

Technology to Boost CVT Fuel Efficiency through Series Connection of Mechanical and Electric Oil Pumps

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ABSTRACT

A new oil pump system that combines mechanical and electric oil pumps has been developed in order to optimize oil pump work under low-load driving conditions. The use of the new system has reduced energy loss in the oil pumps, increased CVT transmission efficiency, and increased vehicle fuel efficiency.

A distinctive feature of the hydraulic circuit employed in the developed system is the positioning of the electric oil pump downstream from the mechanical oil pump, realizing a configuration in which it is possible to switch between the sources of hydraulic pressure supply to the pulleys and other parts. In addition, the configuration helps prevent any decline in driving performance in the event that the electric oil pump loses function.

The rated output of the electric oil pump employed was selected in order to realize benefits under a variety of driving conditions, and the control applied in the actual vehicle was optimized. As a result, an increase of 0.71% in CVT average transmission efficiency and an increase of 1.23% in fuel efficiency in an actual vehicle were realized.

1. Introduction

CVTs realize shift operation with lower fuel consumption than stepped transmissions⁽¹⁾. Due to this, their application has rapidly become more widespread as a means of boosting fuel efficiency, with a particular focus on city driving⁽²⁾. When the vehicle is cruising however, a driving mode in which shift is not necessary, there are occasions on which the transmission efficiency and fuel efficiency of CVTs are lower than those of stepped transmissions⁽³⁾. The realization of increased transmission efficiency in CVTs will, therefore, be an important factor in realizing further increases in fuel efficiency.

One reason that the transmission efficiency of CVTs falls behind that of stepped transmissions is the fact that oil pump work is high^{(4), (5)}. Figure 1 shows ratios of causes of energy loss in a CVT in JC08 mode. In a CVT, hydraulic pressure produces belt clamping force, and the unit transmits torque by means of friction between the metal pushing V-belt and the pulleys as a result of this belt clamping force. It is necessary for the maximum belt clamping force to be several times greater than the maximum pressure employed in a stepped transmission.

Figure 2 shows a schematic diagram of the allocation of oil pump work in a CVT. The required oil pump work in a CVT can be broadly divided into two types. One of these is the work necessary for the belt clamping force mentioned above and for the engagement of the clutch. High pressure, P_H , and a necessary flow rate, Q_A , including shift, are demanded for torque transmission. The other is the work necessary for supply of oil to the torque converter and for lubrication. In this case, hydraulic pressure P_1 is demanded. As this indicates, it is necessary for the CVT oil pump to satisfy two demands. As it is necessary for the oil pump to supply high-pressure working oil to generate belt clamping pressure at high flow rates that maintain a state of lubrication, oil pump work during operation occurs within the maximum scope that incorporates these two demands. The region within this maximum scope in which there is no effective oil pump work represents energy loss.

Against this background, two-rotor-type trochoid pumps^{(4), (5)} and variable-discharge oil pumps⁽⁶⁾ have been put into practical application in order to reduce oil pump energy loss. However, these units switch the discharge rate in response to the CVT operating state in stages, and they therefore do not realize the optimization of oil pump work.

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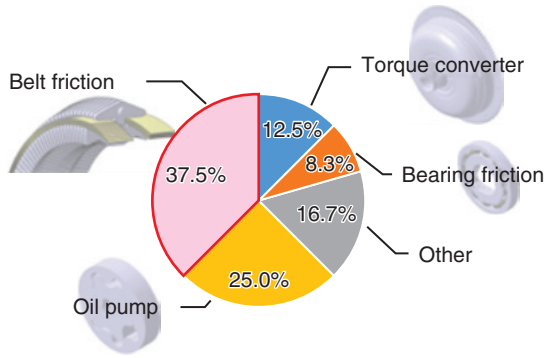


Fig. 1 Breakdown of power consumed in CVT (during operation in JC08 mode)

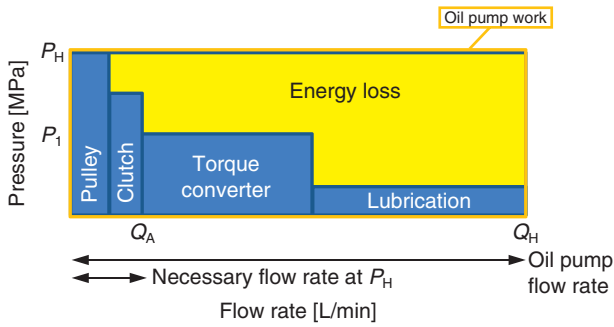


Fig. 2 Oil pump work in CVT

This paper will discuss a technology for the realization of increased fuel efficiency using an oil pump system developed in the quest for a higher level of oil pump work optimization.

2. Development Aims

CVTs realize stepless shift by inducing a stroke in the pulley axial direction and changing the winding diameter of the belt. During shift, changes in the pressure of the working oil and the pulley stroke occur simultaneously. When a pulley stroke is generated, the capacity of the pulley oil chambers fluctuates, and the necessary flow rate for the working oil also changes. As a result, the absence of any reduction in the theoretical discharge rate of the oil pump is a precondition for the ability to respond to sudden shifts.

When the oil pump is driven by the engine crankshaft, the discharge rate of the mechanical oil pump (termed “MOP” below) is proportional to the engine speed⁽⁷⁾. Even at the same discharge pressure, energy loss will be greater at higher engine speeds. In order to optimize oil pump work, it would be necessary to use an oil pump drive method that is independent of engine speed. The research under discussion selected an electric oil pump (termed “EOP” below).

A design concept encompassing the selection of EOP specifications and EOP operation in order to minimize oil pump work was developed. The necessary consideration in applying an EOP was to realize a hydraulic circuit that would help prevent any interruption to hydraulic pressure supply even when the EOP lost function.

Mainly, the following three items were set as the development aims:

- (1) The development of a hydraulic circuit that would help ensure that driving performance was not impeded even if the EOP lost function.
- (2) Selection of a rated output for the motor used in the EOP that would minimize oil pump work.
- (3) Optimal control of EOP in response to driving conditions.

3. Basic Structure

Figure 3 shows a schematic diagram of the operation of the EOP in relation to driving conditions. In sections A and B, when the system’s maximum pressure, P_H , is low, and the demanded flow rate, Q_A , is also low, the two oil pumps are operated together in order to reduce energy loss. In section C, when both the maximum pressure and the demanded flow rate are high, the EOP is stopped and hydraulic pressure is supplied by the MOP alone.

3.1. Hydraulic pressure Circuit with Oil Pumps in Series

Figure 4 shows a schematic diagram of the hydraulic pressure circuit proposed in this research. The two oil pumps are positioned in series, and the source of supply of high hydraulic pressure is switched by the opening and closing of a bypass valve. The hydraulic channel pressure between the EOP intake side and the MOP is designated P_1 , and the hydraulic pressure in the EOP discharge section is designated P_H . In addition, the system is provided with a P_1 sensor to measure pressure P_1 in order to realize hydraulic pressure feedback for EOP control.

Due to the series connection of the oil pumps, the electric

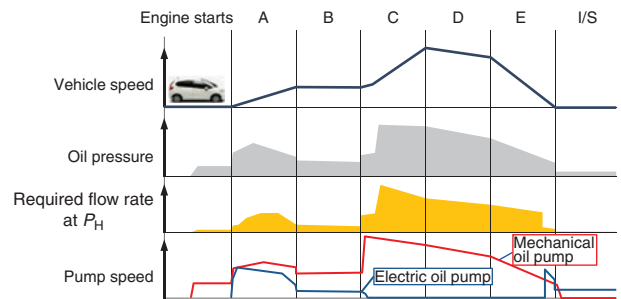


Fig. 3 Schematic image of electric oil pump operation

OP receives discharge pressure from the MOP, and outputs further increased pressure. When the bypass valve is closed, the MOP is responsible for the low-pressure sections, and the EOP is responsible for differential pressure from the low-pressure to the high-pressure sections. Figure 5 shows a breakdown of the work for the oil pump system at this time. Figure 5 indicates that the MOP energy loss area, shown in Fig. 2, has decreased in size. When high pressure is demanded for belt clamping or the driving scenario necessitates sudden shift, it is not possible to maintain the allocation shown in Fig. 5, due to the EOP's limited discharge performance. Therefore, in driving scenarios in which the MOP's discharge performance exceeds that of the EOP, the bypass valve is opened and the MOP is exclusively responsible for hydraulic pressure supply. The system returns to the allocation of the oil pump work shown in Fig. 2.

Due to the series configuration, the EOP is not able to increase the discharge flow rate of the MOP. Since the theoretical discharge flow rate of the MOP remains unchanged from a conventional hydraulic circuit, there is no effect of friction reduction in the MOP unit.

Conventional EOP employed in idling stop systems (termed "I/S" below) do not realize the basic hydraulic characteristic or durability that were necessary

specifications for the development under discussion. An EOP was, therefore, newly developed.

It was necessary to consider the fact that EOP may become unable to operate due to malfunctioning of the power source, the harness employed for communications, or constituent electric parts such as the motor control circuits. Since the developed system adopts a configuration in which the bypass valve is switched by front-rear differential pressure, steady-state operation is possible even in the event of electric malfunctions. Therefore, even if the EOP fails to function, normal driving will not be impeded because the system will switch to supply from the MOP in isolation.

3.2. Selection of EOP Output Specifications

The following equation expresses the oil pump work when hydraulic pressure is supplied by a conventional MOP in isolation, as shown in Fig. 2:

$$W^* = \frac{P_H Q_H}{\eta_p} \quad (1)$$

The following equation expresses the oil pump work when hydraulic pressure is supplied by the EOP and the MOP, as shown in Fig. 5:

$$W = \frac{P_1 Q_H}{\eta_p} + \frac{(P_H - P_1) Q_A}{\eta_{sys} \eta_a} \quad (2)$$

The first term on the right-hand side is MOP work; η_p is pump efficiency. The second term is EOP work; η_{sys} is the system efficiency of the EOP system, and η_a is generation efficiency. Since the power necessary for driving the EOP is supplied by the engine, it is important to consider its generation efficiency. Given this, the degree of reduction in energy loss, ΔW , is expressed by the following equation:

$$\begin{aligned} \Delta W &= W^* - W \\ &= (P_H - P_1) \left(\frac{Q_H}{\eta_p} - \frac{Q_A}{\eta_{sys} \eta_a} \right) \end{aligned} \quad (3)$$

As Eq. (3) indicates, because $P_H - P_1 > 0$, when $Q_A / \eta_{sys} \eta_a \geq Q_H / \eta_p$, ΔW is negative. In other words, system efficiency η_{sys} has an important meaning in terms of making maximal use of the EOP.

System efficiency η_{sys} is given by the following equation:

$$\eta_{sys} = (P_H - P_1) * Q / (V * I) \quad (4)$$

Q: EOP discharge flow rate

V: EOP input voltage

I: Current consumed by EOP

As system efficiency η_{sys} is mainly dependent on the rated output of the EOP motor, the selection of motor output is an important factor.

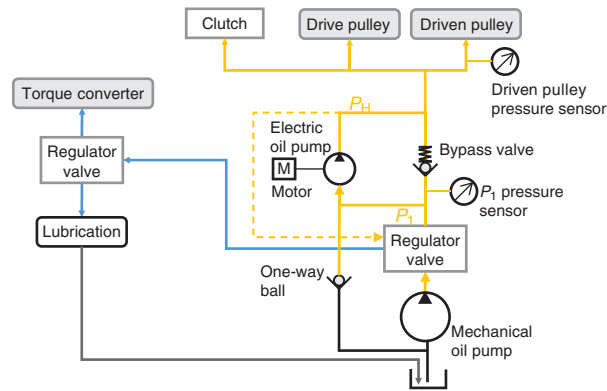


Fig. 4 Hydraulic circuit

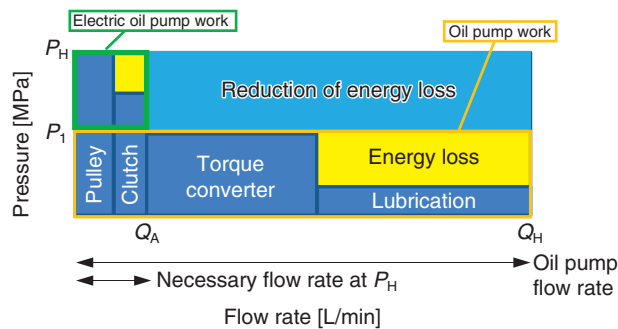


Fig. 5 Developed oil pump system

Figure 6 shows an example of the system efficiency of the EOP employed in the developed system. The horizontal axis shows the EOP discharge flow rate, and the vertical axis shows discharge pressure. The more the area is shaded red, the higher the system efficiency. The broken lines show equivalent output diagrams that multiply the horizontal and vertical axes. As Fig. 6 indicates, system efficiency is higher on the high-output side.

The higher the rated output of the motor employed in the EOP, the higher the pressure and flow rate that the state shown in Fig. 5 can be maintained. This means that energy loss in the MOP can be reduced across a wider range of driving conditions. However, as the motor's rated output increases, its mass increases and it becomes larger, with the result that it becomes unsuitable for vehicle fitting⁽⁸⁾. It was, therefore, desirable to select the lowest motor rated output for the wanted effect.

Figure 7 compares the degree of reduction of energy loss at vehicle takeoff for different EOP motor rated outputs. As this graph shows, motor rated output and the effect of motor rated output are non-linear characteristics.

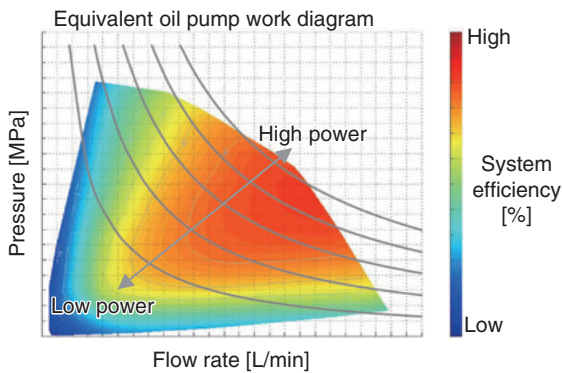


Fig. 6 System efficiency of electric oil pump

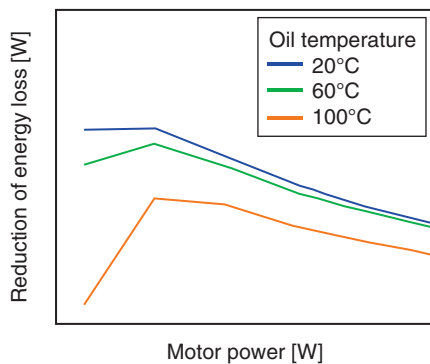


Fig. 7 Comparison of reduction of energy loss at vehicle takeoff for differing electric oil pump rated outputs

Figure 8 shows the results of a comparison, in desktop studies, of the degree of reduction of energy loss under conditions that combine a variety of driving states. At the highest motor rated output of 300 W, energy loss in the MOP was less than 100 W. This means that CVT efficiency is highest at a motor rated output of 300 W. However, because the power loss due to the operation of the EOP is highest at 300 W, the motor rated output that is preferable from the perspectives of low energy loss and good fuel efficiency as considered for the vehicle as a whole is 100 W.

Figure 9 shows the results of a comparison in desktop studies of the degree of energy loss reduction for different motor outputs rated in representative fuel consumption modes. These results show that for each of the fuel economy modes, which involve different driving patterns, a rated output of 100 W offers an adequate effect of energy loss reduction.

3.3. EOP Control

The EOP employed in the developed system receives command values for motor speed from a host CPU via CAN communications. The EOP applies control in accordance with the received motor speed command values.

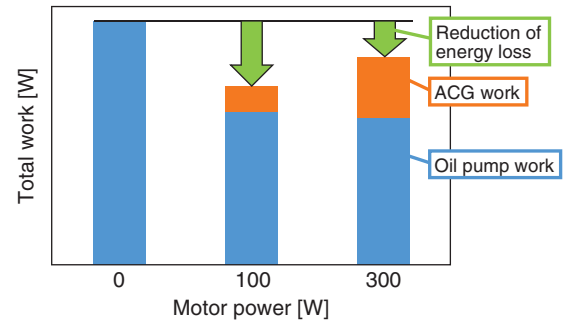


Fig. 8 Effect of reduction of energy loss

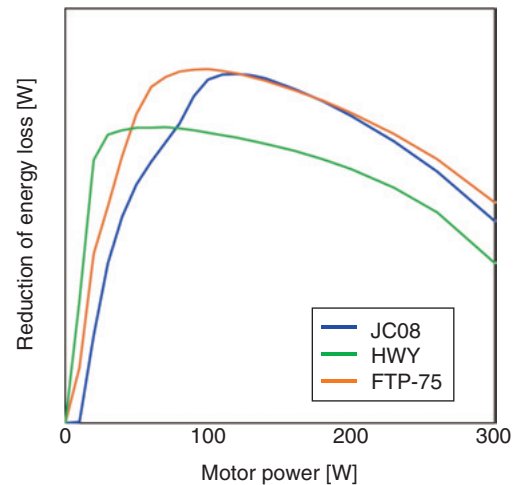


Fig. 9 Fuel economy improvement

Figure 10 shows the state transition diagram that represents the control concept for the developed system. Five states are defined; Driving and I/S are the basic states for EOP operation. In cases in which the system state transitions from Driving to a state in which it is not possible to reduce energy loss in the MOP, or the necessary hydraulic pressure or flow rate exceeds the performance of the EOP (state changes that would cause operational or other issues in different systems), the system changes state by reverting to Standby. The system does not return from Driving to Standby directly, but passes through Stop transition. This continues until the system has transitioned into a state in which the bypass valve is open.

Figure 11 shows the flow of calculation of the EOP motor speed command value in the Driving state. This control flow is positioned as subordinate to the control flow in which the CVT pulley clamping force is calculated, and employs the physical values for hydraulic pressure and the discharge flow rate necessary for shift determined by the governing flow as variables. It can be assumed that due to variations in the parts making up the hydraulic pressure

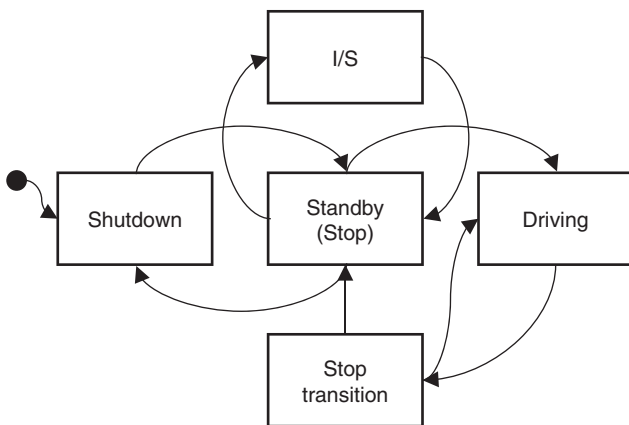


Fig. 10 State transition diagram

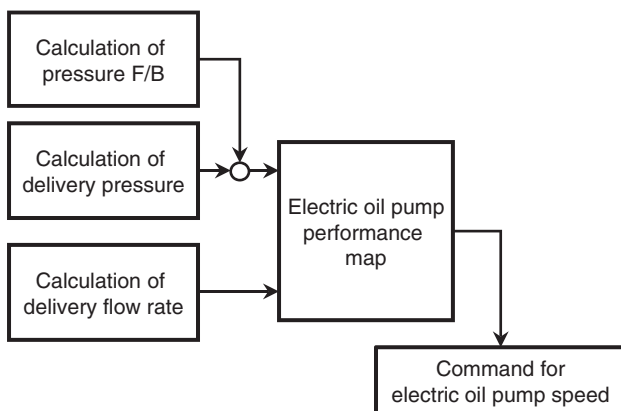


Fig. 11 Calculation of command value

control system, there will be occasions on which it is not possible to reduce pressure to the target MOP discharge pressure based on the command motor speed. In order to make it possible to also respond to these situations, the pressure difference between the P_1 pressure sensor value and the target pressure is employed as feedback pressure. The necessary variables are compared with the EOP performance map, and the motor speed command value is calculated.

In the Stop transition state, in response to the driving status, the EOP motor speed command value that will minimize fluctuations in hydraulic pressure when the bypass valve is open is calculated.

Figure 12 shows changes in hydraulic pressure when the EOP is in operation in the developed system. At point A, the EOP motor speed is increasing, and the discharge flow rate is also increasing. Since the bypass valve is open at this time, there is no change in pressure P_1 , the MOP discharge pressure. When motor speed reaches point B, the flow passing the bypass valve is also entirely passing the EOP side. As there is no front-rear pressure differential, the bypass valve is closed by spring force, and pressure P_1 begins to decline. When point C is reached, pressure P_1 becomes the same as the torque converter circuit pressure, and does not decline any further. In this driving state, the optimal value for the EOP motor speed is the speed at point C, at which the MOP discharge pressure ceases to decline, and any motor speed in excess of this would be undesirable from the perspective of increasing energy loss as generation proceeds.

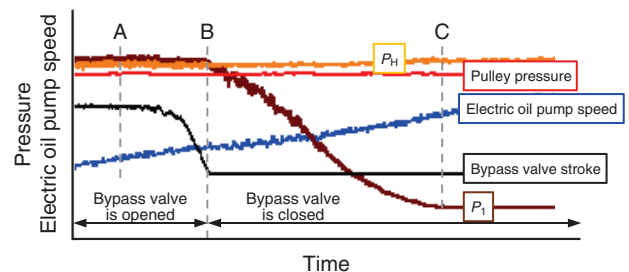


Fig. 12 Reduction in P_1 pressure

4. Effect in Boosting Performance

This section will discuss the performance increases against a conventional system obtained in an actual vehicle fitted with the developed system.

4.1. Transmission Efficiency

Figure 13 shows transmission efficiency by shift ratio during steady-state driving. Transmission efficiency is increased when the EOP is operating, and the average rate

of increase in efficiency under driving conditions within the normal operating range is 0.71%.

4.2. Current Consumption during I/S

The EOP employed for I/S in a conventional vehicle maintains a fixed motor speed using ON-OFF control; the system is configured to exhaust the portion that exceeds the flow rate that is necessary to maintain clutch pressure. While the necessary flow rate displays a positive correlation with oil temperature⁽⁹⁾, because the discharge flow rate of the EOP does not change, the current consumed by the motor has no relation to oil temperature, and is basically constant.

In the developed system, oil pump work has been optimized by employing motor speed commands in accordance with the necessary flow rate for each oil temperature, reducing current consumption. Figure 14 shows discharge flow rates for the EOP during I/S. Despite the fact that the output of the EOP motor employed in the developed system was high compared to existing motors for I/S, optimization of control has reduced the current consumed, contributing to the realization of increased fuel efficiency in an actual vehicle.

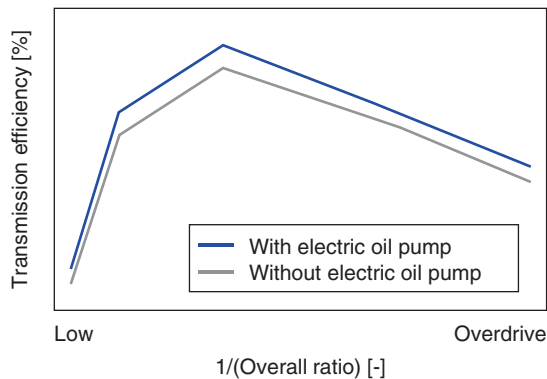


Fig. 13 Transmission efficiency

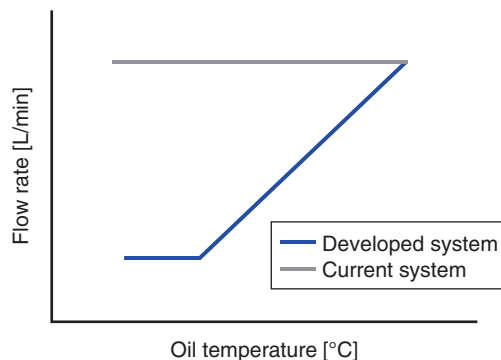


Fig. 14 Flow rate during Idling stop

4.3. Effect in Increasing Fuel Efficiency

Figure 15 shows the increase in fuel efficiency in NEDC mode in a vehicle fitted with the developed system. It was verified that vehicle total fuel efficiency was increased by 1.23% even when the increase in generation work for the operation of the EOP is considered.

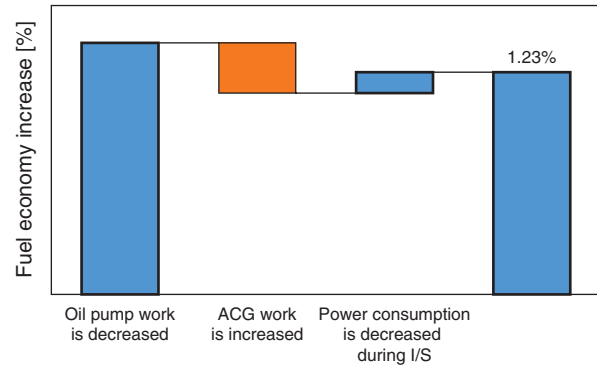


Fig. 15 Fuel efficiency increase (NEDC mode)

5. Conclusion

An oil pump system has been developed as a technology to boost fuel efficiency in CVTs for small vehicles. This development produced the following results:

- (1) A hydraulic circuit structure was developed that helps to ensure that driving performance is not impeded even if the EOP loses function.
- (2) The optimum motor rated output was selected, helping realize an average increase of 0.71% in transmission efficiency and an increase of 1.23% in fuel efficiency in NEDC mode.
- (3) An EOP control was developed that feeds back the MOP discharge pressure and is able to vary the EOP motor speed from suspension to optimum speed irrespective of variations in the constituent parts of the hydraulic pressure control system.

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