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Performance Evaluation of a Driving Power Transmission System for 50 kW Narrow Tractors

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Abstract

Purpose: The development of compact tractors that can be used in dry fields, greenhouses, and orchards for pest control, weeding, transportation, and harvesting is necessary. The development and performance evaluation of power transmission units are very important when it comes to tractor development. This study evaluates the performance of a driving power transmission unit of a 50 kW multi-purpose narrow tractor. **Methods:** The performance of the transmission and forward-reverse clutch, which are the main components of the driving power transmission unit of multi-purpose narrow tractors, was evaluated herein. The transmission performance was evaluated in terms of power transmission efficiency, noise, and axle load, while the forward-reverse clutch performance was evaluated in terms of durability. The transmission's power transmission efficiency accounts for the measurement of transmission losses, which occur in the transmission's gear, bearing, and oil seal. The motor's power was input in the transmission's input shaft. The rotational speed and torque were measured in the final output shaft. The noise was measured at each speed level after installing a microphone on the left, right, and upper sides. The axle load test was performed through a continuous equilibrium load test, in which a constant load was continuously applied. The forward-reverse clutch performance was calculated using the engine torque to axle torque ratio with the assembled engine and transmission. **Results:** The loss of power in the transmission efficiency test of the driving power unit was 6.0-9.7 kW based on all gear steps. This loss of horsepower was equal to 11-18% of the input power (52 kW). The transmission efficiency of the driving power unit was 81.5-89.0%. The noise of the driving power unit was 50-57 dB at 800 rpm, 70-77 dB at 1600 rpm, and 76-83 dB at 2400 rpm. The axle load test verified that the input torque and axle revolutions were constant. The results of the forward-reverse clutch performance test revealed that hydraulic pressure and torque changes were stably maintained when moving forward or backward, and its operation met the hydraulic design standards. **Conclusions:** When comprehensively examined, these research results were similar to the main driving power transmission systems from USA and Japan in terms of performance. Based on these results, tractor prototypes are expected to be created and supplied to farmhouses after going through sufficient in-situ adaptability tests.

Keywords: Agricultural tractor, Driving power transmission, Performance test, Efficiency, Durability

Introduction

Although the domestic agricultural machinery market maintains a production of 2.4 trillion won, unfavorable

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conditions, such as shrinkage of the domestic market and weakening of the competitiveness of domestic products, are continuously increasing. Furthermore, a change is required because of the expansion of the agricultural product market and the aging of the farming population (KAMICO and KSAM, 2016). The decrease in the number of farmhouses and farmers has also affected the cultivation acreage per capita (rice paddies and dry-fields). The cultivation acreage per capita increased from 6,434.5 m^2 in 1970 to 13,799.8 m^2 in 2013. The aging ratio of the farming population (more than 65 years) increased by 1.9% over the previous year, reaching 40.3%, and will continue to increase (SK, 2016). With the goal of achieving a mechanization ratio of 60% by 2017, agricultural mechanization policies have recently focused on the agricultural mechanization of dry-fields because of factors, such as increasing cultivation acreage per capita, aging of the farming population, and opening of agricultural product markets (e.g., FTA) (KAMICO and KSAM, 2016).

Small-scale farm works that lack mechanization, such as dry-field farming, require precision and small sizes. Small-sized machinery is being widely used in dry-fields because the land area is small (Chang et al., 2010). However, the domestic tractor market lacks mechanization of small models because of a technology development oriented toward enlargement and stability. Small farm works, such as fruit farming, present a low mechanization ratio of 30%. Fruit farming particularly works with a high labor input time, including harvesting (17.2%), fruit thinning (14.6%), pruning (11.8%), and pest control (11.5%). Therefore, mechanization for these works is urgent (Capalbo and Denny, 1986). The development of 50 kW multi-purpose narrow tractors that can work on dry fields, greenhouses, and orchards is necessary. Power transmission systems, such as driving power, PTO (Power Take-Off) power, and three-point hitch hydraulic power, are the core of tractor development (Lee, 1990; Jung, 2013; Ha, 2015).

Design of agricultural tractors has changed over times (Guzzomi and Rondelli, 2013) and significant technological innovations have been applied to agricultural tractor industries to satisfy farmers need (Cavallo et al., 2014a and 2014b; Kabir et al., 2014). Chung et al. (2016) developed and evaluated the performance of a hydraulic power transmission system for the 3-point hitch of a 50 kW narrow tractor targeting spraying, weeding, and transportation operations. Being off-road vehicle, power transmission system of tractor has a great importance compared to general vehicle. Several studies have been carried out to improve the efficiency of power transmission system of agricultural tractors to suit different farm works. An integrated engine hydro-mechanical transmission control algorithm was developed for improving tractor efficiency (e.g., Choi et al., 2013). Simulation results showed that the integrated engine hydro-mechanical transmission could improve the fuel economy by 7.5% compared with the existing engine optimal operating line control (Ahn et al., 2015). A network model of hydro-mechanical transmission (HMT) was developed for tractor to estimate the power performance and efficiency of HMT and also can be used for designing an HMT configuration (Park et al., 2016a). Raikwar et al. (2015) presented a modeling and simulation technique of a power shuttle transmission system for an agricultural tractor that could be beneficial in designing different types of power transmission systems. Lee et al. (2016) simulated the fatigue life of the PTO gear according to the operating point of the tractor engine and found minimum fatigue life of the PTO gear as 19.61 hours at 70% of the maximum engine power. Improvement in the design of a mechanical transmission was analyzed by Kim et al. (2016) to increase the gear strength and to reduce the gear weight for a small 4.8 kW tracked agricultural transporter. Park et al. (2016b) developed a simulation model to investigate the effect of pinhole position errors in the planet carrier of a planetary gear set (PGS) on load sharing among the planet gears in the hydromechanical transmission (HMT) system of an agricultural tractor.

Power transmission systems are composed of various parts, among which, the performance of transmissions and forward-reverse clutches are very important. Hence, various performance tests regarding power transmission efficiency, noise, axle load, and durability are conducted. Kim et al. (2009) constructed a test device composed of motor, transmission, and dynamometer to test the life cycle of tractor transmissions. The motor used was a 74.6 kW DC motor, while the dynamometer was a pneumatic disc-brake type with a maximum applicable torque of 15 kN·m. It was set such that the desired torque could be controlled using a pneumatic regulator. The life cycle test was conducted using the dynamometer to apply the desired torque to the output shaft while maintaining the rotational speed of the DC motor at the rated revolutions of the tractor's input shaft. The speed step was set to have the biggest traction load, and the life cycle test torque was set to be 1.2 times the rated torque, which was the engine's maximum torque.

Noise is the very significant hazard in the working environment of agriculture and also a source of occupational health problems (Nelson et al., 2005), therefore, in the recent decades, efforts have been made to reduce exposure of noise for better performance of tractor operator and to enhance the competitiveness of domestic tractor industry (Kabir, 2015). Transmission system is one of the main sources of noise (Kechayov and Trifonov, 2003). Bilski (2013) assessed the audible noise levels ranged on an average 68.2 to 83.8 dB-A (A-weighted) and infrasonic noise levels from 87.3 to 111.3 dB-G (G-weighted) for 20 types of modern tractors during typical working conditions. International Labor Organization (ILO) accepted 85 dB (A) as warning limit as over this limit have effects such as temporary or permanent hearing disabilities and 90 dB (A) as a danger limit for continuous work for 8 h (ILO, 1987). OECD code 2 (OECD, 2018) and ISO 5131 (ISO, 2015) stated the requirements and noise measurement techniques at the operator's position of an agricultural tractor where, at least 2 measurements should be made with no-load condition and at least 3 measurements for ISO 5131 with no load or with a load applied to the drawbar.

For the driver's sake, Harris and Jensen (1964) augmented a forward-reverse power shift with a reverse speed of approximately 5.6 km/h. Yun (1998) established a mathematical model on a clutch-independent hydraulic control system that can increase the shift quality and analyzed its static and dynamic characteristics. By performing control strategies to increase the shift quality through the hydraulic control system and the integrated simulation of the vehicle, the shift transient torque was confirmed to decrease. Lee et al. (1996) proposed a new pressure regulator to improve the shifting characteristics of powershifts in construction vehicles. They also predicted and confirmed its performance through simulation and test. They claimed that the shifting characteristics can be effectively controlled by controlling the characteristics of the hydraulic pressure supplied to the clutch. Meanwhile, Cho (2003) presented a typical hydraulic pressure curve to control the pressure modulation of forward-reverse clutches of tractor powershifts. Regarding pressure modulation, the clutch transmits power as the frictional force increases as result of the piston applying a vertical force on the clutch pack. However, the power is cut off if this happens too late. Therefore, the power must initially be set, such that it is transmitted at a low pressure. Modulation normally starts at approximately 15% of the system's pressure and ends at 80%. After this, power is transmitted after the system's maximum pressure is applied to the piston.

The present research was conducted to evaluate the performance of driving power transmission systems for 50 kW multi-purpose narrow tractors designed and created in previous research to be used in dry-fields, greenhouses, and orchards (Chang et al., 2010; Ha, 2015).

Materials and Methods

Prototype of the driving power transmission part

The driving power transmission part used in this research was developed in a precedent research (Chang et al., 2010; Ha, 2015). Based on market survey of competitive brands, target transmission efficiency and cabin noise of the 50 kW narrow tractors were set as 85% and 80 dB, respectively, which were world-best levels. The power flow of the driving axle power transmission part was such that the power generated in the engine was transmitted to the forward-reverse clutch located at the front of the power transmission device. The forward-reverse clutch allowed shifting while driving without stopping the tractor. The transmitted power was determined as forward or reverse direction by the forward-reverse clutch shift. Subsequently, the step was determined after being transmitted to the main shift. The main shift was composed of a synchronizer, which allowed a shift through speed synchronization below a certain speed. Power was transmitted to the auxiliary shift once the speed step is determined in the main shift. In the auxiliary shift, shifting was performed with the tractor stopped using a color shift. Power was then transmitted from the auxiliary shift to the differential, and finally to the rear axle. Figure 1 shows the prototype photo of the driving power transmission used in the study. Important issues of the driving transmission are transmission efficiency and noise, axle load durability, and clutch inertia, to secure global competitiveness.

Transmission efficiency contributes the output power and fuel consumption, and the cabin noise level should be maintained at the levels lower than drivers' safety criteria. Axle load durability and clutch inertia are important to obtain longer life time under working conditions.

Performance test method

Transmission efficiency

The transmission efficiency test measured the efficiency of the power transmitted from the engine to the axle, which was the final output shaft. This test measured the power losses occurring in the transmission gear, bearing, and oil seal, excluding those occurring in the cooling fan, hydraulic pump, and flywheel housing during the engine matching process. In the transmission efficiency test, the rotational speed and the torque were measured in the final output axle for the motor's power in the transmission input axle without assembling the engine into the transmission, but by using a separate motor as the power source. All the shift-steps from the lowest to the highest were tested herein. The transmission efficiency was obtained by calculating the power losses inside the transmission.

The transmission's horsepower loss was calculated as shown in Eq. (1):

$$H_{loss} = T \times n/7160 \quad (1)$$

Where, H_{loss} is the power loss (kW); T is the transmission's torque (Nm); and n is the number of revolutions per minute (rpm).

The transmission efficiency was calculated as shown in Eq. (2) using the power loss calculated earlier.

$$E_{trans} = (1 - H_{loss}/H_{input}) \times 100 \quad (2)$$

Where, E_{trans} is the transmission efficiency (%); H_{loss} is power loss (kW); and H_{input} is the power input (kW).

Figure 2, and Tables 1 and 2 present the images and the main specifications of the motor and torque-measuring instrument used in the transmission efficiency test.

Figure 1. Photo of the transmission system used in the study (Chang et al., 2010; Ha, 2015).

Figure 2. Photos of the motor (left) and the torque transducer (right) used in the transmission efficiency test.

Table 1. Specifications of the motor used in the transmission efficiency test

Item	Specifications
Model	KAC-100
Capacity	100 kW
Torque	150 N·m
Rotational speed	8,000 rpm
Manufacturer	SIEMENS

Table 2. Specifications of the torque transducer used in the transmission efficiency test

Item	Specifications
Model	YDR-500 K
Capacity	4903 N·m
Rated output	1.5 mV/V
Non-linearity	0.09% rated output
Hysteresis	0.08% R.O.
Manufacturer	SETech Co., Ltd

Table 3. Specifications of the noise tester

Item	Specifications
Model	MK II
Manufacturer	MULLER-BBM
Maximum data rate, resolution	204.8 kSa/s, 24-bit
Number of channels	16–128

Transmission noise

The transmission noise was measured with a tester (Table 3) at all shift-steps from the lowest to the highest without assembling the transmission into the engine, but by using a separate motor as a power source. Microphones were installed on the transmission's left side, right side, upper side, and rear PTO output shaft for noise measurement at each point. The transmission noise must be maintained below 80 dB, which was lower than the noise standard inside the cabin. The transmission's noise was measured at step 3 of the auxiliary shift and step 4 of the main shift, which were the maximum speed steps at which driving is normally performed. The test was conducted under fixed auxiliary shift and main shift conditions while changing the input number of revolutions. The transmission was operated not with an engine, but with a motor; hence, the test was conducted at three different input rpm conditions (i.e., 800, 1600, and 2400 rpm).

Axle load durability

The axle load test was conducted to ensure reliability by performing a durability test and verifying the strength of the transmission's driving system gear, axial flow, differential gear, and rear axle. A continuous uniform load test was conducted. During the test, the transmission's rear axle was connected to the tester, and load was applied to the transmission by braking with the brake in the rear axle. The load applied to the rear axle was determined based on the vehicle weight, vehicle load sharing ratio, maximum slip torque and deceleration ratio.

The maximum slip torque was calculated as follows using Eq. (3):

$$T_S = W_{tractor} \times C_{rear} \times R_{rear} \times \mu \tag{3}$$

Where, T_s is the slip torque (N·m); $W_{tractor}$ is the vehicle weight (N); C_{rear} is the rear wheel load sharing ratio (0.6); R_{rear} is the rear tire effective radius (m); and μ is the friction coefficient.

During the axle load test, the temperature of each part of the engine, transmission oil temperature, axle's rotational speed, and axle torque were measured to check for transmission failure. The test was stopped whenever the engine coolant temperature exceeded 100℃ or whenever the transmission oil temperature exceeded 90℃. Cooling was then performed before checking for any abnormal phenomenon.

The continuous uniform load test was conducted after installing an engine-mounted transmission on the axle load tester and setting the speed step, in which the engine torque and the axle torque became 1:1. No gear failure should occur during 400 h of test when applying a uniform load to both left and right sides with the differential locking device turned on. Table 4 shows the main specifications of the dynamometer used in the axle load test.

Table 4. Specifications of the chassis dynamometer

Item	Specifications
Model	TrCD 224
Capacity	402.7 kW
Torque	25,000 N·m
Max. slip speed	715 rpm
Max. air pressure	550 kPa
Max. coolant pressure	270 kPa
Manufacturer	DTS Co.

Forward-reverse clutch inertia

Ensuring the durability of the forward-reverse clutch is the top priority to ensure the reliability of the transmission's forward-reverse system. The clutch is one of the main components of power transmission systems. In addition to transmitting the engine power to the next stage of the power transmission system and providing a driving force to the tractor, it also reduces torque variations, which inevitably occur in the engine's intermittent combustion process and in the inertia resistance caused by the reciprocating motion of parts constituting the engine. Problems related to power transmission, which is the main role of the clutch, include wear caused by repetitive use, power transmission performance degradation caused by fatigue, and power transmission performance degradation caused by the frictional changes from heating in harsh use conditions.

The forward-reverse clutch performance and durability test was conducted on a test bench with the transmission and engine assembled. The safety factor was calculated using the ratio of the engine torque to the axle torque. In the transmission installed on the test bench, a circular plate corresponding to the inertia moment of the axle was mounted on the final deceleration shaft to resemble the vehicle weight. The inertia moment of the axle was calculated using Eq. (4):

$$M_{shaft} = (W_{tractor} \times R_{rear}^2)/i^2 \tag{4}$$

Where, M_{shaft} is the inertia moment of the axle (N·m s^2); $W_{tractor}$ is the vehicle weight (N); R_{rear} is the rear tire effective radius (m); i is the deceleration ratio of the final deceleration shaft.

The shift step in the forward-reverse clutch durability test was set to be that in which the ratio of the engine torque to the axle torque became 1:1. The engine's number of revolutions was fixed as the rated number of revolutions. Considering forward to reverse and reverse to forward as 1 cycle (2 minutes), 5,000 cycles (167 h) were repeated (Figure 3).

The testing device consisted of an engine, transmission, flywheel, pressure gauge, and torque meter (Table 5). As in the axle load test, the temperature of each part of the engine, the temperature of the transmission oil, the rotational speed of the axle, the axle's torque, and the forward-reverse pressure were measured in the forward-reverse clutch test to check for transmission failure. The test was terminated whenever the temperature of the engine coolant exceeded 100℃ or whenever the temperature of the transmission's oil exceeded 90℃. Cooling was then performed before checking for any abnormal phenomena.

Considering shifting from forward to reverse, then from reverse to forward as 1 cycle (2 min), 5,000 cycles (more than 150 h) were repeated in the forward-reverse clutch inertia test. The engine's number of revolutions was maintained at its rated value of 2,300 rpm. The forward-reverse clutch inertia test showed that instead of the axle's load, the inertia force corresponding to the axle was replaced by a flywheel; hence, a flywheel was attached to the axle while maintaining the engine's number of revolutions at the rated number of revolutions of the tractor's input shaft. A test to verify the axle's torque and the hydraulic pressure of the forward-reverse valve was conducted using a torque meter on the output shaft. The shift step was set to be H1, which was the step with the highest towing load. During the test, the forward-reverse hydraulic pressure and torque were sampled at 100 Hz. The transmission temperature should not exceed 130℃. Moreover, the exhaust gas temperature

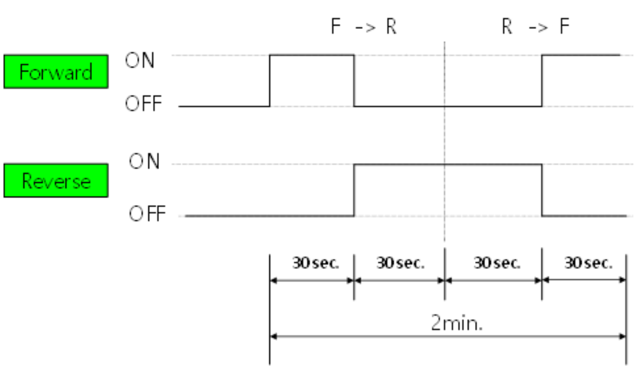

Figure 3. Cycle time for the wet clutch durability test.

Table 5. Specifications of the major components of the clutch durability and inertia tests

Item	Specifications
Engine	Model: V3307-DI-T-E3B; Kubota, Japan; 3331 cc; 50.7 kW
Transmission	Power shuttle type; main shift-4 × range shift-3
Flywheel	Diameter- 70 cm; thickness- 4 cm; mass-120 kg
Pressure gauge	Model: PH-200 KB; Kyowa Co., Ltd., Japan
Torque meter	Model: YDR-500 K; SETech Co., Ltd., Rep. Korea

should not exceed 660℃. The test should be stopped whenever the forward-reverse valve's hydraulic pressure deviates from 1961.33 ± 196.13 kPa, as this is considered as an abnormality of the tester.

Figure 4 shows a picture of the speed sensor (MP-981, Ono Sokki Co. Ltd., Japan) and the strain amplifier (YSA-50A, SETech Co. Ltd., Rep. Korea) used. The output of the engine used in the forward-reverse inertia test was 52.2 kw. The capacity of the torque meter (YDR-500KM, SETech Co. Ltd., Rep. Korea) of the measuring device was 600 Nm with a non-linearity of 0.09% R.O. and a hysteresis of 0.08% R.O. The capacity of the pressure sensor (PH-200KB, Kyowa, Japan) was 20 MPa, with a non-linearity of 0.04% R.O. and a hysteresis of 0.14% R.O. A data recorder (EDX-1500A, 16CH, Kyowa, Japan) was used to verify the axle's number of revolutions, the hydraulic pressure of the forward-reverse valve, and the axle's torque. The measuring range of the RPM sensor (MP-981, Ono Sokki Co. Ltd., Japan) used additionally was 1 Hz-20 kHz.

Results and Discussion

Transmission efficiency

The transmission efficiency test was divided into the driving system and the PTO. The driving system test was conducted by shifting the main shift based on the step of the auxiliary shift. Figures 5-7 show that at steps 1 and 2 of the auxiliary shift, a transmission efficiency of 56-89% was observed at 52 kW of input power, with a loss of 6.0-7.5 kW. At step 3 of the auxiliary shift, which was a driving speed step, the power loss seemed to increase with the increasing main shift speed. A transmission efficiency of 81.5-87.0% was observed at step 3 of the

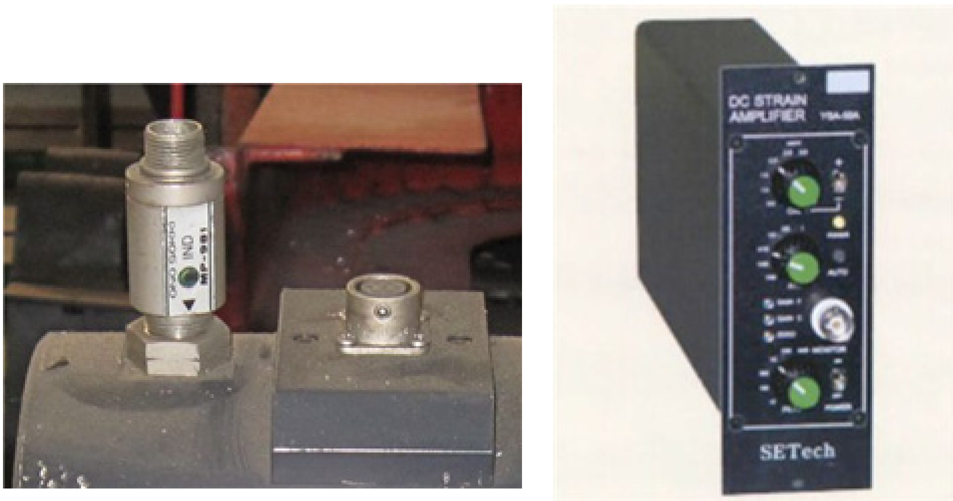

Figure 4. Photos of the speed sensor (left) and the DC strain amplifier (right) used in the test.

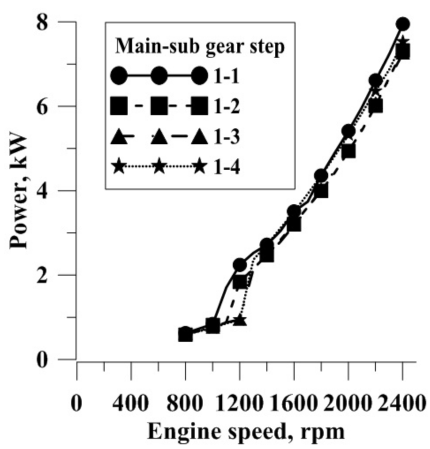

Figure 5. Transmission output by the engine speed at a gear setting of the range 1st step.

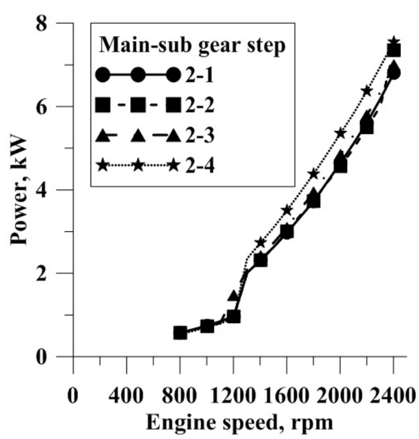

Figure 6. Transmission output by the engine speed at a gear setting of the range 2nd step.

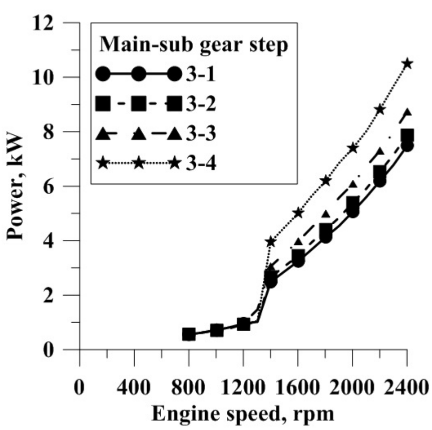

Figure 7. Transmission output by the engine speed at a gear setting of the range 3rd step.

auxiliary shift, with a power loss of 6.7-9.7 kW. The efficiency was greater than the target efficiency of 80.0 %

The PTO system test was conducted by shifting the PTO steps based on step 2 of the auxiliary shift, which was the step mainly used during work. A performance test was conducted at steps 1 and 2 of the PTO. Figures 8 and 9 show that the power loss at step 1 of the PTO was 6.7-9.0 kW; hence, the actual power loss should be 0.7-2.2 kW considering that the driving system's power loss was 8-10 hp. The PTO's power loss displayed a transmission efficiency of 96.0-98.5% at 52 kW input power. At step 2 of the PTO, the power loss was 6.7-11.9 kW; thus, the actual power loss should be 0.7-4.5 kW considering that the driving system's power loss was 6-10 kW. The PTO's power loss displayed a transmission efficiency of 91.5-98.5% at 52 kW of the input power. Overall, the driving and PTO transmission efficiency was greater than the target efficiency of 80.0 %

Transmission noise

Figures 10-12 present the results of measuring the transmission noise at each input number of revolutions. The noise at 800 rpm was 55.4 dB (right side), 55.4 dB (left side), 50.2 dB (rear), 55.3 dB (top), and 59.6 dB (forward-reverse valve). The noise at the right and left sides was the same because of the rotating shift gears. The noise at the rear was relatively low because of the low number of PTO gears. Overall, the transmission noise was good compared to the standard target level of 80 dB.

The noise at 1600 rpm was 73.0 dB (right side), 74.3 dB (left side), 69.6 at the rear, 75.5 at the top, and 77.2 dB at the forward-reverse valve. The noise was generally high compared with that in the previous case (800 rpm) because the rotational speed of the transmission gears also generally increased as the input number of revolutions increased.

The noise at 2400 rpm was 79.7 dB (right side), 80.3 dB (left side), 76.2 dB (rear), 83.3 dB (top), and 84.1 dB (forward-reverse valve). It was generally high when compared to those in the previous cases (800 and 1600 rpm) because the rotational speed of the transmission gears also increased as the input number of revolution increased. In the future, a sound-absorbing material should be added to the bottom part when designing the cabin.

Axle load

Figure 13 shows the results of the continuous uniform load test, in which 1/2 of the maximum axle load was applied to one axle. Considering that the left/right axle number of revolutions and the input torque persisted in the continuous uniform load test, the strength of the rear wheel differential was fully usable. Moreover, considering that the output persisted, the transmission of power was running well without wet clutch of the driving system or slip of the synchronizer. The temperature of the transmission oil remained constant without becoming high because no heat was generated by abnormal friction or wear inside the transmission. The engine oil and the exhaust gas temperature remained

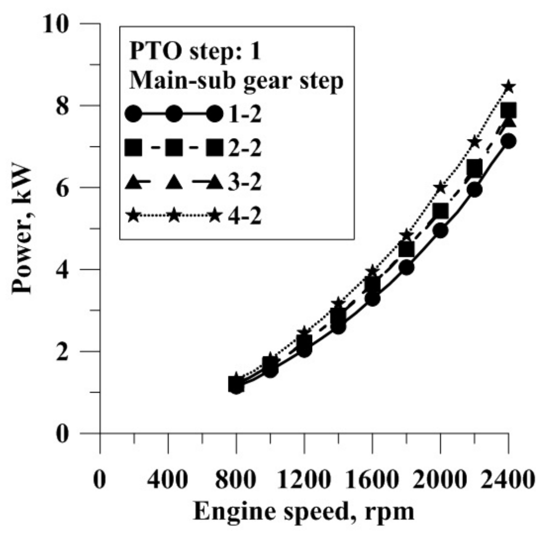

Figure 8. Transmission output by the engine speed at a gear setting of range 2nd step and PTO 1st step.

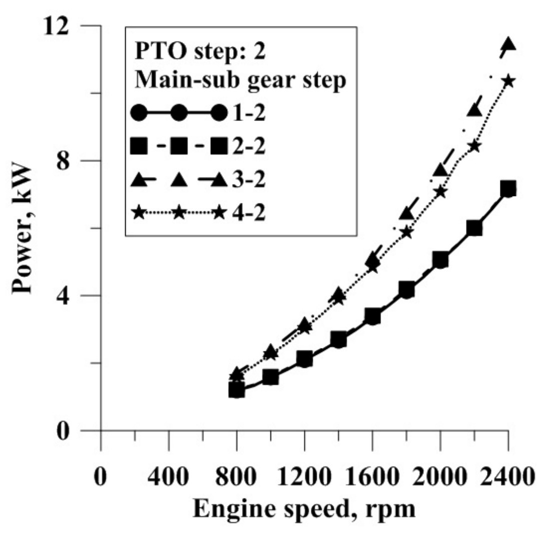

Figure 9. Transmission output by the engine speed at a gear setting of range 2nd step and PTO 2nd step.

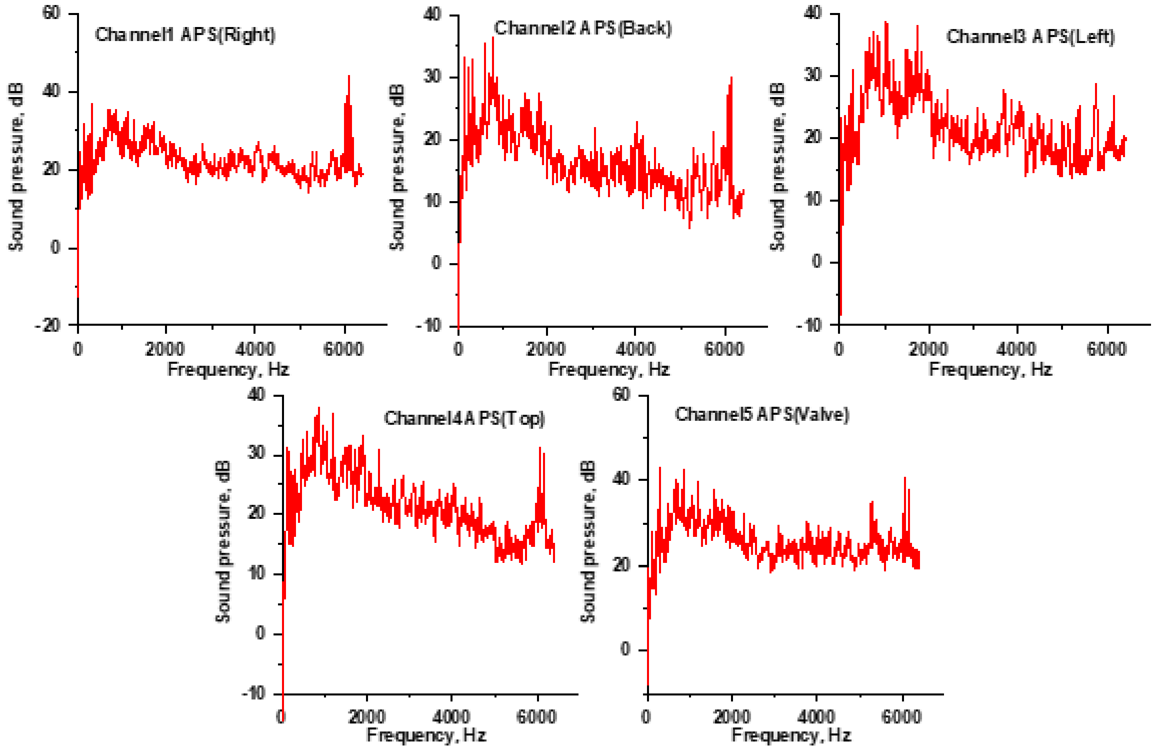

Figure 10. Noise at a rotational speed of 800 rpm.

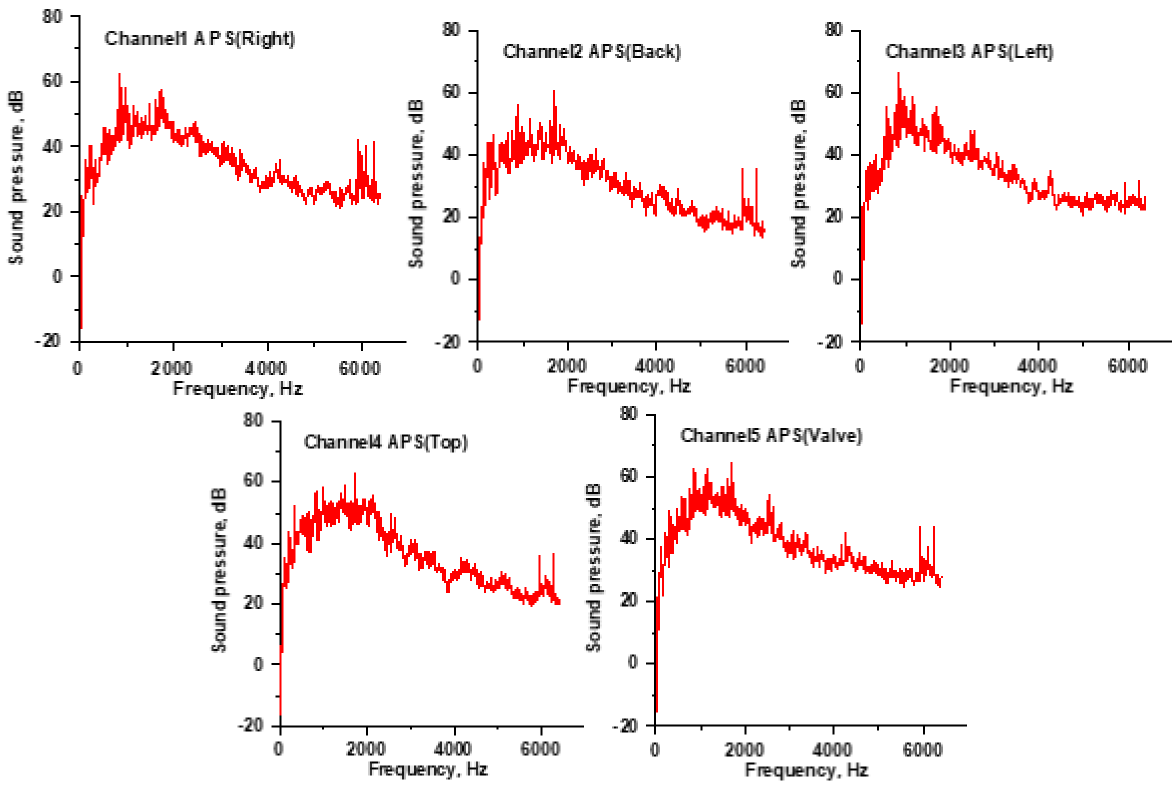

Figure 11. Noise at a rotational speed of 1600 rpm.

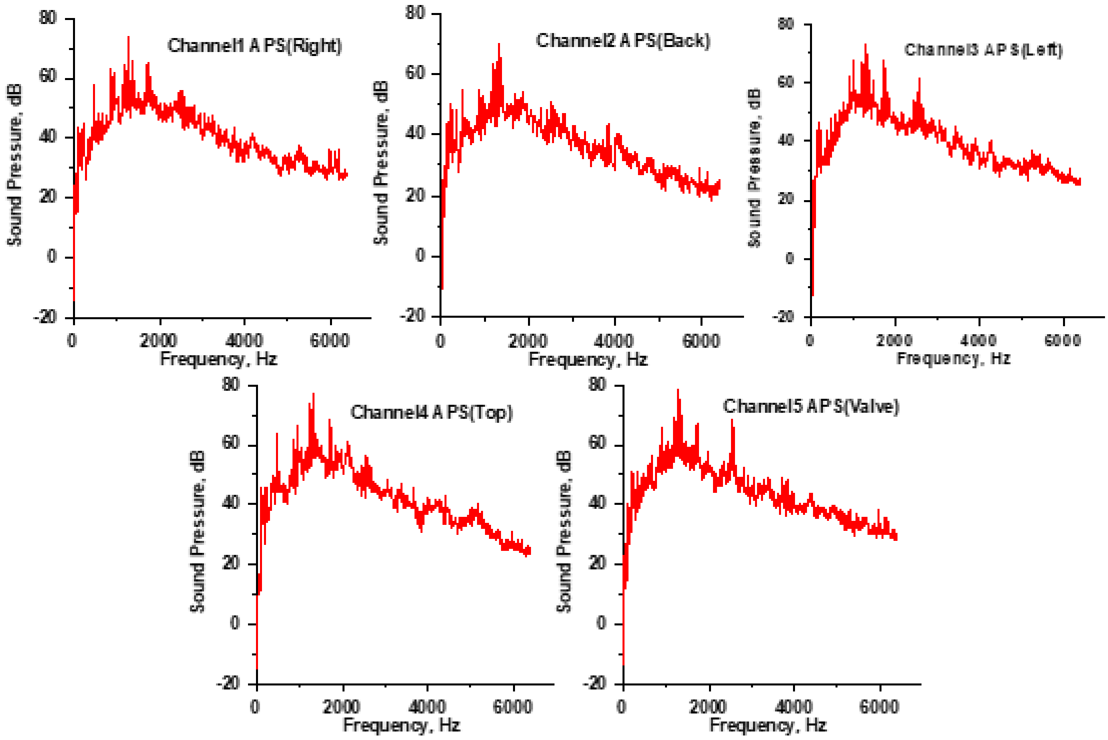

Figure 12. Noise at a rotational speed of 2400 rpm.

constant as a proof that the engine was sufficiently good.

Forward-reverse clutch durability and inertia

Figure 14 shows a graph of the hydraulic pressure and the torque observed during the forward-reverse clutch durability tests. The changes in the hydraulic pressure and the torque remained constant when moving forward or backward. The hydraulic pressure standard was well designed to be 1961.33 ± 196.13 kPa, and the transmission's forward-reverse clutch operated well and stably. The hydraulic pressure was lower than 1961.33 kPa when moving forward/backward because it decreased owing to valve friction and foreign matter. Torque overshooting occurred when moving forward/ackward because when the operation was performed in the opposite direction, in which the axle rotated, the fluid flow drastically changed

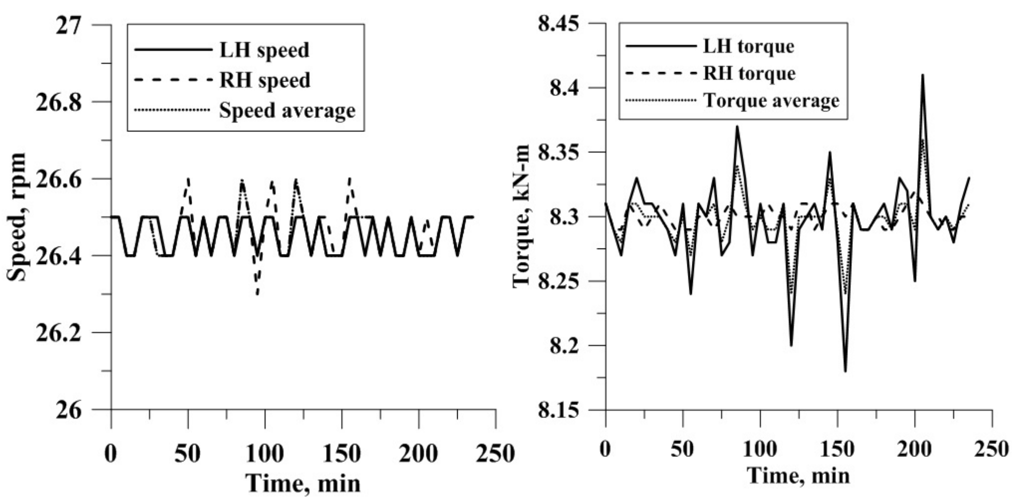

Figure 13. Results of the axle load test.

owing to the control valve operation. The kinetic energy of the fluid changed into pressure energy; hence, abrupt pressure fluctuations occurred, generating a surge pressure. Although a significant impact on the transmission was not observed in this test, the fact that the surge pressure was high indicated that the damage caused to the parts was high. The surge pressure was observed when the oil flow suddenly changed because of the operation of the electronic switching valve or the delayed action of the relief valve in the hydraulic circuit. In addition, measures should be taken for each mitigation because it occurred owing to the viscosity or degree of air contained in the oil. The transmission torque of the forward-reverse clutch is an important design variable in power transmission systems. Hence, accurate calculation and verification of this design variable are important in securing the durability of power transmission devices.

Figure 15 shows the data showing the inertia during the durability tests. The left/right transmission torque appearing to be reversed was caused by the mounting position of the inputoutput of the torque meter and showed the real torque size. The transmission torque was generated from the point when pressure was supplied, and the wet clutch was combined. The torque decreased in the point when the actual engine torque and the final driving shaft torque became equal. When designing wet multiple-disk clutches, the calculated transmission torque should be 588 Nm. The results of the actual forward-reverse clutch inertia test revealed that the transmission torque applied to the forward-reverse clutch was approximately 559 Nm.

Conclusions

This study conducted a performance test on a power transmission system for 50 kW narrow tractors that can be used in dry-fields, greenhouses, and orchards.

The driving power transmission efficiency test showed that the horsepower loss was 6.0-9.7 kW at all the shift steps. Such horsepower loss was equivalent to 11-18% of the input power (52 kW). The transmission efficiency of the driving power part was 81.5-89.0%. The noise of the driving power part was 50-57 dB at 800 rpm, 70-77 dB at 1600 rpm, and 76-83 dB at 2400 rpm. The axle load test verified that the axle number of revolutions and the input torque remained constant. The results of the forward-reverse clutch performance test revealed that the changes in the hydraulic pressure and the torque when moving forward/reverse were constant. Furthermore,

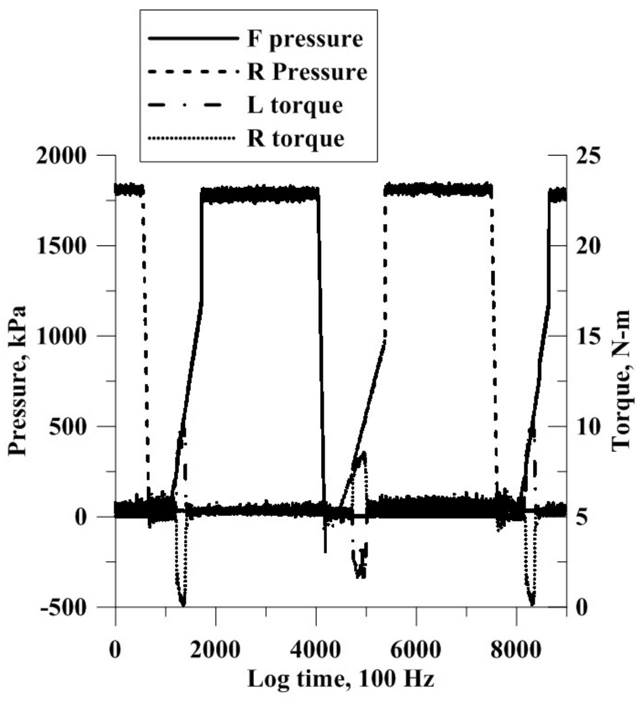

Figure 14. Pressure and torque curve of the power shuttle during the forward-reverse durability tests.

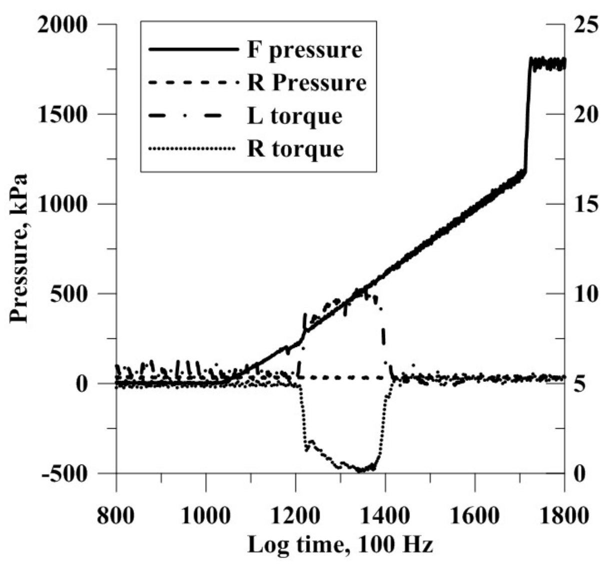

Figure 15. Pressure and torque curve of the power shuttle showing the inertia.

the forward-reverse clutch worked according to the hydraulic design standards. From a comprehensive point of view, these research results were similar to those obtained for the performance of the main driving power transmission systems of USA and Japan. Thus, this research showed a potential to allow the creation of tractor prototypes and their subsequent distribution to farmhouses after sufficient field adaptability tests.

Conflict of Interest

The authors have no conflicting financial or other interests.

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References

Ahn, S., J. Choi, S. Kim, J. Lee, C. H. Choi and H. Kim. 2015. Development of an integrated engine-hydro-mechanical transmission control algorithm for a tractor. Advances in Mechanical Engineering 7(7): 1-18.

Bilski, B. 2013. Audible and infrasonic noise levels in the cabins of modern agricultural tractors - does the risk of adverse, exposure-dependent effects still exist? International Journal of Occupational Medicine and Environmental Health 26(3): 488-493.

Capalbo, S. M. and M. Denny. 1986. Testing long-run productivity models for the Canadian and U.S. agricultural sectors. American Journal of Agricultural Economics 68(3): 615-625.

Cavallo, E., E. Ferrari, L. Bollani and M. Coccia. 2014a. Strategic management implications for the adoption of technological innovations in agricultural tractor: the role of scale factors and environmental attitude. Technology Analysis & Strategic Management 26 (7): 765-779.

Cavallo, E., E. Ferrari, L. Bollani and M. Coccia. 2014b. Attitudes and behaviour of adopters of technological innovations in agricultural tractors: A case study in Italian agricultural system. Agricultural Systems 130:44-54.

Chang, D. I., M. S. Kim, K. D. Kim, Y. K. Huh, S. O. Chung and B. K. Cho. 2010. Development of a 50 kW tractor for European orchards. Annual Research Report. Jeonju, Republic of Korea: Center for IT Convergence Agricultural Machinery.

Cho, J. M. 2003. Analysis of pressure modulation control and design parameters of powershift shuttle hydraulic clutch for tractor. MS thesis. Cheonan: Korea University of Technology and Education, Department of Mechanical Engineering.

Choi, S. H., H. J. Kim, S. H. Ahn, S. H. Hong, M. J. Chai, O. E. Kwon, S. C. Kim, Y. J. Kim, C. H. Choi and H. S. Kim. 2013. Modeling and simulation for a tractor equipped with hydro-mechanical transmission. Journal of Biosystems Engineering 38(3): 171-179.

Chung, S. O., Y. J. Kim, M. C. Choi, K. H. Lee, J. K. Ha, T. K. Kang and Y. K. Kim. 2016. Development of a hydraulic power transmission system for the 3-point hitch of 50-kW narrow tractors. Korean Journal of Agricultural Science 43(3): 450-458.

Guzzomi, A. and V. Rondelli. 2013. Narrow-track wheeled agricultural tractor parameter variation. Journal of Agricultural Safety and Health 19(4): 237-260.

Harris, K. J. and J. K. Jensen. 1964. John Deere power shift transmission. SAE Technical Paper 640052.

Ha, J. K. 2015. Development of power transmission system for orchard tractor. PhD diss. Daejeon: Chungnam National University, Department of Agricultural Machinery Engineering.

ILO, 1987. Safety in The Working Environment. International Labour Conference, Geneva, Switzerland: ILO.

ISO. 2015. ISO-5131: Acoustics - Tractors and machinery for agriculture and forestry - Measurement of noise at the operator's position - Survey method. Geneva, Switzerland: International Organization for Standardization.

Jung, G. H. 2013. Gear train design of 8-speed automatic transmission for tractor. Journal of Korean Society Fluid Power Construction Equipment 10(2): 30-36 (in Korean with English abstract).

Kabir, M. S. N. 2015. Comfort evaluation and classification of agricultural tractors considering safety factors. PhD diss. Daejeon, Republic of Korea: Chungnam National University, Department of Agricultural Machinery Engineering.

Kabir, M. S., Ryu, M. J., Chung, S. O., Kim, Y. J., Choi, C. H., Hong, S. J and J. H. Sung. 2014. Research trends for performance, safety, and comfort evaluation of agricultural tractors: A review. Journal of Biosystems Engineering 39(1): 21-33.

KAMICO (Korea Agricultural Machinery Industry Cooperative) and KSAM (Korean Society for Agricultural Machinery). 2016. Agricultural Machinery Yearbook Republic of Korea. Korean Society for Agricultural Machinery. Available at http://www.ksam76.or.kr.

Kechayov, D. and A. Trifonov. 2003. Noise effect of tractor stayer 942 upon the subsidiary agricultural workers. Journal of Environmental Protection and Ecology 4(4): 831-835.

Kim, B. G., W. O. Lee, S. Y. Shin, H. K. Kim, C. H. Kang and J. Y. Rhee. 2009. Analysis of determining factors for power size of a tractor. Journal of Biosystems Engineering 34(1): 8-14 (in Korean, with English abstract).

Kim, H. G., Y. J. Jo, C. S. Kim, Y. H. Han and D. C. Kim. 2016. Design improvement of mechanical transmission for tracked small agricultural transporters through gear strength analysis. Journal of Biosystems Engineering 41(1): 1-11.

Lee, K. H. 1990. Spur and helical gear design for manual transmission. Journal of the Korean Society of Automotive Engineers 12(6): 3-10 (in Korean).

Lee, I. Y., S. N. Yun, K. U. Yang, P. Y. Huh and D. G. Lee. 1996. Performance improvement of clutch actuating hydraulic control system at semi-automatic transmission for construction vehicles. Transactions of Korea Society of Automotive Engineers 4(3): 10-21 (In Korean, with English abstract).

Lee, P. U, S. O. Chung, C. H. Choi, Y. J. Park and Y. J. Kim. 2016. Analysis of the effects of operating point of tractor engine on fatigue life of PTO gear using simulation. Korean Journal of Agricultural Science 43(3): 441-449.

Nelson, D. I., R. Y. Nelson, M. Concha-Barrientos and M. Fingerhut. 2005. The global burden of occupational noise-induced hearing loss. American Journal of Industrial Medicine 48(6): 446-458.

OECD. 2018. OECD code 2: Standard code for the official testing of agricultural and forestry tractor performance. Parish, France: OECD. Available at www.oecd.org.

Park, Y. J., S. C. Kim and J. G. Kim. 2016a. Analysis and verification of power transmission characteristics of the hydromechanical transmission for agricultural tractors. Journal of Mechanical Science and Technology 30 (11): 5063-5072.

Park, Y. J., J. G. Kim and G. H. Lee. 2016b. Characteristic analysis of planetary gear set of hydromechanical transmission system of agricultural tractors. Journal of Biosystems Engineering 41(3): 145-152.

Raikwar, S., V. K. Tewari, S. Mukhopadhyay, C. R. B. Verma and M. S. Rao. 2015. Simulation of components of a power shuttle transmission system for an agricultural tractor. Computers and Electronics in Agriculture 114:114-124.

SK (Statistics Korea). 2016. Agricultural area statistics in 2015. Available at http://kostat.go.kr/portal/korea/index.action.

Yun, C. J. 1998. Analysis and control of the clutch hydraulic control system for a variable speed drive independent quality. MS thesis. Seoul: Seoul National University, Department of Mechanical Design and Production Engineering.