New Technology Employed for the Latest 13B-Rotary Engine

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ABSTRACT

The 1990 RX-7's 13B rotary engines have been designed to give the spirited performance of a genuine sports car engine. This engine with improved response provides the operating ease, plus the oil and fuel economy expected by today's consumer.

This paper describes the new technology and improvements employed in the 13B rotary engines including:

- the variable dynamic effect intake system
- the independent twin scroll turbo system (These are incorporated to improve intake.)
- optimization of transient fuel control
- improved control over knock and idle feedback

- a lighter rotor
- an ion-nitried gear
- inertia moment reduction of the rotating system
  (to increase engine power and response)

also including an electronically controlled oil metering system for improved lubrication.

INTRODUCTION

In 1978, the RX-7 made a sensational debut in the sports car market, featuring beautiful styling, superior handling and high performance. All of this was enhanced with the incorporation of a lightweight, compact, and powerful rotary engine.

The rotary engine has since undergone a number of modifications which have improved the quality of its emissions, increased its fuel economy and boosted its power output.

Some of those modifications included,
(A) replacing the 12A's thermal reactor emission purification system with a catalyst system
(B) replacing the carburetor equipped 12A engine in 1984 with an electronic gasoline injected 13B-EGI engine, and
(C) creating the 13B T/C engine by adding a turbocharger in 1986.

To make the RX-7's 13B rotary engine a powerplant suitable for the sports car of the 1990's, various improvements have been made, including further increasing power output, shortening response time, and improving fuel & oil economy. The following represents an outline of these improvements:

I OBJECTIVES

The 13B-EGI and 13B-T/C for the new RX-7 have been developed, with special emphasis placed on the following points:

(1) Enhancement of output performance over all ranges of operation
(2) Increasing the 13B-EGI's maximum RPM range
   To give the sports car driver a greater participation and more excitement through increased control over the vehicle.
(3) Improvement of response
   To offer quick and effective response to match the aggressive demands of sports car driving. This combined with a gentle smoothness to meet the needs of comfortable city driving.
(4) Improvement of fuel economy
   To not only improve the marketability of the car via improved mileage, but also to allow the vehicle to be equipped with various car options.
(5) Reduction of lubricating oil consumption (LOC)
   To realize improved oil economy, thus reducing maintenance costs.
II ENGINE SPECIFICATIONS AND PERFORMANCES

The principal specifications of the 13B-EGI and 13B-T/C are shown in Table 1. The main technological improvements are listed in Fig. 1. Performance comparisons of the existing models are given in Figs. 2 and 3.

The 13B EGI’s output performance is 119kW (160HP) at 7,000rpm, 190N-m (140lb-ft) at 4,500rpm. The 13B T/C’s performance is 169kW (200HP) at 6,500rpm, 265N-m (196lb-ft) at 4,000rpm.

![Torque vs. RPM Graph]

**Fig. 2 Performance of 13B EGI & T/C**

**Fig. 3 Output performance and running performance**

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**Table 1 Main data of new engines**

<table>
<thead>
<tr>
<th></th>
<th>13B-EGI</th>
<th>13B-T/C</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>TYPE</strong></td>
<td>Water-cooled 2-rotor inline</td>
<td></td>
</tr>
<tr>
<td><strong>Displacement</strong></td>
<td>654X2</td>
<td></td>
</tr>
<tr>
<td><strong>Compression ratio</strong></td>
<td>9.7</td>
<td>9.0</td>
</tr>
<tr>
<td><strong>Max output (kW) /rpm</strong></td>
<td>119(160) / 7000</td>
<td>140(200) / 6500</td>
</tr>
<tr>
<td><strong>Max torque (N-m) /rpm</strong></td>
<td>190(140) / 4500</td>
<td>265(196) / 4000</td>
</tr>
</tbody>
</table>

**Port Intake**

<table>
<thead>
<tr>
<th></th>
<th>Open</th>
<th>Primary</th>
<th>Secondary</th>
<th>ATDC</th>
<th>ATDC</th>
<th>ATDC</th>
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<tbody>
<tr>
<td><strong>Exh</strong></td>
<td>Close</td>
<td>75 BBDC</td>
<td>48 ATDC</td>
<td>32 ATDC</td>
<td>45 ATDC</td>
<td></td>
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</tbody>
</table>

**Charging method**

- Dynamic supercharging
- Exhaust turbo

**Supercharger**

- F-side Air/Oil 
- R-side Air/Oil

**Charged Air Cooler**

- Air-cooling

**Fuel supply**

- Electronically-controlled fuel injection

**Oil cooler**

- Air-cooling

**Radiator**

- Corrugated type

**Ignition**

- Full transistor ignition
- Electronically-controlled

**Spark Plug**

- Air-Gap type

**Exhaust emissions control**

- 3-way catalyst

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**Fig. 1 Principale technologies and the objectives**

- Variable dynamic effect intake system
- Independent twin-scroll Turbo
- Electronic boost pressure control
- Optimization of transient fuel control
- Knocking control improvement
- Idle speed feedback system
- Higher compression ratio (with machined combustion chamber)
- Lighter rotor
- Ion-nitrified gears
- Inertia moment reduction of rotating parts
- Electronically controlled oil metering system

**Technologies**

- 13B EGI
- 13B T/C

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**Max Output (kw)**

<table>
<thead>
<tr>
<th></th>
<th>13B EGI</th>
<th>13B T/C</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>100</td>
<td>100</td>
</tr>
</tbody>
</table>

**9-60 miles/hr (sec)**

<table>
<thead>
<tr>
<th>13B EGI</th>
<th>10-sec</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>7</td>
</tr>
</tbody>
</table>
These figures have been realized by the following technological changes; 1) To improve the intake system, a variable dynamic effect intake system was used on the 13B-EG1 and an independent twin scroll turbo system was incorporated into the 13B-T/C. 2) To perfect control system, transient fuel control was optimized, knocking control was improved (this is for the 13B-T/C only) and idle speed feedback system was added. 3) To improve engine's internal system, a lighter rotor and ion-nitrided gears were used. In addition to those, the rotating parts' inertia moment has been reduced. 4) To improve the lubricating system, an electronically controlled oil metering system was adopted.

III PRINCIPAL TECHNOLOGY FOR THESE NEW ENGINES

III-1 VARIABLE DYNAMIC EFFECT INTAKE SYSTEM

The 13B-EG1 has a variable port timing system called 6PI (six port induction) system and a dynamic effect intake system.

Its function is based on the existence of interfered pressures between the cylinders. Engines with those systems experience a high flat torque curve from low RPM to high RPM. The dynamic effect intake system has since been further improved to offer the higher performance at both low and high RPM with an expanded operating range at high RPM. (The engine speed limit has since been increased from 7,000rpm to 8,000rpm.) The improved system is called "Variable Dynamic Effect Intake System", which makes two changes to the length of the air passages transferring air pressure between the cylinders (See Fig.4.)

III-1-a Principle of the Previous Dynamic Effect Intake System

The dynamic effect intake system which utilizes the mechanisms of a two-rotor rotary engine enhances the volumetric efficiency by mutually transmitting the pressure waves generated in the intake passages into the opposite cylinder. Fig.5 shows two types of pressure waves. One pressure wave is generated when the air flow inside the intake passage is suddenly stopped (when the intake port is closed) and pressurized by its inertia. "A" in Fig.5) The other is generated when the intake port ("B" in Fig.5) opens causing the air inside the intake passage to surge backwards and become highly pressurized by the combustion chamber's exhaust gas.

Fig.4 Variable dynamic effect intake system

Fig.5 Pressure wave transmission timing of dynamic effect intake system and its effect
When these two pressure waves enter the adjoining cylinder through the intake passage just before the intake port is closed, it elevates the amount of air pressure and enhances the volumetric efficiency of combustible air.

III-1-b The Principle of the New Variable Dynamic Effect Intake System

The two pressure waves travel through the intake passage at the speed of sound. Therefore, in the high speed range where the travel time is shorter, the intake passage is necessarily shorter, and in the low RPM range, it is necessarily set longer.

Fig. 6 shows the changes to the torque curve obtained by extending the air passage length between the cylinders at 200mm increments. As seen from the figure, an increase in the length of the air intake passage causes an increase in maximum torque. As indicated, there is a tendency for the peak to occur at lower RPM, and for the torque to drop radically at higher RPM. This figure shows improvements to the 1988 model intake system. For the 1990 models, we found that the best combination for obtaining the optimum torque curve, included making the passage length "a" 200mm shorter for high RPM and "b" 400mm longer for low RPM ranges. When the combination offered by "a" for maximum torque and "a" for maximum output are considered, acceleration within the medium RPM range deteriorates. This effect destroys the "sports-car" feeling, and results in a final acceleration time which becomes inferior to the optimum combination of "a" and "d".

III-1-c The Structure

In Fig. 7 an additional passage with a control valve was installed in the primary/secondary intake passages. At high RPM, as shown in Fig. 7a, the control valve is opened, and the pressure wave is transmitted through a shorter passage. At low RPM, as shown in Fig. 7b, the control valve is closed, and the pressure is transmitted through a longer passage. The intake passages are situated directly opposite each other as shown in Fig. 7a, in order to enhance the efficiency of pressure wave transmission especially at high RPM.

Meanwhile, as indicated by the shaded area in Fig. 7b, if the passage develops an area of stagnant air when the valve is closed, the pressure wave becomes weakened, creating a lower dynamic effect, which in turn causes the torque to drop. To prevent this occurrence, a rotating type control valve was incorporated. Normally a rotating valve is heavy and inefficient. Those problems were overcome by using a light weight, plastic, rotating valve.

Fig. 8 Control of control valve
III-1-d Control of the Intake Passage

To optimize the effect obtained by combining the SPI and the variable dynamic effect intake system, the opening and closing of the control valves are controlled by a microprocessor as shown in Fig. 8. The timing of the closure/opening varies according to engine RPM and the volume of intake air. In the lower RPM range, to the left of the changeover line, the SPI auxiliary port valve (APV) is kept closed to advance the timing of the inlet port's closure (IC). For low RPM, the rotary valve of the variable intake is also closed to make the air passage longer. For the higher RPM range, to the right of the changeover line, both valves are opened, and a delayed closing of the inlet port plus the use of a shorter air passage are selected. Employing this system allows a higher average torque to be used overall regardless of the RPM.

III-2 THE INDEPENDENT TWIN-SCROLL TURBO SYSTEM

A wide range engine RPM and load are required during normal operation, and when supercharging with a conventional single-scroll turbo, it is difficult to obtain optimal performance over the entire range. Therefore emphasis is usually put on either low RPM, high RPM, or on a compromising point between high and low RPM. To resolve this issue, the 13B-T/C has an independent twin-scroll turbo system which realizes a dramatic improvement in performance over the entire operating range. This newly developed system is particularly suited to the unique features of the rotary engine. As shown in Fig. 9, the system is composed of a turbocharger with completely independent scrolls which use an electronic control system to boost air pressure.

III-2-a The Structure of the Independent Twin-Scroll Turbo

The exhaust passage composed of an exhaust manifold and a turbine scroll is divided completely into two independent passages. In an effort to increase and maintain turbo effect the unit's passage length from the exhaust port to the turbine blade has been minimized and tapered smoothly. These changes prevent interference between the exhaust pulses of the two exhaust passages and prevents sudden expansion or attenuation in the exhaust manifold. As shown in Fig. 10, the structure has an area over radius (A/R) of 0.5 inches in each scroll, (1.0 inch combined) and has a turbine with a diameter of approx. 2.5 inch. Each scroll has an independent waste gate with the bypass gas being simultaneously controlled by one valve.

III-2-b The Principle of the Independent Twin-Scroll Turbo

The turbocharger by using the turbine scroll convert the energy of the heat and the pressure of the exhaust gas into kinetic energy. To create the power for turbocharging, the turbine further converts them into a rotating force before the gas exits from the turbo blades. It is noted that the magnitude of kinetic energy is proportional to the square of the velocity of the exhaust gas flow. Therefore, maintaining the movement of the exhaust gas would increase the peak value of the exhaust pulse pressure which would in turn increase kinetic energy and substantially increase the turbine's workload. The method of effectively utilizing this fact is called "pulse turbocharging". Exhaust pressure pulsation is much greater in rotary engines than in conventional reciprocating engines.
This is due to the unique design of the rotary engine which has no exhaust valves but instantaneously opens and closes its ports. These features in turn make the rotary engine particularly suited to the use of pulse turbocharging. When a common type exhaust manifold is used, the intense blow-down exhaust pulse suddenly expands and diffuses into the cylinder (that is in the midst of an exhaust stroke) and into the exhaust manifold, causing the exhaust pulses to weaken. This further causes the peak value of the pressure to drop and reduces turbine output.

To the contrary, the completely independent twin-scroll turbo is designed to optimize exhaust pulse energy. With the new twin-scroll turbo, at high RPM, there is now no exhaust interference between cylinders and the volume of the exhaust manifold is minimized, allowing exhaust pulse to enter the turbine without attenuation. As a result, at high RPM when gas volume is particularly large, turbine efficiency is enhanced, while the exhaust resistance is lowered and the engine back pressure is lowered. The independent twin-scroll turbo has thus made it possible to significantly improve output as compared to that of a single-scroll turbo. Briefly, using independent twin scroll turbo with a total A/R equal to that of the single-scroll turbo, realizes a much higher power output at high RPM than by using a single-scroll turbo.

Even at low RPM which has a smaller gas volume, less thermal energy and lower energy pressure, the twin-scroll makes full use of the exhaust pulse, creating high boost pressure in spite of the large size of the turbo.

III-2-c The Electronic Boost Pressure Control System

This system represents the technique for obtaining the exact amount of boost pressure desired. It is done by controlling the opening angle of the waste gate valve by using a microprocessor. As shown in Fig.9, the opening angle is controlled by changing the amount of pressure relief allowed by the duty solenoid. The amount of relief allowed is according to the amount of pressure applied on the diaphragm type actuator that drives the valve. The amount of relief determined by a microprocessor which takes into consideration engine RPM, the degree of the throttle opening and the amount of pressure in the exhaust manifold.

According to the conventional method, when approaching maximum boost pressure, total boost pressure was applied on the actuator, thereby lowering the pushing force on, or the sealing performance of, the waste gate valve. When using the conventional method, the waste gate valve is opened by the exhaust gas pressure in the exhaust manifold causing the maximum boost pressure timing to be forced into the high RPM operating range. By contrast, according to this new system, the microprocessor keeps the amount of pressure relief at the maximum level until the boost pressure reaches its highest level. This lowers the pressure applied on the actuator which in turn forces to the waste gate valve to remain closed, which makes it possible to generate maximum boost pressure at low RPM as shown in Fig.11.

![Fig.11 Effect of electronic control of boost pressure (using independent twin scroll turbo)](image_url)

Fig.12 illustrates by comparison the performance obtained when using the independent twin-scroll turbo, with the performance achieved by when using only a single-scroll turbo having the same turbine diameter and A/R. (The both twin and the single scroll turbo possessed a boost pressure electronic control.) At high RPM, power output is approx. 5% larger due to weakening back pressure. At low RPM, boost pressure and torque are improved. For example, Fig.12 indicates an approx. 60% increase in pressure with an approx 112% increase in torque at 1500rpm. At 2000rpm, the improvements are about 60% and 15% respectively. In Fig.13, we note that such improvements are realized even during transient operation. During transient operation, the increase in boost pressure is accelerated, thus producing excellent acceleration response.
III-3. Optimization of Transient Fuel Control

Control over transient fuel has been improved to quicken acceleration response time during transient operation. Generally, the response lag in the initial stage of acceleration (excluding the engine's inertia moment factor) is caused by a delay in torque elevation due to a drop in air/fuel ratio. This drop is caused by the amount of time required by the engine's computer system to measure sudden changes to intake air volume and fuel requirements etc. during initial acceleration. In order to realise a better acceleration response, a new system has been added to the EGI system to improve fuel control. During the initial stage of acceleration, this new system first analyses engine RPM, the throttle opening and throttle opening changing rate, then commands the fuel injectors to shoot once only a precisely calculated amount of fuel into the cylinders. This injection is given in addition to the engine's specified injection only during the initial stage of acceleration. If two or more such commands are given, the air/fuel ratio becomes overrich, and torque deteriorates instead of improving. (Excess fuel supply also causes torque to deteriorate.)

As shown in Fig. 14, using this new system now causes indicated mean effective pressure to be raised simultaneously with acceleration instead of first suffering a loss during the initial stage of acceleration.

These improvements to fuel control in addition to a reduction in inertia moment (as described later in this paper) have led to dramatically improved response times. For example; the no-load racing operation time from idling RPM to 6,000 RPM has been shortened from 0.95 to 0.85 seconds in the 13B-EGI, and from 1.5 to 1.1 seconds in the 13B-T/C.

This 13B-T/C with the new fuel control system has been tested in a road test and the result is shown in Fig. 14.

- A microprocessor has been incorporated in order to achieve precise control over a wide range of RPM. To accommodate the use of a microprocessor, the following improvements have been made on the conventional control systems.
III-3-b Knock Control Improvements
In the 3B-T/C, increasing boost pressure raised the compression ratio from 8.5 to 9.0, thus making the 3B-T/C very sensitive to engine knock at all RPMs. To resolve this problem for a significant improvement of the power output and fuel economy, the position of the knock sensor has been changed as shown in Fig. 15. It has been moved from the conventional intermediate housing to the upper part of the T-plug hole of the rotor housing closest to the origin of the knock. Previously, due to its position, excessive background noise often made the sensor ineffective due to a low signal to noise (S/N) ratio.

By contrast, in its new position the S/N ratio is improved, enabling engine knock to be more easily recognized.

Fig. 15 Position of knock sensor

Further improvement was made with the incorporation of a microprocessor which uses the faster, more efficient and accurate digital system versus the previously used analog circuitry. With the analog circuitry, the sensitivity level for detecting knock was fixed at the mid point between the lowest knock signal and the highest level of background noise to cover all engines. In using the new digital system, background noise and the knock sensor's signals are now individually analyzed for each engine by the microprocessor. The microprocessor then ensures optimum knock detection by adding a predetermined (fixed) value to the now known value of the background noise.

It now concentrates on the sensor's signals which peak above this newly calculated value. This method thereby allows for noise and signal level differences of each engine and gives precise knock control over all engine RPMs.

III-3-c The Idle Speed Feedback System
The required air/fuel ratio during the idling operation changes according to environmental conditions such as altitude and ambient temperature. Therefore, when using the conventional EGI system, significant changes in environmental conditions sometimes causes great changes in idle speed or a reduction in torque at low RPM. These changes in idle speed or torque occurred because theoretically, the conventional system can only compensate for reduced air density but not for a reduction in exhaust gas residual. To correct this, an idle speed feedback system has been developed to stabilize the idling operation under changeable environmental conditions. This system works by sensing engine RPM changes during idling by using a crank angle sensor. When the value exceeds a tolerable limit, feedback is provided to the microprocessor to keep the speed variations within a specified range. When the idle speed variation exceeds the permissible range due to a lean fuel to air ratio as shown at point "a" in Fig. 16, the ratio is made richer until it reaches point "c" which is 2% richer than the tolerable limit. When the idle speed variation is at point "b" due to an over rich mixture, the air/fuel ratio is made leaner until it again reaches point "c". This control system automatically sets the air/fuel ratio during idling for maximum fuel economy and tolerable level engine vibration. Stabilizing the idle results in a higher burn temperature which in turn effectively prevents spark plug fouling created by an over rich or too lean fuel/air ratio.

Fig. 16 Air/fuel ratio control by idle feedback system
III-4 IMPROVEMENTS TO INTERNAL ENGINE MECHANISMS

III-4-a The Rotor

In order to enhance power output and fuel economy via improved combustion, the compression ratio was raised in both the 13B-EGI and 13B-T/C engines.

The specifications of rotor and gears are shown in Table 2. The modifications of rotor and gears are shown in Fig. 17 and Fig. 18 respectively.

Table 2 Specifications of rotor & gears

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<thead>
<tr>
<th></th>
<th>13B-EGI</th>
<th>13B-T/C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compression ratio</td>
<td>9.7</td>
<td>9.0</td>
</tr>
<tr>
<td>Recess</td>
<td>Machined</td>
<td></td>
</tr>
<tr>
<td>Rib</td>
<td>2mm</td>
<td></td>
</tr>
<tr>
<td>Rotor gear</td>
<td>Ion-nitrided</td>
<td></td>
</tr>
<tr>
<td>Stationary gear</td>
<td>Ion-nitrided</td>
<td></td>
</tr>
</tbody>
</table>

Because conventional rotors have "as-cast" rotor recesses which function as the engine's combustion chambers, compression ratio tolerable variants of approx ± 0.3 can occur. Increasing the compression ratio therefore often leads to an increase in engine knock. By machining the recesses, the differences in compression ratios has been reduced to the insignificant figure of ± 0.06. Because of this low variation, the compression ratio is now changed from 9.4 to 9.7 in the 13B-EGI and from 8.5 to 9.0 in the 13B-T/C engine. That is to say, the compression ratio is set at the upper limit of the variations. The improved compression ratio makes combustion more efficient and allows the ignition timing to be improved thereby improving torque at low RPM and overall fuel economy. Further, in the 13B-T/C engine since an improvement in the compression ratio automatically leads to an improved expansion ratio, the exhaust gas temperature is lowered. Subsequently fuel used to cool the exhaust gas at high RPM can be saved. As a result, the combustion efficiency is further improved and contributes to an enhanced output.

On the 13B-EGI engine, the bearing load increases due to the rise in its maximum RPM. In order to lower this load increase, the weight of the rotor has been reduced. By improving the casing precision, the thickness of the rib was changed from 3mm to 2mm and the weight was reduced by 14%. At the same time, the stationary gear for determining the phase of the rotor is ion-nitrided for increased strength. On the 13B-T/C, the bearing load also increases due to an increase in peak combustion pressure. To counter this, the rotor gear and stationary gear have been ion-nitrided and the rotor weight reduced.

III-4-b The Reduction of Rotating System's Inertia Moment

As mentioned earlier, to further improve acceleration response, it is necessary to decrease the inertia moment of the engine's internal systems. Such reductions caused fluctuating idle speeds and vibrations. Incorporating an idling speed feedback system corrected these problems and allowed for a reduction in the weight of the rotor, flywheel and balance weight. The end result is a smoother running engine with a 24% reduction in the moment of inertia in the 13B-T/C and a 12% reduction in the 13B-EGI.
III-5 IMPROVEMENT TO THE LUBRICATING SYSTEM

When using a conventional mechanical oil metering system to change the oil flow rate according to the degree of the throttle opening cannot be accurately done over the entire operating range. In addition to that, it is difficult to satisfactorily reduce lubricating oil consumption (LOC). In lieu of that, an electronic oil metering system using a microprocessor to optimally control the oil flow rate according to engine operating conditions, has been developed and is now used in both engine types. Thanks to this system, oil consumption has been markedly reduced by 80% while maintaining engine reliability.

III-5-a System structure

Fig.19 is a diagram of the new electronic oil metering system. A cross section view of the oil pump is shown in Fig.20.

This new oil metering system retains the original oil pump concept. Under that concept, the amount of oil pumped is measured according to plunger rotation. However, a stepping motor is now included to control oil discharge and a microprocessor is added to control the flow rate. A sensor has been incorporated to monitor and report back to the microprocessor the operating angle of the stepping motor. This ensures an optimum oil flow as determined by the microprocessor which also considers engine RPM, intake air flow and water temperature.

IV-5-b Determining the Rate of Oil Flow

Having achieved control over the rate of oil flow via the electronic oil metering system, the rotor's apex seal temperature was measured to determine the minimum rate of oil flow required.

Fig.21 illustrates the effects on the apex seal temperature according to variations in oil flow. As noted Point "A" represents an equilibrium point of proper apex seal temperature while using the lowest possible rate of oil flow. Increasing the oil flow, as indicated to the right of Point "A", created little change to the apex seal's temperature and needlessly increased oil consumption.

Decreasing the oil flow radically increases seal temperature. The range between Point "A" and "B" represents an area titled "Boundary Lubrication" in which occasional metal contact is experienced. The sudden increase in seal temperature, indicated to the left of Point "B" indicates constant metal contact.

![Diagram of electronic oil pump system](image)

![Diagram illustrating the relationship between apex seal and lubricating oil volume](image)