Dynamic Modeling and Simulation on Handling Characteristics of Tracked Vehicle

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***Abstract*—Soil Traction is the emerging research area in the field of terramechanics. Over the last decades many terramechanics engineers investigated the problem related to terrain interaction from empirical model to theoretical model. This is mainly because of nonlinear behavior of traction between ground and track during higher endurance. One of the major problems in off road vehicle is pretending to achieve the better handling characteristics for different terrain conditions. The work carried out in this paper is for ATV scale down model, involving skid steer system. The proposed work aims at estimating the traversibility for different soil conditions and obtaining performance analysis in scale down model. In order to calculate required angular velocity of different motor. General dynamic equations by bekkers model are used for computations. The future work will be implementing the developed system in UGV Tracked vehicle.**

Keywords—Terramechanics; endurance; ATV; shear analogy; bekkers model

I. INTRODUCTION

For the past years, there has been many dynamic force analysis & vehicle designing methodologies are adopted in designing the real military tracked vehicle, even though the research at higher flow still it occurs with very less flaws. This paper fully comprises of dynamics related to skid steering. In skid steer considering problem is large frictional moments opposing the turn. If in the case having larger track it has to slew on the ground during steer[1]. The traversibility and tractive effort of the designed vehicles is completely deviated on the real off-road drive conditions due to different terrain conditions and their cohesive nature. The properties of terrain are divided into bearing and shearing, whereas bearing takes place in normal direction and shearing in tangential direction[2].

Scenario's and trends in solving problem related to terrain properties is solved by Hock and kitano, which depicts actual data very similar to theoretical data. Kitano

The control methodology for slip is either by longitudinal control or lateral control, this paper mainly focus on lateral control. Slip control is achieved quicker and more precisely without need of mathematical computation by controlling the torque in the individual drive motor, this will be even

implemented for high speed cornering without loose of traction.

model is very unique in calculating actual turn radii[3]. The main correlation between terrain and track mainly depends upon the pressure sinkage relationship with respect to penetration velocity[4]. Analysis for the real ATV is checked with drawbar pull experiments, so that the effective tractive force and coefficient of friction between the soil and track will be obtained. It is clear that increase in vehicle weight increases maximum mobility pull[5].

Different terrain has different properties like cohesiveness, repose angle, pressure sinkage characteristics to specific load. In order to estimate interaction between grounds and track the appropriate key role is maintained that is prediction methods and assessment on terrain using soil mechanics. To make the model more appropriate and accurate plate-sinkage tests have been applied to acquire data for studying off road conditions. It is clear that pressure sinkage tests will provide correct young's modulus of soil, which is usual for further analysis. But the drawback is exploring the different terrains in battlefield is so tough[6].

It is very hard to determine trajectory for vehicle if it have higher slip phenomenon and unknown terrain conditions. So the nature for soil must be identified in real time for developing actual model. Use of basic reference control algorithms is suitable for known terrains whereas in unknown area it is done by some filtering algorithm and sensors[7].

Many researchers are still evaluating bekkers model and checking the soil compaction. General pressure sinkage relationship developed by Bernstein and Goriatchkin is adopted overall. i.e.

$$P = kz^n \tag{1}$$

The above equation is completely dependent in soil stiffness constant. Bekker combined two basic concepts sinkage relations and soil stiffness constant[8].

$$P = (k_c/b + K_\phi)\, z^n \tag{2}$$

II. TRACKED VEHICLE MODEL

Off road vehicles are classified either by wheeled model or tracked model, which describes the type of contact between terrain and vehicle. Vehicle with tracks provide greater traction and equal pressure distribution on the ground. In soil mechanics, the tracked vehicle generates tractive effort by deforming soil in longitudinal shear.

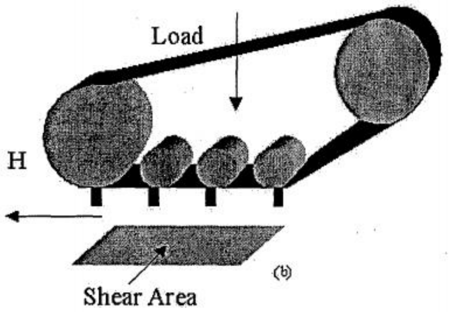


Fig. 1. Track shear occurrence.

A. Conceptual Design

Initially the track model has to be designed and considering some assumptions. Fig 2 shows the slip occurrence in track during constant turn radius R which is sliding longitudinally and laterally on the ground.

- Assume ground is flat and rigid, No slip occurrence.
- Uniform track loading.
- Isotropic columb friction occurring between track and ground.
- Lateral acceleration is negligible since there is no weight transfer.

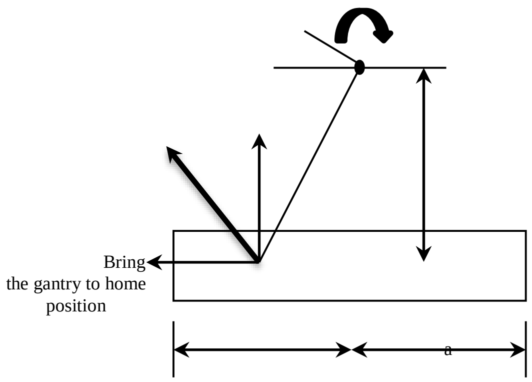


Fig. 2.Slip occurrences in track.

In skid steer outer track skids backwards on the ground, since it exerts a forward thrust. The inner track skids forwards to resist the motion of vehicle. This eventually generates slewing couple.

B. CAD model for proposed ATV prototype

To analyze the slip occurrence and mobility of the vehicle it is must to check in real time scenario. This paper comprises of fabricating the model and estimating the required speed ratio to perform constant radius test. Fig 3 shows the CAD model, which is designed as tracked vehicle model involving basic dynamics of skid steer and armed military vehicles.

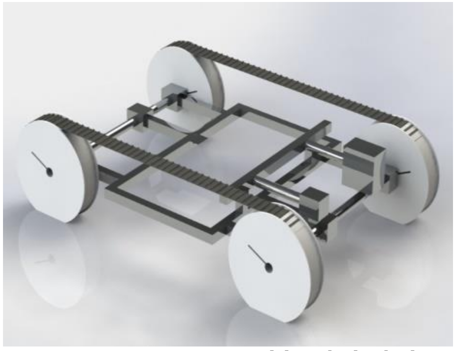


Fig. 3. CAD model tracked vehicle

Tracked vehicle is designed in such a way that it should have a better traction, so geometric ratio of the model is set too high (C/L ratio). Initial belt tension is nominal so that there is no slip between the wheel surface and belt. A very small consideration, air drag with reference to front area of the vehicle is considered while designing, which makes a profound idea to make rigid bars with less frontal area, the drag is minimized.

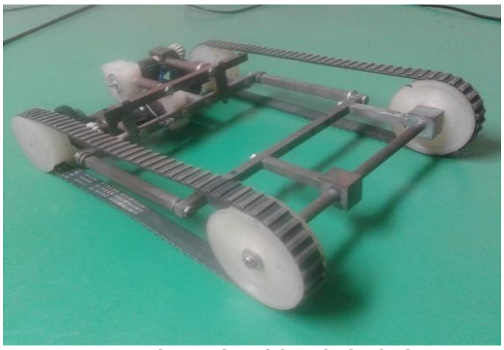


Fig. 4. Fabricated model tracked vehicle.

TABLE 1 DIMENSIONS FOR ATV SCALE DOWN MODEL

Specifications	Dimensions
2L- Length	0.3 m
2C- Center distance	0.27 m
r – Sprocket radius	0.035 m
l - Track length	0.3 m
w - Track width	0.015 m
Total Load	39.24

III. DESIGN CALCULATION

In this section design calculations is done, which provides the information for selecting a required specifications for scale down model.

A. Estimating traction & coefficient of friction

In order to estimate the traction and coefficient of friction, the model uses columb equation at idealized elasto-plastic soil conditions considering theory of plastic of equilibrium. This idealized consideration is not valid for all terrain, it is suitable for modeling clay and dry sand[9]. It is necessary to determine the maximum tractive force that the scale down model can produce with respect to load.

To validate the estimated friction value Draw bar pull test must be conducted to check the theoretical value in comparison with experimental value. This theoretical calculation is done considering dry sand.

It is necessary to determine the total tractive effort requirement of the vehicle. Total tractive effort determines the total effort that must be used to propel the vehicle to desired velocity; however the slip occurrences will occur.

RR = Force required to overcome rolling resistance.

GR = Force required to climb an inclination.

FA = Force required to accelerate to final velocity.

Total tractive force = RR (kg) + GR (kg) + FA (kg)

Estimating top speed of the vehicle,
Assume total distance to be covered = 100 m
Maximum speed of motor = 75 rpm, considering 50 rpm
Speed =Circumference * RPM
= 2*3.14*0.035*50 = 10.99 m/min
= 10.99*(1/60) = 0.183166 m/s
=~0.2 m/s

Gross vehicle weight = 4 kg
Radius of wheel = 0.035 m
Top speed (V_{max}) = 0.2 m/s
Desired acceleration time (Ta) = 1s
Maximum incline angle (θ) = 10^0
C_{rr} = 0.04 (wet soil) 0.2 (sand)
Resistance factor = 1.12 between (1.1 -1.15)

Rolling Resistance (RR) is necessary to propel a vehicle over a particular terrain. The worst possible surface type to be encountered by the vehicle should be factored into the equation.

$$RR=GVW\ (kg) * C_{rr} \tag{3}$$

= 4*0.2 = 0.8 kg

GR is the amount of force necessary to move a vehicle up a slope or “grade”. This calculation is made using the maximum inclination angle that will be expected to climb. In our case the terrain is straight so the maximum inclination angle is limited to 10^0 at the worst case.

$$GR= GVW\ (Kg) * \sin(\theta) \tag{4}$$

= 4* 0.1736 = 0.6944 kg

$$FA,\ \text{Acceleration force} = [GVW\ (Kg) * V_{max}\ (m/s)) / (9.81\ (m/s^2) * Ta\ (s)] \tag{5}$$

= (4*0.2)/ (9.81*1) = 0.0815 kg

Total tractive force = 1.5759 kg = 15.4595 N
Coefficient of friction,

$$\mu = \frac{\text{Tractive force}}{\text{Normal force}} = \frac{15.4595}{39.24} = 0.39 \sim 0.4$$

B. Grouser selection

Grousers in vehicle play significant role in providing traction on continuous tracks, especially in loam and dry sand. It is so clear that to get a better traction the grouser with higher shear force mobility must be considered. Rubber pad grouser is selected for scale down model since it is having better traction when compared to metal grouser.

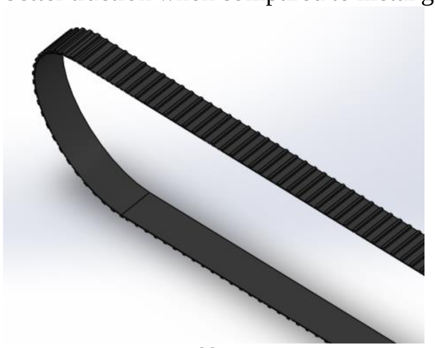


Fig. 5. Rubber grouser.

IV. KINEMATICS OF SKID STEERING

This section describes radius of turn as function of speed ratios that includes the hard level going with accounting Non dimensional slip factors into it. This is extended to provide the required torque and speed for the sprockets. Fig 6 shows tracked model involving in constant radius test, where the turn radius R is set constant. It is tough to consider whole model to solve kinematic ally, Semi track model is considered in this paper. It is necessary to take some assumptions while estimating the track model, considering ground is frictional and track link is non-directional.

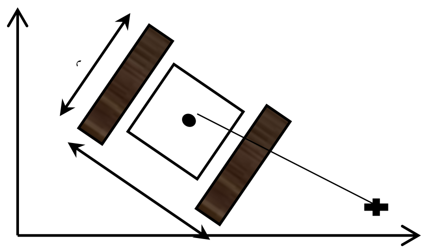


Fig. 6. Constant radius turn.

Under these circumstances the laws for dry friction is assumed and frictional forces are equal to the coefficient of friction multiplied by normal reaction will act in a direction such as to oppose sliding.

For the constant velocity these forces and moments must balance,

$$Z_o + Z_i = Z_e \tag{6}$$
$$X_o + X_i = X_e \tag{7}$$
$$(X_o - X_i)\, C = M_o + M_i \tag{8}$$

equ(6) Consider that the weight of vehicle is shared equally, ignoring any lateral weight distribution due to cornering at high speed or side slope.

equ(7) says that sum of propulsive forces on tracks is used to overcome drag. If there is no drag in ATV i.e, in rare case occurrence, X_o is equal and opposite to X_i.

equ(8) shows that it is the difference of the propulsive forces that needs to be high, in order to overcome the resistance to yawing.

$$\frac{M'_o + M'_i}{X'_o - X'_i} = \frac{C}{L} \quad (9)$$

Estimate the longitudinal forces for inner and outer track using the initial drag coefficient, which is decided by frontal area of the vehicle and the velocity.

$$X_o = \mu\, Z_o X_o' \quad (10)$$

$$X_i = \mu\, Z_i X_i' \quad (11)$$

The torques at the sprockets required for the vehicle during a turn, considering the SRR at no slip condition will be,

$$T_O = X_o * r \quad (12)$$

$$T_i = X_i * r \quad (13)$$

Yaw rate Ω is calculated by velocity of the moving body divided by turn radius R.

For the inner and outer wheels the velocity will be,

$$V_o = \Omega(R + C) \quad (14)$$

$$V_i = \Omega(R + C) \quad (15)$$

Considering wheel slip the relations be,

$$V_s = \beta L \Omega \quad (16)$$

Here, β is the non-dimensional slip radii that occurs during a turn. The data set for slip radii is taken from [10].

The corresponding wheel angular velocities will be,

$$\omega_{nso} = \frac{V_O}{r} \quad (17)$$

$$\omega_{nsi} = \frac{V_i}{r} \quad (18)$$

Outer angular velocity

$$\omega_o = (R/L + C/L + \beta_O) L\Omega / r \quad (19)$$

Inner angular velocity

$$\omega_i = (R/L - C/L + \beta_i) L\Omega / r \quad (20)$$

The mean sprocket speed maintains the same proportion of engine speed as in the straight condition.

$$\text{Slip factor} = \left(\frac{\text{Mean sprocket speed with slip}}{\text{Mean sprocket speed without slip}}\right)$$

$$\text{Slip Factor} = 1 + \frac{(\beta_o + \beta_i)}{2R/L} \quad (21)$$

i. Engine speed remains same if $\beta_o = -\beta_i$
ii. Slip factor exceed unity, engine speed must rise $\beta_o > -\beta_i$
iii. Engine speed reduction, $\beta_o < -\beta_i$

Sprocket speeds and sprocket ratio

$$n = \frac{\omega_O}{\omega_i} = (R/L + C/L + \beta_O) / (R/L - C/L + \beta_I) \quad (22)$$

Normalized turn radius (For no slip assume $\beta_o = \beta_i = 0$)

$$R/L = \{(n+1)C/L + \beta_o - n\beta_i\} / (n-1) \quad (23)$$

Rolling resistance Xe = SRR x W (24)

Sprocket powers

$$P_o = T_o \omega_o \quad (25)$$

$$Pi = T_i \omega_i \quad (26)$$

V. RESULTS AND DISCUSSIONS

Simulink model is developed with kinematic equation (10-26) including the slip factor non dimensionally. The model is having various subsystems to make the model simple, each and every equation in the subsystem is stated in above equations.

TABLE 2 SIMULATION INPUT VALUES

Simulation parameter name	Value (Unit)
Vehicle velocity	0.3 (m/s)
Turn Radius	10 (m)
Friction	0.3
Wheel Radius	0.035 (m)
Half load	0.01962(KN)

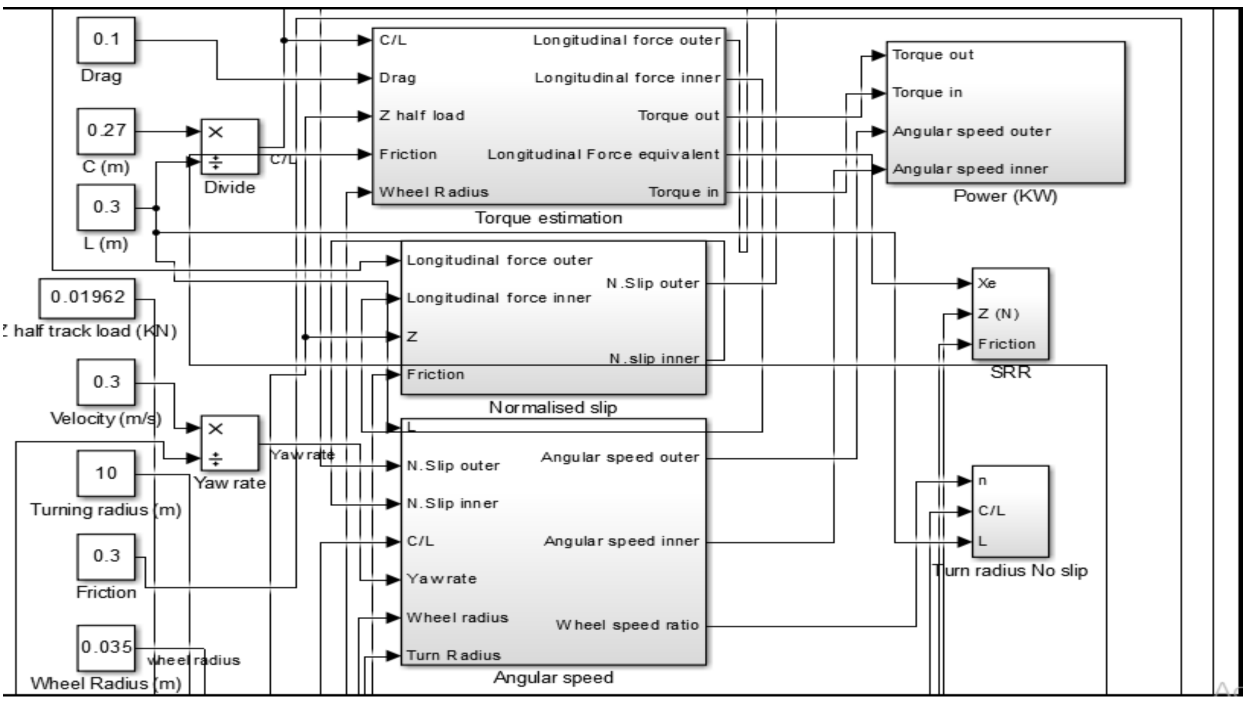


Fig.7 .Simulink model.

Simulation input parameters are stated in table 2. In calculated value V_{max} is 0.2 m/s. In simulation it is 0.3 m/s to check condition at higher velocity. Fig 7 shows the complete Simulink model to estimate the speed difference in inner and outer sprockets. The fabricated model dimensions are incorporated into the model to perform actual simulationwith prior to real data. However friction coefficient is mathematically calculated.In order to test model using HIL simulation the friction coefficient cannot be found during simulation. Separate terrain test has to be conducted i.e. draw bar pull test using load cell which estimates the maximum tractive pull. Fig 8 shows the given turn radius for simulation, which respect to constant turn radius 10m actual turn radius is found and results are shown in Fig 9.

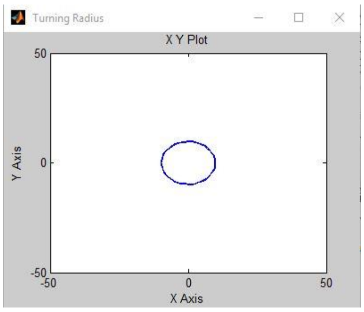


Fig. 8. Given turn radius.

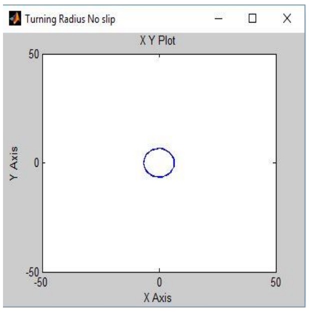
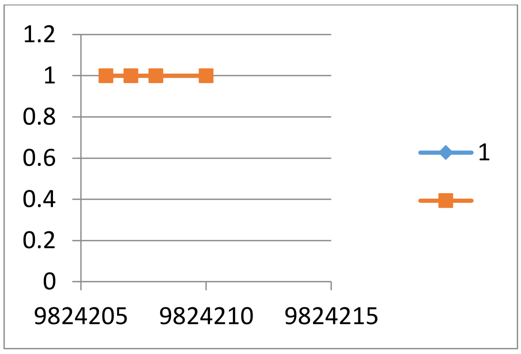


Fig. 9. Actual turn radius.

Simulation is started with given turn radius to be achieved, but due to lateral slip the turn radius may either increased or decreased or wavier. Considering the slip occurrence into the model and checking for results. Look up table is made for already experimented data with non-dimensional slip parameters and model is analyzed. Graph 1 shows that the turn radius gets decreased for the increase in drag, so correspondingly for different turn radius sure lateral slip occurs which decreases the turn radius.



Graph. 1. Turn radius for different drag.

TABLE 3. SPEED RATIO FOR DIFFERENT DRAG

Drag	Angular speed outer (rad/s)	Angular speed inner (rad/s)	Outer sprocket speed (RPM)	Inner sprocket speed (RPM)	Speed ratio
0.1	8.728	8.435	83.35	80.55	1.035
0.2	8.746	8.443	83.52	80.62	1.036
0.3	8.766	8.450	83.71	80.69	1.037
0.4	8.805	8.454	84.08	80.73	1.041

VI. CONCLUSION

Based on the previous work and methodologies carried out in terramechanics, a Simulink model is developed with dynamic equation to estimate the required power, torque and actual turn radius of the robot. Graph 1 depicts the fact that due to slip occurrence towards lateral side, the turn radius in actual is minimized for increase in drag. The turn radius varies in accordance with drag. In actual it will be 1.6-1.7 times larger turn radius achieved in real time when compared with calculated[11]. This proposed model will be used for further odometry and to map desired motion with respect to angular rate and speed required for the sprockets. Future work will be continued with actual set point RPM obtained from model and testing in different terrain conditions with IMU sensor installed in it.

ACKNOWLEDGMENT

The authors would likely thank Mr. G.Venkatesan Sct ‘E’ DRDO CVRDE Chennai & Mr. A. Jothilingam, Head of Department, Production Technology, MIT for their valuable support and guidance in making the project, a successful one.

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