Approach for Improving Fuel Consumption of City Cranes

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Two control systems have been newly developed for fuel saving in hydraulic wheel cranes: namely, a one-way-clutch system and an advanced engine control system. The former allows one-way transmission of power only in the direction from the motor to the axle and require neither an actuator nor a controller to control the motor action. The advanced engine control system determines the engine rotation speed in response to partial stroke operations of the lever, preventing the unnecessary generation of engine power. It was found that fuel savings of 12% in driving and 20% in crane operation were achieved for the former and the latter systems, respectively, in the practical operation of the cranes.

Introduction

Energy-saving performance has become increasingly important with the rise of fuel costs and the environmental consciousness of the users. In the field of construction machinery, machine makers have been developing new models that feature lower energy consumption to meet new emissions regulations[1), 2)].

The wheel cranes manufactured by KOBELCO CRANES CO., LTD. mainly rely on hydraulic control systems to transmit power from their engines to actuators for traveling and working. A hydraulic control system has a pump and valves that control the behavior of the actuators, and significant power losses occur in various parts of the system. Therefore, minimizing these losses by optimizing the control system and mechanism is a key to improving the energy saving performance.

In 2008, KOBELCO CRANES CO., LTD. launched a wheel crane, the RK250-7. In 2011, this model underwent a minor change in which newly developed technologies were incorporated to further improve its energy saving performance. This paper describes the approach taken to improve the energy saving performance of the wheel crane and introduces those newly developed technologies.

1. Hydraulic systems in wheel cranes

1.1 Hydraulic system for traveling

The RK250-7 comprises a traveling system including a hydraulic static transmission (hereinafter denoted as "HST")[3), 4)]. Unlike general torque converters and mechanical transmissions, the HST unit transmits power hydraulically from an engine to a drive shaft.

A wheel crane comprises an upper slewing body having actuators for crane work/jobs and a lower traveling body containing a drive shaft. In a system that employs a torque converter or a mechanical transmission, almost all of the connections, from the engine to the output shaft, are made mechanically. Therefore, the engine must be placed in the lower body, and the actuators, located on the upper body, must receive the hydraulic power via a center swivel joint. The RK250-7, on the other hand, has its engine mounted on the upper slewing body to achieve downsizing while maintaining the lifting capacity of the crane. The power is transmitted from the engine to the drive shaft via the HST unit, allowing a high degree of freedom in the equipment layout.

As depicted in **Fig. 1**(a), the RK250-7 has three hydraulic systems: namely, a traveling system, crane-operating system and auxiliary system. In the traveling system (Fig. 1(b)), the engine drives a variable displacement pump via a power divider. The pump discharges oil to actuate a variable displacement motor that rotates an axle and tires to run the vehicle. The hydraulic circuit is closed such that the oil discharged from the motor is returned to the pump.

When the pump is operated at a constant discharge rate, an increase in the motor displacement increases the driving torque, while a decrease in the motor displacement increases the traveling speed. Increasing the pumping rate, on the other hand, increases the power input to the HST unit. In other words, the HST unit exploits the displacement controls of the pump and motor to control the increase and decrease of the driving torque and traveling speed.

1.2 Hydraulic system for crane operation

Fig.1(c) depicts the hydraulic system for operating the crane. The engine drives a variable displacement pump via a power divider. The oil discharged from the pump is delivered, via control valves, to a motor for winching up and down, a cylinder for telescoping the boom and a cylinder for hoisting the boom. The control valves select actuators to be supplied with hydraulic oil and control the moving direction and

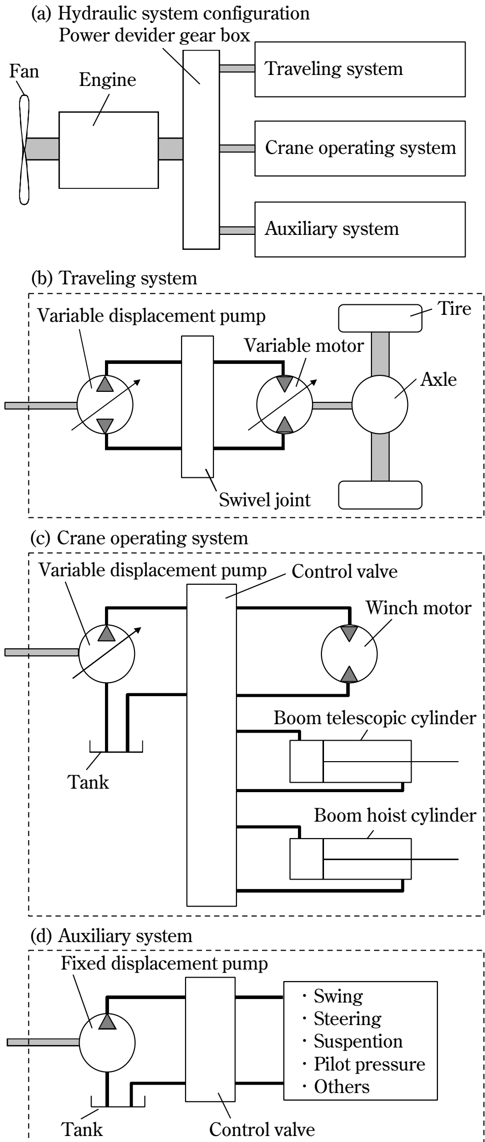

Fig. 1 Hydraulic system configuration of wheel crane

speed of the actuators. Unlike the oil in the HST unit, the oil discharged from the actuators passes through the control valves and returns to a hydraulic oil reservoir.

2. Predicting driving fuel consumption by simulation

Prediction by simulation is useful when adopting a new system and it also serves to shorten the time taken by development and specification changes. KOBELCO CRANES CO., LTD. developed a technology for predicting driving fuel consumption. The technology was, however, only applicable to simple running conditions, such as constant-speed driving. In reality, the running conditions are more complex, involving constant-speed driving, acceleration driving, inertia driving and uphill driving. This called for a simulation technology to predict the driving fuel consumption, using data measured on actual runs. Therefore, a new technique has been developed to calculate the fuel consumption of an engine on the basis of the load imposed on, and the loss caused by, each system device during actual traveling.

As shown in **Fig. 2**, actual measurement data for such factors as vehicle speed and road inclination are used to calculate the running resistance, and to this figure is added the loss caused by each system device. The resulting sum is used to determine the revolutions per minute (rpm) and torque required for the engine and thus to estimate the fuel consumption. Variable smoothing, corresponding to the traveling conditions, was performed on the traveling data to reduce noise, which has enabled the accurate analysis of fuel consumption. The losses caused by the pump and motor were determined by a recursive algorithm based on the Newton-Raphson method. For a pump, the following state equations, (1) to (3), hold:

$$W_p = W_{EG} - W_{add} \quad (1)$$

$$Q_p P_p = W_p \eta_m \quad (2)$$

$$Q_p = q_p N_p \eta_v \quad (3),$$

wherein W_p, W_{EG}, and W_{add} represent the power of the pump, engine and auxiliary pump, respectively; q_p is the pump displacement; P_p is the pump pressure; Q_p is the pumping rate; N_p is the rpm of the pump; and η_v and η_m are, respectively, the volume efficiency and mechanical efficiency of the pump. Calculations

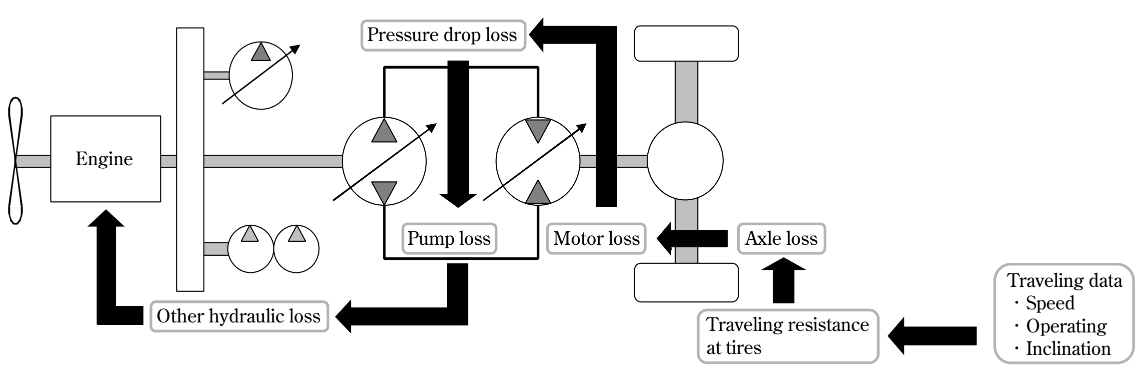

Fig. 2 Flow of engine load calculation

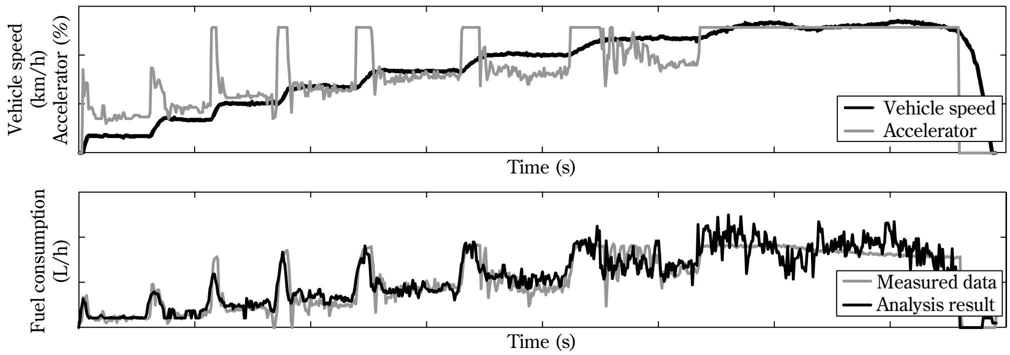

Fig. 3 Comparison of measured data and simulation

were conducted using the pumping rate Q_p as an evaluation function and revolution ω_p as a variable.

To verify the accuracy of this simulation model, the analysis results were compared with the data obtained from an actually running machine, as shown in **Fig. 3**. The traveling data in this figure were obtained for speed increased in a stepwise manner. The results indicate that the analysis accurately reproduces the fuel consumption during acceleration driving and constant-speed driving. The total fuel consumption deduced by the analysis deviates less than 5% from the actual value, verifying the validity of the technique.

3. Improvement of fuel consumption during driving

3.1 Pump displacement control [5)]

During travel, the engine drives not only the pump for the HST unit, but also the pumps and auxiliary pumps for cooling hydraulic oil, for assisting steering and for providing pilot pressure. These pumps are designed with the minimum possible capacities and run unloaded when they are not in use. However, power loss is inevitably caused by the pressure loss of oil as it passes the control valves and piping. **Fig. 4** shows the pressures of unloaded pumps rotating at various rpms. As shown in this figure, the pump pressure increases with an increasing rpm, which raises the burden on the pump. In addition, as shown in **Fig. 5**, the fuel efficiency of an engine is highest when the engine rotates slower than its rated rpm. Therefore, the fuel consumption is effectively reduced by decreasing the rpm. In consideration of these characteristics, a pump displacement control has been developed as described below.

For the pump in the traveling system, a displacement command value is determined for a given rpm according to a function as shown in **Fig. 6**(A). Here, the displacement command value increases with an increasing engine rpm. This is

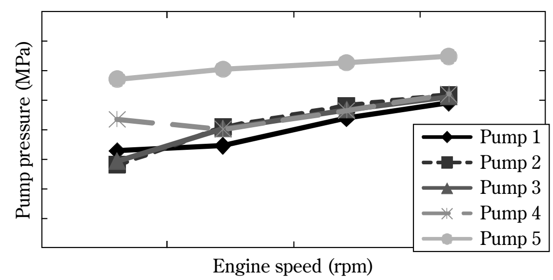

Fig. 4 Unloaded pump pressure

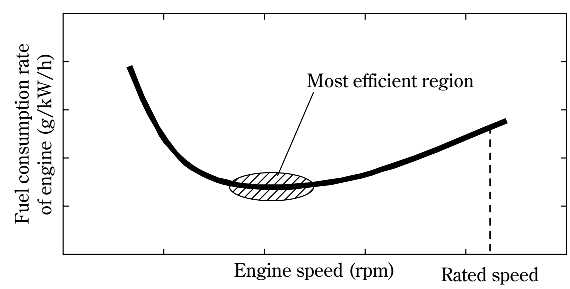

Fig. 5 Engine specific fuel consumption

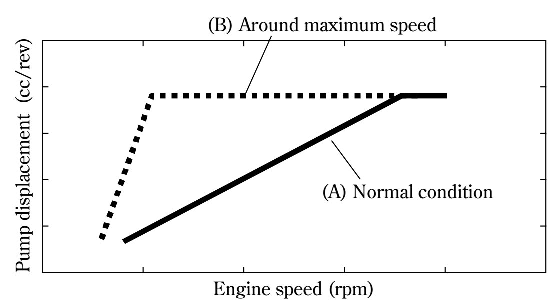

Fig. 6 Pump displacement control

because a large pump displacement at the time of engine start may cause an increased torque to be abruptly imposed on the engine, destabilizing the initial revolution and stalling the engine.

An engine generates increasingly greater power as its rpm increases. When full power is required during, for example, acceleration and uphill-traveling, the engine must operate at a high rpm. On the other hand, traveling on level ground at a constant speed imposes a small load and requires less power, and the engine can run at a low rpm.

The engine rpm, N_E, at a speed V is given by Equation (4).

$$N_E = \frac{q_m \varepsilon_{axle} \varepsilon_{pd}}{q_p \eta_p \eta_m} V \quad \cdots\cdots \quad (4),$$

wherein ε_{axle} is the reduction ratio of the axle, ε_{pd} is the reduction ratio of the power divider, q_m is the motor displacement, and η_m and η_p are the efficiencies of the motor and pump, respectively. Equation (4) indicates that decreasing the motor displacement and/or increasing the pump displacement decreases the engine rpm. However, if the pump displacement is controlled according to Fig. 6(A), the pump displacement decreases with the decreasing engine rpm, and the engine rpm may not be sufficiently decreased. Therefore, when the vehicle runs near or at the maximum speed, the pump displacement is controlled according to Fig. 6(B) so as to decrease the engine rpm.

The above control has been adapted not only for speed, but also for acceleration. A displacement control similar to Fig. 6(B) is applied to conditions such as inertia driving and low-speed driving that require a small amount of driving power. The engine rpm has thus been optimized for a wide range of speeds.

Fig. 7 shows the changes in vehicle speed and engine rpm during a test in which the vehicle was accelerated to its maximum speed. The pump displacement was controlled under two different conditions, i.e., the conventional condition (condition 1) and the condition depicted by dotted line (B) in Fig. 6 (condition 2). Although both conditions 1 and 2 yield almost the same vehicle speed, condition 2 resulted in a lower engine rpm for the maximum vehicle speed, which improved the momentary fuel consumption by 11%, verifying the validity of the newly developed control.

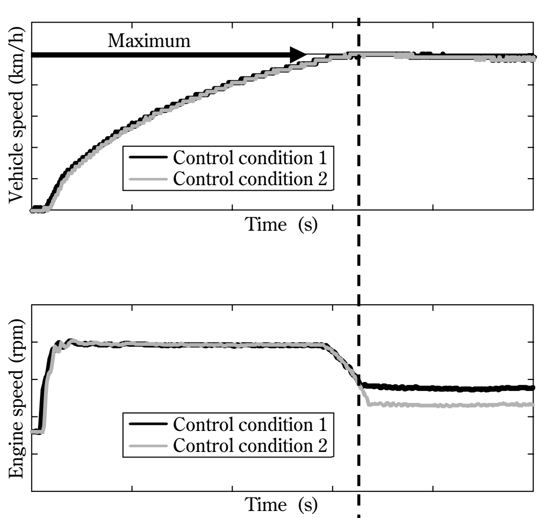

Fig. 7 Effect of engine speed reduction

3.2 One-way clutch system [6)]

A heavy vehicle such as a wheel crane requires a large torque during acceleration and uphill driving. Because the output torque of a motor increases proportionally to the motor displacement, an actual machine must be driven by a large displacement motor, which causes difficulty in mounting. Therefore, an HST unit includes more than one motor. On the other hand, low-load driving (e.g., constant-speed driving) requires only a small motor torque. Thus, a certain amount of energy can be saved by temporarily unloading at least one motor; however, as shown in **Fig. 8**, a loss torque remains even when the motor displacement is set to zero. Furthermore, resistance is caused by the gears rotating with the motor in the mechanical power combiner.

To reduce the idling loss of the motors that are not in use, we developed a system comprising a one-way clutch (hereinafter referred to as "OWC") on the motor axis connected to the mechanical power combiner (**Fig. 9**). The OWC transmits power from motor 2 only in the direction of the mechanical power combiner and prevents any power transmission in the reverse direction. When the displacement of motor 2 is turned to zero during running and no driving force is generated, the clutch is disconnected and the motor stops rotating. For uphill driving and acceleration, the clutch is coupled automatically to drive motor 2 and the power is transmitted in

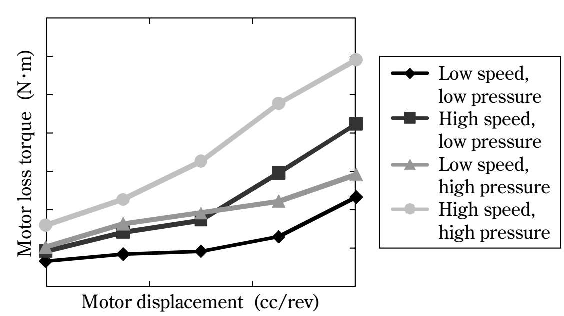

Fig. 8 Motor leak and mechanical loss

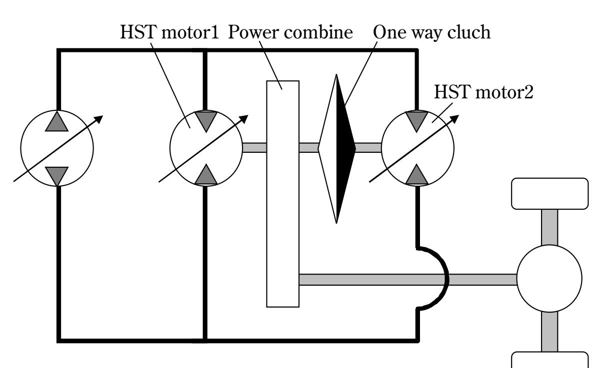

Fig. 9 OWC system configuration

combination with the power from motor 1. Because the OWC requires neither an actuator, nor a control pressure, nor a signal, a conventional motor mechanism can be adapted for this power transmission system with minimum modification.

A bench test was conducted to verify the effect of the OWC. The result is shown in **Fig.10**. A motor was rotated with zero displacement at different rpms, and the drag torque was measured on the motor axis. The result indicates that the loss torque is decreased to approximately one third in the vicinity of the maximum speed.

The effect of loss reduction by the OWC system was incorporated into the simulation model discussed in Section 2 to analyze the fuel consumption during driving. The result is shown in **Fig.11**. Due to the characteristics of the device, the OWC has no effect until the motor displacement reaches zero, and no improvement is observed in fuel consumption in the low speed region. However, once the high speed region is reached, the motor with zero displacement stops, decreasing the loss and improving fuel consumption. According to a test pattern simulating city driving, the fuel consumption has been found to improve by about 15%. An actual running test using a prototype vehicle equipped with the OWC resulted in an improvement of 12%, verifying the effectiveness of the system.

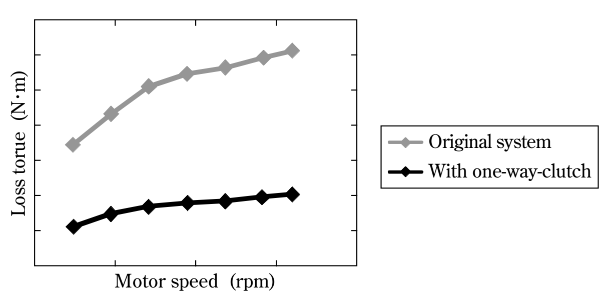

Fig.10 Shaft resistance in motor

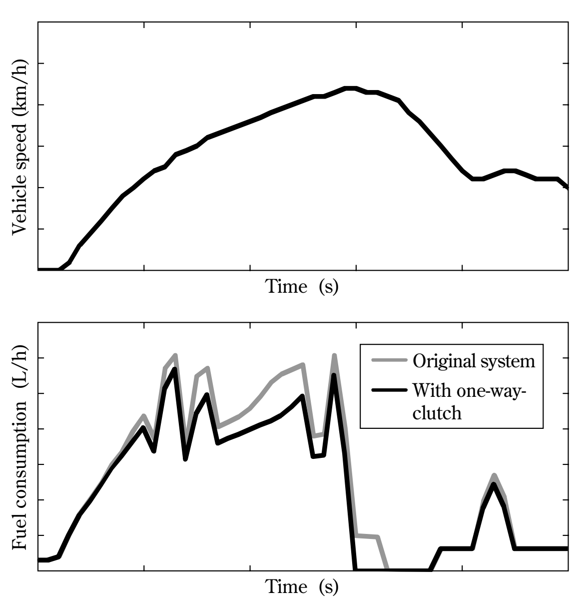

Fig.11 Simulation of OWC system

4. Improvement of fuel consumption during crane work

4.1 Positive control

During crane work, the speeds of actuators are controlled by control valves. The flow of oil to an actuator is determined by the stroke of a control lever. In a case such as that shown by the curve (C) in **Fig.12**, excess oil, which is discharged from the pump but not utilized for work, is returned to the reservoir. In the case of the conventional pump control (shown by the line of dashes (A) in Fig.12), the pump displacement is constant regardless of the lever stroke, causing a significant loss in the region where the lever stroke is small. A positive control (polygonal line (B) in Fig.12) increases the pump displacement according to the lever stroke, while reducing the loss.

4.2 Lever-sensitive control of engine revolution

Crane work involves the operation of control levers and an accelerator pedal to control the actuators' speeds. The acceleration is controlled according to the type of work and is usually fixed during a job. When a job is performed with full acceleration, the engine is maintained at its maximum rate of revolution regardless of the lever stroke. However, as described in Section 3.1, increasing the engine rpm increases the load imposed on the auxiliary pump, which can deteriorate the efficiency of the engine. Crane work, in particular, consumes a relatively small amount of power compared with traveling, and increased loss has a more significant impact on the fuel consumption.

With this background, a lever-sensitive engine

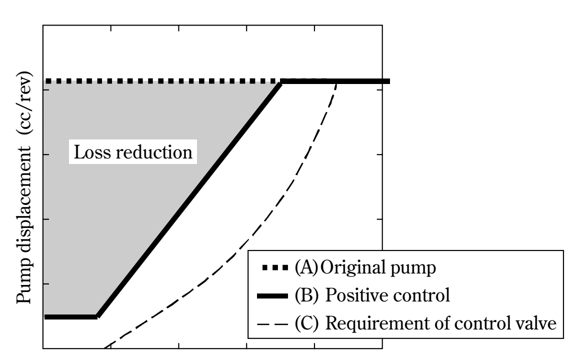

Fig.12 Pump displacement in positive control

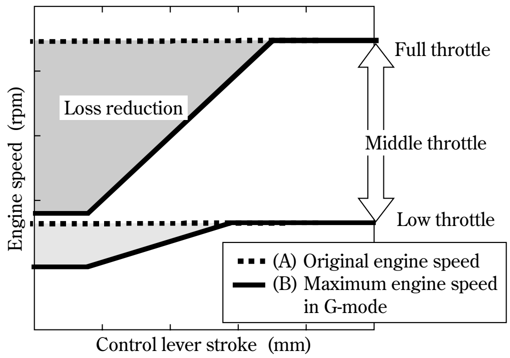

Fig.13 Engine speed in G-mode

revolution control, "G mode" (**Fig.13**), has been newly developed. The G-mode is designed such that the engine rpm can be controlled by small lever strokes and is useful for jobs that do not require speed. Conventionally, the engine rpm has been controlled by an accelerator pedal alone. The newly developed technology enables the levers to control the engine rpm. More specifically, the engine rpm is set either to the allowable rate of revolution corresponding to the lever stroke, or to the rate of revolution called for by the accelerator work, whichever is smaller. This enables the engine rpm to be reduced frequently in coordination with the lever operation, even during full acceleration. Another feature is that the operator can easily control and change the work speed at will, from low to high, simply by operating a lever without using the accelerator pedal.

Fig.14 shows the changes in engine rpm and fuel consumption during actual crane work (combining the operations of winching up/down, moving uphill/downhill and slewing). In this test, jobs other than idling were performed with full acceleration. Job (A) was performed with the lever in the full stroke position. In this case, the engine rpm is comparable with that of the conventional control. On the other hand, jobs (B) and (C) were performed with the levers in partial stroke positions. The result confirms that the engine rpm is decreased significantly by the G mode, and fuel consumption is improved by about 20%.

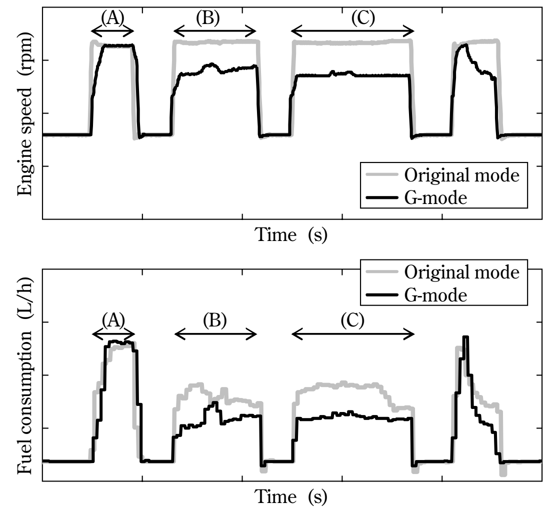

Fig.14 Engine speed and fuel consumption in G-mode

Conclusions

A new technology has been developed to control pump displacement for a traveling system to improve the fuel consumption of a wheel crane. A one-way clutch (OWC) system has also been developed to reduce motor loss, resulting in a 12% reduction in fuel consumption during driving.

On the other hand, a positive control has been introduced to improve fuel consumption during crane work. We have developed a lever-sensitive engine revolution control, "G-mode", that makes it possible to lower the engine rpm with the lever in a partial stroke position, reducing fuel consumption during crane work by 20%. We will continue to strive to improve the fuel consumption.

References

1) Kagoshima, et.al. *R&D KOBE STEEL ENGINEERING REPORTS.* 2007, Vol.57, No.1, p.66-69.
2) H. Shimokakiuchi. *Kensetsu no Sekou Kikaku.* 2007, No.683, p.37-40.
3) M. Umeda. *Hydraulics & pneumatics.* 1999, Vol.38, No. 5, p. 24-26.
4) S. Abe. *Construction machinery and equipment.* 2011, Vol.47, No.1, p.38-41.
5) *Jpn. Patent.* No.4589979.
6) *Jpn. Unexamined Patent.* No.2011-84094.