International Journal of Environment and Climate Change

Volume 14, Issue 1, Page 871-881, 2024; Article no.IJECC.112263
ISSN: 2581-8627
(Past name: British Journal of Environment & Climate Change, Past ISSN: 2231–4784)

Design Considerations in Electric Multi-Tool Carrier for Protected Cultivation

Yamagar S. G. [a++*], Adarsh Kumar [a#], H. L. Kushwaha [b†], Awani Kumar Singh [c#] and Ramasubramanian V. [d†]

[a] Division of Agricultural Engineering, ICAR-IARI, New Delhi, 110012, India.
[b] Division of Agricultural Engineering and Renewable Energy, ICAR-Central Arid Zone Research Institute (CAZRI), Jodhpur, Rajasthan, 342003, India.
[c] Division of Vegetable Science, ICAR-IARI, New Delhi, 110012, India.
[d] Division of Research System Management, ICAR-NAARM, Hyderabad, 500030, India.

Authors' contributions

This work was carried out in collaboration among all authors. All authors read and approved the final manuscript.

Article Information

DOI: 10.9734/IJECC/2024/v14i13905

Open Peer Review History:
This journal follows the Advanced Open Peer Review policy. Identity of the Reviewers, Editor(s) and additional Reviewers, peer review comments, different versions of the manuscript, comments of the editors, etc are available here: https://www.sdiarticle5.com/review-history/112263

Original Research Article

Received: 10/11/2023
Accepted: 15/01/2024
Published: 25/01/2024

ABSTRACT

The design considerations of a prototype electric multi-tool carrier rely heavily on the selection of a prime mover, where the Brushless DC (BLDC) motor emerges as the optimal choice. The BLDC motor's compact size, reduced weight, and high torque output make it well-suited for the intended agricultural operations. The transition from the prime mover to the power transmission system involves determining the ground wheel diameter and achieving a target speed of 10 km/h. The calculated speed ratio of the BLDC prime mover to the power transmission system (25:1) ensures efficient and synchronized multi-tool carrier operation. During testing, the BLDC prime mover

---

[++] *Research Scholar;*
[#] *Principal Scientist;*
[†] *Head;*
**Corresponding author: E-mail: ysujitrao@gmail.com;*

exceeded expectations, producing a torque of 90.24 Nm, surpassing the specified requirement of 67.53 Nm. This robust performance highlights the prime mover capability to meet and exceed operational demands across varying speeds. The meticulous consideration of torque requirements in the design phase validates the BLDC motor's efficiency for the proposed electric multi-tool carrier, emphasizing its suitability for the envisioned application.

Keywords: Electric multi-tool carrier; BLDC Prime mover; torque requirements; power transmission system and operational efficiency.

1. INTRODUCTION

The escalating global concern over environmental degradation and the pressing need to mitigate the adverse effects of climate change have thrust the transportation sector into the spotlight as a significant contributor to greenhouse gas (GHG) emissions. In this context, the imperative transition to electric vehicles (EVs) has emerged as a pivotal solution for fostering sustainable mobility in the present era. The combustion of fossil fuels in traditional internal combustion (IC) engine vehicles not only releases carbon dioxide, a major greenhouse gas, but also contributes to air pollution, posing severe threats to public health. Against this backdrop, the advent of EVs, powered by electricity stored in advanced battery technologies, presents a promising alternative that addresses both environmental and health concerns.

The primary motivation for embracing electric vehicles lies in their inherent ability to reduce greenhouse gas emissions and dependence on finite fossil fuel resources. The transport sector accounts for a substantial portion of global carbon dioxide emissions, primarily due to the widespread use of gasoline and diesel-powered IC engine vehicles. EVs, on the other hand, operate on electricity, which can be generated from a diverse range of sources, including renewable energy such as solar, wind, and hydropower. This shift towards electrification represents a paradigmatic change that aligns with global efforts to transition to a low-carbon economy and mitigate the impacts of climate change. Scientific research in the field of electric vehicle technology aims to not only optimizes existing systems but also to explore innovative solutions for enhancing the efficiency and sustainability of electric transportation [1,2].

Apart from the environmental benefits, the incorporation of electric vehicles into the mainstream automotive market is critical for reducing dependence on finite fossil fuel reserves. The conventional reliance on oil and gas for powering internal combustion engines not only perpetuates geopolitical tensions but also exposes economies to the volatility of oil prices. By transitioning to electric vehicles and fostering the growth of a sustainable energy infrastructure, nations can enhance energy security and reduce vulnerability to geopolitical uncertainties in the fossil fuel market. Scientific research endeavors in the realm of electric vehicle technology are, therefore, crucial for advancing battery storage capabilities, improving charging infrastructure, and optimizing energy management systems to ensure the seamless integration of EVs into the broader energy landscape.

In addition to environmental and energy security considerations, the adoption of electric vehicles holds immense promise for improving air quality and public health. Urban centers around the world grapple with the detrimental impacts of air pollution, primarily driven by vehicular emissions. Electric vehicles produce zero tailpipe emissions, thereby mitigating the release of harmful pollutants such as nitrogen oxides (NO_2) and particulate matter (PM) that contribute to respiratory illnesses and other health complications. As the global population continues to urbanize, the need for clean and sustainable transportation becomes imperative for ensuring the well-being of urban dwellers. Scientific research efforts focus on developing technologies that not only enhance the performance of electric vehicles but also address challenges related to battery recycling and the environmental impact of manufacturing processes, contributing to a holistic and sustainable approach to electric mobility.

The imperative transition to electric vehicles in the present time is a multifaceted solution that addresses pressing environmental, energy, and public health challenges. As governments, industries, and researchers collaborate to advance the science and technology behind electric vehicles, the vision of a sustainable and clean transportation future becomes increasingly attainable. This scientific research paper delves into the pivotal role of electric vehicles in the broader context of global sustainability, exploring advancements in technology, policy frameworks, and interdisciplinary collaborations that collectively pave the way for a transformative shift towards electric mobility.

The green energy has wide application agricultural mechanization and progressively becoming a hot research topic [3-5] Chauhan reported that fossil energy tends to dry up, energy crisis and environmental pollution are becoming more and more obvious, and traditional fuel vehicles turning into new energy vehicles are inevitable. The research and development of electric farm vehicle can effectively alleviate the energy crisis, which is significant for the long is term development of agricultural machinery.

2. METHODOLOGY

2.1 Electric Multi-Tool Carrier

The mass of a suggested prototype electric multi-tool carrier was calculated using compatible equipment (bed former, sprayer and pollinator) to determine the BLDC prime mover required to propel the carrier inside the protected structure. The prior work on self-propelled devices provided empirical evidence that inspired the mass aim of the prototype electric multi-tool carrier. According to several study evaluations, the mass of the vehicles ranges between 300 and 600 kg; hence, based on CAD prototype design model and market survey of the parts, the maximum 280 kg mass of the electric multi-tool carrier for protected cultivation was considered.

2.2 Drive Unit Design

The electric multi-tool carrier's power train consists of an electric prime mover, drive system with a lead acid battery acting as an energy buffer. Typically, only one electric prime mover is linked to the wheel shaft via a gearbox and differential system [6]. The electric energy is chemically stored in the lead acid battery, which is electrically coupled to the BLDC prime mover through regenerative controller and DC power electronic converter. Fig 1 depicts a typical electric multi-tool carrier drive system, which I also the sort of system explored in this inquiry.

Because an electric multi-tool carrier equipped with compatible equipment will be used in a range of field situations, it was determined to design a vehicle with a maximum speed of 20 km/h while the vehicle is not carrying any weight. Taking into account the same Table 1 expands on the carrier’s specification and summarizes the essential facts needed to determine the drive unit's power requirements.

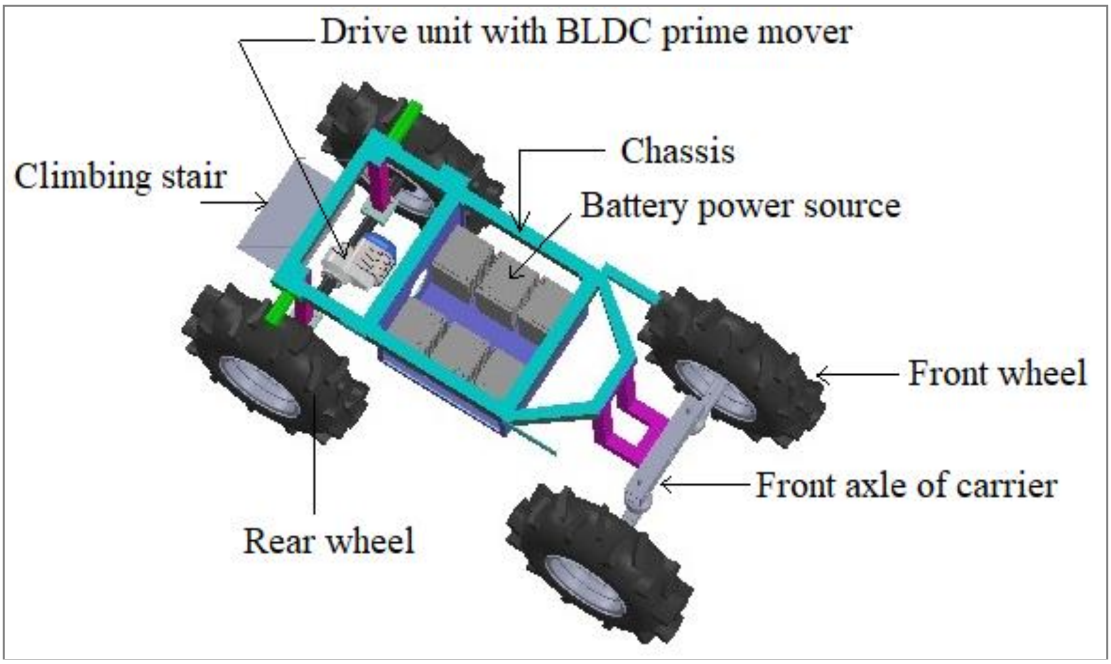

Fig. 1. Schematic sketch of a electric multi-tool carrier power train

Table 1. Details for electric multi-tool carrier specifications

Parameters	Specifications
Overall carrier dimensions ($l \times b \times h$), mm	2030×1575×2000
Track width, mm	660
Wheelbase, mm	1320
Traction device type and size, inches	Pneumatic tyre- 5.0/5.0-12.0
Rim- type and material	Disc type- Stainless steel
Rim dimeter, mm	304.8
Chassis material and size	MS rectangular pipe of 2 inch with 3 mm thickness
BLDC prime mover capacity, W	1500
Rated speed of prime mover, rpm	3300
Torque output of the prime mover, Nm	3.8-4.5
Mechanical transmission system type integrated between rear axle and prime mover	Gear motor type, power train final drive gear ratio 25:1 and torque: 120 Nm
MS rear axle track width, mm	850
Prime mover controller type	Regenerative DC 1500W
Power source	Lead acid battery
Battery pack capacity	72 V 42 Ah
Battery pack size ($l \times b \times h$), mm	500×560×300
MS front axle track width, mm	830
Loading capacity of rear axle, kg	2000
Braking system	Regenerative braking and drum braking system
Steering gearbox type	Worm gear and peg
Steering column height, mm	1000
Operator climb step material	Mild Steel
Operator climb step dimension ($b \times h$), mm	300×200
Operator's platform material	Wooden platform
Operator's platform dimensions ($l \times b \times t$), mm	$960 \times 660 \times 10$
MS Safety guard's dimensions ($l \times b \times h$), mm	860×735×880
Weight of the carrier, kg	280

2.3 Design Calculations for Electric Multi-Tool Carrier Power Train

2.3.1 Selection of transmission system

The dynamics of a vehicle aim to define vehicle movement on and off the road while it is under the influence of forces between the tyre and the road, aerodynamics, and gravity. As per Newton's second law of mechanics, the dynamical movement of the vehicle in one coordinate axis is generally determined by the sum of all forces acting on it in the same direction axis. A general form of the total tractive effort of a vehicle can be expressed as the following summation of the terms.

$$\mathrm{TTE} = \mathrm{RR} + \mathrm{GR} + \mathrm{FA} + \mathrm{F_{air}} + \mathrm{R_1} \qquad \text{Eq (1)}$$

Where *RR* is the rolling resistance, *GR* is the grade resistance, F_A is the acceleration force, F_{air} is the drag force due to air resistance and R_l is the declaration resistance. With change in the driving conditions the general equation should be modified. If a vehicle mass at a constant speed on a level surface then the term *GR* and R_l are ignored.

2.3.2 Factors affecting the required torque

Fig. 2 illustrates the various forces operating on moving vehicles. There were two significant power requirements. The average power needed under typical circumstances is the first and aids in determining the overall amount of energy storage needed. The following formulas, for which various researchers [7-10] have used different nomenclature, are used to determine the power.

1. Rolling resistance (*RR*)
2. Grade resistance (*GR*)
3. Acceleration force (*FA*)
4. Aerodynamic drag (F_{air})

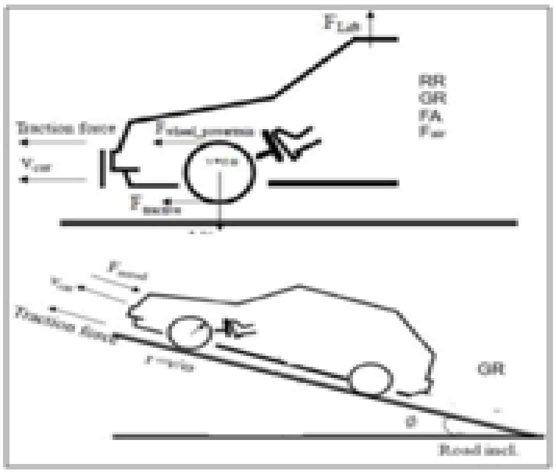

Fig. 2. Forces acting on moving vehicle

2.3.3 Calculating the rolling resistance

RollingResistanceistheopposingforcethatthevehiclemustovercomeduetotherollingmotionbetween the wheels and the surface of motion of the vehicle which is calculated as

$$\mathrm{RR} = \mathrm{GCW} \times \mathrm{Cr} \qquad \text{Eq (2)}$$

Where, RR is the rolling resistance GVW is the gross vehicle weight and C_r is the Coefficient of rolling resistance

Now in this case from Table 1 *GVW* = 2775 N and coefficient of rolling resistance C_r= 0.03 for off-road purpose hence putting the values in equation 2

PR=2775X0.03= 83.25 N

2.3.4 Calculating the grade resistance

The gradient resistance of the electric multi-tool carrier is the amount of gravitational resistance that it generates when it moves up or down the gradient surface. When the electric multi-tool carrier goes down from the gradient surface, the gradient resistance value is negative, i.e., the total resistance power required is reduced. Instead, the inertia energy is restored with an energy recovery system. Grade resistance is the form of gravitational force that tends to pull the vehicle back when climbing an inclined surface. The operating gradient of 0 to 3 % is very common; the grade resistance acting on the electric multi-tool carrier can be calculated as

$$\mathrm{GR} = \mathrm{GCW} \times \mathrm{Sin}\emptyset \qquad \text{Eq (3)}$$

Where, GR = Grade resistance, N; GCW = Gross multi-tool carrier weight, 2775 N; Ø= Grade or inclination angle (2 deg in the normal field)

$$\mathrm{GR} = 2775 \times \mathrm{Sin}2° = 96.85 \text{ N}$$

Hence, the grade resistance for the electrical multi-tool carrier was 96.85 N.

2.3.5 Calculating the acceleration force

The force helps the vehicle to reach a pre-defined speed from rest in a specified period. The motor torque bears a direct relationship with the acceleration force. The higher the torque, the less time the vehicle requires to reach a pre-defined speed. It is a function of the mass of the vehicle and is calculated as

$$\mathrm{FA} = \mathrm{m} \times \mathrm{a} \qquad \text{Eq (4)}$$

$$\mathrm{m} = \frac{\mathrm{GCW}}{\mathrm{g}}$$

Where, FA = Acceleration force; m = Mass of multi-tool carrier; g = Acceleration due to gravity (9.81 m/s^2); a = Required acceleration

Consider the maximum speed of the electric multi-tool carrier as 10 km/h at a lower speed switch on the accelerator to perform the operation and return safely to the destination when the vehicle is empty. In this case, the gross weight is reduced but the normal condition during the spraying operation; the estimated gross carrier weight is 283 kg, assuming due to the spray tank capacity, the maximum speed is 3.5 km/h. Hence, the phenomenon of these two cases of carrier weight must be determined for maximum acceleration force calculation for an empty carrier with a maximum speed of 10 km/h and maximum weight during spraying operation with a speed of 3.5 km/h. The required maximum acceleration force considered for the power requirement from these two cases (loaded and unloaded conditions) and calculated as follows:

The empty electrical multi-tool carrier runs to the destination at a maximum speed of 10 kemps; in this condition, the weight of the vehicle is reduced by eliminating the weight of water from the sprayer tank and it should be 2531 N (GCW of the unloaded vehicle) and electric motor accelerating force required to reach maximum speed in10 seconds.

$$\therefore \mathrm{m} = \frac{\mathrm{GCV}}{\mathrm{g}} = \frac{2531}{9.81} = 258\ \mathrm{kg}$$

a = 10 $\frac{\mathrm{km}}{\mathrm{h}}$ (at maximum speed reached at 10 sec)

a = 0.277 m/s

So, acceleration is $\frac{0.277}{10}$ = 0.0277m/s^2

So, the acceleration force for the unloaded electrical multi-tool carrier is calculated from the above values of a and m,

$$\therefore \mathrm{FA_{unload}} = \mathrm{m} \times \mathrm{a} = 258 \times 0.0277 = 7.14\ \mathrm{N}$$

Hence, the acceleration force for the unloaded electrical multi-tool carrier was found to be 7.14 N.

Similarly, the loaded electrical multi-tool carrier runs to its destination at a maximum speed of 3.5km/h; in this condition, the gross carrier weight of the vehicle should be 2777 N and the electric motor accelerating force required to reach maximum speed in 5seconds.

$$\mathrm{m} = \frac{\mathrm{GCV}}{\mathrm{g}} \quad \text{Eq (5)}$$

$$= \frac{2777}{9.81} = 283\ \mathrm{kg}$$

a = 3.5 $\frac{km}{h}$ (at maximum speed reached at 5 sec)

a = 0.194 m/s

So, acceleration is $\frac{0.194}{5}$ = 0.038 m/s^2

So, the acceleration force for the loaded electrical multi-tool carrier is calculated from the above values of a and m,

$$\mathrm{FA_{load}} = 283 \times 0.0388 = 10.75\ \mathrm{N}$$

Hence, the acceleration force for the loaded electrical multi-tool carrier was found to be 10.75 N. In the above unload and load condition, the acceleration force required in the load condition is more than that of the unload condition. Hence, the values of FA at load condition, *i.e.,* 10.75 N considered for calculating the total tractive effort.

2.3.6 Aerodynamic drag

As an electrical multi-tool carrier runs on a road, the relative air movement occurs opposite to the driving directions of the multi-tool carrier, even if no wind is there. Because of this airflow, the relative experiences aerodynamic forces such as drag and lift on the vehicle and operator's body. As the electrical multi-tool carrier is designed for indoor low-speed operations, only drag force is considered and uplift forces are eliminated. An air resistance (F_{air}) on the electric multi-tool carrier for drag force is given by

$$\mathrm{F_{air}} = \frac{\mathrm{S_a}}{2} \times \mathrm{C_d} \times \mathrm{V}^2$$

Where, S_a = Multi-tool carrier frontal surface area; C_d = Drag coefficient; V = multi-tool carrier speed

As the effect of air resistance on the electrical multi-tool carrier is negligible due to very low operating speed and is designed especially for the protected structure where wind velocity is almost absent, aerodynamic drag was not included in further calculations.

2.3.7 Finding the total tractive effort

To determine the total tractive effort to select the electric motor capable of producing enough torque to run the electrical multi-tool carrier. It is the total force required to move the vehicle with the desired characteristics and is the sum of the forces calculated in the above sections. Therefore, the total tractive effort from the modified total tractive effort general equation calculated is as

$$\mathrm{TTE} = \mathrm{RR} + \mathrm{GR} + \mathrm{FA_{load}}$$

Where, TTE = Total tractive effort; RR = Rolling resistance; GR = Grade resistance and FA_{load} = Acceleration force of load

So, the total tractive effort for the electrical multi-tool carrier is calculated from the above calculated values of RR, GR and FA_{load}

$$\mathrm{TTE} = 83.28 + 96.88 + 10.75 = 190.91\ \mathrm{N}$$

Hence, the total tractive effort for the electrical multi-tool carrier was found to be 190.91 N and used for the further design of the multi-tool carrier.

2.3.8 Determination of wheel motor torque

The required ground wheel motor torque computation based on the tractive effort is given below.

$$\tau = \mathrm{R_f} \times \mathrm{TTE} \times \mathrm{R_w} \qquad \text{Eq (7)}$$

Where,

R_f = resistance factor
$\mathrm{R_w}$ = tractor device radius, m
τ = wheel motor torque, Nm

The resistance factor value typically ranges between 1.1 to 1.15 for friction losses between the wheels, axles and drag on the motor bearings. So, the motor wheel torque for the electrical multi-tool carrier is calculated from the values of R_f (1.1) and R_w $(0.236\ \mathrm{m})$.

$$\tau = 1.1 \times 190.91 \times 0.236$$

$$\tau = 49.50\ \mathrm{Nm}$$

Hence, the motor wheel torque for the electrical multi-tool carrier was found to be 49.50 Nm and used for further design.

2.3.9 Verification of the model

The prime mover transmits the required torque from the drive wheels to the ground. The maximum wheel tractive torque ($\tau_{\max}$) is equal to the normal load times the friction coefficient between the wheel, hub and the radius of the driving wheel and are calculated as,

$$\tau_{\max} = \mu \times \mathrm{GCW} \times \mathrm{R_w} \qquad \text{Eq (8)}$$

$$= 0.8 \times 283 \times 0.236 = 53.45\ \mathrm{Nm}$$

For satisfactory performance of the electrical multi-tool carrier, the following conditions must be satisfied,

$$\tau_{max} \geq \tau \qquad \text{Eq (9)}$$

So, the maximum wheel tractive torque and motor wheel torque for the electrical multi-tool carrier able to satisfy the above condition is as

$$53.45\mathrm{Nm} \geq 49.50\ \mathrm{Nm}$$

From the above, total tractive effort is the net horizontal force the drive wheels apply to the ground. As the design has two drive wheels, the force applied per drive wheel is half the calculated value of total tractive effort. This quantity does not change with the number of drive wheels. Hence, the wheel torque calculated above is the total torque. The motor torque must be greater than or equal to the computed wheel torque, *i.e.,* 53.45 Nm, while selecting the motor.

The maximum tractive torque represents the maximum torque applied for each drive wheel before slipping occurs. The total wheel torque calculated 0 $r$must be less than the sum of the maximum torques for all drive wheels, *i.e.,* r_{max}. Now, in this case, from the above calculated values, it is observed that it satisfies all conditions. Hence, the design of an electrical multi-tool carrier is safe.

2.3.10 Calculating the motor power requirement

Considering the power required as 1.5 kW, a 3300 rpm BLDC motor can drive the electrical multi-tool carrier. To determine whether the motor specification whether suitable or not, it is necessary to check load calculations. The power requirement is calculated as

$$\mathrm{P} = \frac{2\pi \mathrm{NT}}{60} \qquad \text{Eq (10)}$$

Where, P = Power, W; N = Motor, rpm and T = Torque, Nm

$$\therefore \mathrm{T} = \frac{60 \times \mathrm{P}}{2 \times \pi \mathrm{N}} = \frac{60 \times 1500}{2\pi \times 3300} = 4.34\ \mathrm{N}$$

The maximum velocity of the electric multi-tool carrier is 10 km/h on unloaded conditions, which is converted in m/s as

$$= 10 \times \frac{5}{18} = 2.78\ \mathrm{m/s}$$

Whereas velocity is written as

$$\mathrm{V} = \mathrm{R_w} \times \omega \qquad \text{Eq (11)}$$

Where, V = Velocity, m; Rw = Radius of the wheel; w = Angular movement, rad/sec

$$\therefore \omega = \frac{v}{R_w} \quad \text{Eq (12)}$$

$$= \frac{2.78}{0.236} = 11.77 \text{ rad/sec}$$

To find out the motor speed, we know that

$$\omega = \frac{2\times\pi\times N}{60} \quad \text{Eq (13)}$$

$$\therefore N = \frac{11.77 \times 60}{2 \times 3.14} = 112 \text{ rpm}$$

The available torque on the shaft from the above calculated rpm by using the following equation

$$T = \frac{60\times P}{2\times\pi\times N} \quad \text{Eq (14)}$$

$$\therefore T = \frac{60 \times 1500}{2 \times 3.14 \times 112} = 127.95 \text{ Nm}$$

2.3.11 Relation between torque, power and speed

The basic formulation of power is shown in the equation. Taking P=1500 W and rpm = 3300 with 90 percent transmission efficiency.

$$\therefore T = \frac{60 \times 1500}{2 \times \pi \times 2970} = 4.82 \text{ Nm}$$

2.3.12 Equations for rpm at different stages of transmission

N is the motor speed in rpm

Where,

N = 3300 rpm

As the motor is 90 percent efficient, N_1 is given as

$$N_1 = 0.9 \times N \quad \text{Eq (15)}$$

$$\therefore N_1 = 0.9 \times 3300 = 2970 \text{ rpm}$$

The output with gearbox is

$$\frac{N_1}{N_2} = \frac{D_2}{D_1} \text{ Or } N_2 = \frac{N_1}{GR} \quad \text{Eq (16)}$$

Where, D_2:D_1 is the gear ratio (GR) or speed ratio and 1:25 is the ratio assumed to get the desired torque for an electrical multi-tool carrier.

Therefore,

$$\therefore N_2 = \frac{2970}{25} = 118 \text{ rpm}$$

Now, calculating the efficiency of the transmission system as 90 %, we have,

$$\therefore N_3 = 0.9 \times N_2$$

$$\therefore N_3 = 0.9 \times 118 = 106.2 \text{ rpm(say 107 rpm)}$$

Torque at the wheel on this rpm by putting the values in the equation

$$\therefore T_w = \frac{60 \times 1500}{2\pi \times 107} = 133 \text{ Nm}$$

Velocity at the wheel is given by

$$\therefore v = \frac{\pi d \times N_3}{60} \quad \text{Eq (17)}$$

$$= \frac{3.14 \times 0.45 \times 107}{60} = 2.51 \text{ m/s}$$

The tractive effort developed by the electric BLDC motor on the driven wheel is shown in Fig 3.

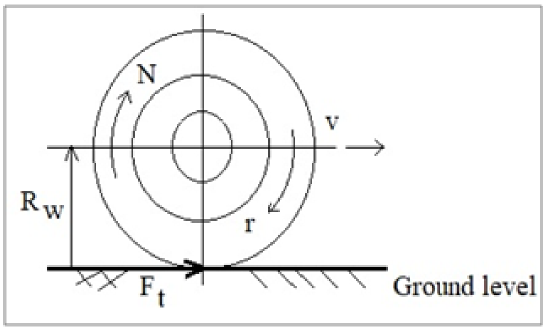

Fig. 3. Tractive effort developed by a driven wheel

The tractive effort developed by the motor is calculated as

$$T_w = F_t \times R_w \quad \text{Eq (18)}$$

Where, T_w = Torque at the wheel; F_t = Normal tractive force acting on the wheel; R_w = Radius of wheel

$$F_t = \frac{133}{0.236} = 563.55 \text{ N}$$

In addition, the force required on the road for a maximum speed of 10 km/h is given as:

$$\text{Acceleration} = \left(\frac{\text{Final velocity} - \text{Initial velocity}}{\text{Time}}\right) \qquad \text{Eq (19)}$$

It was noticed that the machine reached 10 km/h in 10 seconds. At the initial stage, the initial speed is 2 km/h and within 10 sec, the speed reaches 10 km/h. For this phenomenon, the acceleration is given as Therefore,

$$\mathrm{A} = \frac{10-2}{10} = 0.8\ \mathrm{m/s^2}$$

Thus calculating the acceleration force for this condition,

$$\mathrm{F} = \mathrm{m} \times \mathrm{a} \qquad \text{Eq (20)}$$

$$\mathrm{F} = 283 \times 0.8 = 226.4\ \mathrm{N}$$

Again, to find out the traction or total tractive force developed by the motor with gear ratio and motor efficiency as below

$$\mathrm{TTE_{motor}} = \frac{\text{Motor torque} \times \text{motor efficiency}}{\mathrm{R_w}} \qquad \text{Eq (21)}$$

$$= \frac{4.82 \times 25 \times 0.90}{0.236} = 459.53\ N$$

The calculated F_t (563.55N) and TTE_{motor}(459.53 N) represent the maximum tractive force developed by the motor, which is higher than the required total tractive force TTE (190.91 N). Hence, the motor with 1.5 kW power and 3300 rpm is suitable for the electrical multi-tool carrier. If the load increases because of unexpected load, the motor can sustain load until 1.50 percent of rated TTE.

A traction equation (William and Gill 1973) is used to determine whether the motor can sustain the load of the electrical multi-tool carrier.

$$\mathrm{w} = \frac{\mathrm{H}}{\mu} \qquad \text{Eq. (22)}$$

Assuming the maximum rolling friction between the electrical multi-tool carriers in extreme conditions is 0.075 (for soft grass)

$$\mathrm{w} = \frac{459}{0.075}$$

3. RESULTS AND DISCUSSION

The total tractive effort developed by a motor on the ground wheel can be calculated using Eq. 7. Since the design incorporates two rear drive wheels, the force applied per drive wheel is determined to be half of the calculated total tractive effort. This quantity remains constant regardless of the number of drive wheels, emphasizing its significance. Consequently, the wheel torque, as calculated above, represents the total torque for the design, given the presence of two drive wheels.

It is crucial to ensure that the prime mover torque exceeds or is equal to the computed wheel torque, which stands at 53.45 Nm. Therefore, when selecting the prime mover, its torque must be greater than or equal to the calculated wheel torque (i.e. 53.45 Nm). This criterion is essential for optimal performance and efficiency. Moreover, the maximum tractive torque signifies the maximum torque applicable before slipping occurs for each drive wheel. It is imperative that the total wheel torque (τ) is less than the sum of the maximum torques for all drive wheels (τ_{max}), ensuring operational stability (53.45 Nm ≥ 49.50 Nm).

Upon careful examination of the calculated values, it is evident that they satisfy all the specified conditions. The prime mover torque, exceeding the computed wheel torque, aligns with the stipulated requirement, affirming a judicious selection of the BLDC prime mover. This meticulous consideration of torque parameters contributes to the overall reliability and performance of the design, reflecting a comprehensive understanding of the intricate interplay between prime mover capabilities and wheel torque in the context of tractive effort.

The values derived from Eq. 18 (563.55 N) and Eq. 21 (459.53 N) signify the maximum tractive force developed by the prime mover, surpassing the required tractive force of 190.91 N. Consequently, the prime mover with a power of 1.5 kW and operating at 3200 rpm emerges as the more suitable choice for the electric multi-tool carrier. This selection is based on its ability to generate tractive forces exceeding the specified requirement.

In anticipation of diverse field conditions, should rolling friction values escalate to 0.055, particularly in soft grassy fields under extreme circumstances, the motor demonstrates its capability to efficiently carry a load of up to 400 kg. This resilience underscores the adaptability of the proposed electric multi-tool carrier in challenging terrains.

Furthermore, in practical working conditions, if there is an unexpected increase in load, the motor exhibits robust performance by sustaining the load up to 1.30 % of its rated Total Tractive Effort (TTE). This characteristic resilience ensures operational reliability, even in scenarios where the load deviates from the anticipated norm.

The determined parameters substantiate the suitability of a Brushless Direct Current (BLDC) prime mover with the following specifications: 1.5 kW power, 48V operating voltage, and 3200 rpm, for propelling the envisioned prototype of an electrical multi-tool carrier. The thorough assessment of tractive forces, load-bearing capacities, and adaptability to diverse conditions emphasizes the compatibility of the prime mover with the designated application, thereby reinforcing its appropriateness for integration into the conceptualized electric multi-tool carrier.

4. CONCLUSION

The selection of an electric BLDC prime mover stands as a pivotal parameter in the design of the prototype electrical multi-tool carrier, crucially dependent on factors such as the estimated weight and the initial effort required to set the prototype in motion. A meticulous analysis, considering the pros and cons of various prime mover types such as Induction motors, Brushless DC motors, and permanent magnet DC motors, led to the conclusion that the BLDC motor is the most optimal choice for the prototype model. The decision is rooted in its compact size, reduced weight, and the capacity to deliver high torque, making it exceptionally suitable for the envisioned application.

Transitioning from the electric motor to the power transmission system, the ground wheel's diameter was established at 450 mm. In order to achieve a target speed of 10 km/h, the estimated rotational speeds of the ground wheels were determined to be 120 rpm. Consequently, the speed ratio of the BLDC motor to the power transmission system was calculated as 25:1, considering the ground wheel diameter and the desired speed. This ratio ensures an efficient and synchronized power transmission from the BLDC motor to the ground wheels, a critical aspect for the overall performance and mobility of the electrical multi-tool carrier prototype.

In essence, the choice of the BLDC motor is a strategic decision based on a comprehensive evaluation of size, weight, and torque considerations, affirming its status as the most suitable prime mover for propelling the envisioned electrical multi-tool carrier prototype. The calculated speed ratio further underscores the precision in the design, ensuring an optimal match between the BLDC motor and the power transmission system for seamless and efficient operation.

The BLDC motor, with a dynamic operational range of 0 to 3000 rpm, attains its maximum speed through a full-throttle voltage supply. Consequently, we successfully propelled the vehicle, achieving speeds ranging from 0 to 8.29 km/h across various throttle settings. Considering the calculated parameters, it is imperative for the motor to generate torque exceeding 67.53 Nm, accounting for the final drive gear ratio. Remarkably, during testing, the motor demonstrated exceptional performance by producing a torque of 90.24 N-m, surpassing the specified requirement. This robust torque output reaffirms the motor's capacity to meet and exceed operational demands, highlighting its efficiency in powering the vehicle across the entire speed spectrum. The ability to surpass the torque threshold not only ensures the smooth acceleration of the vehicle but also underscores the resilience and capability of the BLDC motor in demanding operational scenarios. This outcome substantiates the meticulous consideration of torque requirements in the design phase, ultimately resulting in a motor that not only meets but surpasses performance expectations for the envisioned application.

ACKNOWLEDGEMENTS

This research was financially supported by ICAR-Indian Agricultural Research Institute (IARI), New Delhi (India). The authors would like to thank the Division of Agricultural Engineering, for the use of research facilities.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

REFERENCES

1. Lehnert C, English A, McCool C, Tow AW, Perez T. Autonomous sweet pepper harvesting for protected cropping systems. IEEE Robotics and Automation Letters. 2017;2(2):872-9.

2. Lehnert C, McCool C, Sa I, Perez T. A sweet pepper harvesting robot for protected cropping environments. Arxiv Preprint Arxiv:1810.11920; 2018.
3. Saurabh Chauhan Motor torque calculations for electric vehicle. International Journal of Scientific & Technology Research. 2015;4(8): 126-127.
4. Satnam Singh. Autonomous robotic vehicle for greenhouse spraying. Unpublished Thesis University of Florida; 2004.
5. Xiaofei Zhang. Design theory and performance analysis of electric. Tractor drive system. International Journal of Engineering Research & Technology. 2017;6(10):235-238.
6. Emma Arfa Grunditz. Design and assessment of battery electric power train, with respect to performance, energy consumption and electric motor thermal capacity. Published thesis for the degree of Doctor of Philosophy; 2016. (ISBN 978-917597-412-5).
7. Amin Ghobadpour, LoicBoulon, Hossein Mousazadeh, Ahmad Sharifi Malvajerdi, Shahin Rafiee. State of the art of autonomous agricultural off-road vehicles driven by renewable energy systems. Energy Procedia. 2019;162:4–13.
8. Bawden Owen John. Design of a lightweight, modular robotic vehicle. for the sustainable intensification of broadacre agriculture. Unpublished thesis, School of Electrical Engineering and Computer Science and Engineering Faculty, Queensland University of Technology; 2015.
9. Chethan MR, Krishna Reddy PC, Gopala Krishna SG; 2014.
10. Design analysis and develop a low-cost tractor operated Engineering Research and Technology. 2014;2(1):177-184.

Peer-review history:
The peer review history for this paper can be accessed here:
https://www.sdiarticle5.com/review-history/112263