Theoretical transmission analysis to optimise gearbox for a 2.6 kW automatic pepper transplanter

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Abstract

A gearbox is an essential component of an automatic transplanter to transmit engine power to the transplanter components. It is necessary to find the appropriate gearbox dimensions and materials for the pepper transplanter to minimise transmission losses. Therefore, the objectives of this research were to simulate the power transmission efficiency of the gearbox and to determine a suitable number of stages, materials, and the dimensions of the spur gears. A 2.6 kW gasoline engine was considered as the prime source to power the entire transplanter. The available maximum length between the engine and transplanter subsystem was 422 mm. Considering design issues, a simulation model was created to determine the efficiency of the pepper transplanter gearbox, including various types of mechanical losses in the gearing system. Three different modules (1, 2, and 3 mm) and two materials were used to evaluate the effects on transmission. The analysis results indicated that the gearbox transmission efficiency levels of seven to twelve stages were in the range of 93.0–98.7%, whereas the eight-stage gearbox yielded a maximum efficiency of 98.7%, more significant than the target efficiency of 98.0%. Therefore, an eight-stage gearbox was selected for power transmission to the components. The power transmission simulation results showed that the overall efficiency from the engine to the transplanting mechanism shaft varied in a range of 95.2-95.9% owing to contact of the gear meshes. The analysis results also indicated that the 25CrMo4 carbon steel material with a 2-mm module gear was appropriate for the pepper transplanter. Therefore, the analysis in this paper can be used as a reference in the design of pepper transplanter gears and gearboxes with suitable material properties to provide the desired efficiency.

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Introduction

Pepper (*Capsicum annuum* L.) is one of the most cultivated vegetables, and worldwide pepper production increased by 8656 thousand tons (23.5%) from 2008 to 2020 (FAOSTAT, 2020). Although the pepper cultivation rate and cultivated area are increasing worldwide, decreases in pepper production were recorded in some countries (*e.g.*, the Republic of Korea and Japan) over the past years, possibly due to lack of mechanisation, farm labour shortage, small-scale agricultural land, and aging of farmers (Islam *et al.*, 2020). Developing a low-powered and power-efficient pepper transplanter is important to overcome these difficulties and improve cultivation quality and efficiency in pepper seedling transplanting (Kim *et al.*, 2018). The power delivery efficiency of agricultural machinery during operations depends on the performance of the power driveline and the design of the gearbox parameters (Kim *et al.*, 2000; Kim *et al.*, 2018). Hence, the initial steps in power transmission design are understanding the key parameters affecting the gearbox efficiency and quantifying their roles. Gears are the most common power transmission parts in machinery, automobiles, and aircraft (Budynas and Nisbett, 2008; Kuria and Kihiu, 2011). The efficiency of these power transmission systems is an important design objective. Extensive research on the efficiency and friction modelling of gear pair systems was conducted by many researchers (Kuang and Lin, 2001; Vaishya and Singh, 2003; Xu *et al.*, 2007). Xu *et al.* (2007) proposed a computational model for the friction-related mechanical efficiency losses of parallel-axis gear pairs. The model incorporated a gear load distribution model, friction model, and mechanical efficiency formulation to predict the mechanical efficiency of a gear pair under typical loading, surface, and lubricating conditions. The results showed that the mechanical efficiency of a parallel-axis spur gear pair was in the range of 98.0-99.5%. Anderson and Loewenthal (1986) analysed the effect of modified addendum, tooth thickness, and gear centre distance on the efficiency of non-standard and high contact ratio involute spur gears. The study considered sliding friction, rolling friction, and clearance losses. It showed that despite their high sliding velocities, high-contact ratio gears could be designed through proper selection of gear geometry to achieve efficiencies comparable to conventional standard gears while retaining their unique advantages.

Optimal gear transmission design is not a recent research idea. Various studies proposed guidelines for selecting gear transmission stages (Selfridge, 1980; Thompson *et al.*, 2000; Yokota *et al.*, 1998). Selfridge (1980) proposed an iterative calculative method for optimum multistage transmissions and minimum transmission rotational inertia. Thompson *et al*. (2000) formulated a numerical multi-criterion optimisation method for transmissions, which considered objective functions, such as the gear train volume and efficiency. Various optimisation algorithms were formulated to determine the stage ratios of a gear train, in which each gear was constrained to have a predetermined number of teeth (Gandomi *et al*., 2013). Bartlett *et al*. (2018) proposed a technique to select the number of stages in a multistage transmission with a given desired total transmission ratio for maximum efficiency, maximum acceleration, or minimum mass of the transmission. The power transmission design criteria largely depend on surface phenomena, including friction, lubrication, wear, and environmental deterioration (Juvinall and Marshek, 2006). The technique considered several implications for the gear train design. For the off-field vehicles, the minimum rotational inertia and mass can be considered as objectives for optimal selection of the stage ratios and can often be minimised by increasing the total number of stages above a minimum realisable value. However, for agricultural machinery for which an optimal operation implies a constant rotational speed, the power transmission design criteria would differ from those for the off-field vehicles. Therefore, to select the number of stages in a gear train for agricultural machinery, an analytical procedure would be necessary to calculate the stages and minimise the power loss of transmission. An automatic pepper transplanter is under development, and the power transmission efficiency of the gearbox needs to be evaluated. Therefore, the objectives of this research were to analyse the power transmission efficiency of the gearbox and to determine a suitable number of stages, materials, and dimensions of the spur gears.

Materials and methods

Power transmission system of the automatic pepper transplanter

The overall structure and power transmission system of the automatic pepper transplanter under development are shown in Figure 1. A two-row pepper seedling dibbling unit is attached to the conveying unit to plant the seedlings (Figure 1, b). During the operational period, the picking mechanism picks five seedlings simultaneously from the tray and transfers them to the release tray. Meanwhile, a push bar mechanism pushes the seedling to the conveying unit. Finally, the conveying unit drops the seedlings into the two-row hopper-type dibbling units by rotating the sprocket and chain. A 2.6 kW engine (Figure 1, e) (SUBARU Industrial Products Co. Ltd., Japan) was considered as the power source of the transplanter. The engine specification is described in Table 1. The engine power was transmitted to the transplanter components (wheel and dibbling-picking mechanism) through a belt-pulley transmission system (Figure 1, f). The distance between the two pulleys was 125 mm. From the gearbox pulley, the power was separated into two drivelines. One driveline was used to rotate the

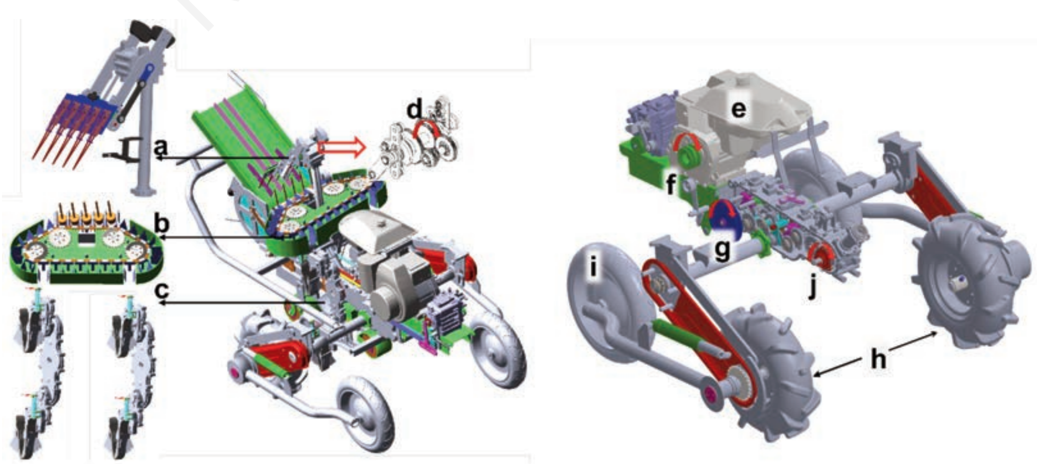

Figure 1. The overall structure of the automatic pepper transplanter: a- picking mechanism; b- conveyer mechanism; c- dibbling mechanism; d- picking gear; e- engine; f- engine pulley; g- gearbox pulley; h- rear wheel; i- front wheel; and, j- dibbling and picking shaft.

wheel shaft, and the other was used to operate the picking and dibbling mechanisms. The prototype pepper transplanter had two 400-mm diameter rear wheels (Figure 1, h) and two 300-mm diameter front wheels for moving across the field (Figure 1, i). A gearbox with a multi-stage reducer was used to transmit power from the engine to the working components of the transplanter.

Procedure to determine the appropriate number of gearbox stages

The major step to consider in designing a multi-stage transmission for a given overall transmission ratio is selecting an appropriate number of stages and the stage ratio associated with each selection. Figure 2 shows the flow diagram of the working principle of theoretical transmission analysis to optimise the gearbox for the pepper transplanter. First, we selected the gear based on the considered parameters. The minimum threshold size of the gearbox was 422 mm. After selecting the gearbox, we performed a simulation including different materials to calculate the power loss of the transmission. Table 2 indicates the considered variables in the transmission analysis for the pepper transplanter.

Table 1. Technical specifications of the engine used for this study.

Item	Specification
Model	EY15-2B
Type	Air-cooled 4-cycle single cylinder, side valve, horizontal shaft, gasoline engine
Maximum output [HP (kW)/rpm]	3.5(2.6)/2000
Continuous output [HP (kW)/rpm]	2.7(2.0)/1800
Fuel	Automobile gasoline
Dry mass(kg)	14.2
Dimensions L×W×H (mm)	324×311×368

Table 2. Variable notations, definitions, and units used for the transmission analysis.

Notation	Definition, unit, and value
μ	Coefficient of friction between the gears, typically between 0.05 and 0.15
s_i	Transmission ratio corresponding to the i^{th} stage of the transmission, decimal
k	Constant containing information about the gear geometry, integer
Z_1	Number of teeth on the pinion in the mesh, integer
β	Pressure angle, radian
ε_α	Profile contact ratio, integer
$\varnothing$	Loss factor, integer
η_T	Calculated as the product of the individual stage efficiencies (η_i), %
C_1	Constant, 1.12×10^{-8}
C_2	Constant that depends on the face width to diameter ratio, integer
ρ	Density of the gear operational environment, kg m^{-3}
n	Rotation speed, radian s^{-1}
D	Pitch diameter of the gear, m
ν	Kinematic viscosity of the lubricant, m^2 s^{-1}
λ	Constant related to the type of housing surrounding the gear (λ=1 for open enclosure, 0.7 for loose enclosure, 0.5 for closed enclosure)
P_s	Sliding power loss, N
V_s	Sliding velocity, m s^{-1}
F_s	Sliding force, N
w	Load parameter, integer
C_3	Constant, 29.66
b	Face width, mm
μ_0	Ambient viscosity at ambient temperature, m^2 s^{-1}
VT	Rolling velocity, m s^{-1}
F_r	Rolling force, N
C_4	Constant, 9×10^{-7}
h	Central film thickness, mm
φ_t	Thermal reduction factor, integer
P_r	Rolling power loss, N

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Gear train efficiency is important when selecting the number of stages of a gearbox, in which the gearbox power consumption or output torque is of fundamental concern. This study determined the range of the available transmission ratios for each gear train. First, the final stage gear set ratio should be calculated to select the suitable stage of the gearbox. The transplanter gearbox input speed was 625 rpm, and the final gear ratio was 10.42:1 to maintain the transplanting operation at 60 rpm. Initially, the efficiency of each stage was calculated to evaluate the total transmission efficiency and select the range of the number of teeth of the final stage gear. Our target minimum efficiency was 98.0%, based on the results of Xu *et al.* (2007).

The gear train efficiency (η_i) of the single stage of a spur gear train can be calculated using the analytical approach proposed by Velex and Ville (2009), as described in Equations 1-3. The primary considerations to determine a suitable multistage gearbox for the transplanter in the calculation are listed in Table 3.

Efficiency at each stage, $\eta_i = 1 - (\mu \cdot k) - (\mu \cdot k \cdot s_i)$ (1)

$$k = \frac{\pi}{Z_1} \cdot \frac{1}{\cos\beta} \cdot \varepsilon_\alpha \cdot \Delta \quad (2)$$

$$\eta_T = \prod_{i=1}^{n} \eta_i \quad (3)$$

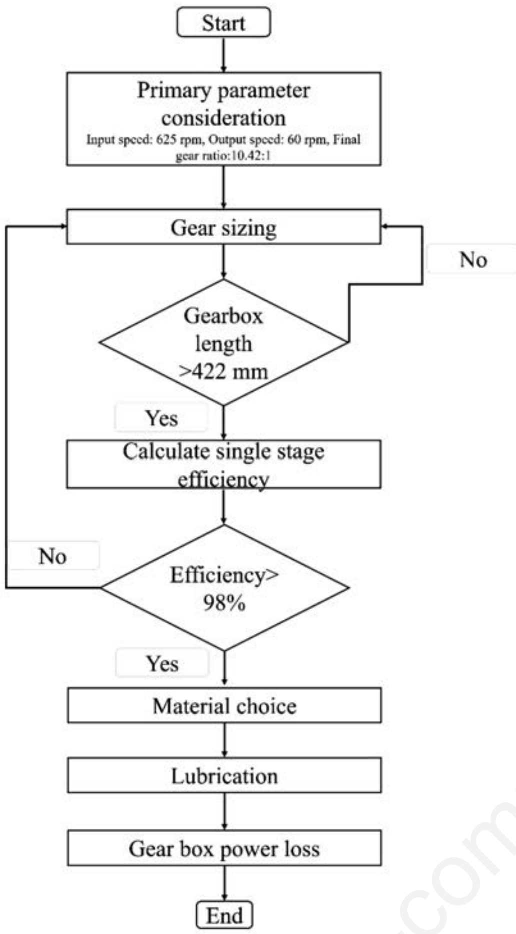

Figure 2. Flow diagram of the theoretical transmission analysis to optimise the gearbox for the pepper transplanter.

Powertrain simulation to calculate the efficiency of the gearbox

A three-dimensional simulation model was constructed for the powertrain investigation using a commercial computer-aided gear efficiency calculation software package (KISSsoft, Version 2017, KISSsoft AG, Bubikon, Switzerland). The simulation was performed using the input data fed into the first stage (reducer) at 1.7

Table 3. Primary parameter consideration to determine a gearbox (multistage) for pepper transplanter power transmission from engine to other components.

Parameter	Specification
Input speed	625 rpm
Output speed	60 rpm
Final gear ratio	10.42:1
Gearbox available length	422 mm

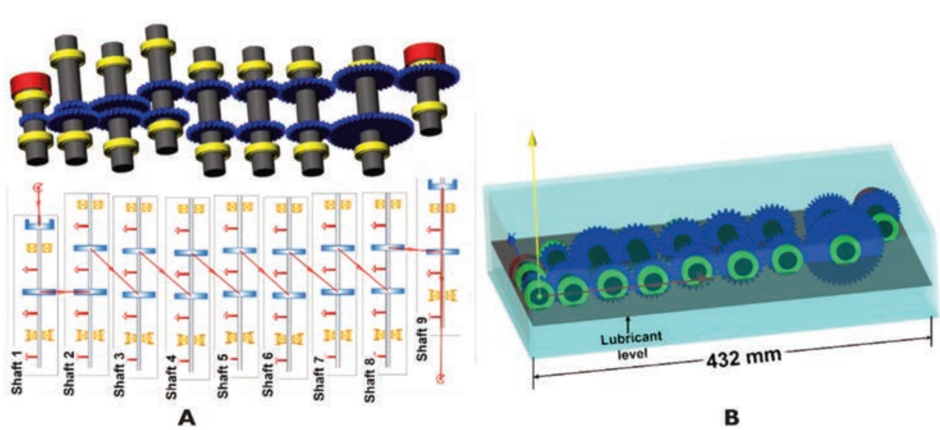

Figure 3. Simulation model of the pepper transplanter gearbox: three-dimensional diagram (A) and gearbox with lubricant (B).

kW and 625 rpm. Figure 3 shows a three-dimensional simulation model of the pepper transplanter gearbox. Power from the engine was applied to the input shaft (shaft 1) of the gearbox. Once coupling 1 is closed and coupling 2 is open, the whole power flow will go through coupling 1. On the other way, when coupling 1 is open, and coupling 2 is closed, the power flow flows through coupling 2. These two conditions represent the forward and reverse rotation of the transmission, which have a similar impact on the strength of the output shaft. After making the geometry of the gearbox, the gears and shaft materials were assigned. The efficiency of the transmission system was evaluated based on the materials and face width of the gear. Steel materials were used to manufacture the spur gears, as recommended by many researchers (Choi and Choi, 1999; Townsend *et al*., 1978). Several types of carbon steel materials are available at the research and industrial level. Two of the most commonly used materials, namely steel composite material 25CrMo4 (medium carbon steel) and C60 steel (high carbon steel), were selected for this analysis, as used by Islam *et al*. (2021). Lubricant temperature is a major issue in effective viscosity. In this study, thermal conductivity was considered as 50 W/(m·K), and lubricant oil temperature was set as 80°C.

For the validation, the results of our simulation were compared with the output shaft power of the field experiment data using the picking and dibbling mechanisms of the transplanter prototype reported previously (Islam *et al*., 2020; Islam *et al*., 2021; Iqbal *et al*., 2021a, 2021b). A test bench was fabricated to measure the load of the picking device gear. A three-phase electric motor was used as an external power source for the direct load applied to the test bench. Similarly, a prototype of the dibbling mechanism was fabricated, and transplanting was performed on a test bench with a 30 mm × 30 mm aluminium profile. A three-phase electric motor and chain-sprocket were used to move the test bench at the desired speeds on parallel steel pipe rails. The previous study measured the power levels required to operate the picking and dibbling mechanisms. The required power level of the picking device was found to be 18.59 W (Islam *et al*., 2020). The power consumption of the dibbling mechanism operating at a speed of 300 mm/s was recorded as 40.91 W (Iqbal *et al*., 2021a, b). In this study, the output shaft power was recorded as 70 W. After generating the simulation model, the accuracy and power loss were calculated using the ISO TR 14179-2 simulation environment. During the simulation, the housing wall thickness was considered as 5 mm to quantify the heat losses toward the environment. The gross volume of the gearbox was 49,980 mm^3 by considering the gears, shaft, and lubricant filling coefficient. The gearbox efficiency varies depending on several types of operational losses, including windage, lubricant churning, sliding friction, and rolling friction of gears. The major loss, namely windage, occurs due to the lubricant being flung off the gear teeth as the gears rotate and displacement of the lubricant during gear mesh (Handschuh and Kilmain, 2008; Heingartner and Mba, 2003). Lubricant churning losses result from gears moving the lubricant inside the gear case (Townsend *et al*., 1978). All rotating gears in direct contact with the lubricant, which partially submerged them, contribute to lubricant churning losses, and the deeper the components are submerged, the higher the losses. The Dawson loss Equation (4) considered the gear diameter, pitch, face width, and housing effects.

$$P_{WL} = C_1 \cdot C_2 \cdot \rho \cdot n^{2.85} \cdot D^{4.7} \cdot \nu^{0.15} \cdot \lambda \tag{4}$$

The motion of the gears is achieved by rolling and sliding. The instantaneous sliding friction loss is a function of the sliding velocity and friction force, which is also a function of the average tooth load and coefficient of friction. The magnitude of the sliding velocity depends on the position of the contact along the contact path with a peak velocity (Kuria and Kihiu, 2011). The rolling friction loss depends on the rolling velocity and lubricant film thickness (Heingartner and Mba, 2003). As the gear teeth move into the mesh, a lubricant film is developed between the teeth in contact with the mesh. The action of the gear teeth during the engagement draws the lubricant into the contact zone. The parameters influencing the rolling friction loss are the lubricant film thickness, the angular velocity of the gears, the working pressure angle, and the point of contact along its contact path. Equations 5-9 describe the rolling and sliding losses of the gear train. The parameters considered in the power train simulation to determine the appropriate overall multistage gearbox power loss and efficiency are listed in Table 4.

$$P_s = 10^{-3} \cdot V_s \cdot F_s \tag{5}$$

$$F_s = \mu \cdot w \tag{6}$$

$$\mu = 0.0127 \log \frac{C_3 \cdot \frac{w}{b}}{\mu_0 \cdot V_s \cdot V_T} \tag{7}$$

$$F_r = C_4 \cdot h \cdot \varphi_t \cdot b \tag{8}$$

$$P_r = 10^{-3} \cdot V_T \cdot F_r \tag{9}$$

Results and discussion

Selected number of stages

The number of final gear stage teeth was determined to obtain a total ratio of 10.42:1. For this transplanter, the dibbling and picking shaft rotating speeds should be the same. Therefore, for the final stage (picking shaft), the tooth numbers of the driven and driven gears need to be the same. Figure 4A shows the number of teeth of the final stage for the total gear train efficiency. The efficiency range was 93.0-98.7%. The range of efficiency of the multistage gear train for each stage varied from 94.0% to 99.5% (Kuria and Kihiu, 2011). This indicated that 13 to 50 teeth could maintain the efficiency in the range of 94.0-99.5%. For a desired total ratio of 10.42:1, the stage ratios were determined with respect to effi-

Table 4. Gearbox parameters are considered in the power transmission simulation.

Item	Specification
Gear material	25CrMo4 and C60 (ISO standard)
Gear module	1, 2, and 3 mm
Shaft diameter	12 mm
Gearbox geometry	Rectangular
Coupling	Inside the housing
Thermal conductivity	50 W m^{-1} K^{-1}
Gearbox wall thickness	5 mm
Lubricant oil temperature	80°C
Input speed	625 rpm
Input torque	16.42 N·m
Output shaft power	70 W

ciency for various values of n ($7 \leq n \leq 12$). The results of this analysis are plotted against the number of stages, as shown in Figure 4B. This analysis offers insight into the selection of the number of stages in a transmission. Specifically, the efficiencies of various transmission peaks for eight stages are shown in Figure 4B. The transmissions were designed to maximise, but the difference between the overall efficiency values was insignificant. Therefore, the range of transmission efficiency obtained by varying the number of stages in the transmission was also very small (approximately 1.5% change in efficiency from n=7 to n=12). A length objective function was formulated considering the length of the gears and gearbox. The length of the gearbox considered for the pepper transplanter was 422 mm. The number of teeth for three different gear modules (1, 2, and 3 mm) is presented in Table 5.

Power loss and efficiency of the gearbox

In a gear transmission system, power loss is a function of the angular rotation of the gear sets, and this will only be known as a function of time if the mean rotational speed is assumed to be constant (Oh *et al.*, 2005). A cubic spline interpolation was employed to determine the magnitude of each component of the power loss for each gear mesh at each point of contact as a function of time for one mesh period of the output gears (Kuria and Kihiu, 2011). The power loss of the pepper transplanter gearbox varied according to the number of stages. Each stage had a separate gear set that included an individual dimension. Figure 5 shows the effect of power loss for 3 different gear modules (*i.e.*, 1, 2, and 3 mm) and two different materials. The 3-mm gear module yielded a power loss of 91.9 W for the C60 steel material.

In contrast, the 25CrMo4 carbon steel material produced the lowest power loss of 67.3 W. Approximately 1.5% of the excessive loss was obtained owing to the difference in the materials. On the other hand, for the 2-mm gear module simulation, the overall maximum and minimum power losses were 92.0 W and 91.3 W for C60 and 25CrMo4 carbon steel materials, respectively. For the 1-mm module, the overall maximum and minimum power losses recorded were 91.3 W and 91.1 W, respectively. Power loss is also related to the speed condition. Figure 5 shows that a speed of 156 rpm for all conditions resulted in a higher power loss than at the other speed conditions. The power loss characteristics also indicated that in the middle stages (3-7 no.), the recorded power loss was higher owing to the higher shaft speed and higher number of gear teeth, as shown in Figure 5.

The average power loss and efficiency of different modules for the two materials are shown in Figure 6. It shows the gear transmission performances for the 1-mm, 2-mm, and 3-mm modules. The efficiency of the 1-mm module was high, with minimal power loss. In contrast, the 3-mm face width yielded a comparatively lower efficiency with a higher power loss. As a result, the highest

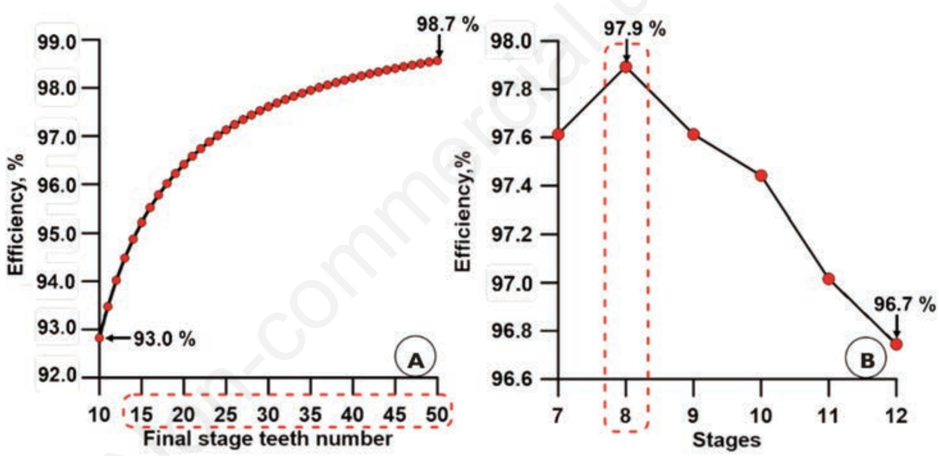

Figure 4. Gearbox stage selection based on efficiency: number of teeth of the final stage (A) and number of stages (B).

Table 5. Considered number of teeth based on stages and available gearbox length.

Stage	Number of teeth					
	1-mm module		2-mm module		3-mm module	
	Driver gear	Driven gear	Driver gear	Driven gear	Driver gear	Driven gear
1	32	64	16	32	10	20
2	32	64	16	32	10	20
3	48	48	24	24	16	16
4	48	48	24	24	16	16
5	48	48	24	24	16	16
6	42	54	21	27	17	22
7	46	92	23	46	15	30
8	68	68	34	34	22	22

average efficiency levels were obtained for the 1-mm gear module, with 95.9% and 94.0% values for the 25CrMo4 and C60 carbon steel materials, respectively.

On the other hand, the lowest average efficiency levels were obtained for the 3-mm gear module, with values of 95.3% and 95.2% for the 25CrMo4 and C60 carbon steel materials, respectively. The average power loss levels of the pepper transplanter gearbox for the 1-mm, 2-mm, and 3-mm gear modules were 7.5±3.0, 7.6±3.0, 10.1±4.0 W, respectively, for 25CrMo4 material. They were 10.2±4.0, 10.2±6.9, and 10.3±6.9 W, respectively, for C60 carbon steel materials. Figure 6 shows the average power loss and efficiency for all the conditions. No statistically significant difference was observed between the two materials, as shown in Figure 6. The power loss was directly inversely proportional to efficiency, and the results indicated that the efficiency decreased by an average of 1.0% for 1-mm gear module increments.

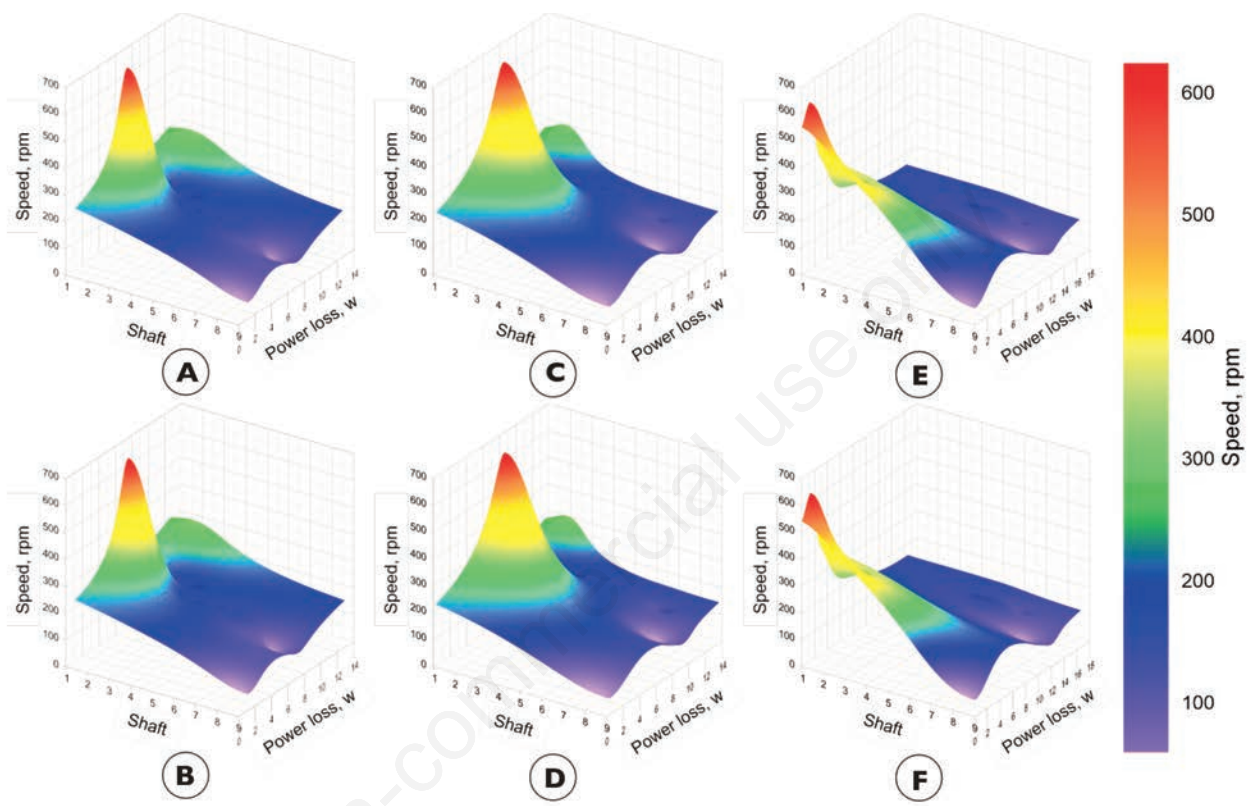

Figure 5. Power loss for different materials and gear modules: 1 mm, 25CrMo4 (A), 1 mm, C60 (B), 2 mm, 25CrMo4 (C), 2 mm, C60 (D), 3 mm, 25CrMo4 (E), and 3 mm, C60 (F).

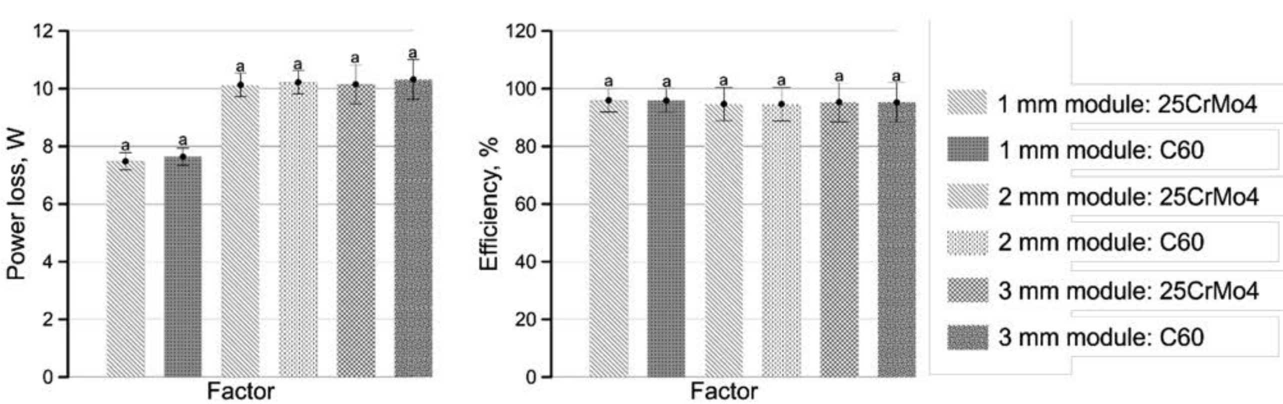

Figure 6. Average power loss (left) and efficiency (right) for different modules and materials of the gearbox. The mean values in the graphs indicated with different letters are significantly different based on Tukey's one-way comparisons (P≤0.05).

Conclusions

This study proposed a method to determine the optimal gear stage for a desired overall transmission ratio that yielded maximum efficiency, investigating several related implications regarding gear train design. For a gear train with a given total gear reduction ratio (10.41:1), the minimum gearbox length was used as the objective function in selecting the number of stages. Increasing the total number of stages could decrease gear efficiency. The overall power transmission performance can be significantly improved by adding stages, sometimes more remarkable than the minimum feasible number of stages. A power transmission analysis was performed for three different gear modules and two different materials. Several losses, including windage, lubricant churning, sliding friction, and rolling friction, were considered by creating a computer-based simulation environment. The overall efficiency of the system was found to be a function of power loss and ranged from 95.2% to 95.9%. The results also demonstrated that the high and medium carbon steel materials used in this study were suitable for this type of gearbox design. However, the high-carbon steel material was more expensive than the medium-carbon steel material. For instance, a medium-carbon steel material (such as 25CrMo4 carbon steel) with a carbon percentage of 0.3-0.6% can be used to design the gearbox of a 2.6-kW automatic pepper transplanter. The results presented in this paper can be used as a reference in designing optimal pepper transplanter transmissions, thereby improving design reliability and reducing material costs.

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