APPLICATION NOTE ON REGENERATIVE BRAKING OF ELECTRIC VEHICLES AS ANTI-LOCK BRAKING SYSTEM

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Abstract

Anti-locking brake systems (ABS) are well known in the automotive industry and studied under safety heading. ABS improves vehicle safety by reducing longitudinal breaking distance. This occurs by on-off control of the wheel slip. In this study, a basic modeling approach has been introduced on a quarter car model by using ANSOFT Simplorer for the following braking modes; hydraulic braking and all electric vehicle regenerative braking concept.

Keywords:

ABS, Electric vehicle, Quarter Car Model

1-Introduction

Electric or hybrid electric vehicles propose not only better fuel economy and less environmental pollution but also superior performance of braking, traction control and stability control systems employing motoring and regenerative braking capability of electric machines.

A car braking system is one of the major factor for the driving safety. The introduction of the Anti-Lock Braking Systems has contributed to improve the security of modem cars decisively by automatically controlling the brake force during braking in potentially dangerous conditions such as braking on iced or wet asphalt, panic braking, etc. . . [1]

This paper starts with development of quarter car model (QCM). First a hydraulic ABS model is applied to the QCM. Later modification of permanent magnet (pm) brushed dc machine model for field weakening region is introduced. Finally this model is applied to QCM to investigate regenerative braking performance of electric vehicles by means of ABS.

2-Quarter Car Model

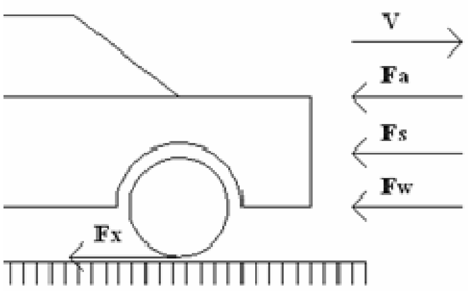


Figure 1 Forces acting on the vehicle

Forces acting on a vehicle is shown in Figure 1, which are wheel friction force (Fw), aerodynamic drag force (Fa), slope friction force (Fs) and force due to vehicle inertia (Facc). Fx denotes the tire braking force.

Forces acting on one wheel of a vehicle;

$$Fw = c_t.m.g.\cos\alpha/4 \qquad (1)$$

$$Fs = m.g.\sin\alpha/4 \qquad (2)$$

$$Fa = 0.5.c_r.\delta.A_f.V^2/4 \qquad (3)$$

$$Facc = (m/4).dV/dt \qquad (4)$$

where c_t, m, α, c_r, δ, A_f and V are wheel rolling resistance coefficient, total vehicle mass (kg), slope angle (rad), aerodynamic coefficient, air density (kg/m^3), vehicle frontal area and vehicle speed (m/s) respectively.

Longitudinal vehicle dynamics of quarter car during braking can be given as;

$$-Fx-Fw-Fs-Fa = \frac{m}{4}.\frac{dV}{dt} \qquad (5)$$

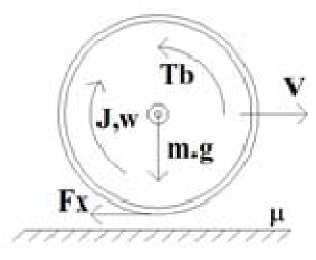


Figure 2 Wheel longitudinal dynamics

$$Fx = \mu.m.g/4 \qquad (6)$$

Fx is tire braking force and μ can be calculated based on a Pacejka magic tire formula [2] or taken from a table of μ vs. slip ratio (s). Slip ratio is defined as;

$$s = \frac{wv - ww}{max(wv,ww)} \qquad (7)$$

where wv and ww represents vehicle and wheel angular speeds respectively.

For this study μ is calculated based upon the following graph in Figure 3, which represents a dry road condition.

Tire model can be given as;

$$Fx.r-Tb = J.\frac{d\omega}{dt} \qquad (8)$$

where r, Tb, J and w are wheel radius, braking torque, wheel inertia and wheel angular velocity respectively.

Tabel 1 Vehicle Parameters used in model

Vehicle Weight (m)	1700 kg
Wheel radius (r)	0.325
Vehicle Frontal Area (m^2)	3.1
Tire rolling resistance Coef. (cr)	0.3
Aerodynamic resistance Coef. (ct)	0.01
Wheel inertia (kg.m^2)	0.5

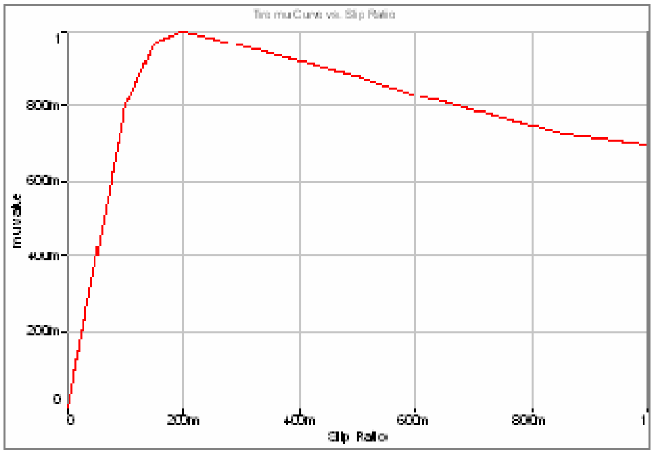


Figure 3 μ vs. s graph

Finally, quarter car model during braking is represented in the following figure. For this model, definition of initial speeds is crucial. Initial speed is selected as 100 km/h for both vehicle and wheel for all of the simulations.

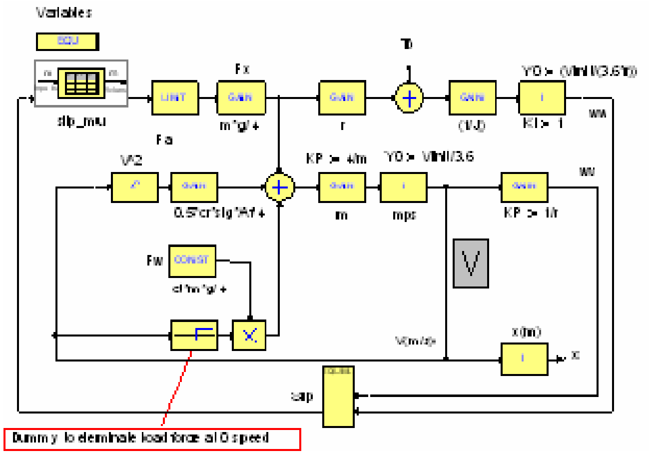


Figure 4 Quarter car model during braking

3-Hydraulic ABS Braking

The purpose of ABS is to optimize the braking effectiveness and maintain vehicle stability under various road conditions. It is achieved by controlling the slip ratio at the point where maximum braking force can be applied to the wheels.

For the control of the ABS, optimum slip ratio is entered to the controller as reference value. Slip error then is feed to hydraulic actuator.

The dynamic model of hydraulic fluid lag of braking system is used as the following first order transfer function:

$$G(s) = \frac{k}{\tau.s + 1}$$

where for this study k and τ are selected as 100 and 0.01 respectively.

Then braking torque is simply achieved by integrating the hydraulic fluid and multiplying by a constant as show in figure 5.

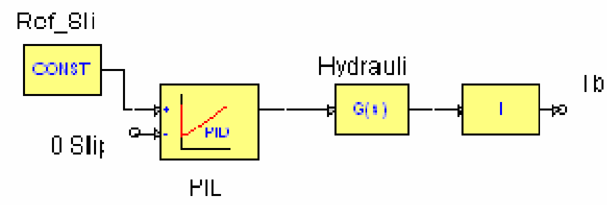


Figure 5 Hydraulic ABS model

Integration of the hydraulic ABS model and QCM is given in figure 6.

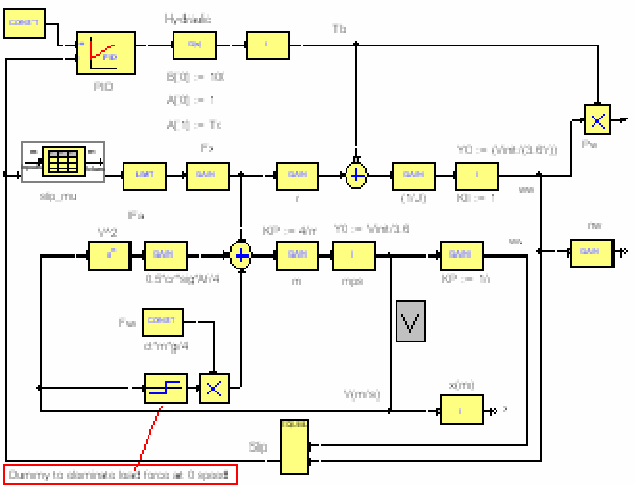


Figure 6 Integration of hydraulic ABS to QCM

During braking simulation, maximum braking torque of the system is limited to 1500 Nm. Reference slip value is entered as 0.2 seeing that maximum braking force occurs at this point as seen in figure 3.

In figure 7, vehicle and wheel speeds (km/h) are plotted during simulation. Total braking time is 3.5 s. Around 1.5 s, wheel slip ratio reaches 0.2 where maximum braking force is achieved as indicated in figure 8 (m on the wheel slip axis indicates 10^{-3}). Total distance traveled during braking is 57.27 m shown in figure 9.

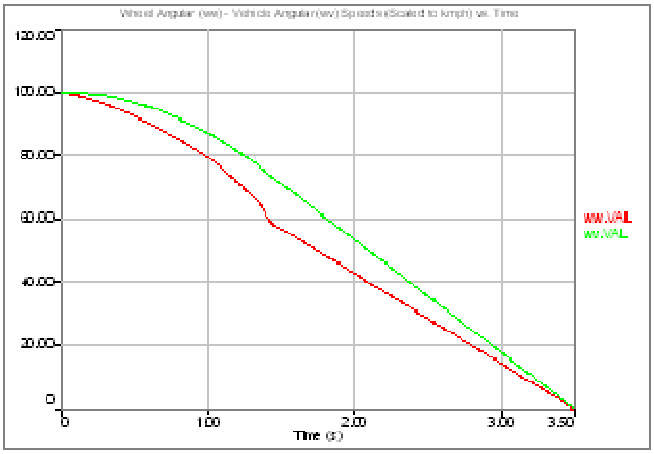


Figure 7. Vehicle and wheel speeds vs. time

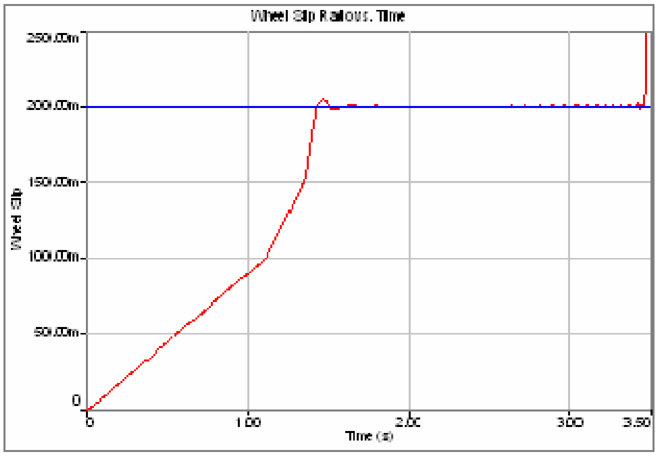


Figure 8. Wheel slip vs. time

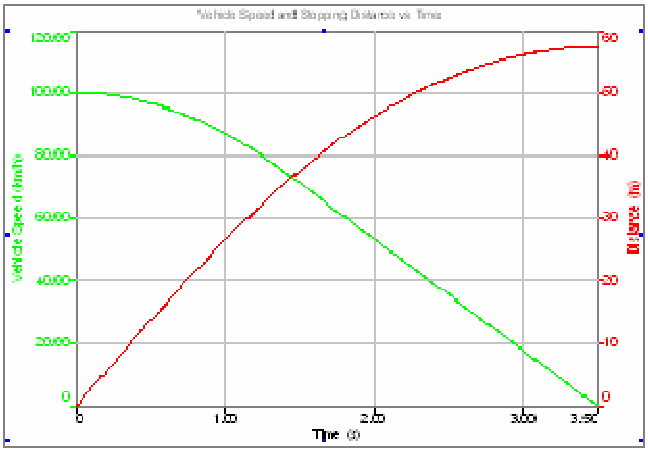


Figure 9 Vehicle speed and traveled distance

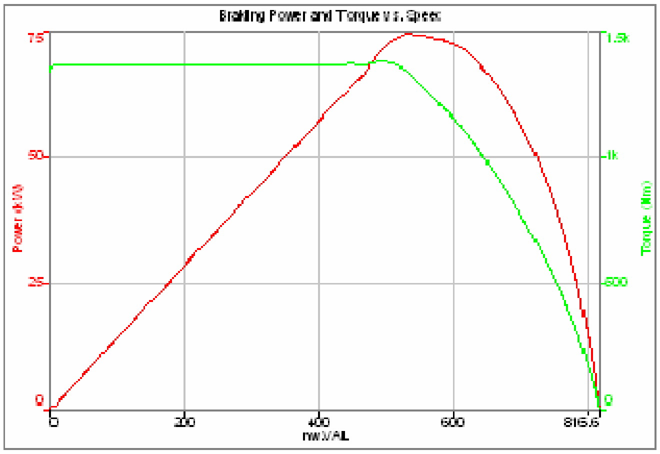


Figure 10. Braking power and torque vs. speed

Braking power and torque vs. wheel rotational speed is plotted in figure 10. As can be seen from the figure nominal power is 75 kW at 530 rpm and nominal applied braking torque is around 1400 Nm.

4-Modification of PM Brushed DC Machine Model for Field Weakening Region

Electric vehicle applications require high constant power to constant torque ratio, typically in the range of 3 to 5 for better performance at lower power consumption [3].

For simplification of the overall electric traction system modeling, a dc motor model will be used looking from system engineering point of view.

Conventional permanent magnet stator dc machine model simplified equations can be modified as below to simulate the constant power region of field oriented controlled ac machines;

$$Va = Ea + Ri.Ia + Li.dia/dt \qquad (9)$$

$$Ea = k_e(\omega_r).\omega_r \qquad (10)$$

$$Te = k_t(\omega_r).Ia \qquad (11)$$

where Va, Ea, Ri, Li, k_e, k_t and ω_r represent supply voltage, back EMF voltage, winding resistance, winding inductance, back EMF constant (rotor flux), torque constant and rotor speed respectively.

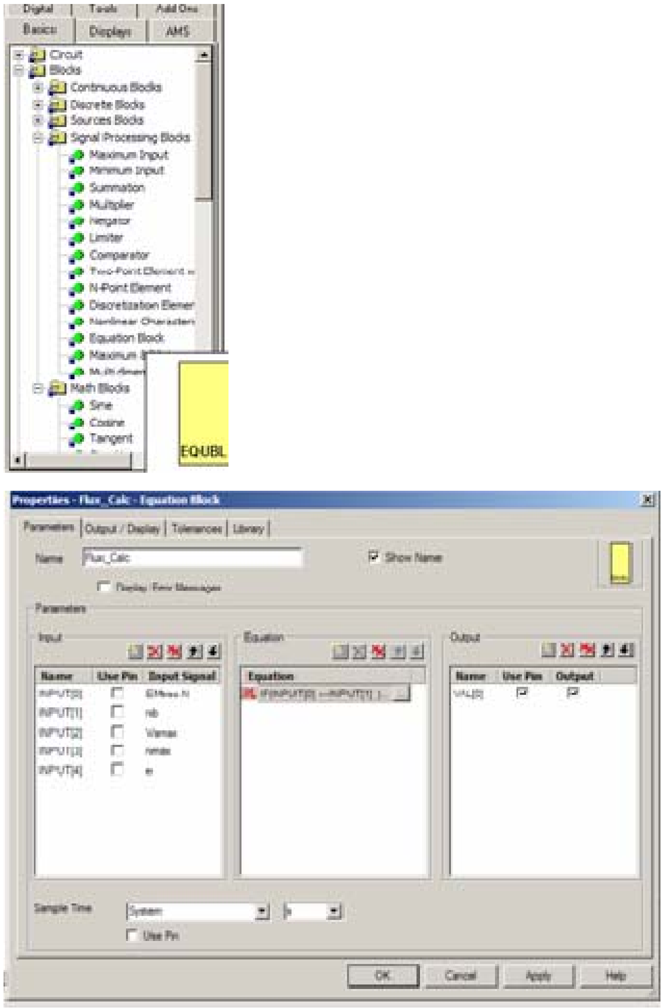
Considering electromechanical power equality;

$$Te.\ \omega_r = Ea.Ia \qquad (12)$$

$$k_t(\omega_r).Ia.\ w_r = k_e(\omega_r).\omega_r\ Ia \qquad (13)$$

$$k_t(\omega_r)= k_e(\omega_r) \qquad (14)$$

In this modified version of the model, rotor flux value is calculated by an equation block. Until the base speed of the motor, ke is kept constant, above base speed it is decreased proportional to angular rotor speed such that overall back EMF stays at the same value. Above maximum speed, motor is considered as it is in the natural mode and ke is decreased proportional to square of rotor angular speed.



Name: Flux_Calc

Name	Use Pin	Input Signal
INPUT[0]	☐	[illegible]
INPUT[1]	☐	nb
INPUT[2]	☐	Vamax
INPUT[3]	☐	nmax
INPUT[4]	☐	[illegible]

Equation
IF(INPUT[0] <=INPUT[1]) ...

Name	Use Pin	Output
VAL[0]	☑	☑

Sample Time: System

Figure 11 Modification of rotor flux

Code entered under equation section of the block for the rotor flux calculation is given below;

```
IF(INPUT[0] <=INPUT[1]  )
{
VAL[0] :=INPUT[4]  *INPUT[2] /(2*MATH_PI*INPUT[1] /60);
}
ELSE IF((INPUT[0]>INPUT[1]) AND (INPUT[0]  <=INPUT[3])  )
{
VAL[0] :=INPUT[4] *INPUT[2] /(2*MATH_PI*INPUT[0] /60);
}
ELSE
{
VAL[0] :=INPUT[4] *INPUT[2] /((2*MATH_PI*(INPUT[0]) /60))^2;}
```

To control the output torque of the electric motor, a current feedback loop is used. PI controller output is amplified by a gain block, which controls voltage source.

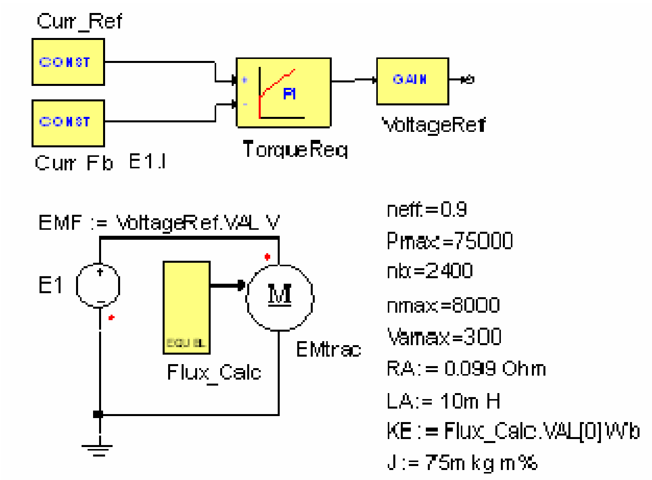


Figure 12 Modified dc motor and torque loop model

Resultant torque vs. speed graph of the electric motor and corresponding rotor flux curve have been given in the following figures.

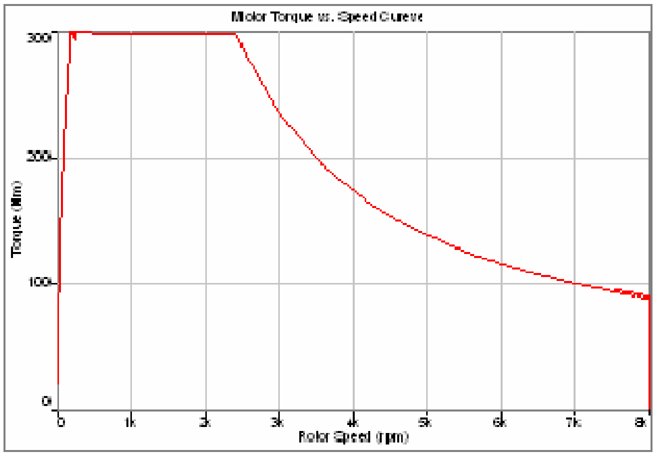


Figure 13 Torque vs. speed graph

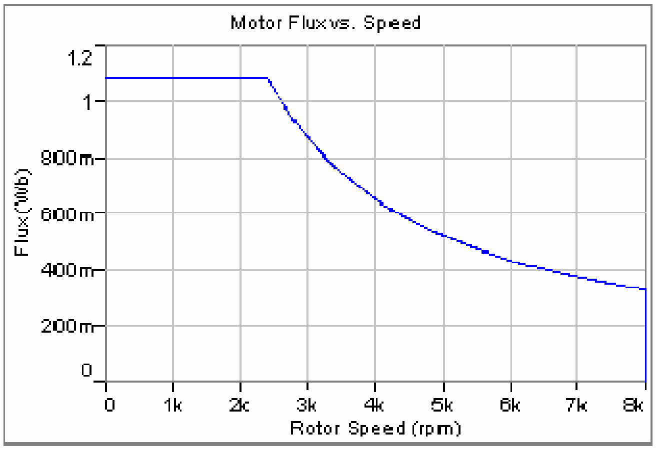


Figure 14 Rotor flux curve

5-All Electric ABS Braking

For the simulation of all electric abs braking, an electric motor connected to a wheel by a reduction gear is applied to QCM. The ratings of the electric motor are selected as seen in figure 12. This motor can supply 300 Nm torque at its shaft. A 5:1 reduction gear is considered to match 1500 Nm torque of hydraulic braking.

The QCM is modified such that total load force is reduced to the motor shaft. So the load torque is reduced by a factor of 5. In the same manner, EM rotor speed is amplified by 5 to reach to the wheel speed. Tire inertia is accepted to be referred to the rotor. Modified version of QCM is given in Figure 15.

Simulation results are given in the following figures. Stopping time is 3 seconds and total traveled distance is 43.95 meters. Electric motor power and torque curves vs. rotor speed during braking can be seen in figure 19.

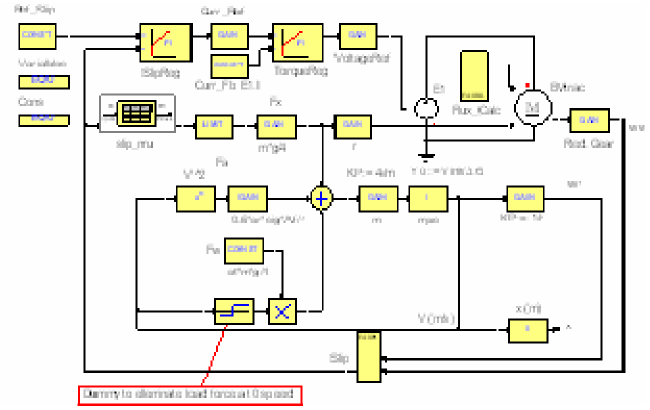


Figure 15 QCM modified for all electric braking

In this case, %20 wheel slip ratio has been achieved at the first second. Reaching maximum braking force faster yielded in reduced braking time and distance compared to hydraulic model.

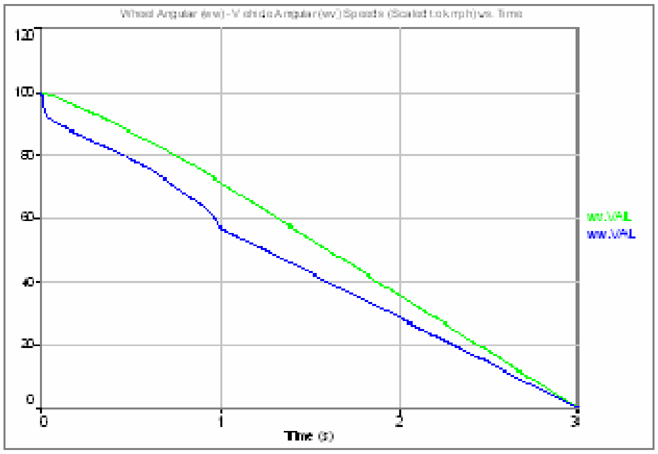


Figure 16 Vehicle and wheel speeds vs. time

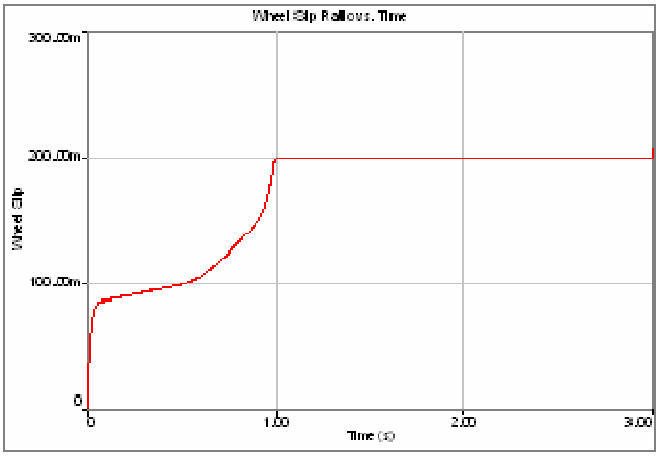


Figure 17 Wheel slip

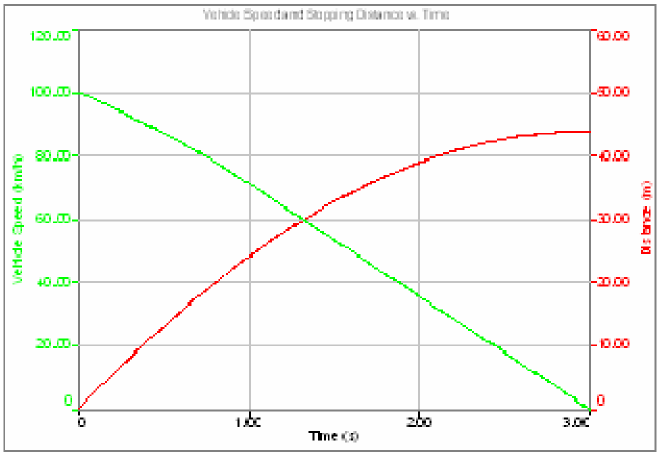


Figure 18 Stopping distance

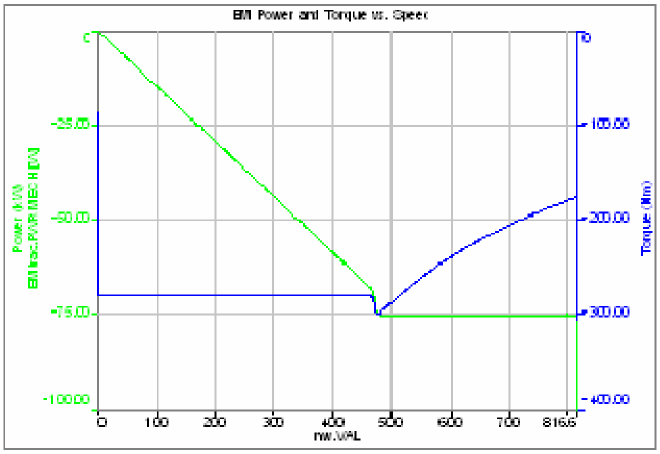


Figure 19 EM power and torque vs. rotor speed

6-Conclusion

In this study, basic modeling effort on ABS braking has been given considering hydraulic and all electric approaches on quarter car model. For all electric ABS application, a conventional dc motor model has been modified for field weakening operation.

Although simulation results show that all electric ABS response is better for panic stop situation, comparison of both systems under this simple approach is not possible. Reliability, cost and sizing issues and required energy storage device of "all electric ABS" should be considered carefully for such an analysis.

Acknowledgments

This study is supported by Ansoft Europe - EM (ElectroMechanical) Division. Special thanks go to Mr. David J. Dupuis from ANSOFT French Office.

References

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Further Readings

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