat

(2) HEAT EXCHANGER

Reactivity in the thermal reactor system is mostly governed by temperature. To keep a high exhaust gas temperature, a port insert is provided at the inlet of the thermal reactor and at the exhaust port of the rotor housing. Further, for better reactivity, the secondary air preheat system is effective.

This system consists of a heat exchanger and a secondary air passage covering the exhaust pipe, as shown in **Fig. 4.59.**

The secondary air from the air control valve enters into the heat exchanger to be heated by the exhaust gas. The heated air is jetted into the exhaust port insert from the air nozzle in the rotor housing through the secondary air passage covering the exhaust pipe (**Fig 4.57**). Such pre-heated secondary air greatly improves the reactivity of the thermal reactor.

Fig. 4.60 shows the thermal reaction limit from secondary air temperature.

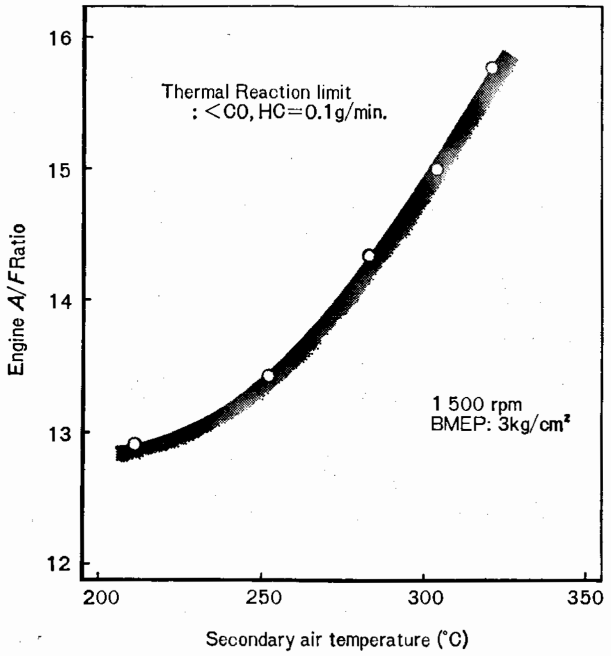

Fig. 4.60 Effect of secondary air temperature on engine A/F ratio for thermal reaction

4.8.2 CATALYST SYSTEM

The catalyst system has made possible the fuel economy improvement of the rotary engine to a great extent in place of the thermal reactor system. It is used in engines where unleaded fuel is available such as Japan, U.S.A., etc.

The thermal reactor system was first adopted as an exhaust emission control system for the rotary engine for reasons described in 4.8.1, and has contributed to fuel economy by improvement of the engine and the system itself.

The continuous improvement of the engine extended the lean limit of air-fuel mixture for engine operation, but was soon encountered with a limit for leaner

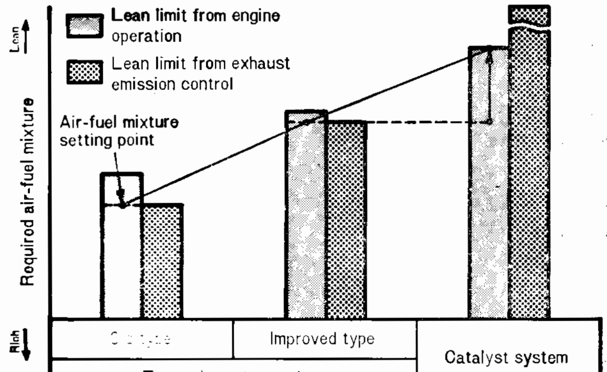

Fig. 4-61 Approach to leaner air-fuel mixture

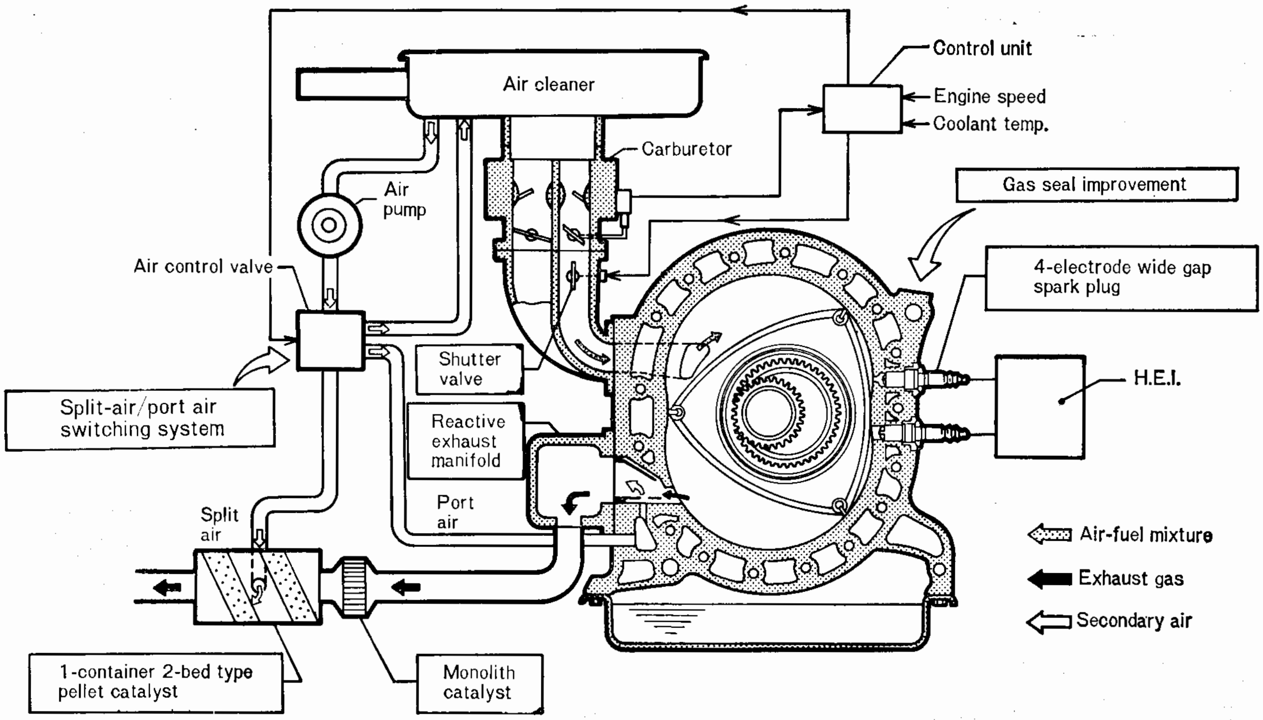

Fig. 4.62 Catalyst system

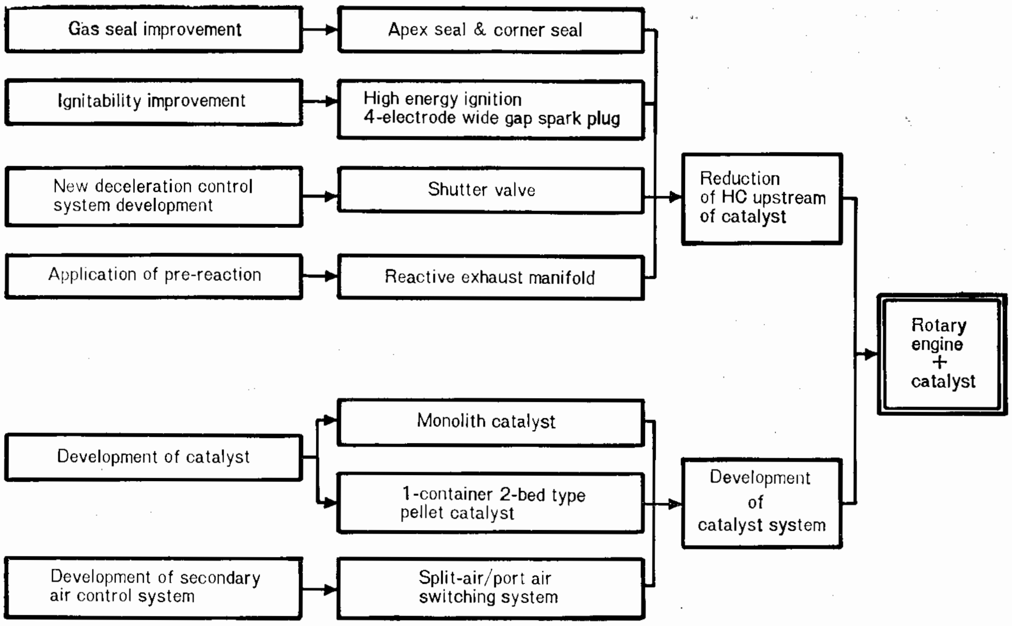

Fig. 4.63 Process of catalyst adoption

mixture by means of the lean limit from exhaust emission control. Then, the catalyst system was required for further improvement of fuel economy. **Fig. 4.61** shows an approach to leaner air-fuel mixture for rotary engines.

Since the catalyst reacts even at low temperatures, it is possible to make the mixture leaner. If it is used for the rotary engine, a great amount of HC must be purified, which will cause problems such as thermal deterioration, etc. of the catalyst, requiring the following: ① Development of HC reduction technique before reaching the catalyst to relieve its load. And ② Development of a catalyst system and a secondary air control system so that a high rate of purification can be kept.

The previous techniques have been improved and new techniques developed, as well as new catalysts far better than previous ones.

Fig. 4.62 shows the catalyst system used in the 1982 model RX-7 for the U.S. market.

Fig. 4.63 shows the process of catalyst system adoption in the rotary engine.

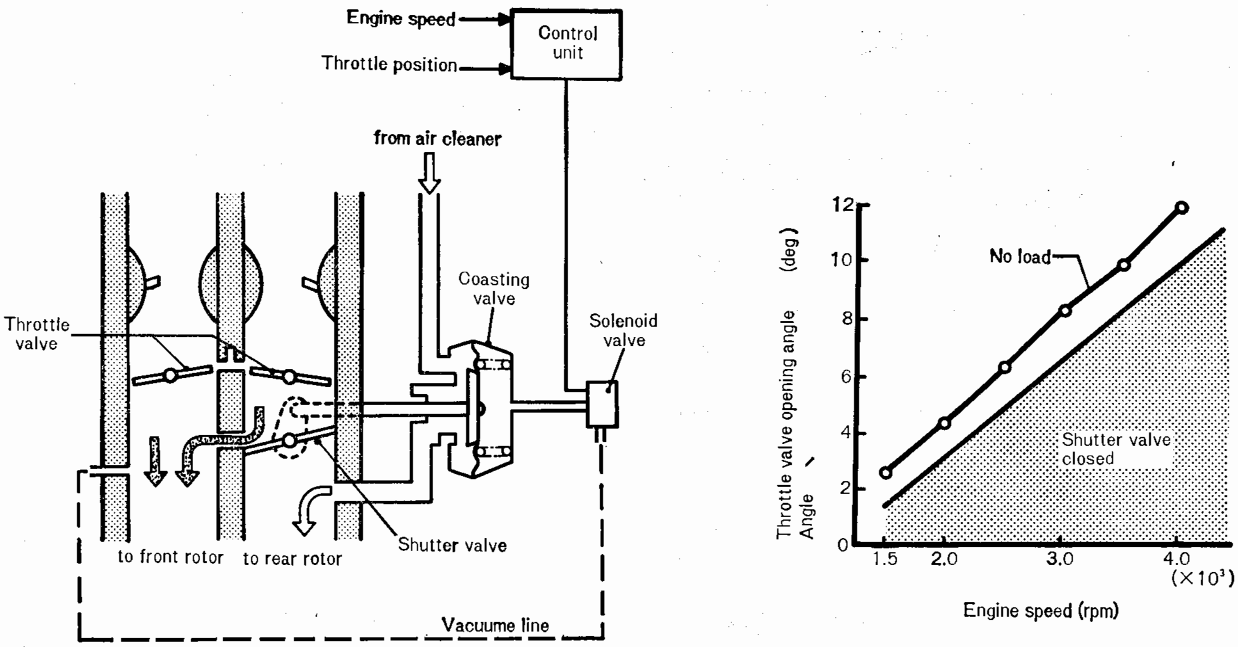

Fig. 4-64 Shutter valve system

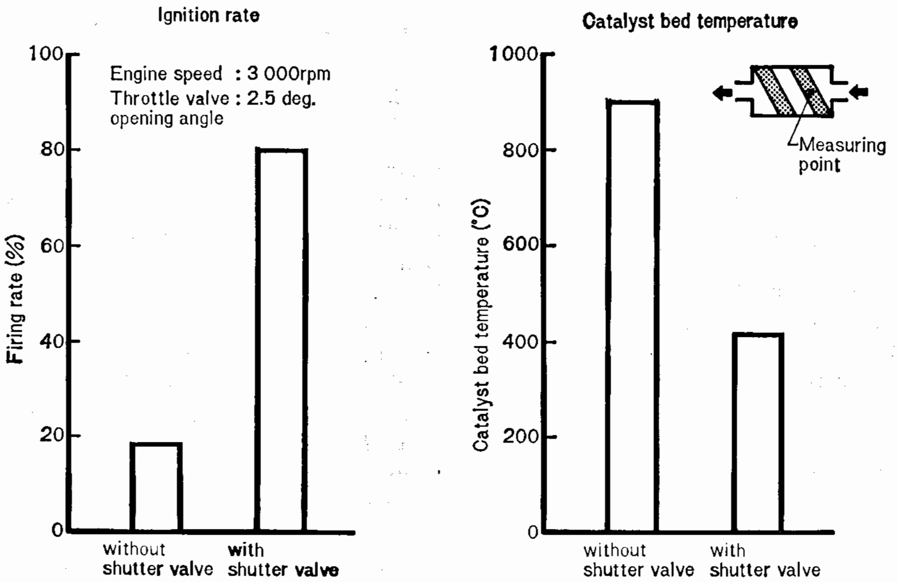

Fig. 4-65 Effects of shutter valve during deceleration

(1) SHUTTER VALVE

In the region of low charging efficiency of the engine during deceleration, the high rate o[illegible]tion gas will lower ignitability and cause misfiring. The engine torque will fluctuate and impair driveability. If unburned HC emissions are purified by the catalyst, temperatures will be rise causing heat deterior[illegible]. **Fig. 4.64** shows the shutter valve system that improves ignitability by increasing the charging efficiency during deceleration.

The shutter valve is installed beneath the rear side throttle valve of the inlet manifolds independent of the two working chambers. The valve is fully closed during deceleration. The mixture flows into the front side through a bypass hole above the valve. Thus, the charging efficiency and ignitability in the front side working chamber will be improved, and the catalyst temperature lowered. Simultaneously, air supplied to the rear side through the coasting valve directly connected to the shutter valve will keep the proper manifold boost to prevent the mixture from leaking.

Fig. 4.65 shows the effect of the shutter valve on ignitability and catalyst temperature.

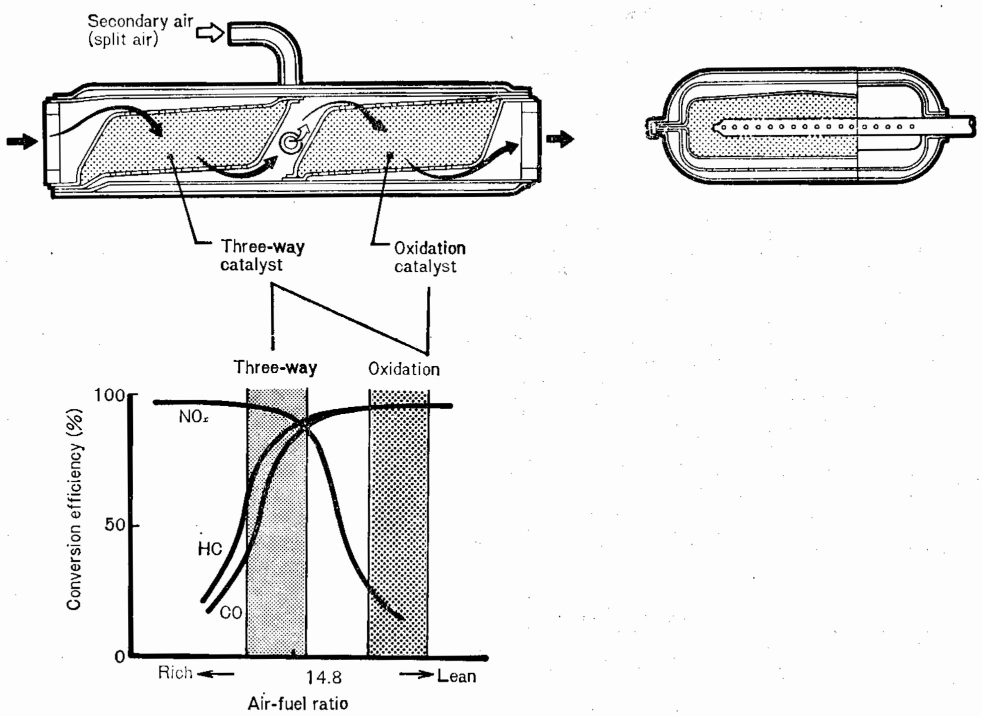

Fig. 4.66 Pellet type catalytic converter (1-container 2-bed type)

(2) CATALYTIC CONVERTER

The catalytic converter consists of a pre-converter and a main converter. The pre-converter plays a part of purification of exhaust gas. A monolith-type catalyst of small capacity is used to facilitate reaction in the main converter.

The main catalyst is incorporated with two beds, as shown in **Fig. 4. 66**, to efficiently purify HC, CO and NO_x in the exhaust gas. An air nozzle of a multi-hole type is provided between the two beds to supply secondary air. The main converter is a compact pellet converter of 1-container 2-bed type capable of obtaining a high purification rate, as a whole.

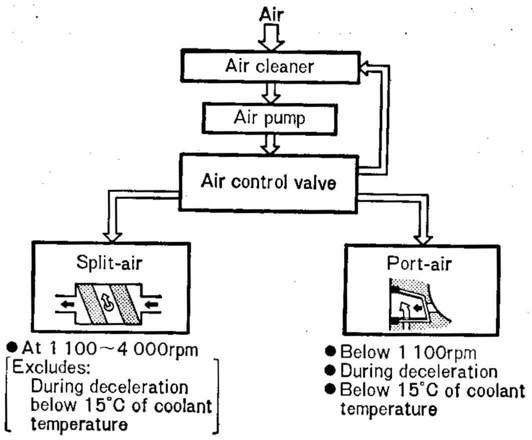

Fig. 4.67 Secondary air control system (point in/split air)

(3) SECONDARY AIR CONTROL SYSTEM

For efficient purification of exhaust gas, the air control valve is used to change over the secondary air between the port air injected into the exhaust port and the split air injected in between the two beds of the catalytic converter (**Fig. 4. 67**), according to the operating condition.

During low speed or deceleration operation and engine warming-up when the rate of HC and CO is relatively high, port air is supplied to mainly purify HC and CO with the entire catalyst as an oxidation catalyst.

During normal driving when the rate of NO_x is relatively high, the split air is supplied to reduce NO_x and purify a part of HC and CO with the monolith catalyst and the front pellet catalyst as a 3-way catalyst. The rear pellet catalyst mainly purifies HC and CO as an oxidation catalyst.

4.9 SIX PORT INDUCTION SYSTEM (RE 6PI)

In order to improve fuel economy, the selection of port timing for preventing misfiring and improving torque at low speed, review of the shape of passage for the mixture for improving vaporization and atomization of the mixture, and efforts to make the area of the intake port smaller have been performed, hitherto. However, there is a limitation in pursuing fuel economy due to the drawback of lowering the high speed performance; therefore, the only alternative was to seek a point where fuel economy and output

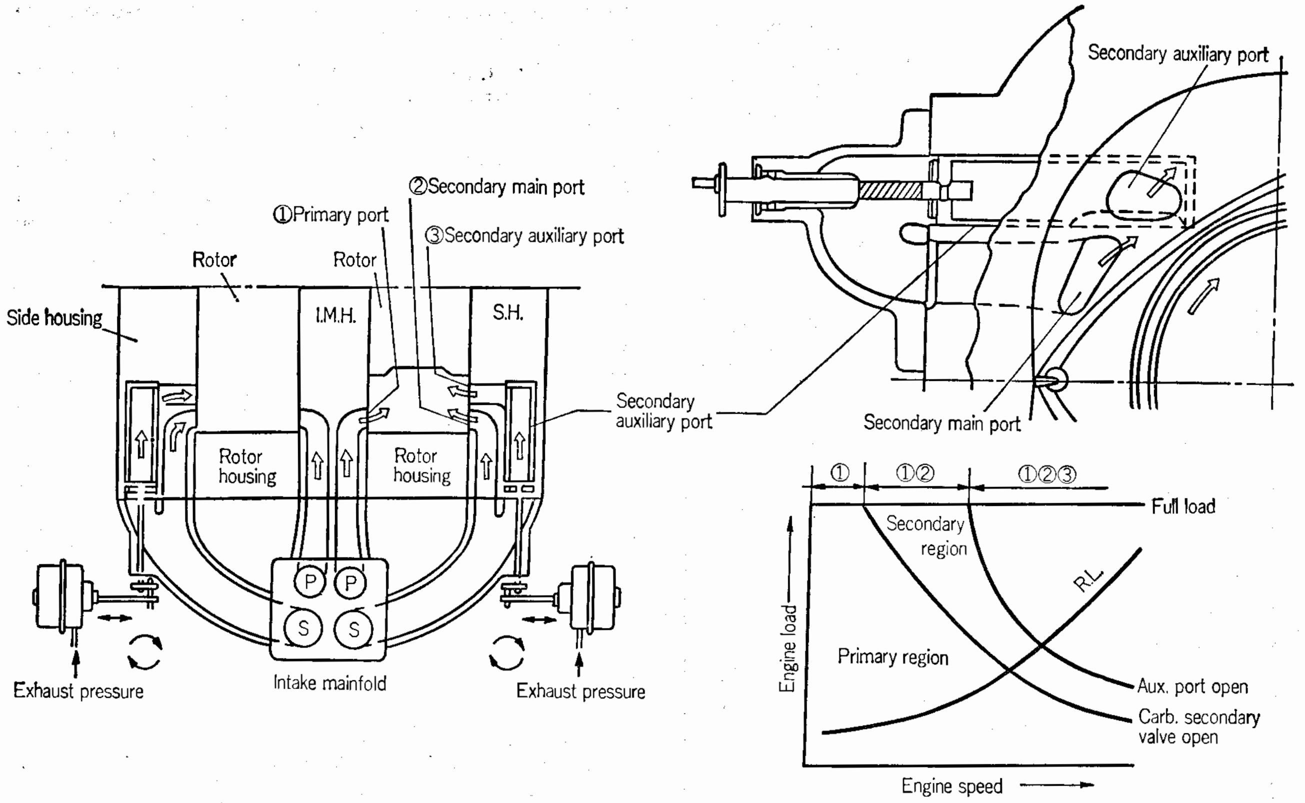

Fig. 4.68 Structure and operation of RE6PI

performance are well balanced.

Fig. 4.68 shows an intake system called the RE 6PI which enables fuel economy and high speed performance to be compatible by providing variable port timing, corresponding to valve timing for reciprocating engines.

4.9.1 STRUCTURE AND OPERATION

As for the intake ports, there are three ports for one rotor: the primary port, the secondary port, and the secondary auxiliary port. So, for a 2-rotor rotary engine, the intake system consists of 6 ports. The conditions of supply of mixture from each port changes in three steps as follows:

① Primary port.
② Primary port+Secondary main port.
③ Primary port+Secondary main port+ Secondary auxiliary port.

A cylindrical valve (auxiliary port valve) with a notch is inserted into the auxiliary port, and the valve is rotated by an actuater. The exhaust pressure works on the actuater, and, at the time of high speed, heavy load operation when the exhaust pressure becomes higher, the notch on the valve gradually opens the secondary auxiliary port.

4.9.2 AIM AND EFFECT

The aim of this system is to provide the most effective port shape and port timing for the primary port

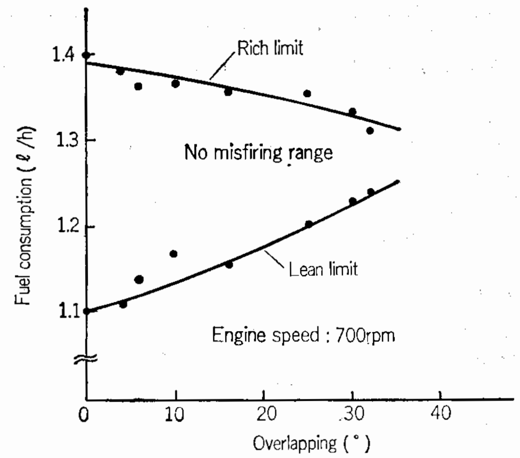

Fig. 4.69 Overlapping and idling performance

in terms of fuel economy, and secure output performance during high speed, heavy load operation by providing a secondary auxiliary port.

The primary port is of a radical shape attaching importance to low speed operation. In other words, ignitability is improved by reducing the amount of residual gas carried into the intake stroke by retarding the intake port opening time and eliminating overlapping. **Fig. 4.69** shows an example of relations between overlapping and ignitability in the state of idling.

The above figure indicates that ignitability for both lean and rich sides is improved as the overlapping

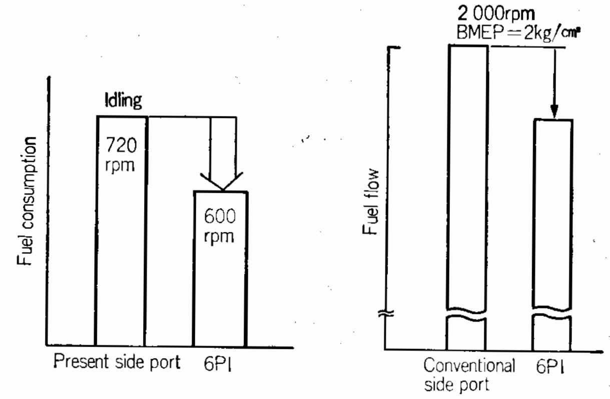

Fig. 4.70 Fuel saving results of RE6PI

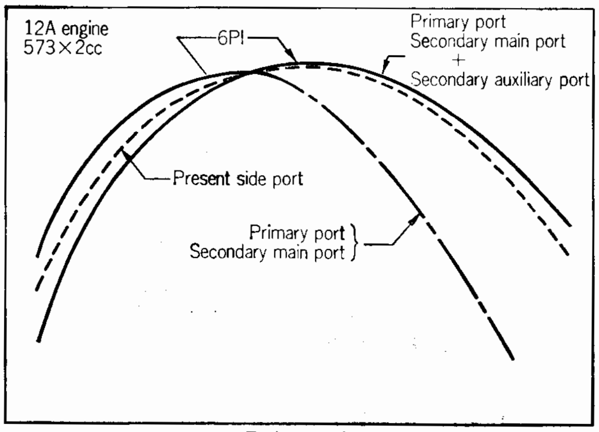

Fig. 4.71 W.O.T. performance of RE6PI

is reduced.

Also, torque at low speed is improved by preventing the mixture from blowing back into the intake stroke from the compression stroke, during low speed revolution, by quickening the closing time of the intake port.

At the same time, improvement of ignitability is occurs by promoting atomization of mixture through the reduction of intake manifold diameter and the port area.

When driving at a constant speed it is possible to run using only the primary port, and a lean mixture because misfiring has been eliminated.

Fig. 4.70 shows the effects of improvements made by RE6PI in terms of fuel consumption.

As can be seen from Fig. 4.70, significant improvements have been attained by increasing the torque in the lower speed range and improving the ignitability, which cause a stable operation of the engine at low speed, especially when the idling speed is further lowered.

On the other hand, during high speed operation when the ports on the secondary side are also used, the auxiliary ports are opened to increase the amount of mixture intake, thereby improving the charging efficiency and maintaining high speed performance. **Fig. 4.71** shows the effect of performance at W.O.T. The torque at low speed is improved while maintaining the torques at medium and high speeds.

Furthermore, increase in output can be expected through the selection of auxiliary ports and by adequate control of the auxiliary port valves. It is also possible to improve fuel economy by changing the F.G.R., reducing the engine volume, etc.

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CHAPTER 5

FUTURE DEVELOPMENT

Various types of prime movers for motor vehicles, such as gasoline and diesel engines, gas turbines, electric motors, etc. have been so far developed and applied for practical use solely or as competing with one another in each specific field according to their output characteristics, weight and cost.

These prime movers are now encountering the social requirements of the modern age which must be met, including purification of exhaust emission, improvement of fuel economy, use of low quality fuel, etc. To meet such severe requirements, research and development of new techniques for each of the prime movers are being actively performed.

The rotary engine is no exception. Development of new techniques is "on the go" to meet social requirements. The results are reflected on the rotary engine for more applicability and potentiality. The rotary engine is, thermodynamically, a displacement type internal combustion engine the same as the 4-cycle reciprocating engine. There are many new techniques of the rotary engine under study based on common concepts with the reciprocating engine. However, from inherent difference in construction and operating mechanisms, approaches unique to the rotary engine are in the process of materialization.

This chapter presents several new techniques of significance under study on the rotary engine and views the future diversification of the rotary engine.

5.1 FUEL INJECTION TYPE ROTARY ENGINE

The advantage of the fuel injection system over the carburetor system is the possibility of controlling the amount and timing of fuel supply more freely and accurately. The development of the fuel injection system for the rotary engine is under way for the purposes of improvement of performance and fuel economy, purification of exhaust emission, etc.

The fuel injection system can be generally classified by the control method into the mechanical control type and the electronic control type ; by the injection method used into the continuous injection type and the intermittent injection type ; and by the injecting position into the intake manifold injection type and the working chamber injection type. These types combined as desired will bring about characteristic systems, respectively.

Of these, three typical systems will be presented.

Fig. 5.1 shows a system using the electronic control and the intake manifold injection. The same fuel metering system as for the reciprocating engine is used. In the reciprocating engine, the fuel is injected toward the heated intake valve for better vaporization. On the other hand, the rotary engine has no such heated part in its intake system. To provide such a heated part in it is also difficult. In this respect, to facilitate atomization and vaporization of fuel, special considerations must be given to the configuration of intake manifold, location of the injection nozzle, spraying characteristics of the nozzle, etc.

Figs. 5.2 and **5.3** show the construction of a working chamber injection system for the rotary engine.

Fig. 5.2 shows a low-pressure continuous injection type rotary engine developed by Audi-NSU (West Germany). It is designed so that injection takes place in the intake stroke to prevent compression pressure being applied on the injection nozzle.

Fig. 5.3 is also a development by Audi-NSU. It is of high pressure intermittent injection type timed for injection in the compression stroke.

As described above, the rotary engine gives much wider freedom than the reciprocating engine for selecting

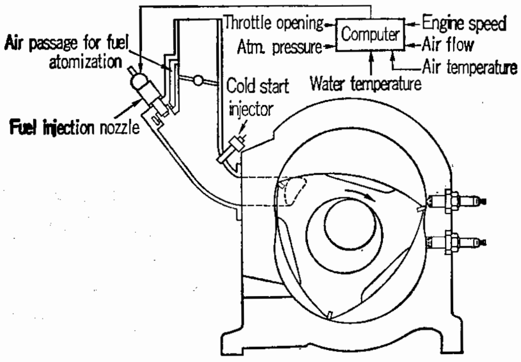

Fig. 5.1 Electronic control fuel injection system

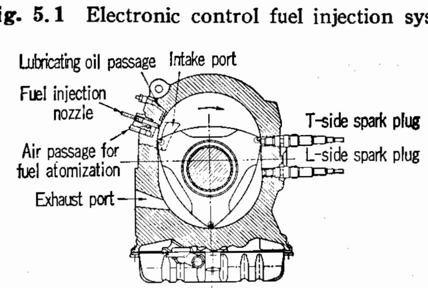

Fig. 5.2 Continuous injection type rotary engine (Audi-NSU, West Germany)

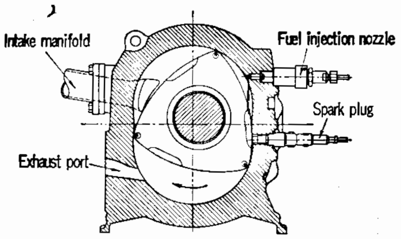

Fig. 5. 3 Intermittent injection type rotary engine (Audi-NSU, West Garmany)

the location of the injection nozzle, injecting direction, injection timing, etc. Especially, many studies on the working chamber injection type are also being performed as a means of forming a stratified mixture, as described in 5. 2.

5. 2 STRATIFIED CHARGE TYPE ROTARY ENGINE

The stratified charge rotary engine is being actively studied as an effective engine to satisfy requirements, contrary to one another, such as to control ignition and combustion, to reduce fuel consumption, to purify exhaust gas, to improve operability, etc., by properly distributing the mixture of different air-fuel ratios over the working chamber.

In general, the mixture is stratified by distributing easily ignitable rich mixture around the spark plug and lean mixture surrounding the former so that a properly lean mixture fills in the entire working chamber. When the mixture of a smaller air-fuel ratio is stratified, the amount of intake air will be increased to reduce the manifold boost in the intake pipe. Then, the following two effects will function geometrically for improving fuel economy and driveability.

One is the improvement of ignitability by reducing the ratio of burned gas diluting the mixture. The other is the reduction of intake resistance loss (pumping loss).

Thus, the stratified charge engine reduces the emission of unburned HC and CO by more complete combustion of the supplied fuel as well as improving the fuel economy and driveability. In addition, generation of NO_x can be controlled by burning the mixture under the condition of its two strata, rich and lean. In the combustion-stroke of the rotary engine, the flame propagates in the working chamber faster in the leading direction than in the trailing direction, as described in 4. 3. Making use of this characteristic, a unique stratified charge system to the rotary engine is also being studied, in which a rich mixture is distributed on the leading side of the working chamber and a lean mixture on the trailing side and surrounding wall.

To stratify the mixture, there are two fundamental concepts generally applied: One is to form the strata in the stream in the working chamber, and the other is to provide a pre-combustion chamber for the rich mixture with the main combustion chamber used for the lean mixture.

The intake and exhaust ports of the rotary engine are automatically opened and closed by the rotor revolution. Its fundamental differences from the reciprocating engine are that the port timing is determined by the location and configuration of the ports, that the working chamber moves in rotation in a certain direction, etc. Therefore, special considerations have been given to the stratification of mixture.

Several examples of the stratified charge rotary engine presently under study are given below.

(a) TWO-STRATUM CHARGE SYSTEM

Fig. 5. 4 shows the construction of SCRE (Stratified Charge Rotary Engine) developed by Toyota.

The rich mixture is supplied from the peripheral port and only air from the side port in slightly retarded timing. Thus, the mixture is stratified as an easily ignitable rich mixture is distributed over the central area and leading side of the working chamber and the lean mixture over both sides and the trailing side.

Such a system as changing the mixture of different air-fuel ratios through the two intake systems is generally called the two-stratum charge system. Several other systems are being studied on the location and configuration of the intake port, and method for supplying rich and lean mixtures.

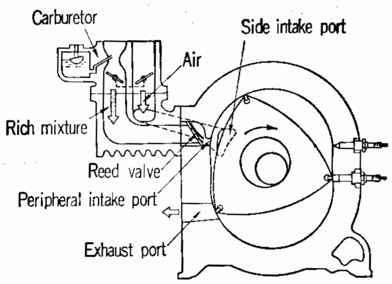

Fig. 5. 4 SCRE (Toyota)

(b) ROSCO SYSTEM

Fig. 5. 5 shows the construction of ROSCO (ROtat-ing Stratified COmbustion) system developed by Toyo Kogyo.

At light-load, air is supplied from the peripheral port only, and fuel is injected into this swirl as timed. The fuel injected from the nozzle hits the swirl and is

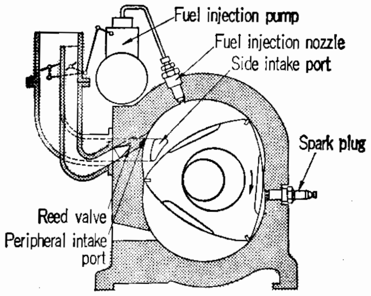

Fig. 5. 5 ROSCO system (Toyo Kogyo)

atomized. At the same time, it is carried to the leading side of the working chamber to form the strata of rich mixture in the leading side and lean mixture in the trailing side. A reed valve is provided in the peripheral port to prevent the burned gas from leaking into the intake manifold. At heavy-load, the insufficient amount of air intake from the peripheral port only is covered by a great amount of air supplied from the side port to obtain a high performance.

(c) CURTISS-WRIGHT's SYSTEM

Fig. 5.6 shows the construction of a stratified charge system developed by Curtiss-Wright.

Fuel is injected toward the rotor recess through a multiple jet nozzle, as appropriately timed, during the compression-stroke, to form the rich mixture in the rotor recess. The configuration of the rotor recess, direction of fuel injection, number of jets, etc. are properly determined for stabilized combustion, improvement of thermal efficiency, prevention of emission of unburned HC, etc.

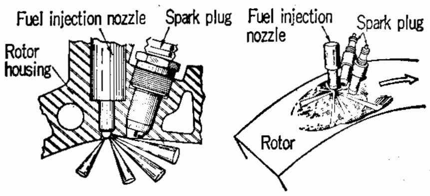

Fig. 5.6 Stratified charge rotary engine (Curtiss-Wright, USA)

(d) SCP SYSTEM

Fig. 5.7 shows the construction of SCP (Stationary Combustion Process) of pre-combustion chamber type developed by Toyo Kogyo.

At light-load, fuel is supplied into the spherical pre-combustion chamber only where it will be ignited and burned. The burned gas in the pre-combustion chamber blows and expands into the working chamber to accomplish the work. Any unburned element in the blown gas will be completely burned with the fresh air in the working chamber.

During heavy load operation, sufficient power cannot be obtained only by supplying fuel to the pre-combustion chamber, so a high output is obtained by supplying air-fuel mixture also to the working chamber.

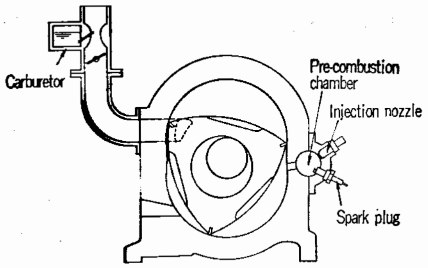

Fig. 5.7 SCP system (Toyo Kogyo)

(e) PDI SYSTEM

Fig. 5.8 shows the construction of PDI (Partial Direct Injection) system developed by Audi NSU.

The main fuel for combustion is supplied through a conventional carburetor, in which the air-fuel ratio is adjusted for lean mixture. A small amount of additional fuel is injected toward the vicinity of the spark plug to form easily ignitable rich mixture. It is ignited to burn the lean mixture in the working chamber which is difficult to ignite.

Other than the above, a torch ignition system is being studied, in which a small pre-combustion chamber is provided to ignite the lean mixture in the working chamber.

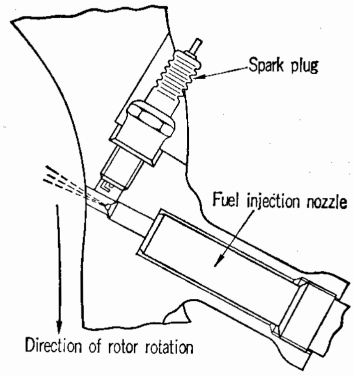

Fig. 5.8 PDI system (Audi-NSU, West Germany)

5.3 SUPERCHARGE TYPE ROTARY ENGINE

Pressurized air or mixture may be supercharged into the engine to obtain higher charging efficiency and performance. This supercharge system is widely used in the reciprocating engine for racers and large diesel trucks. Recently, attention is being paid to the combination of a small displacement engine for ordinary automobiles with the supercharge system as an effective means for improving fuel economy. It has been put into practical application in a certain field. This supercharge system will allow a small engine to develop performance comparable to that of a large displacement engine when a high performance is required to save fuel during partial load operation.

Two methods are generally used to pressurize the intake air. One is to drive a centrifugal compressor by turning a turbine by exhaust energy (exhaust-turbo system). The other is to drive a displacement type compressor by the output shaft torque (air-pump system). Fuel is supplied by either carburetor or injection system either before or after its pressurization.

Two typical supercharge rotary engines are introduce as follows:

Fig. 5.9 shows the construction of a typical exhaust-turbo system. This system gives little supercharge effect at low-speed due to a small amount of exhaust gas, but develops a greater supercharge effect by the increasing exhaust gas as the speed increases. In the exhaust gas passage, a bypass passage with a valve (waist

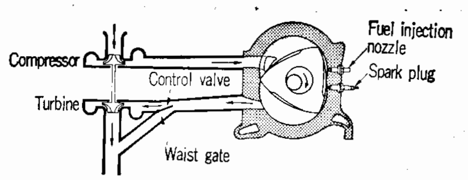

Fig. 5.9 Exhaust gas-turbo supercharge rotary engine

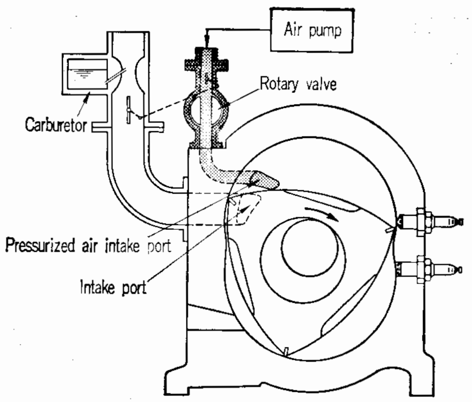

Fig. 5.10 TISC system (Toyo Kogyo)

gate) controlled by the supercharge pressure and exhaust gas pressure is provided to bypass a part of exhaust gas through the exhaust gas turbine. This waist gate prevents the supercharge and exhaust gas pressures from being excessively increased.

Fig. 5.10 shows the construction of TISC (Timed Induction with Supercharge) system developed by Toyo Kogyo. This system is designed for a greater output especially at medium- and low-speeds. A part of intake air is pressurized and forced into the working chamber, timed by the rotary valve. An air pump belt-driven by the output shaft is used to pressurize the intake air, which will allow the engine to develop a sufficient supercharge effect even over medium- and low-speeds. In this system, a small air pump is available for pressurizing only a part of the intake air with little loss in driving it. Fuel is controlled to the amount of supercharged air for an optimum air-fuel mixture in the working chamber.

5.4 MULTI-ROTOR TYPE ROTARY ENGINE

The rotary engine of single-rotor unit is basically sufficient for normal operation. In the early stage of development, the single-rotor engine was aimed at.

Later, a 2-rotor engine with two single-rotor units arranged in line was developed to satisfy the output characteristics and driveability required for automobile engines. It has proven performance characteristics comparable to that of a 6-cylinder reciprocating engine.

The advantages of the multi-rotor rotary engine are:

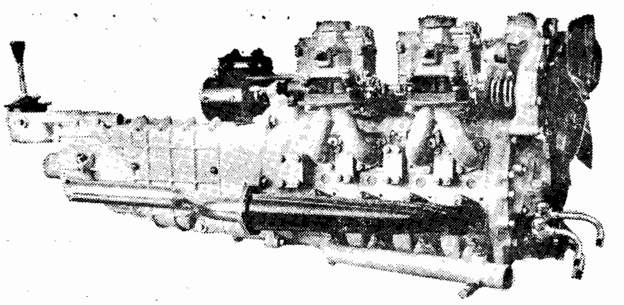

Fig. 5.11 4-rotor rotary engine (Toyo Kogyo)

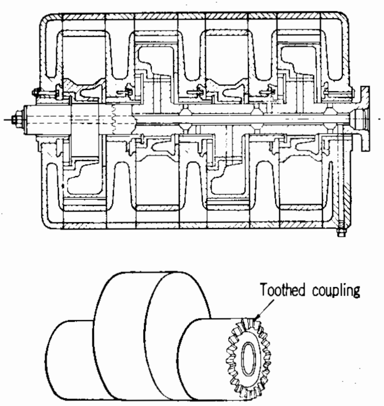

Fig. 5.12 Split type output shaft

① Improvement of performance by increasing the total displacement.
② Reduction in torque fluctuation.
③ Easier balance with inertia force and couple.
④ Capability of increasing the allowable engine revolutions by reducing the bearing loads and the sliding speed of gas seals in case the total displacement is kept the same.

The rotary engine can be rather easily made into a multi-rotor engine by arranging the single-rotor units in line. For in-line multi-rotor engines with more than three rotors, another stationary gear is required also for the central engine unit and another main bearing also on the middle part of the eccentric shaft. The following method is adopted for assembling convenience:

① The main bearing and stationary gear to be of split-type construction.
② The output shaft to be of separate-type (**Fig. 5.12**).

Also, a multi-rotor engine with more than two 2-rotor units arranged in parallel and connected by gearing is being studied. Such a type of construction will make it possible to easily shift the full-unit operation to the single-unit operation at light-load for effective fuel economy.

5.5 DIESEL ROTARY ENGINE

The diesel engine is characterized by use of low

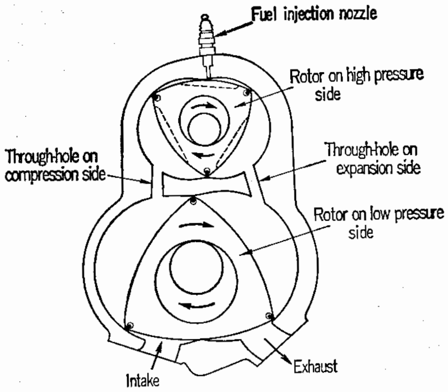

Fig. 5.13 Two-stage diesel rotary engine (Rolls Royce, Great Britain)

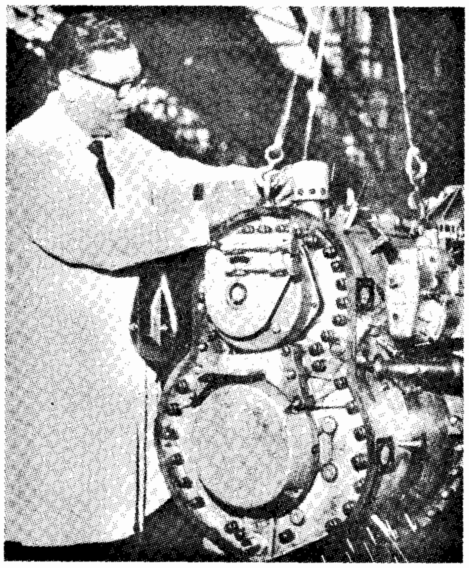

Fig. 5.14 Diesel rotary engine (Rolls Royce, Great Britain)

quality fuel and low rate of fuel consumption. Application of the rotary engine system to the diesel engine is also being studied.

In such an application, the first problem lies in how to obtain as high a compression ratio as required for the diesel cycle.

The compression ratio of the rotary engine is determined by the trochoid constant (generating radius R/eccentricity e) and the volume of the rotor recess. A higher compression ratio can be obtained by combining a trochoid of greater trochoid constant with a rotor recess of smaller volume.

However, a single stage compression to obtain a sufficiently high compression ratio will make the working chamber shallower and narrower at the top dead center. Proper distribution and combustion of fuel will be difficult.

Then, a two-stage rotary engine is being studied to obtain a high compression ratio as keeping the optimum configuration of the rotor recess and trochoid.

Fig. 5.13 shows a two-stage diesel rotary engine developed by Rolls Royce (Great Britain).

Two vertically arranged rotors connected by gearing rotate in the same direction. The upper smaller unit is used for ignition and initial combustion and the lower unit for main combustion and initial air intake. The total compression ratio is expressed by

$$\frac{\text{Displacement of lower unit}}{\text{Displacement of upper unit}} \times \begin{array}{l}\text{Compression ratio}\\ \text{of upper unit}\end{array}$$

Such a two-stage rotary engine will become more complicated in construction and greater in weight and volume than the single-stage rotary engine. Nevertheless, when compared with the reciprocating diesel engine, it holds priority with respect to weight, compactness, vibration, noise, etc.

5.6 APPLICATIONS OTHER THAN TO AUTOMOBILES

The rotary engine has been popularized mainly for automobile engines. Its features of light weight, compactness, less vibration and noise, etc. attract attention not only for automobile engines but also as prime movers in various fields. Therefore, various types of rotary engines have been developed for application to fields other than automobiles.

Some applications of the rotary engine will be introduced and future applicability viewed as follows:

(a) Motorcycle

Rotary engine motorcycles of high performance

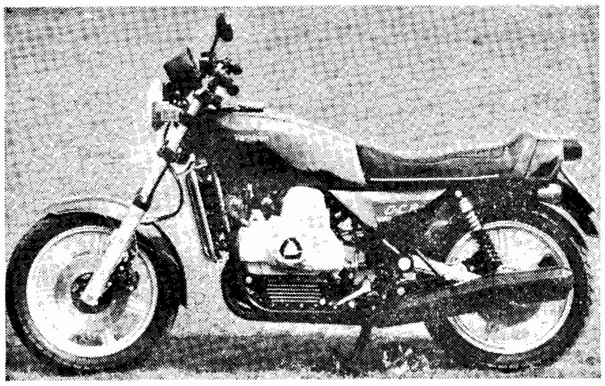

Fig. 5.15 Motorcycle (Comotor, Luxemburg, 900cc × 2-rotor)

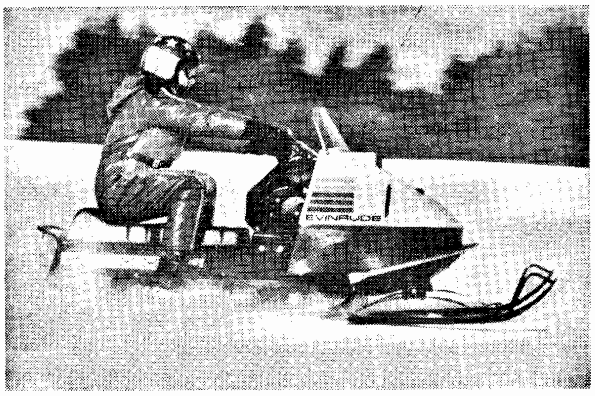

Fig. 5.16 Snowmobile (Outboard Marine, USA, 530cc × 1-rotor)

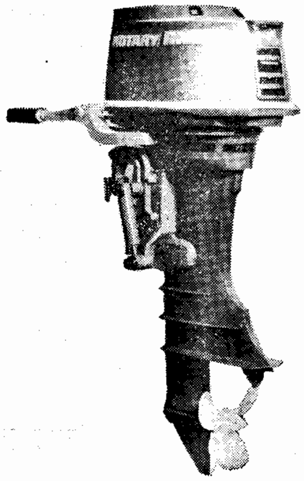

Fig. 5.17 Outboard engine (Yanmar Diesel, 300cc×2-rotor)

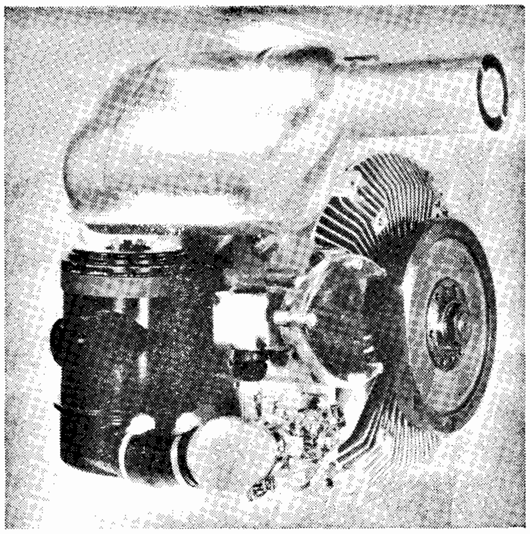

Fig. 5.18 General-purpose engine (Fichtel & Sachs, 300cc×1-rotor)

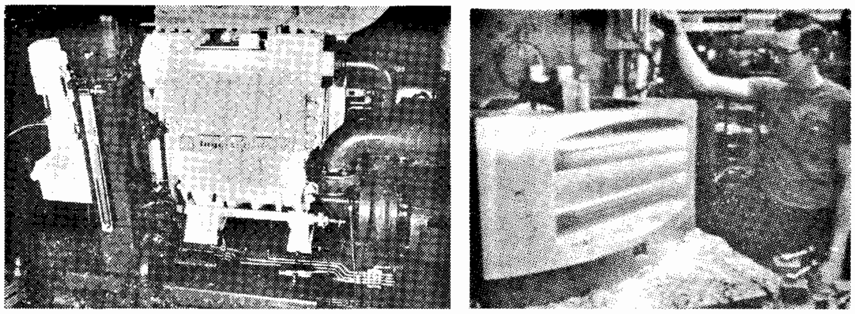

Fig. 5.19 Industrial engine (Ingersoll-Rand, USA, 41ℓ×2-rotor)

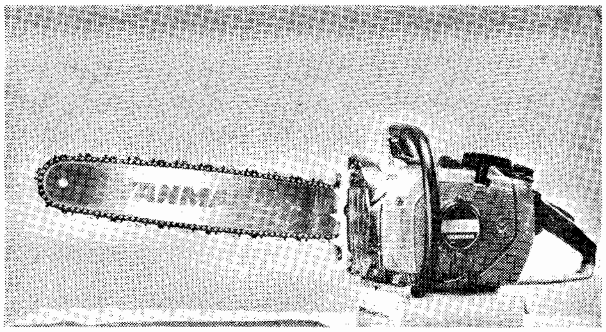

Fig. 5.20 Chain saw (Yanmar Diesel, 60cc×1-rotor)

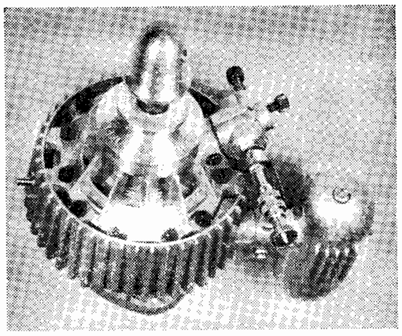

Fig. 5.21 Model engine (Ogawa Seiki, 5cc×1-rotor)

shows the comfort feeling unique to the rotary engine resulting from less vibration, high power, etc.

(b) Snowmobile

Snowmobiles equipped with a rotary engine develop high performance and provides comfort in operation.

(c) Outboard Engine

Outboard rotary engines are more compact consume less fuel, and provide higher performance and comfort than the ordinary 2-cycle reciprocating engines.

(d) High-speed Boat

Rotary engine high-speed boats have already won championships in many races.

(e) Light Plane

Light planes equipped with rotary engines of light weight and high outrut are reputed to have less vibration and noise.

(f) Helicopter

Helicopters installed with a rotary engine by Curtiss-Wright are used as training helicopters in the U.S. Air Force.

(g) General-purpose Engine

Small, light weight and portable rotary engines are widely used for generators, fire pumps, agricultural machines, etc.

(h) Industrial Engines

Very large rotary engines are also developed for driving compressors and pumps.

(i) Chain Saws

The very low vibration of the rotary engine is effective in preventing occupational diseases, such as vibration disease.

(j) Model Engines

The smallest sized rotary engine ever developed when installed on a model plane, has the advantages of compactness and a centrally lacated output shaft.

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