

ENGINEERING

# Theoretical analysis of power requirement of a four-row tractor-mounted radish collector

Khine Myat Swe<sup>1,†</sup>, Mohammad Ali<sup>1,†</sup>, Milon Chowdhury<sup>1,2</sup>, Md Nasim Reza<sup>1,2</sup>, Md Ashrafuzzaman Gulandaz<sup>2</sup>, Sang-Hee Lee<sup>3</sup>, Sun-Ok Chung<sup>1,2,\*</sup>, Soon Jung Hong<sup>4,\*</sup>

<sup>1</sup>Department of Agricultural Machinery Engineering, College of Agriculture and Life Sciences, Chungnam National University, Daejeon 34134, Korea

<sup>2</sup>Department of Smart Agricultural Systems, College of Agriculture and Life Sciences, Chungnam National University, Daejeon 34134, Korea

<sup>3</sup>National Institute of Agricultural Sciences, Rural Development Administration, Jeonju 54875, Korea

<sup>4</sup>Korea National College of Agriculture and Fisheries, Jeonju 54874, Korea

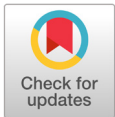
<sup>†</sup>These authors equally contributed to this study as first author.

\*Corresponding authors: sochung@cnu.ac.kr, hsj43333@korea.kr

## Abstract

Development of radish collectors may enhance radish production and promote upland crop mechanization in the Republic of Korea. Theoretical analysis of power is crucial to ensure the optimum design of agricultural machinery. The aim of the present study is to analyze theoretically the power requirement of a tractor-mounted radish collector under development and to propose design guidelines. The important components of the radish collector were belt-type conveyors, three hydraulic motors, and a direct current (DC) winch motor to operate the total radish collecting process. Theoretical equations were used to calculate the hydraulic motor's power, winch motor power, and draft power at loaded and unloaded conditions. A variety of tractors (44 - 74 kW) and different soil characteristics (hard, firm, tilted, and sandy) were considered to investigate the appropriate drawbar power. Variations of the power requirement of the tractor-mounted radish collector were observed due to modifications of the design parameters. The required hydraulic power of the stem cutting conveyor, stem cutting blade, and transfer conveyor of the radish collector were 0.23 and 0.24, 0.18 and 0.19, and 0.19 and 0.22 kW under unloaded and loaded conditions, respectively. The maximum draft power was calculated as 0.89, 1.07, 1.25, and 1.61 kW at a 30° tilted angle for hard, firm, tilted, and sandy soil, respectively. The calculation showed 2.07 kW DC power was required for unfolding or folding the stem-cutting conveyor. A maximum power of 4.78 kW was prescribed for conducting the whole process of the tractor-mounted radish collector. The analysis of power introduced in this study will be helpful to select the appropriate design parameters for the successful development of a tractor-mounted radish collector.

**Key words:** agricultural mechanization, hydraulic power, radish collector, power analysis, upland crop harvesting



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## Introduction

Radishes (*Raphanus raphanistrum*) are grown almost everywhere globally and are one of the major root vegetables in the Republic of Korea, having various medicinal and socio-economic values (Shen et al., 2013; Banihani, 2017). White radishes have been used as a leading ingredient in combination with a variety of fermented vegetables (i.e., cabbage, cucumber, chillies) for preparing the spicy Korean traditional dish, Kimchi (Jung et al., 2014). Despite its importance, the area of land under white radish cultivation has steadily decreased from 24,989 ha in 2010 to 21,391 ha in 2020 (KOSIS, 2020) because of the labor-intensive, time-consuming harvesting process, and the effect of low mechanization (Hong and Lee, 2012). Radish tillage, plastic mulching, and disease control are mostly mechanized, while the seeding and harvesting sectors are not, and remain heavily dependent on human labor. The population of farm households is also continuously decreasing due to the aging of farmers, rapid urbanization, and industrialization. The total mechanization rate of upland crops reached up to 56% (Shin et al., 2015; Yu et al., 2015), which is insufficient. It is very important to boost radish crop mechanization by promoting the use of farm machinery to minimize time and labor dependency in the production of this key upland crop.

Because of the traditional cultivation and harvesting process (e.g., manual harvesting, compiling, loading, unloading, packaging, and transportation), radish harvesting technology has the advantages of simple structure, low cost, and market-friendliness (Guolong et al., 2016). Various studies have been conducted to develop a variety of harvesting technologies, particularly for upland crops (Kataoka, 2016; Ali et al., 2019; Han et al., 2021), which have created interest in mechanized harvesting and collecting processes to decrease crop management workloads and increase production. Japan has begun developing a radish harvester by evaluating its performance in different field conditions (Yaegashi et al., 2001). Lee et al. (2019) developed a Japanese radish harvesting assistance device by occupying the grasping and digging mechanism and evaluating its performance with field experiments. In China, pre-existing radish harvesting and collecting technology using a digging shovel model was tested by applying the theoretical methods before the development of the desired harvester (Guolong et al., 2016). Several types of research are ongoing in the Republic of Korea to boost the development of different categories of radish harvesting and collecting technologies. The research mainly investigates the proper design factors for the different parts (i.e., radish carrying conveyor, depth controlling plate, digging conveyor, digging blade, radish grabbing gearbox, cylinder, and secondary blade by field tests) (Choi et al., 2001). The theoretical assessment to develop radish collection technologies is still lacking. The advanced radish harvesting and collecting technologies are composed of driving, main harvesting, collecting, power transmission, and discharging parts. The main focus of the studies was designing and evaluating the performance of the system for the development of a self-propelled radish harvesting technology (Kook et al., 2021). Jung et al. (2018) analyzed the strength of the harvesting part of the radish harvester by applying different theoretical methods to identify the weakest part. In an effort to enhance the design dependability and dimensions of the materials, Swe et al. (2022) assessed the stress and fatigue of significant components of a tractor-mounted radish collector under variable load circumstances. Although the radish harvesting and collecting technologies have not been well mechanized and still under development in the Republic of Korea. To develop advanced radish-collecting technology, design considerations such as power source, sufficient power distribution, and arrangement of the necessary components of the collector should be investigated theoretically.

Theoretical analysis of power is crucial to avoid failure and ensure the better operation of the radish collector. Some preliminary technical aspects of the machine's power can be formulated using mathematical and simulation models. The power or energy requirement calculation of the machine aids design of the optimal operational parameters (Chen et al., 2004).

Systematic designs of radish collectors often need to investigate power issues to estimate engine load, mechanical execution, and fuel usage. Inappropriate management of power increases fuel consumption and overall costs, which decreases the performance of tractor-mounted radish collectors. The generation of sufficient power to handle radish harvesting or collecting operations in agricultural fields should be confirmed. Precise prediction of the required power is essential for manufacturing workable designs for any kind of machinery (Ali et al., 2021a). A multipurpose platform is required to carry radish collectors in the radish field. A tractor can be used as this common platform to handle radish collectors where the variety of tractors relies on available required power to accomplish the desired activities (ASABE standards S390.5, 2011; Ali et al., 2021b). Hence, estimation of required power is essential prior to the development of a tractor-mounted radish collector, as this makes it simple to identify an appropriate tractor to use with collectors under development.

Tractors (middle to high power) transmit sufficient draft power as the main power source to haul agricultural implements on typical farms (Howard et al., 2013; CCS, 2021). McLaughlin et al. (2008) studied a tractor with attached implements using different models for justifying the output power in agricultural fields. The studies revealed that the calculative power satisfied the power requirements to carry the workable implements. Hence, the theoretical equations related to row crops could predict the average power requirement of tractors performing radish-collecting operations in the field.

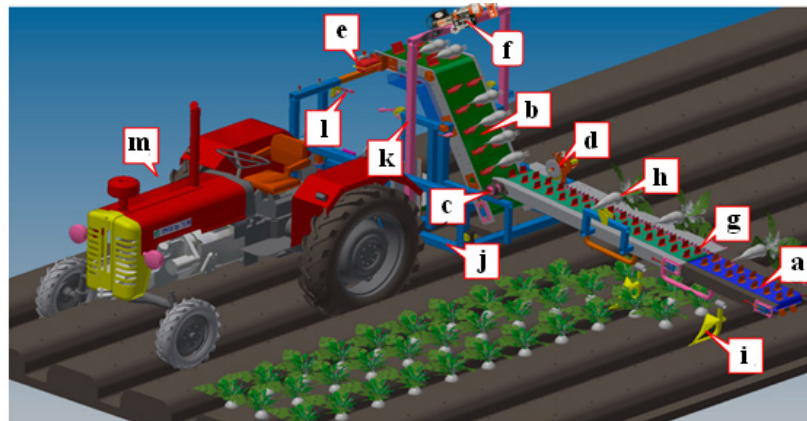
A tractor-mounted radish collector is under development in the Republic of Korea. The analysis of the required power was determined to be crucial in designing the optimal parameters of the radish collector. The design parameters of the tractor-mounted radish collector required to deliver suitable power according to the actual field conditions are taken into consideration. The tractor configurations and motors are introduced and selected based on the theoretical calculations of the required power. The overall power consumption is presented in terms of the tractor hydraulic and DC-motor power sources, and the draft power of the tractor to pull the radish collector. In this study, having access to suitable power-calculating methodologies helped overcome the problem of insufficient power during the radish collection. The objectives of the study were to analyze the theoretical estimation of the draft and hydraulic motor power required to operate the tractor and radish collector as well as the winch motor power for folding and unfolding the stem-cutting conveyor and suggest design guidelines for the manufacturers.

## Materials and Methods

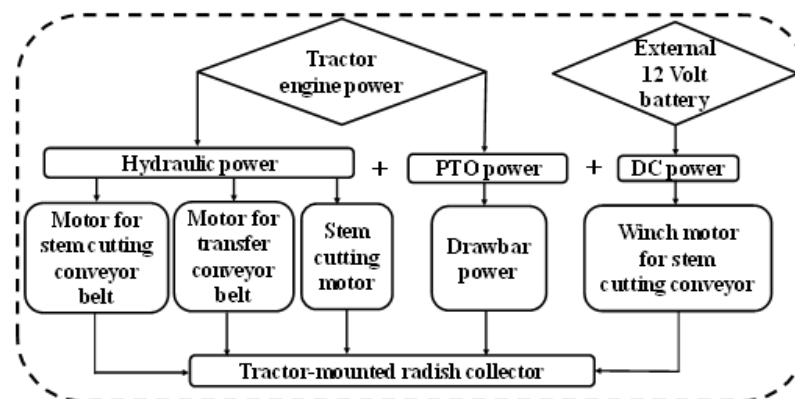
### Structure and working principles of a tractor-mounted radish collector

The four-row tractor-mounted radish collector under development was designed with consideration for the manual harvesting and automatic collection process. The overall mass and dimensions of the radish collector were 460 kg and 4,210 mm × 1,200 mm × 2,000 mm (length × width × height), respectively. It was composed of a stem-cutting conveyor and blade, transfer conveyor, and winch motor attached to the frame as shown in Fig. 1. In the radish-collecting process, radishes are first harvested manually and placed on the stem-cutting conveyor. A circular rotating stem-cutting blade was installed at the end of this conveyor. The blade cut the stems of the radishes one-by-one randomly and then transported them up through the radish transfer conveyor. Finally, the fresh radishes were placed into boxes or polypropylene bags. The designed radish transfer conveyor was able to run at 0.18 m·s<sup>-1</sup> speed with a capacity of 0.549 t·h<sup>-1</sup>. At the end of the radish-collecting process, the stem-cutting conveyor was required to fold with the safety frame of the radish collector. The radish collector was attached to the 3-point hitches of the tractor, which pulled it. Sufficient power take-off (PTO) power was needed to carry out the radish

collector along with the applied loads during the field operation. Two hydraulic motors were used to produce sufficient power to perform the radish conveying process, and power from another hydraulic motor was used to cut the radish stems. The tractor's hydraulic power worked as the main power source to support the three hydraulic motors, including the required draft power to pull the radish collector. To complete the radish-collecting operation, a suitable tractor was required. Additionally, winch motor power was used to fold and unfold the stem-cutting conveyor in a vertical position. To select the desired design parameters, the power requirement analysis had to consider expected field conditions to successfully perform the radish collection process. Therefore, the overall power requirement calculation was performed through various conditions such as soil type, conveyor rotational speed, and tractor driving speeds with and without load conditions. Fig. 2 presents a block diagram of the power source and power transmitting distributions of the tractor-mounted radish collector. The design variables (shown in Tables 1 - 5) were selected for calculating the maximum power required to successfully run the tractor-mounted radish collector in real field conditions. The design variables for analyzing the maximum power requirements for the tractor-mounted radish collector that is under development were considered in accordance with the manufacturer's design guidelines and radish field sizes and patterns.



**Fig. 1.** A 3D model of the tractor-mounted four-row radish collector under development (Chowdhury et al., 2020): (a) stem cutting conveyor; (b) radish transfer conveyor; (c) stem cutting conveyor motor, (d) stem cutting blade; (e) transfer conveyor motor; (f) DC winch motor; (g) rubber bit; (h) radish; (i) furrow guide; (j) collector base frame; (k) safety frame; (l) hanging rods of the radish collecting bag; (m) tractor.



**Fig. 2.** A block diagram of power source and power requirement components. PTO, power take-off; DC, direct current.

**Table 1.** Design variables were used for calculating the hydraulic power of the conveyor belts.

Variable	When fixed	Range (interval)
Stem cutting conveyor		
$M_m$ (kg)	63.00	40 - 80 (10)
$M_B$ (m)	30.00	10 - 50 (10)
$U_T$	0.033	0.11 - 0.55 (0.11)
$U_R$	0.33	0.11 - 0.55 (0.11)
Transfer conveyor		
$L_t$ (m)	1.93	1 - 3 (0.05)
$H$ (m)	1.16	1 - 5 (1)
$S_i$ (m)	1.13	1 - 5 (1)
$W_b$ (kg·m <sup>-1</sup> )	5.17	2 - 10 (2)
$W_m$ (kg·m <sup>-1</sup> )	10.86	5 - 25 (5)

$L_c$  = length of radish transfer conveyor;  $L_{cc}$  = length of radish transfer conveyor and stem cutting conveyor.

$M_m$ , total load of conveyed materials (radish weight);  $M_B$ , mass of the belt;  $U_T$ , friction coefficient on support rollers;  $U_R$ , friction coefficient on skid plate;  $L_t$ , length of the transfer conveyor;  $H$ , vertical height of ascending and descending lift;  $S_i$ , idler spacing;  $W_b$ , weight of belt;  $W_m$ , weight of conveyed material.

**Table 2.** Design variables used for calculating the stem cutting motor power.

Variable	When fixed	Range (interval)
$F_x$ (N)	32	10 - 50 (10)
$r_b$ (m)	0.08	0.05 - 0.50 (0.05)
$\omega_b$ (rad·s <sup>-1</sup> )	251	100 - 600 (100)
$w_{bl}$ (mm)	0.15	0.05 - 0.25 (0.05)
$rek$ (mm)	0.45	0.20 - 1.00 (0.20)
$\phi_{bk}$ (°)	30	10 - 50 (10)

$F_x$ , stem cutting force;  $r_b$ , radius of stem cutting blade;  $\omega_b$ , radial velocity of stem cutting motor;  $w_{bl}$ , width of the blade;  $r_{ek}$ , radius of blade edge;  $\phi_{bk}$ , bevel angle of blade edge.

**Table 3.** Design variables used for calculating the power requirement of winch motor.

Variable	When fixed	Range (interval)
$L_2$ (m)	0.5 L1	0.1 - 1.0 L1 (0.1 L1)
$L_4$ (m)	0.2 L1	0.1 - 0.5 L1 (0.1 L1)
$H_1$ (m)	2	1 - 3 (0.5)
$\theta_2$ (°)	0	0 - 90 (0.5)

$L_2$ , horizontal distance between pivot point and winch tie bracket;  $L_4$ , horizontal distance between winch motor and pivot point;  $H_1$ , vertical distance between pivot point and winch motor fullname;  $\theta_2$ , angle.

**Table 4.** Soil variables used under different soil conditions.

Soil type	Soil water content (%)	Cone index (kPa)
Sand		
Silt	10 - 22	670 - 4,030
Clay		

**Table 5.** Design parameters for calculating tractor draft power.

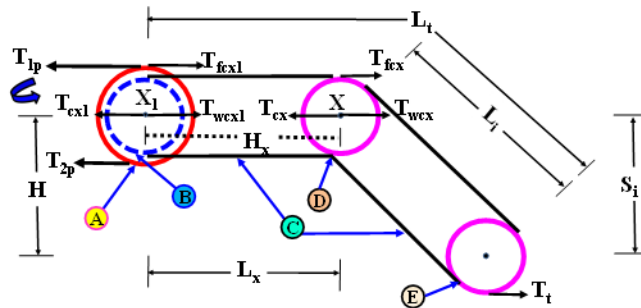
Variable	Range (interval)
$W_{ax}$ (kN)	22 - 27 (1)
$\phi$ (°)	0 - 20 (4)
$d_w$ (m)	0.50 - 3 (0.5)
$b_w$ (m)	0.10 - 0.60 (0.1)
$W_w$ (kN)	4 - 32 (4)
$\beta$ (°)	0 - 30 (3)

$W_{ax}$ , applied vertical load to axle;  $\phi$ , soil angle for shearing resistance;  $d_w$ , wheel diameter;  $b_w$ , wheel breadth;  $W_w$ , weight of tractor rear wheel;  $\beta$ , tilted angle between wheel load and soil.

## Theoretical procedure for power requirement analysis

### Calculation of hydraulic power required to run the stem-cutting conveyor

In this study, the stem cutting conveyor belt was driven by a variable-speed hydraulic motor. The major components of the stem-cutting conveyor are shown in Fig. 3. The tensile load was inserted on the conveyor due to the belt operation. The force requirement for transporting the radish conveyor was influenced by different factors; the total path length of the conveyor, pulley arrangement, belt engagement location, loading state, and operating conditions. The hydraulic power at the drive pulley of the stem-cutting conveyor was calculated using the following equations 1 and 2 under loaded and unloaded conditions (Daniyan et al., 2014; Afolabi et al., 2017).



**Fig. 3.** A schematic diagram of forces on transfer conveyor. (A) Hydraulic motor, (B) drive pulley, (C) conveyor belt, (D) idler roller, (E) tail pulley.

$$P_s = \frac{F_U \times V_s}{1,000} \quad (1)$$

$$F_U = U_T \times g \left( M_m + \frac{M_B}{2} \right) + U_R \times g \left( \frac{M_B}{2} \right) + M_i \quad (2)$$

Where,  $P_s$  is hydraulic power at the drive pulley of the stem-cutting conveyor (kW),  $V_s$  is the speed of the stem-cutting conveyor ( $\text{m} \cdot \text{s}^{-1}$ ),  $F_U$  is the total tangential force at the periphery of the drive pulley (N),  $U_T$  is the friction coefficient with support rollers,  $g$  is the gravitational acceleration ( $\text{m} \cdot \text{s}^{-2}$ ),  $M_m$  is the total loading materials (kg),  $M_B$  is the belt mass (kg),  $U_R$  is the friction coefficient with the skid plate, and  $M_i$  is the mass of the roller idlers (130 kg).

### Calculation of hydraulic power required to run the transfer conveyor

Belt conveyors play an important role in continuous dry bulk material transport. An engineering design method integrates many different aspects of the logical and comprehensible design process that must be planned systematically for designing the conveyor (Pahl and Beitz, 1996). Belt tension is a key factor in the design of belt conveyor systems, influencing belt selection, location, and size. In this study, Maximum belt tension occurs at the intake and loading points on horizontal or inclined conveyors when the belt is continuously conveying the designed load from the loading point to the discharge point. He et al. (2016) expected that speed control of the conveying system may reduce the total energy consumption of the belt conveyor. According to the design parameters of the transfer conveyor shown in Fig. 3, the power requirement was calculated

for an inclined conveyor with a tail pulley drive using equations 3 to 11 (Piotr, 2011). Design parameters for calculating the hydraulic power of the conveyor belts are listed in Table 1.

$$P_{tc} = \frac{T_e \times V_{tc}}{3.6} \quad (3)$$

$$T_e = T_{1p} + T_{2p} \quad (4)$$

$$T_{1p} = T_{cx1} + T_i \quad (5)$$

$$T_{cx1} = T_t + T_{wcx1} + T_{fcx1} \quad (6)$$

When the belt operated with a normal load, a maximum of 3% belt sag was recommended for simplifying the minimum belt tension during the operation using equation 7.

$$T_t = 4.2 \times S_i (W_b + W_m) \quad (7)$$

$$T_{wcx1} = H_x (W_b + W_m) \quad (8)$$

$$T_{fcx1} = L_x [K_t K_x + K_t K_y (W_b + W_m)] \quad (9)$$

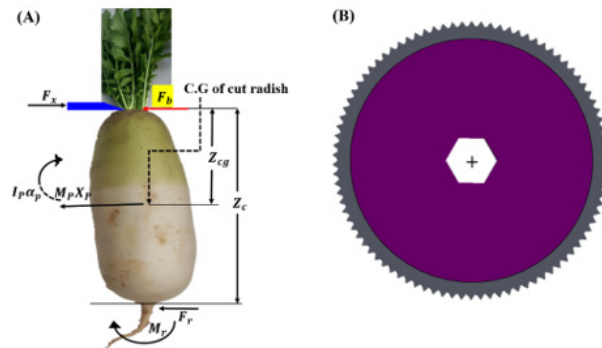
$$T_i = (n \times \text{friction at slacking side pulley}) + (n \times \text{friction at other side pulley}) \quad (10)$$

$$T_{2p} = T_t - L_t (0.015 W_b) + H W_b \quad (11)$$

Where,  $P_{tc}$  is the hydraulic power required to run the transfer conveyor (kW),  $T_e$  is the effective belt tension at the drive pulley (N),  $T_{1p}$  and  $T_{2p}$  are the tight-side and slack-side tension at the pulley (N),  $T_{cx1}$  is the tension force in the belt at point  $x_1$  on a carrying run (5.8 N),  $T_i$  is the non-driving pulley friction (N),  $T_t$  is the back tension at the tail pulley (N),  $n$  is the number of pulleys,  $S_i$  is the idler spacing (m),  $W_b$  is the weight of the belt ( $\text{N} \cdot \text{m}^{-1}$ ),  $W_m$  is the weight of the conveyed material ( $\text{N} \cdot \text{m}^{-1}$ ),  $H$  is the vertical height of the ascending and descending lift (m),  $H_x$  is the vertical distance from the center of the tail pulley to the desired point (m),  $L_i$  is the length of the inclined portion of the transfer conveyor (1.33 m),  $L_t$  is the length of the transfer conveyor (1.93 m),  $L_x$  is the horizontal length of the transfer conveyor (0.60 m), and  $V_{tc}$  is the speed of the transfer conveyor ( $1.61 \text{ km} \cdot \text{h}^{-1}$ ).

#### Calculation of hydraulic power for the stem-cutting motor

Korean radish roots are usually cylindrical, fat, and short. The average mass and length of radishes are usually 0.14 - 2.27 kg and 0.15 - 0.61 m, respectively (Clovegarden, 2020). Several acting forces and moments are considered when calculating the stem-cutting power of radishes (Fig. 4). In that regard, the soil and roots combined provide a force ( $F_b$ ) and a moment ( $M_t$ ) that help keep the stump of the radish in a vertical position. The center of gravity (C.G) acts at a trimmed section of the radish at a specific elevation ( $Z_{cg}$ ) that is equal to the radius of gyration of the cut portion of a plant ( $R_g$ ) in (m). The satisfactory



**Fig. 4.** The schematic diagram of (A) force acting characteristics of radish during the stem cutting process, and (B) circular-shaped radish stem cutting blade.

power for cutting the stem of a radish can be calculated using formulae 12 to 15 (Srivastava et al., 2006). Different design variables for calculating the appropriate stem-cutting motor power were studied (Table 2). A sufficient rotational speed was required for the stem cutter that could lead the stem-cutting process precisely. Therefore, the dimension and thickness of the circular-shaped radish stem cutter were determined to be 240 and 15 mm, respectively (Fig. 4).

As in previous studies, the oscillation of the circular-shaped cutter was determined using experimental methods by reducing the amplitude of the oscillation to avoid adverse situations during operation (Orlowski et al., 2007; Veselý et al., 2012).

$$P_{scm} = \frac{F_x \times r_w \times \omega_b}{1,000} \quad (12)$$

The requirement of initiate cutting force was calculated using the following equation

$$F_x = F_{ek} \quad (13)$$

$$F_{ek} = A_{ek} \times w_{bl} \times P_b \quad (14)$$

$$A_{ek} = r_{ek} [1 + \cos(\Phi_{bk} + \Phi_{ck})] \quad (15)$$

where  $P_{scm}$  is the required hydraulic power for the stem-cutting motor (kW),  $F_x$  is the blade moving force in the x direction (N),  $r_b$  is the radius of the cutting blade (m),  $\omega_b$  is the radial velocity of the stem-cutting motor ( $\text{rad} \cdot \text{s}^{-1}$ ),  $F_{ek}$  is the force inserted by the plant on the blade edge (N),  $A_{ek}$  is the frontal area of the blade edge per mm of width,  $F_b$  is the horizontal reaction force of the stump (N),  $r_{ek}$  is the radius of the blade edge (mm),  $P_b$  is the pressure ahead of the blade edge ( $25 \text{ N} \cdot \text{mm}^{-2}$ ),  $r_g$  is the radius of gyration of the cut portion of the plant (m),  $Z_{cg}$  is the given height of the center of gravity of the cut plant (m),  $\phi_{bk}$  is the bevel angle of the blade edge ( $^\circ$ ), and  $\phi_{ck}$  is the bevel angle of a straight cut ( $0^\circ$ ).

#### Calculation of DC winch power requirement for folding and unfolding the stem-cutting conveyor

Due to the various advantages (i.e., automatic regulation of tension, high breaking power, compact structure, and simple control system), a winch motor was selected for folding and unfolding the stem-cutting conveyor (Fei et al., 2009). The stem-cutting conveyor was vertically folded to the safety frame in the normal position with a DC winch motor. When the operation of the tractor-mounted radish collector began, the stem-cutting conveyor was unfolded into a horizontal position and with the help of a winch motor at variable speeds and was also folded vertically upon completion of operation in the same way (Figs. 5 and 6). The winch motor power required for folding and unfolding the stem-cutting conveyor was calculated using equations 13 to 21.

$$P_{dwp} = \frac{T_w \times \omega_w}{1,000} \quad (16)$$

$$T_w = F \times L_6 \quad (17)$$

$$\sum M_0 = -W_s(L_2 \cos \theta_2) + F \cos \theta_3 [L_3 \cos(\theta_1 + \theta_2)] + F \sin \theta_3 [L_3 \sin(\theta_1 + \theta_2)] = 0 \quad (18)$$

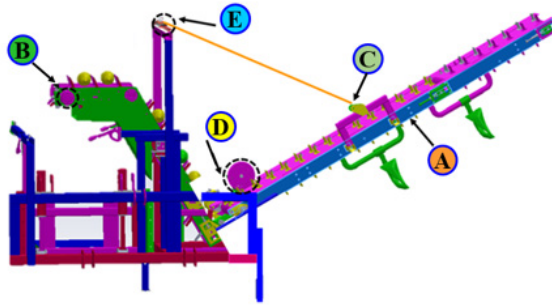
$$F = \frac{W(L_2 \cos \theta_2)}{\cos \theta_3 [L_3 \cos(\theta_1 + \theta_2)] + \sin \theta_3 [L_3 \sin(\theta_1 + \theta_2)]} \quad (19)$$

$$L_2 = 0.5L_1 \quad (20)$$

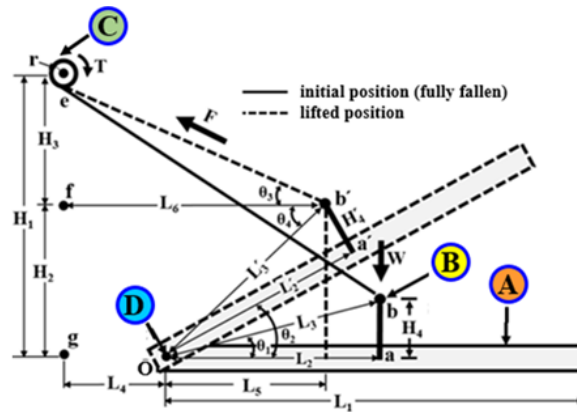
$$L_6 = L_4 + \sqrt{L_2^2 + H_4^2} \cos(\theta_1 + \theta_2) \quad (21)$$

$$\theta_1 = \tan^{-1} \frac{H_4}{L_2} \quad (22)$$





**Fig. 5.** A 3D model side view of radish collector. (A) Stem cutting conveyor, (B) transfer conveyor, (C) winch tie bracket, (D) stem cutting blade with motor, and (E) winch motor.



**Fig. 6.** A schematic drawing explaining variables: (A) stem cutting conveyor, (B) midpoint of which tie bracket, (C) winch motor, and (D) pivot point.

$$\theta_3 = \tan^{-1} \frac{H_3}{L_6} \quad (23)$$

$$H_3 = H_1 - H_2 \quad (24)$$

$$H_2 = \sqrt{L_2^2 + H_4^2} \sin(\theta_1 + \theta_2) \quad (25)$$

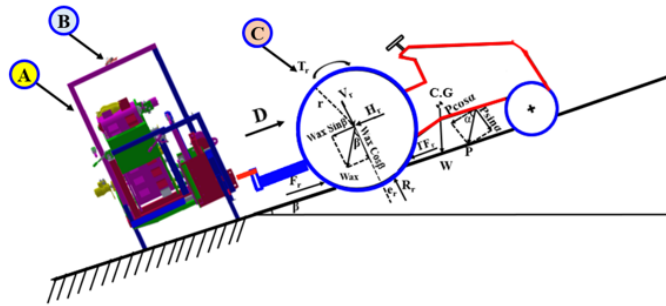
Where,  $P_{dwp}$  is the required power for folding or unfolding the stem-cutting conveyor (kW),  $\theta_1$  is the angle between  $L_2$  and  $L_3$  ( $^\circ$ ),  $\theta_2$  is the angle between  $L_2$  and  $L_2$  ( $^\circ$ ),  $\theta_3$  is the angle between the winch cable (lifted) and the horizontal distance from the mid-point of the lifted winch tie bracket to the winch motor position ( $^\circ$ ),  $F$  is the tension force required to lift the stem-cutting conveyor (N),  $W$  is the weight of the stem-cutting conveyor (800 N),  $T_w$  is the torque of the winch motor (Nm),  $\omega_w$  is the radial velocity of the winch motor ( $\text{rad} \cdot \text{s}^{-1}$ ),  $L_1$  is the length of the stem-cutting conveyor (2.04 m),  $L_2$  is the horizontal distance between the pivot point and winch tie bracket (m),  $L_3$  is the inclined distance between the pivot point and mid-point of the winch tie bracket (m),  $L_4$  is the horizontal distance between the winch motor and pivot point (m),  $L_5$  is the horizontal distance between the pivot point and the lifted mid-point of the winch tie bracket (m),  $L_6$  is the horizontal distance between the winch motor and the lifted mid-point of the winch tie bracket (m),  $H_1$  is the vertical distance between the pivot point and the winch motor (m),  $H_2$  is the vertical distance between the pivot point and the lifted mid-point of winch tie bracket (m),  $H_3$  is the vertical distance between the lifted mid-point of the winch tie bracket and the winch motor (m), and  $H_4$  is the vertical

distance between the pivot point and the mid-point of the winch tie bracket (0.30 m).

The pivot point and central weight of the conveyor were located at positions O and W. When the location of the winch tie bracket ( $L_2$ ) is serially moved to the left under the range of  $0.1 L_1 - 1.0 L_1$ , angle  $\theta_1$  between  $L_2$  and  $L_3$  of the winch tie bracket was changed first. Then, the related parameters and power consumption were also varied. The optimal position of the winch tie bracket was determined by the different power consumption results. The design variables of the DC winch motor are summarized in Table 3.

### Drawbar power of tractor-mounted radish collector

Rolling resistance is an important factor acting on the wheel opposing the direction of wheel movement. It is a function of soil deformation, depending on various parameters such as tire load, inflation pressure, diameter, width, construction, and thread, as well as speed, surface adhesion, sliding, and relative micro-sliding between contact surfaces (Wong, 2001). Many researchers have investigated rolling resistance and defined it as a major determinant when calculating draft power under various soil conditions during agricultural operations (Chung and Sudduth, 2006; Islam et al., 2019). The theoretical relationship of rolling resistance includes soil constants, the bulk unit weight, and the angle of shearing resistance, which could be applied to design the radish collector (Fig. 7).



**Fig. 7.** A free body illustration of draft pull for tractor-mounted radish collector: (A) radish collector, (B) winch motor, and (C) tractor's rear wheel.

The rolling resistance of a towed, rigid wheel over a granular medium could be estimated in terms of the axle load and wheel geometry using the expressions 26 to 27 (Wismer and Luth, 1973).

$$CRR = \left( \frac{1.2}{C_n} \right) + 0.04 \quad (26)$$

$$C_n = \frac{CI \times b_w \times d_w}{W_w} \quad (27)$$

Where, CRR is the co-efficient of rolling resistance,  $C_n$  is the wheel numeric,  $b_w$  is the breadth of the wheel (m),  $d_w$  is the diameter of the wheel (m),  $W_w$  is the axle load (kN), and CI is cone index (kPa). The rolling resistance (R) can be expressed by multiplying the coefficient of rolling resistance with the wheel load (kN) following equation 28.

$$R = \left( \frac{1.2 \times W_w^2}{C_n} \right) + 0.04 W_w \quad (28)$$

The rolling resistance (R) of the tractor wheel was estimated on diverse soil types such as sand, silt, and clay. In the radish field, various soil variables were determined under different soil conditions, as shown in Table 4 (Park et al., 2006).

The design variables (axle load, wheel numeric, rolling resistance co-efficient) were changed to investigate the maximum rolling resistance. Drawbar pull and power demands are vital facts for investigating the tractor efficiency and have thus been reviewed with considerable data when correctly attaching the radish collector to the tractor (Table 5). Draft performance relies on engine power, load distribution on the wheels, hitch type, and the characteristics of the soils. The soil stress-deformation relationship is considered for calculating traction force based on wheel slip or lost motion. A drawbar pull and wheel slip relationship was investigated under field conditions. The drawbar pull was calculated after overcoming the rolling resistance and was expressed in equations 29 to 31 (Macmillian, 2010).

$$D = R_m = \rho W_w \sin \beta \quad (29)$$

$$\rho = \frac{1}{B_n} + 0.04 + \frac{0.5s}{2\sqrt{B_n}} \quad (30)$$

$$S = 1 - \frac{V_a}{V_t} \quad (31)$$

Where,  $D$  is the drawbar pull (N),  $R_m$  is the motion resistance of the machine assist wheel (N),  $W_w$  is the weight of the tractor rear wheel (N),  $\beta$  is the tilted angle between the wheel load and the soil ( $^\circ$ ),  $\rho$  is the coefficient of motion resistance,  $B_n$  is a dimensionless ratio,  $S$  is wheel slip (%),  $V_a$  is the actual velocity, and  $V_t$  is the theoretical velocity.

Drawbar power ( $P$ ) is crucial for the four-row tractor-mounted radish collector under development. However, the draft power was estimated in terms of recommending the appropriate size of the tractor. Shin et al. (2013) reported that approximately 60% of tractors are categorized from 44 to 74 kW in the Republic of Korea, usually used for conducting various farm operations. Drawbar power (kW) on hard, firm, tilled, and soft (sandy) soil conditions were predicted using equation 32.

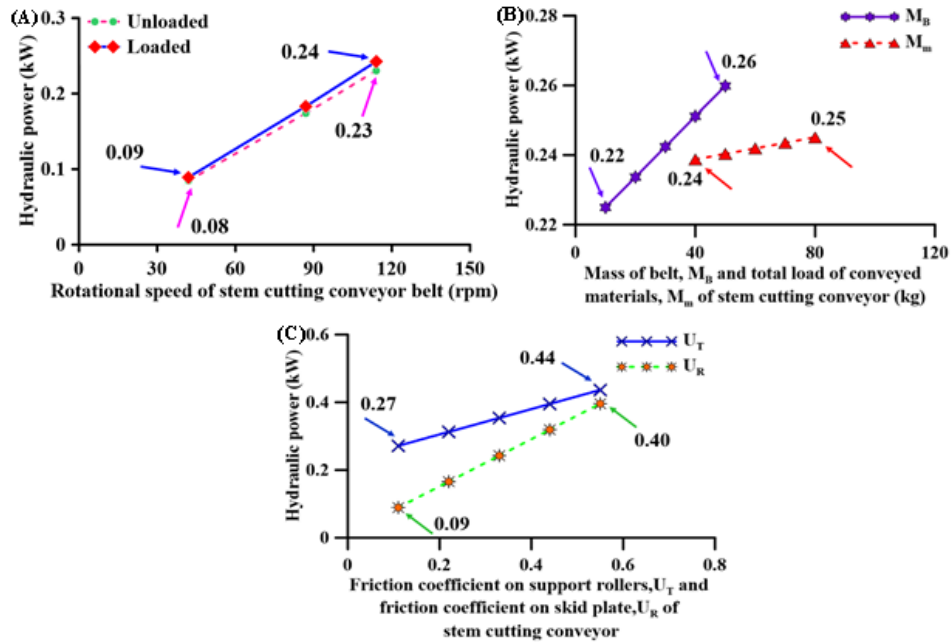
$$P = \frac{D \times V_t}{1,000} \quad (32)$$

The required drawbar power was investigated for the design variables weight of the tractor wheel ( $W_w$ ), and the tilted angle ( $\beta$ ), which were modified and shown in Table 5.

## Results and Discussion

### Hydraulic power for stem-cutting conveyor belt

In the theoretical approach, the average hydraulic power of the stem-cutting conveyor belt was calculated as between 0.08 and 0.23 kW and 0.09 and 0.24 kW under unloaded and loaded conditions, respectively, with conveyor belt motor speeds of 42 to 114 rpm (Fig. 8A). In consideration of design variables, the hydraulic power was slightly increased between 0.22 and 0.26, and 0.24 kW and 0.25 kW due to the modification of the mass of the belt ( $M_b$ ) and the total load of conveyed materials (radish weight) ( $M_m$ ), respectively (Fig. 8B). It is likely that the hydraulic power requirement was significantly raised somewhat between 0.27 and 0.44 kW and 0.09 and 0.40 kW because of the reconsideration of the friction coefficient on the support rollers ( $U_T$ ) and the skid plate ( $U_R$ ), respectively (Fig. 8C). The results showed that when the value of the design variables of the stem-cutting conveyor belt increased, the hydraulic power requirement also increased simultaneously. However, the benefit of a designed hydraulic pump system is that it can operate under different dynamic loads and the theoretical calculations would reflect the real operational conditions in the field (Máchal et al., 2013).



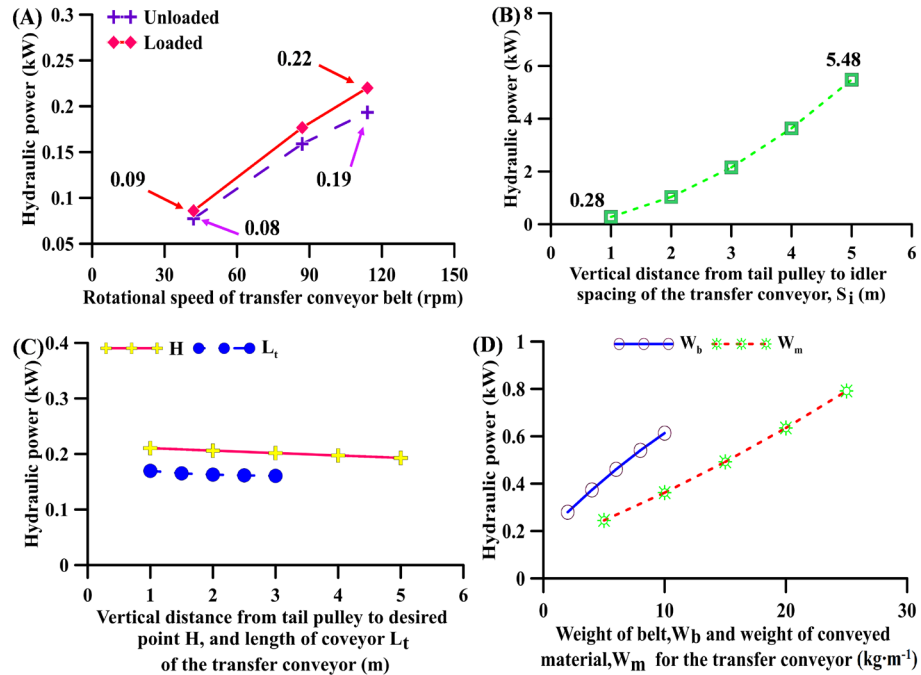
**Fig. 8.** (A) Corresponding hydraulic power on different operational speeds of stem cutting conveyor belt under unloaded and loaded conditions, (B - C) relation to hydraulic power of the stem cutting conveyor belt with the variables with  $M_B$ ,  $M_m$ , and  $U_T$ ,  $U_R$ , respectively.

### Hydraulic power for transfer conveyor belt

The calculated results for the transfer conveyor belt were shown between 0.08 and 0.19 k, and 0.09 and 0.22 kW under unloaded and loaded conditions, respectively, at conveyor belt motor working speeds of 42 - 114 rpm (Fig. 9A). According to Table 1, the design variables were changed for the manufacturers, however, the vertical distance from the central tail pulley to the idler spacing ( $S_i$ ), the desired point ( $H$ ), and the length of the transfer conveyor ( $L_t$ ) were revised and the required power of the hydraulic motor was varied between 0.28 and 5.48, 0.21 and 0.19, and 0.17 and 0.16 kW, respectively (Figs. 9B - C). In addition, when the weight of the belt ( $W_b$ ) and conveyed material ( $W_m$ ) were modified to check the optimal design of the transfer conveyor, it seemed that the hydraulic power requirement significantly increased between 0.28 and 0.61, and 0.24 and 0.79 kW, respectively (Fig. 9D). For designing the transfer conveyor belt, the result showed the hydraulic power requirement decreased due to the variables  $H$  and  $L_t$ , while it increased under the influence of the variables  $S_i$ ,  $W_b$ , and  $W_m$  (the variables are noted in Fig. 9). According to Table 1, when the  $S_i$  value was considered up to the maximum range of 5 m, the maximum requirement of hydraulic power for the transfer conveyor increased up to 5.48 kW (Fig. 9B).

### Hydraulic power for stem-cutting motor

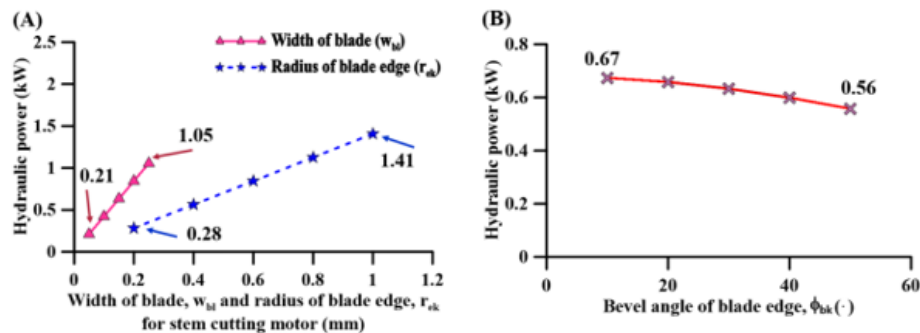
Considering the field conditions, the hydraulic power for the stem-cutting motor was recorded at 0.18 and 0.19 kW at the maximum cutting speed of  $251 \text{ rad}\cdot\text{s}^{-1}$  under unloaded and loaded conditions, respectively. The required power variation when loaded was found at 0.01 kW (a 5.55% increase) compared to the unloaded state (the data not shown). Zhu et al. (2021) investigated three kinds of disc-type cutting blades with a rectangular light edge, triangular tooth edge, and circular blade tooth. They showed that to calculate the power consumption for different blades, cutting effects and resistance, as well as



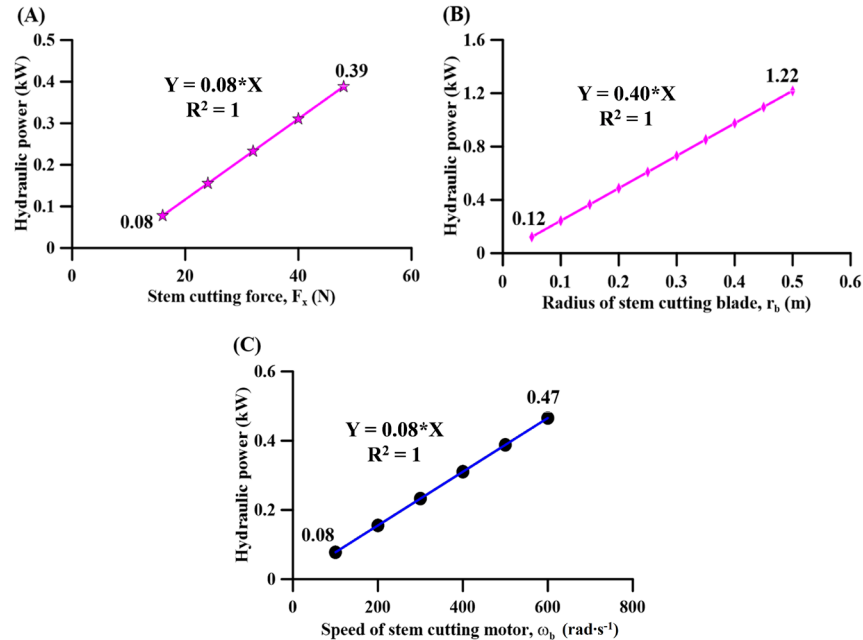
**Fig. 9.** (A) Corresponding hydraulic power and operational speed of transfer conveyor belt under unloaded and loaded conditions and variation of hydraulic power of the transfer conveyor belt to (B)  $S_i$ , (C)  $H$ ,  $L_t$ , and (D)  $W_b$ ,  $W_m$ .

other factors should be considered. Considering the factors, the hydraulic power requirement of the stem-cutting blade was calculated. Thus, the design parameters of the width of the blade ( $w_{bl}$ ) and radius of the blade edge ( $r_{ek}$ ) were revised, as shown in Table 2. The results showed that because of the changing of those parameters, the hydraulic power increased from 0.21 to 1.05, and 0.28 to 1.41 kW, respectively. In contrast, when the bevel angle of the blade edge ( $\Phi_{bk}$ ) was increased, the hydraulic power requirement decreased to between 0.67 and 0.56 kW (Fig. 10). The stem-cutting force ( $F_x$ ), radius of the stem-cutting blade ( $r_b$ ), and speed of the stem-cutting motor ( $\omega_b$ ) were rearranged (Table 2), increasing the requirement of the DC motor power by between 0.08 and 0.39, 0.12 and 1.22, and 0.08 and 0.47 kW, respectively (Fig. 11).

Dange et al. (2012) found a positive correlation between the cutting force and cutting power required for pigeon pea crops and the cross-sectional area of the cutting point and moisture content during the harvesting period. Regression analysis was conducted to determine the effect of stem-cutting force ( $F_x$ ), the radius of the stem-cutting blade ( $r_b$ ), and speed of the stem-



**Fig. 10.** Variation of hydraulic power for stem cutting motor with (A)  $w_{bl}$ ,  $r_{ek}$  and (B)  $\Phi_{bk}$ .



**Fig. 11.** Relation between hydraulic power of stem cutting motor with (A)  $F_x$ , (B)  $r_b$ , and (C)  $\omega_b$ .

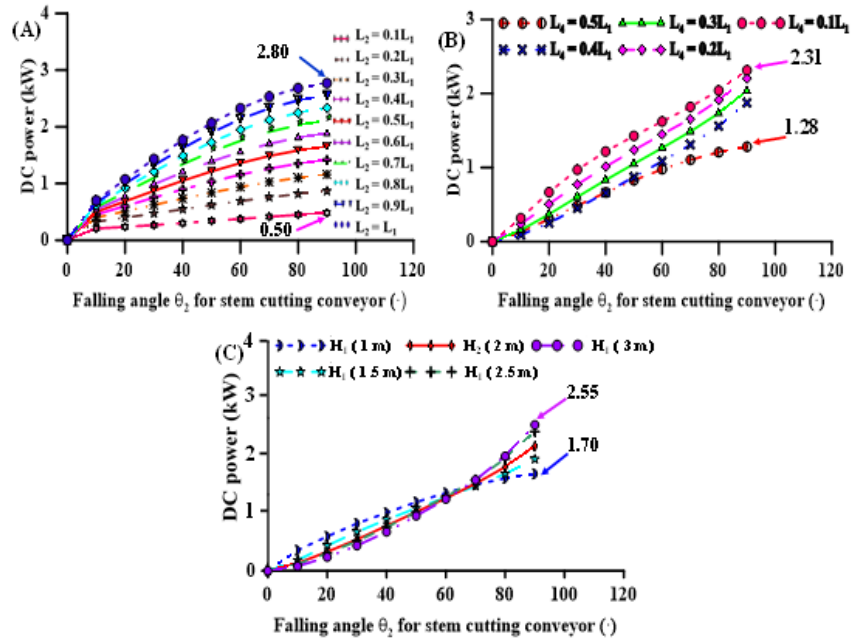
cutting motor ( $\omega_b$ ). The  $R^2$  value of the regression model showed a satisfying correlation between the hydraulic power and the force, radius, and speed of the stem-cutting blade. Linear equations were used to predict the stem-cutting power. The stem-cutting power was increased linearly along with the stem-cutting force and the radius of the cutting blade, and speed increased as shown in the model due to the coefficient of X.

### DC winch power for folding and unfolding stem-cutting conveyor

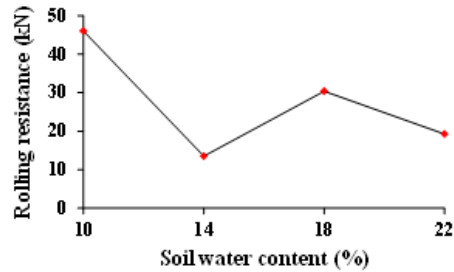
According to Table 3, the design consideration of the DC winch motor power was changed and evaluated. The results showed that the maximum power requirement of the DC winch motor was 2.07 kW for folding or unfolding the stem-cutting conveyor. When the location of the winch tie bracket ( $L_2$ ) for the stem-cutting conveyor was moved to the left side of the conveyor, the maximum average power varied from 0.50 to 2.80 kW with a falling or raising angle ( $\theta_2$ ) in the range of 0–90°. Similarly, when the horizontal distance between the winch motor and pivot point ( $L_4$ ) was shifted from the left to the right side of the conveyor, the maximum power consumption was recorded from 1.28 to 2.31 kW, respectively, as shown in Fig. 12B. When the vertical distance between the winch motor and stem-cutting conveyor ( $H_1$ ) was changed, the maximum power consumption varied from 1.70 to 2.55 kW (Fig. 12C). It was concluded that more power was required when the value of the design variables was at maximum, and the selected DC winch motor was able to produce sufficient power to fold and unfold the stem-cutting conveyor. Due to the belt conveyor length, the dynamic system analysis considered the variation of tensions and transient motion of the components before and after the operation of the conveyor.

### Draft power of a tractor-mounted radish collector

Fig. 13 showed the different results of rolling resistance under various soil conditions (sand, silt, and clay soil). Soil water



**Fig. 12.** Variation of DC falling power and angle to (A) length of  $L_2$ , and (B) length of  $L_4$  for stem cutting conveyor, and (C) length of  $H_1$  for stem cutting conveyor.



**Fig. 13.** Rolling resistance (kN) under different upland crop soil types with their average soil water content values.

content (%) variation impacted the cone index and it changed the rolling resistance from 13.53, 19.27, 30.41, and 46.04 kN, respectively, under different soil conditions (Table 4). In contrast, changing the wheel numeric of the tractor also changed the value of the rolling resistance. When the wheel numeric ( $C_n$ ) was bigger, the rolling resistance values were highest. When the value of the cone index increased, the rolling resistance value decreased. In soft soil conditions, the value of rolling resistance was less compared to other types of soils, and it happened because of the lower inflection pressure and rut depth of the soil. For the same reason, the effect of rolling resistance was comparatively different in hard soil conditions (Kurjenluoma et al., 2009). The rolling resistance was less influenced by tractor speeds in agricultural land but the parameters (i.e., wheel parameters, vertical weight, and soil varieties) played an important role in changing the rolling resistance (Taghavifar and Mardani, 2013; Gharibkhani et al., 2014; Mardani, 2015).

When the heavyweight was applied on the rear wheel ( $W_w$ ) of the tractor, the draft pull and power varied from 0.10 to

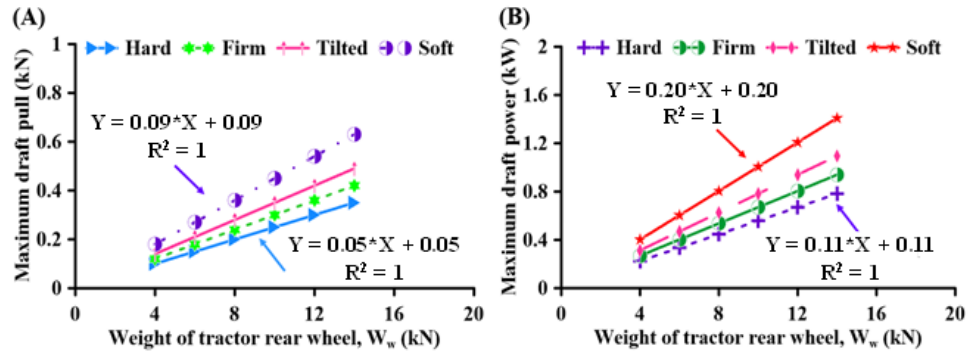


Fig. 14. Various weights of tractor rear wheel to (A) draft pull, and (B) draft power in different soil types.

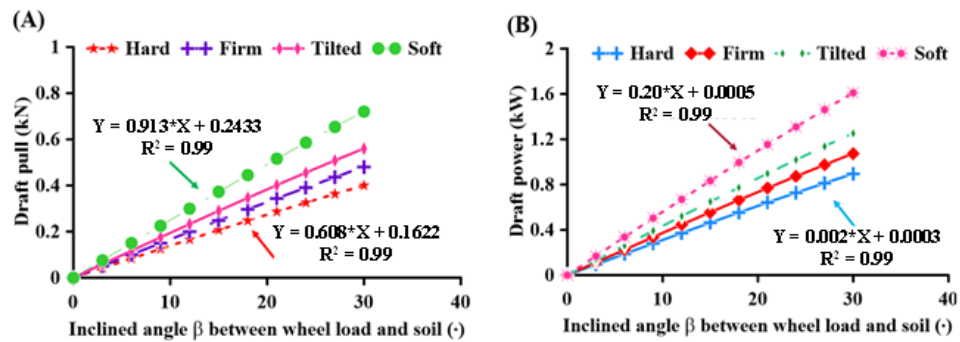


Fig. 15. Various inclined angle  $\beta$  to (A) draft pull, and (B) draft power in different soil types.

0.40, 0.12 to 0.48, 0.14 to 0.56, and 0.18 to 0.72 kN, and from 0.22 to 0.89, 0.27 to 1.07, 0.31 to 1.25, and 0.40 to 1.61 kW, respectively, at a considerable tilted angle of  $30^\circ$  for hard, firm, tilted, and soft (sandy) soils (Fig. 14). The regression equations were developed and calculated the maximum draft pull and power of different-sized tractors (44 to 74 kW) where the coefficient of determination value was higher ( $R^2 = 1$ ), which proved the positive correlation between the applied weight of the tractor rear wheel to draft pull and power.

The coefficients of motion resistance ( $\rho$ ) were 0.05, 0.06, 0.07, and 0.09 in hard, firm, tilted, and soft (sandy) soil types, respectively. If the tilted angle ( $\beta$ ) between total load and soil were partially increased, the draft pull and power were 0.40, 0.48, 0.56, and 0.72 kN, and 0.89, 1.07, 1.25, and 1.61 kW, respectively, at a ground speed of 8.05 kph for the large-sized (74 kW) tractor under different soil conditions (Fig. 15).

Overall, the theoretical total maximum power was observed at 4.78 kW, which consisted of 1.10 kW (23%) of hydraulic power, 1.61 kW (34%) of draft power, and 2.07 kW (43%) of DC power, at the existing operational conditions of the radish collector. However, the total maximum power requirement was determined to be 24.35 kW based on the design parameters for tractor-mounted radish collectors for conducting the operation under field conditions. This consisted of 11.87 kW (49%) of hydraulic power, 4.82 kW (20%) of draft power, and 7.66 kW (31%) of DC power. The motors and tractors (available in the Republic of Korea) selected in this study could produce sufficient power to operate the tractor-mounted four-row radish collector according to the calculated design variables. Furthermore, the power requirement analysis of the radish collector



suggested that the DC winch motor used for unfolding or folding stem-cutting conveyors should be replaced with hydraulic cylinders that can fold or expand the conveyor without the battery. Finally, a single conveyor belt and hydraulic motor without a vertical winch frame and cable should be redesigned to reduce overall power consumption.

## Conclusion

This study was basic research that investigated the power requirement of a tractor-mounted radish collector under development. The overall power of the collector was calculated and evaluated considering different design parameters (i.e., tractor size, soil type, and design structure) to obtain better performance from the introduced radish collector under loaded and unloaded conditions. According to the power analyses, the maximum power was 2.07 kW for running the DC motors. In the normal design considerations, the hydraulic power and draft power were 1.10 kW, and 1.61 kW, respectively. The power requirement of the radish collector increased or decreased due to changes in design variables (i.e., conveyor, conveying belt, stem-cutting blade, etc.). In this regard, the maximum predicting power was observed when the designed variable values were maximum. The radish collector can be operated using the recommended systems with available tractors (44 to 74 kW) and any type of soil conditions (hard, firm, tilted, and soft). Field testing and validation of the tractor-mounted four-row radish collector are necessary to verify the theoretical results and improve its reliability in field conditions. The development of the radish collector could enhance help farmers collect the radishes faster and with minimal manpower.

## Conflict of Interests

No potential conflict of interest to this article was reported.

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## Authors Information

Khine Myat Swe, <https://orcid.org/0000-0002-6565-4932>

Mohammad Ali, <https://orcid.org/0000-0002-1822-3005>

Milon Chowdhury, <https://orcid.org/0000-0002-9887-7980>

Md Nasim Reza, <https://orcid.org/0000-0002-7793-400X>

Md Ashrafuzzaman Gulandaz, <https://orcid.org/0000-0002-6391-1165>

Sang-Hee Lee, <https://orcid.org/0000-0003-2028-9035>

Sun-Ok Chung, <https://orcid.org/0000-0001-7629-7224>

Soon Jung Hong, <https://orcid.org/0000-0003-2584-5379>

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