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MOBILITY OF THE LGP-30 WHEELED VEHICLE ON PEAT TERRAIN IN MALAYSIA

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ABSTRACT
The Low Ground Pressure (LGP-30) wheel vehicle was tested on the 17 kN/m^2 low bearing capacity peat terrain in Malaysia. The number of field test was conducted at three different traveling speeds of 6 km/h, 10 km/h and 12 km/h by mounting the developed torque transducer with the vehicle. The sensitivity of the designed and developed torque transducer was found 20.04 mV/V per Nm. The mean value of the test results on tractive effort was found 11.66 kN, 9.89 kN and 9.06 kN for the travelling speed of 6 km/h, 10 km/h, and 15 km/h, respectively.

Keywords: Peat terrain; LGP-30 wheel vehicle; Tractive effort.

1. INTRODUCTION
The basic purpose of the development of LGP-30 wheel vehicle for performing the transportation operation of palm oil fresh fruit bunches and other goods on the moderate peat terrain in all working conditions. Design and development of the vehicle was made with a comprehensive understanding of the mechanical behavior of the peat terrain under loading conditions similar to those exerted by the vehicle. The peat terrain mechanical properties such as moisture content ω, bulk density γ, cohesiveness c, internal friction angle φ, shear deformation modulus K_W, surface mat stiffness m_m, and underlying peat stiffness k_p used in this study for the simulation of the Seladang LGP-30 wheel vehicle are as stated in Table 1.

Table 1: Peat terrain parameters

Parameters	Un-drained		Drained	
	Mean value	SD	Mean value	SD
ω, (%)	83.51	-	79.58	-
γ, (g/cm^3)	0.156	0.06	0.186	0.08
c, kN/m^2)	1.36	0.21	2.73	0.39
φ, (degree)	23.78	4.56	27.22	2.19
K_w, (cm)	1.19	0.10	1.12	0.17
m_m,(kN/m^3)	27.07	13.47	41.79	13.37
k_p, (kN/m^3)	224.38	52.84	356.8	74.27

Source : Ataur *et al.* (2004)
Key: SD-standard deviation

The specification of the Seladang LGP-30 is stated in Table 2.

Table 2: Seladang LGP-30's specifications

Vehicle Parameters		
Engine power @rpm, kW	E_p	40@3300
Maximum net torque, N-m	Q_n	775.74
Overall length, m	L	4.51
Overall width, m	B_v	2.17
Overall height, m	H	1.74
Weight, kg	W	2650
Load carrying capacity, kg	W_p	1000
Tire	16.9X24 8PR Tube Type	
Ground clearance, m	G_c	0.48
Wheel diameter, m	D	121.38
Wheel base, m	L_w	2.39

To compute the tractive performance in terms of tractive effort and motion resistance, a mathematical model was developed with considering the vehicle in straight motion and turning motion. The mathematical model was then used for simulating the vehicle tractive performance. The vehicle was constructed based on the custum built. The vehicle C.G was placed at the mid point of the wheel base by using 1000 kg counter balanced load at the bucket. The vehicle was tested on the moderate peat terrain in Sepang peat area for evaluating vehicle tractive performance. Before running the test, the terrain moisture content was tested to the laboratory by collecting the undisturbed samples from the terrain. Furthermore, the vehicle tractive effort was measured from the field by mounting the design and developed torque transducer which torque sensitivity was 20.04 mV/V per Nm.

2. KINEMATICS OF A ROLLING WHEEL

The kinematics model development of this study is carried out by classifying the whole kinematics model into two groups. Firstly: considering the rolling of the vehicle tire as a cycloid for determing the slippage of the vehicle and Secondly: considering the tire-terrain interaction mechanism by simplifying the general tractive equations and motion resistances equations of Bekker (1969), Wong *et al.,* (1982), and Wong (2001) for computing the vehicle tractive effort and motion resistance. On a peat terrain, the performance of the vehicle is, to a great extent, dependent upon the manner in which the vehicle interacts with the terrain. The force between the interaction of tire and peat terrain is determined by developing the kinematics model. Figure 1 shows the flow chart of the vehicle kinematics model.

Following assumptions are made in order to valid the mathematical model for wheel vehicle:

(1) Based on the study of peat terrain mechanical properties by Ataur *et al*. (2004), the vehicle critical slip sinkage and ground contact pressure are considered to be 10 cm and 17kN/m^2, respectively.

(2) Terrain reaction at all points on the contact patch is purely radial and is equal to the normal pressure beneath a horizontal contact area of the vehicle.

(3) The perimeter of the tire contact area and the velocity of the vehicle are considered to be constant.

(4) The rotational inertia of the vehicle rotating parts is neglected as it has no significantly affect on the vehicle performance for straight motion.

Sinkage of vehicle causes the power and traction loss. The vehicle performance is severely affected on the vehicle tractive performance if the sinkage of the vehicle is more than or equals to vehicle critical sinkage. The amount of sinkage of the vehicle can be computed by using the equation of *Ataur et al. (*2005):

$$z_0 = \frac{-\left(\frac{k_p D_h}{4m_m}\right) \pm \sqrt{\left[\left(\frac{k_p D_h}{4m_m}\right)^2 + \frac{D_h}{m_m} P_g\right]}}{2} \qquad (1)$$

With $P_g = (k_p z_o + \frac{4}{D_h} m_m z_0{}^2)$

where, P_g is the ground contact pressure of the vehicle in kN/m^2 and z_0 is the sinkage in m, m_m is the surface mat stiffness in kN/m^3, k_p is the underlying peat stiffness in kN/m^3, and D_h is the hydraulic diameter in m.

Slippage is one of the functional parameters for the vehicle traction mechanism. It is significantly affect on the vehicle tractive performance with increasing the vehicle's sinkage. The slippage of the wheel is determined by following the cycloid principle. A cycloid is the curve defined by a fixed point on a rim of the wheel as it rolls, or, more precisely, the locus of a point on the rim of a circle rolling along a straight line. Points of rolling rim describe a cycloid. Consider a wheel of radius R_0 which is free to roll along the x-axis. As the wheel turns, a point P on the tire traces out a curve.

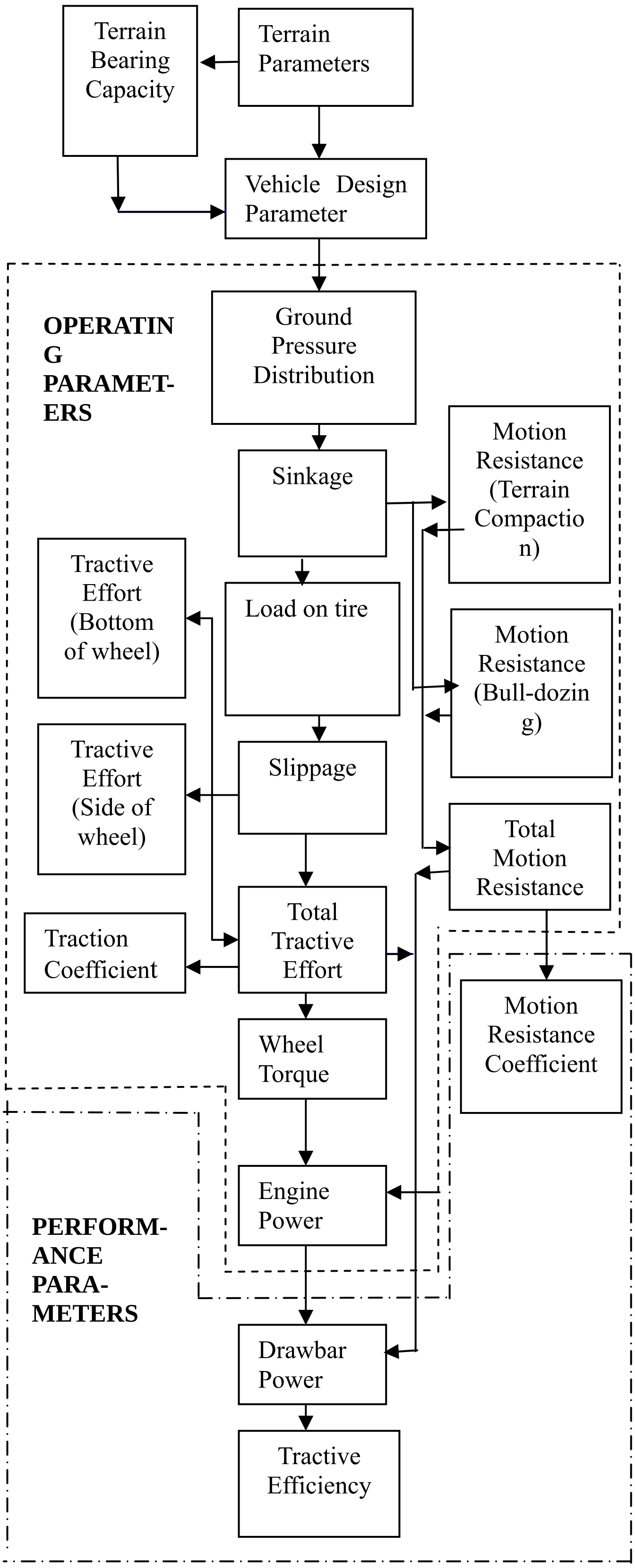


Fig 1: Flow chart of vehicle kinematics model

Assume P is initially at the origin and let C and T is as indicated in the Figure 2, with ϕ denoting the radian measure of angle TCP. Then the arc PT and the segment OT have the same length and so the center C of the rolling circle is at ($R_0\phi$, R_0). Using a trigonometry, we conclude that

$$x = R_0\phi - R_0 \sin\phi = R_0(\phi - \sin\phi) \tag{3}$$

$$y = R_0 - R_0 \cos\phi = R_0(1 - \cos\phi) \tag{4}$$

The equations of the curve of cycloid can be used for the description of the paths of the wheel as shown in Figure 3.

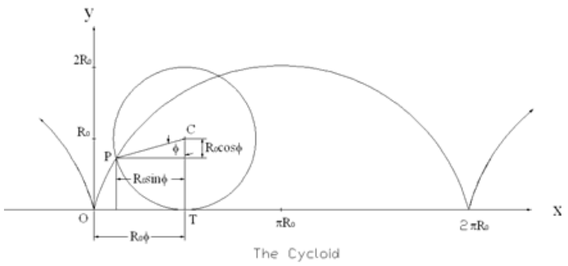


Fig 2: Typical cycloid for the wheeled vehicle

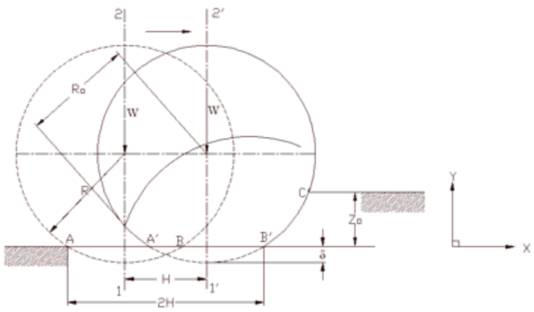


Fig 3: Mathematical analysis of rolling

The cycloid through the origin created by a circle of radius R, consists of the point (x,y) with,

$$x = R(\phi) - R_0 \sin(\phi) \tag{5}$$

$$y = R - R_0 \cos\phi \tag{6}$$

where, R_0 is the distance of the point being examined from the centre of the wheel in m, R is the rolling radius, ϕ is the central angle in degree.

Figure 4 shows a wheel of the vehicle rolls on peat terrain with load and the displacement of the contact point of the wheel relative to the peat terrain. Individual points of the wheel perimeter move along looped cycloid.

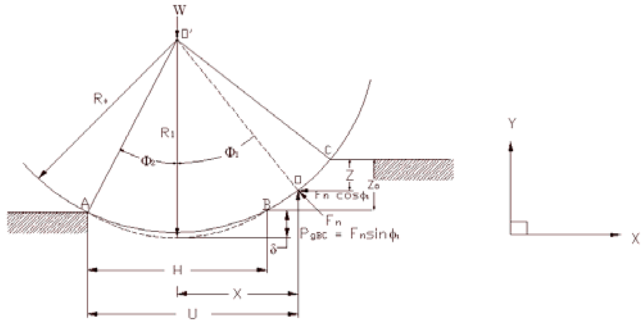


Fig 4: Points on the perimeter of a driving wheel describe a looped cycloid

$$x = U - H/2 \tag{7}$$

with $U = R_1 \sin\phi_1 + R_1 \sin\phi_2$ and $\phi = \phi_1 + \phi_2$

where, R_1 is radius of the point on the perimeter which is being examined, U is the length of the contact surface which is the horizontal projection of the contact arc, and ϕ is the central angle which belongs to U.

It is assumed that the driving wheels operate without exerting any peripheral force, its vertical deformation and the accompanying soil deformations which create a condition as if were rolling without slip with radius R_z. The radius R_z can be measured by using the following equation:

$$R_z = R_0.(1 - i_{rd}) \tag{8}$$

where, i_{rd} is the vehicle wheel tire relative to the terrain in percentage. The slip i_{rd} *will* be induced due to the friction of the driving wheel on the terrain. The resulting slip radius can be calculated by using the following equation:

$$R_i = R_z.(1 - i) = R_0.(1 - i_{rd})(1 - i) = R_0(1 - i_r) \tag{9}$$

where, i_r is the resultant slippage and $i_r = i + i_{rd}$ and $i.i_{rd} \cong 0$. When the driving wheel is under slippage, the slip displacement of the wheel can be represented by rewriting the Eq.(7) :

$$x = U - H/2 = R_i\phi - H/2 = R_0\phi(1 - i_r) - H/2 \tag{10}$$

The resultant slippage i_r is caused by the peripheral force and by tire deformation. The Eq. (10) is represented when considered the kinematics interaction taking place between a deforming tire's contact surface and the peat terrain. When the deformable tire rolls with radius R_0, i.e., $i_r = 0$. The slip displacement of the vehicle tire can be represented as follows:

$$x_0 = R_0\phi - H/2 \tag{11}$$

where, x_0 is the displacement of the tire due to $i_r = o$

When the $i = 0$, the displacement of the tire will be zero i.e $x = 0$. The slip displacement Eq.(10) can be rewritten as follows:

$$x = R_0\phi(1 - i_{rd}) - H/2 = 0 \tag{12}$$

$$i_{rd} = 1 - \frac{H}{(2R_0\phi)}$$

While the inflation pressure of the tire is reduced it is assumed that the deflection of the tire will be δ. The length of the contact surface can be represented as follows by considering the tire deflection δ,

$$H = 2\sqrt{[(\delta)(D - \delta)]} \tag{13}$$

where, H is the contact length of the wheel in m, z_0 is the slip sinkage of the vehicle, and D is the diameter of the tire in m.

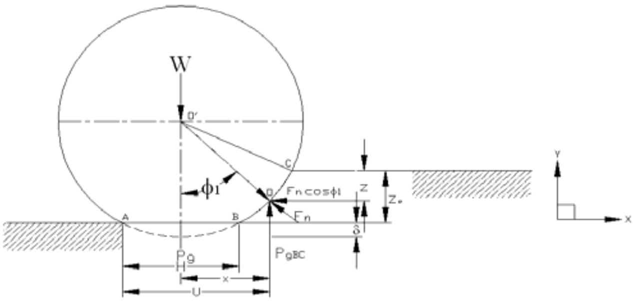


Fig 5: Wheel-terrain interaction model

If we mention that the entry angle of the tire ϕ_1 and exit angle ϕ_2 are made during forward traveling on the terrain, we can compute the entry and exit angle of the tire on the terrain surface based on the Figure 5.

$$\phi_1 = \sin^{-1}\left(\frac{2x}{D}\right)$$

$$\phi = \sin^{-1}\left(\frac{2x}{D}\right) + \sin^{-1}\left(\frac{H/2}{D}\right)$$

(14)

Figure 5 shows the wheel-terrain interaction model. From the Figure 4, we have by applying the geometry:

$$x = \sqrt{\{D(z_0 + \delta - z)\} - (z_0 + \delta - z)^2} \tag{15}$$

$$x = \sqrt{\{(z_0 + \delta - z)(D - (z_0 + \delta - z))\}}$$

For small sinkage,

$$x^2 = D(z_0 + \delta - z) \tag{16}$$

It could be assumed that the $z \cong z_0/2$; since the force F_n acts as the normal force on the curve BC. So that we can get from the Eq. (15) and (16) as follows:

For the excessive sinkage,

$$x = \sqrt{\left\{\left(z_0/2 + \delta\right)\left(D - z_0/2 - \delta\right)\right\}} \tag{17}$$

For small sinkage,

$$x = \sqrt{\left\{D\left|z_0/2 + \delta\right|\right\}} \tag{18}$$

The total entry and ext angle ϕ of the wheel tire can be computed by simplifying the Eqs.(14), (17) and (18) we have:

For excessive sinkage,

$$\phi = \sin^{-1}\left(\frac{2\sqrt{\left\{\left(z_0/2 + \delta\right)\left(D - z_0/2 - \delta\right)\right\}}}{D}\right) + \sin^{-1}\left(\frac{H}{D}\right)$$

(19)

The slippage of the vehicle can be computed by simplifying the Eqs. (12), (18) and (19) we have:

For excessive sinkage,

$$i_{rd} = 1 - \frac{H}{(2R_0\phi)}$$

$$= 1 - \frac{2\sqrt{\{\delta(D - \delta)\}}}{2R_0\left[\sin^{-1}\left\{\frac{2\sqrt{\left\{\left(z_0/2 + \delta\right)\left(D - z_0/2 - \delta\right)\right\}}}{D}\right\} + \sin^{-1}\left(\frac{2\sqrt{[\delta(D - \delta)]}}{D}\right)\right]}$$

(20)

For small sinkage,

$$i_{rd} = 1 - \frac{H}{(2R_0\phi)}$$

$$= 1 - \frac{2\sqrt{\{\delta(D - \delta)\}}}{2R_0\left[\sin^{-1}\left\{\frac{2\sqrt{\left\{D\left(z_0/2 + \delta\right)\right\}}}{D}\right\} + \sin^{-1}\left(\frac{2\sqrt{\{\delta(D - \delta)\}}}{D}\right)\right]}$$

(21)

3. TORQUE TRANSDUCER

The mobility of the vehicle was investigated by measuring the tractive effort of the vehicle. The tractive effort of the vehicle measurement was conducted by developing a torque transducer as shown in Figure 6.

Fig 6: Develop torque transducer

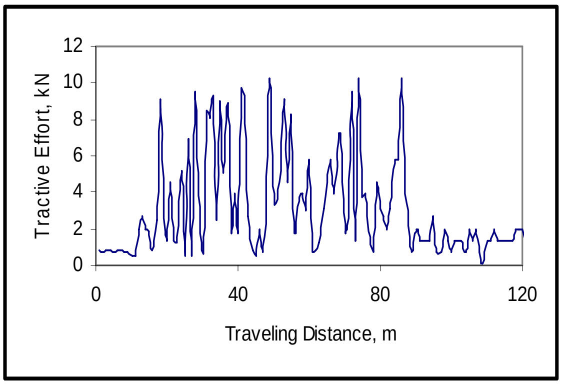
The torque transducer was developed by bonding two set-rossette strain gauge on the hub in between two flanges in order to make a wheatstone bridge and fixing a Michigan S4 slip ring at the end of the extended shaft in order to keep continuation the circuit with the data logger.

4. FIELD TEST RESULTS OF THE LGP-30 WHEEL VEHICLE

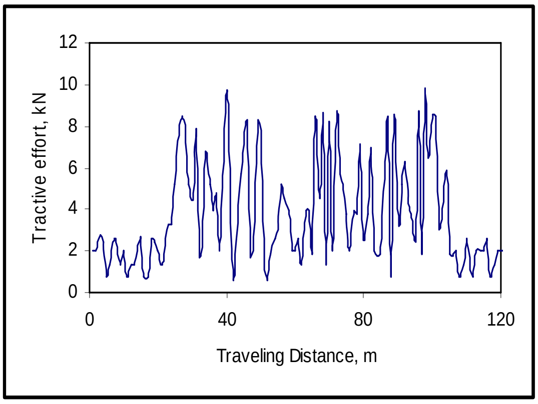
The vehicle testing site was the unprepared moderate peat terrain at the Sepang, DC24 digital panel meter for getting the expected traveling velocity of the vehicle. The straight motion tests of the vehicle were performed at three different traveling speeds of 6 km/h, 10 km/h and 12 km/h. Before each of the test, the vehicle was made ready by installing the portable generator set and the DEWE 2010 on the vehicle.

Fig 7: Instrumentation on LGP-30 wheeled vehicle.

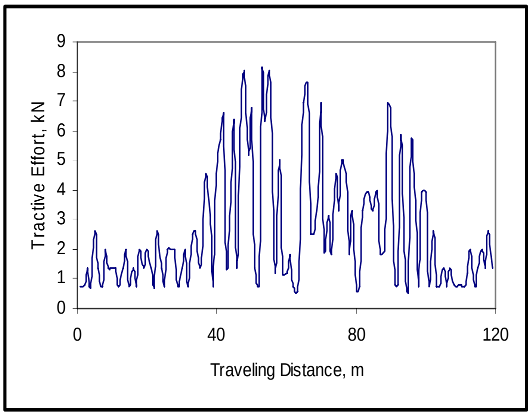
The instrumentation system was tested by executing the developed opposite of the Kuala Lumpur International Airport (KLIA) Malaysia. The vehicle was tested with installing the develop torque transducer as shown in Figure 7 and the speed was set to the K3GN-NDC-FLK programme with DASY Lab 5.6 into the DEWE-2010 on the field. Then, a preliminary run on the terrains was performed for ensuring the expected function of the instrumentation system of the vehicle. The number of tests was conducted on the different vehicle traveling path in the field with the same loading conditions and traveling speeds. Figure 8 shows the results of the *typical* tests. From the tests result statistical analysis it was found that the mean value of the vehicle's tractive effort for Test I was found 11.05 kN, 10.15 kN and 9.5 kN for the travelling speed of 6 km/h, 10 km/h, and 12 km/h excluding starting time and stopping time. While the mean value of the vehicle's tractive effort for Test II was found 11.76 kN, 10.54 kN and 9.46 kN for the travelling speed of 6 km/h, 10 km/h, and 12 km/h. It could be concluded that the different tractive for the vehicle was found for different speed is due to the difference of torque. It is noted that the lower travelling speed of the vehicle is developed higher tractive effort which is due to the higher torque developed in the wheel as the vehicle is equipped with manual transmission system.



(a)



(b)



(c)

Fig 8: Vehicle tractive effort: (a) Traveling speed 6 km/hr, (b) Traveling speed 10 km/h, (c) Traveling speed 15km/h

5. CONCLUSIONS

The Low Ground Pressure (LGP-30) wheel vehicle was tested on the 17 kN/m^2 low bearing capacity peat terrain in Malaysia at three different traveling speeds. Based on the simulation results, the following conclusions could be made:

(1) The sensitivity of the designed and developed torque transducer was found 20.04 mV/V per Nm.

(2) The mean value of the test results on tractive effort was found 11.66 kN, 9.89 kN and 9.06 kN for the travelling speed of 6 km/h, 10 km/h, and 15 km/h, respectively.

6. ACKNOWLEDGEMENTS

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8. NOMENCLATURE

Symbol	Meaning	Unit
ω	Moisture content	(%)
γ	Bulk density	(gm/cm^3)
c	Cohesiveness	(kN/m^2)
φ	Terrain internal friction angle	(degree)
K_w	Shear deformation modulus	(m)
m_m	Surface mat stiffness	(kN/m^3)
k_m	Underlying peat stiffness	(kN/m^3)
P_g	Ground contact pressure	(kN/m^2)
z_0	Slip sinkage of the vehicle	(m)
D_h	Hydraulic diameter	(m)
R_0	Radius of wheel	(m)
ϕ	Central angle	(degree)
U	Length of the contact surface	(m)
i_{rd}	Slippage of the vehicle	(%)
i_r	Resultant slippage	(%)
τ	Shear stress	(kN/m^2)
H	Contact length of the wheel	(m)
D	Diameter of the tire	(m)
Φ_1	Entry angle of the tire	(degree)
Φ_2	Exit angle of the tire	(degree)
δ	Tire deflection	(m)