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***Abstract*—This paper presents a study in the field of electric motor drive testing. One example of propulsion drive load dynamic control system is presented. Mathematical model of the propulsion motor drive of Tesla Roadster is presented. Suggested dynamic load control system is verified using a MATLAB/Simulink computer model and a test bench. The developed methodology can be recommended to adjust the electrical drives for different kinds of testing equipment, including the synchronous, reluctance, induction, and direct current machines.**

Keywords*— *electric vehicles; test equipment; variable speed drives; electric machines; optimization component.

I. Introduction

A design and optimization procedure for electrical machines, accounting for a large amount of natural resources and other minerals as well as the energy needed, is possible to implement and it will result in a lifecycle energy and resource efficient electrical machines. The same is valid for motor drives. However, the development of such design and optimization methodology requires technical design tools as well as technological and market data. The studies on optimization of electrical motor drives [1] shows a proposed methodology that consists of building an optimization procedure based on different simulation tools of the electrical machine as well as the whole motor drive.

Special attention is paid to Electric Vehicle (EV) and Hybrid Electric Vehicle (HEV) modelling. HEVs used instead of Internal Combusting Engine Vehicles (ICEVs) could notably decrease the atmospheric pollution. The effect of using EVs could be even better. To specify the field of study, all vehicles that use batteries for propulsion could be named Battery Electric Vehicles (BEV). The BEVs' market seems to be a promising topic for electric motor drives research, assessment and application. Generic methodology can be recommended for adjusting the electrical motor drives for different kinds of testing equipment, including the synchronous, induction, and direct current machines.

Given paper presents a way to explore the electric motor drives testing possibilities for propulsion systems by using built-in Hardware-in-the-Loop (HIL) features of industrial drives. Proposed methodology could be recommended also for different loads, such as pump and fan, lift, etc.

II. Model of the Vehicle's Propulsion Drive Load

In this section, the vehicle is modelled as a road load. The vehicle and the associated forces are illustrated in Fig. 1. Calculations are based on [2]–[4]. The behavior of the road load model depends on the vehicle geometry, i.e. in this step of the vehicle mode, the modelling type of the propulsion system (ICEV, HEV, EV) does not matter until the road load is applied to the propulsion motor.

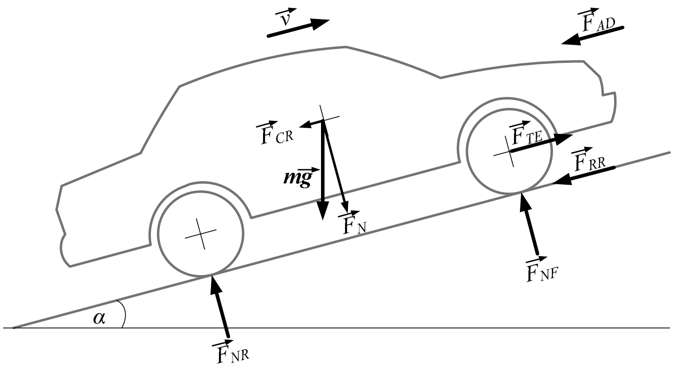

Fig. 1. Forces applied to a vehicle

Consider a vehicle of a mass m, moving at a velocity v, up a slope of an angle α (in degrees). The propulsion force for the vehicle to move forward is determined by the tractive effort F_{TE}. This force has to overcome the rolling resistance F_{RR}, the aerodynamic drag F_{AD}, the climbing resistance force F_{CR}, and the force to accelerate the vehicle (if the velocity is not constant). In that case, the base road load F_{RL} is a sum of rolling resistance, aerodynamic drag and climbing resistance force as follows:

$$F_{RL} = F_{RR} + F_{AD} + F_{CR}\,. \tag{1}$$

The rolling resistance is the force resisting the tire at the roadway surface. Under most circumstances, rolling resistance depends on the coefficient of rolling friction between the tire and the road C_{rf}, the normal force F_N due to the vehicle's weight mg, and the gravitational acceleration g. However, if the vehicle is at rest and the force applied to the road is not strong enough to overcome the rolling resistance, the rolling resistance must cancel out the applied tractive force accurately, to keep the vehicle from moving (2) and (3).

The equation for the rolling resistance can be written as

$$F_{RR} = -F_{TE}\,, \tag{2}$$

if $v = 0$ and

$$F_{TE} < C_{rf}\, mg \cdot \cos\left(\frac{\alpha\pi}{180°}\right), \tag{3}$$

otherwise

$$F_{RR} = -C_{rf}\, mg \cdot \cos\left(\frac{\alpha\pi}{180°}\right). \tag{4}$$

Aerodynamic drag is important, especially at high velocities. The aerodynamic drag depends on the air density ρ, the coefficient of drag C_d, the frontal area of the vehicle A, and the vehicle velocity v (relative to the air):

$$F_{AD} = \frac{1}{2} C_d \rho A v^2 sign(v)\,, \tag{5}$$

where $sign(v) = +1$ if $v>0$
$sign(v) = -1$ if $v<0$.

The climbing resistance force due to the road grade depends on the mass of the vehicle m, road angle in degrees α, and gravitational acceleration g. The equation for this force is

$$F_{CR} = -mg \cdot \sin\left(\frac{\alpha\pi}{180°}\right). \tag{6}$$

The road load curves of a vehicle for varying road angles are shown in Fig. 2. Tesla Roadster with its parameters listed in Table 1 was chosen for illustration. According to the literature review, coefficients of rolling friction for tires are about 0.007 for dry road and 0.004 for wet road. It can be observed that the road load increases with the velocity and with the road angle.

TABLE 1. PARAMETERS FOR THE SIMULATED VEHICLE

Parameters	Value	Unit
Vehicle mass	1235	Kg
Gravitational acceleration	9.81	m/s^2
Rolling friction (dry road)	0.0075	-
Rolling friction (wet road)	0.004	-
Air density	1.225	kg/m^3
Aerodynamic drag coefficient	0.35	-
Frontal area	1.93	m^2
Tire diameter (21 in)	0.53	m

The acceleration force is the force needed to accelerate the vehicle, governed by Newton's second law. This force will provide the linear acceleration of the vehicle

$$F_{ACC} = ma = m\frac{dv}{dt} \tag{7}$$

The total tractive effort is the sum of all the above forces:

$$F_{TE} = F_{RR} + F_{AD} + F_{CR} + F_{ACC}\,. \tag{8}$$

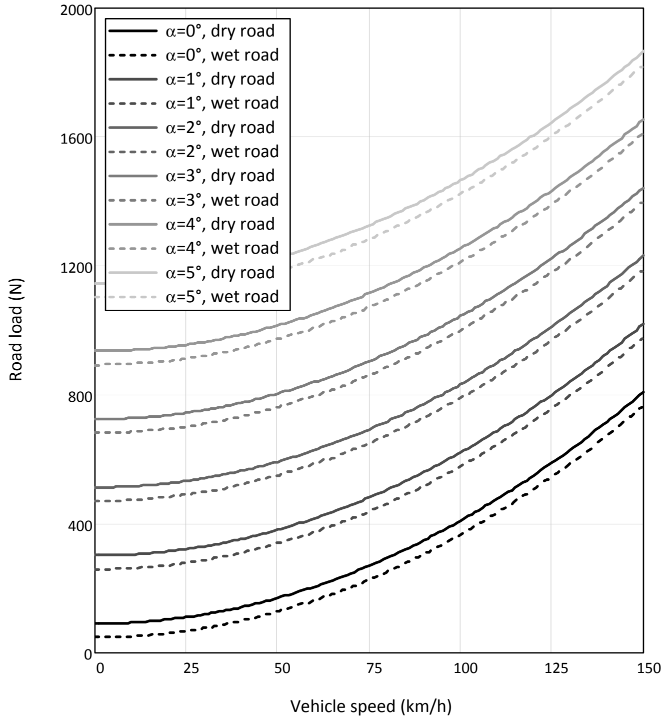

Fig. 2. Road load characteristics for road slope angle α=0°–5°and dry/wet road conditions

The vehicle's velocity is calculated by integrating the vehicle's acceleration with the starting value set to 0 km/h at $t=0$ seconds. It is equal to

$$v = \frac{1}{m}\int_{t=0}^{t}\left(F_{TE} - F_{RR} - F_{AD} - F_{CR}\right)dt\,. \tag{9}$$

In case of an ICEV, the vehicle propulsion force comes from the engine shaft torque and in a BEV, from the traction motor shaft torque. Linear velocity of the vehicle should be referred to the motor (engine) shaft. Fig. 3 shows the mechanical scheme of the motor-to-wheel transmission. T_{motor} is the torque developed by the propulsion motor, ω_{motor} is the propulsion motor's angular velocity and GR_{trans} denotes the transmission gear ratio; T_{axle} and ω_{axle} are wheel axle torque and angular velocity, GR_{diff} is the differential gear ratio; v is the vehicle speed (linear velocity).

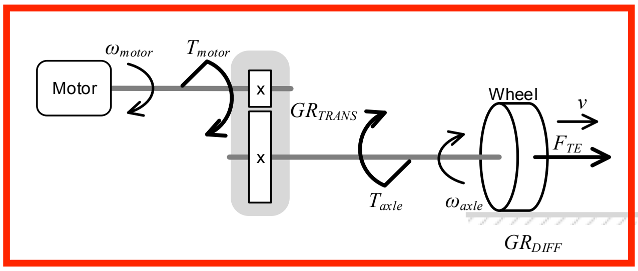

Fig. 3. Mechanical scheme of motor-to-wheel transmission

From Fig. 3 it follows that

$$GR_{TRANS} = \frac{\varpi_{motor}}{\varpi_{axle}} = \frac{T_{axle}}{T_{motor}} \text{ and} \tag{10}$$

$$GR_{DIFF} = \frac{\varpi_{axle}}{v} = \frac{F_{TE}}{T_{axle}}. \tag{11}$$

Equations (10) and (11) are related to the following:

$$T_{motor} = F_{TE} \cdot \frac{1}{GR_{TRANS}} \cdot \frac{1}{GR_{DIFF}}. \tag{12}$$

To simplify the model and prevent nonlinearity, the efficiency of the mechanical transmission is not considered (equal to 1).

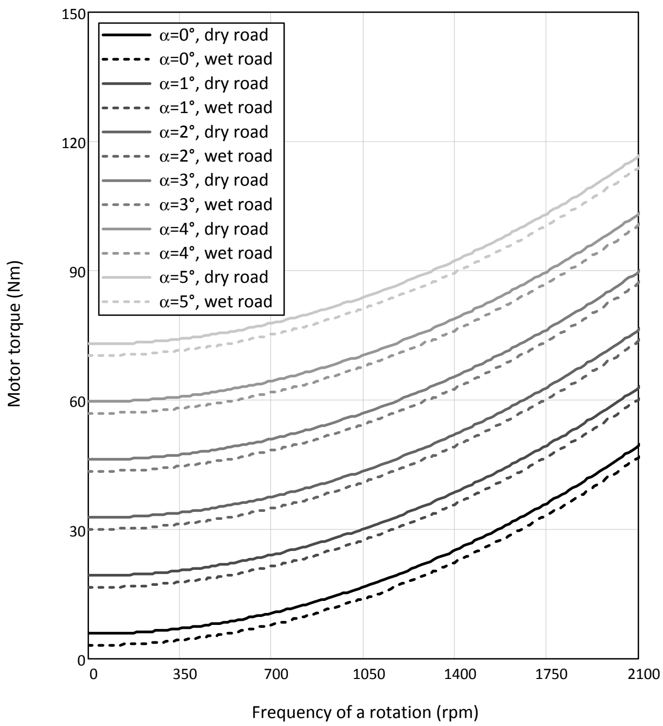

Fig. 4. Propulsion motor load characteristics for the road slope angle α=0°–5°and dry/wet road conditions

Similarly, the frequency of rotation n_{motor} of the propulsion motor could be found:

$$n_{motor} = \frac{60}{2\pi} \cdot v \cdot GR_{TRANS} \cdot GR_{DIFF}. \tag{13}$$

According to (11) and (12), the total tractive effort of the vehicle could be scaled to the propulsion motor. The motor load to the motor frequency rotation characteristics is shown in Fig. 4, the road slope angle $\alpha=0^{\circ}$–5° and dry/wet road conditions are presented. Characteristics shown in Fig. 4 are valid for a steady-state driving mode, or cruising at a constant speed without acceleration.

III. Computer Model of the Propulsion Drive of BEV

There are many different ways available to model a BEV. Mainly, all the models could be classified into three groups [5], [6]:

- Dynamic models that are based on the physical representation of different subsystems; their accuracy depends on the description of different parts of the system;
- Static model that takes into account the steady-state processes of a BEV, i.e., it has no main time constant included in the model;
- The quasi-static model, which is a collaboration between the static and dynamic models; it is similar to the static model but it has the main time included.

The dynamic model gives good simulation accuracy. Proper simulation of power electronic devices with high switching frequencies is important. That makes the dynamic models more complicated for simulation and increases the simulating time. The main drawback of the static model is that no transient processes are included in the model. With a quasi-static model, the study of steady-state and transient modes is possible.

Abundant computer software is available today for computer modelling of the BEV systems. Such software as PSIM, PSCAD/EMTDC, MATLAB/Simulink, Synopsys Saber, ANSYS Simplorer, and Dymola could be used for dynamic models. These electronic circuit simulation software packages are designed for the use in power electronics and electrical motor drive simulations. Any electronic and equivalent circuit can be designed using those simulation packages. Many universities and research institutions have their own toolboxes for EV and HEV studies: SIMPLEV (DOE Idaho Laboratory), MARVEL and PSAT (Argonne National Laboratory), CarSim (AeroVironment Inc.), JANUS (Durham University), ADVISOR (DOE National Renewable Energy Laboratory), ELPH and V-ELPH (Texas A&M University), and Vehicle Mission Simulator (Software Engineering Professionals). Most of those software packages are MATLAB based toolboxes optimized and designed for EV/HEV studies.

To test the motor of the propulsion drive, parameters of the induction motor from ABB M3AA 112m 3GAA 112022-ADC were chosen. The reason to choose the propulsion drive of the BEV based on an induction motor for this study was that it is more interesting and perspective. The computer model of the

BEV propulsion motor drive, created for the study, assumes also that experimental data be confirmed for that case, creating a real motor model that can be used for further research, is more reasonable. MATLAB/Simulink, a more flexible and powerful software package, was chosen to create the propulsion motor drive model. MATLAB/Simulink is presented in Fig. 5 and consist of grid function block 400V 50Hz, library of speed reference signals [7], electrical drive function block, measurement block and toolsets for tracing.

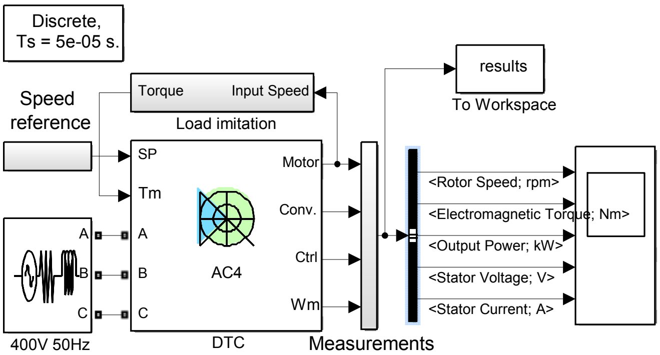

Fig. 5. MATLAB/Simulink model

A test bench, used for computer model verification, was developed in the Laboratory of Electrical Drives at the Tallinn University of Technology, Department of Electrical Engineering and has been presented previously in [8]–[10]. The test bench allows steady-state and transient mode imitation of the propulsion motor drive system of the BEV. The loading drive of the test bench allows imitating different load modes according to the road load model. The sketch of the test bench is presented in Fig. 6.

The test bench incorporates two motor drives. The testing system based on the ABB ACS800 electric drive consists of a squirrel cage induction motor, an active AC/DC power converter, a remote console, control, measurement, and cabling equipment. The testing drive is furnished with a foot pedal to imitate the real driver's habits in vehicle management. The testing motor M3AA 112m 3GAA 112022-ADC has the following parameters: rated speed of the motor - 1455 rpm, rated voltage - 400 V, frequency - 50 Hz, and rated power - 4 kW.

As it follows from Fig. 4, the testing motor of the test bench can cover only small area of real Tesla Roadster drive system, but it will be enough to test the dynamic load performance for selected vehicle.

The loading system built on the ABB ACS611 electric drive consists of an induction motor M3AA132SB 3GAA 138110-ADC, AC/DC power converter with the diode front end, remote console, and measuring and cabling equipment. ABB ACS611 represents a variant of ABB ACS800 drive with a different firmware version and similar functionality. The DTC mode of the drive operation is suitable for simulation of different loads of the real EV. The loading motor has the following parameters: the rated speed of the motor - 2820 rpm, rated voltage - 400 V, frequency - 50 Hz, and power - 4.7 kW.

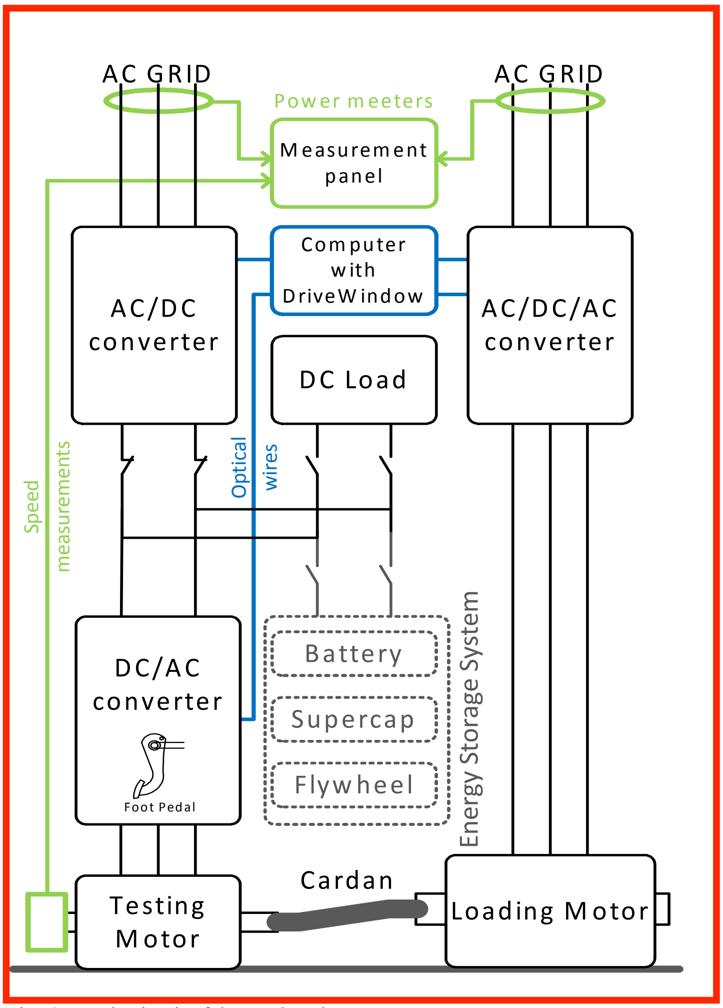

Fig. 6. Basic circuit of the test bench

IV. DYNAMIC LOAD FOR ELECTRIC MOTOR DRIVE TESTING ON THE TEST BENCH

From equation (8) it can be seen that in the tractive effort, resistance components on rolling (3) and climbing (6) do not depend on the speed, but rather on the aerodynamic drag (5) together with the acceleration force (7) varieties under speed changing. It means that together with speed reference, the load should be changed as well. Dynamic load tests could provide more realistic load of the propulsion motor drive. The dynamic load relation to the speed reference could be described as follows:

$$F_{TE}(v) = C^*_1 \cdot v^2 + C^*_2 + C^*_3 \cdot a \ , \qquad (14)$$

where, C^*_1, C^*_2 and C^*_3 are dynamic load coefficients related to the linear vehicle speed, v is the vehicle linear speed and a is the vehicle acceleration.

Dynamic load coefficients could be found as:

$$C^*_1 = \frac{F_{AD}}{v^2} = \frac{1}{2} C_d \rho A \ , \qquad (15)$$

$$C^*_2 = F_{RR} + F_{CR} \text{ and} \quad (16)$$

$$C^*_3 = m \ . \quad (17)$$

According to equations (12) and (13), the dynamic load coefficients should be reduced to the traction motor shaft:

$$T_{motor}(n_{motor}) = C_1 \cdot n_{motor}^2 + C_2 + C_3 \cdot \frac{dn_{motor}}{dt} \ , \quad (18)$$

where C_1, C_2 and C_3 are dynamic load coefficients related to the traction motor shaft, n is the rotation frequency and $\frac{dn_{motor}}{dt}$ is the angular acceleration ($\frac{dn_{motor}}{dt} = r \cdot a$, with the wheel radius r). According to the previous equations, dynamic load coefficients could be found as:

$$C_1 = C^*_1 \cdot \left(\frac{2\pi}{60}\right)^2 \cdot \left(\frac{1}{GR_{TRANS}} \cdot \frac{1}{GR_{DIFF}}\right)^3 , \quad (19)$$

$$C_2 = C^*_2 \cdot \frac{1}{GR_{TRANS}} \cdot \frac{1}{GR_{DIFF}} \text{ and} \quad (20)$$

$$C_3 = J_{eq} \cdot \frac{1}{GR_{TRANS}} \cdot \frac{1}{GR_{DIFF}} \ . \quad (21)$$

J_{eq} is the equivalent inertia moment of the system reduced to the propulsion motor shaft that could be found from the following:

$$J_{eq} \cdot \varpi_{motor}^2 = W_{k,vehicle} + W_{k,trans} + W_{k,motor} \ , \quad (22)$$

where ω_{motor} is the angular speed, W_k is the respective kinetic energy stored in the moving vehicle $W_{k,vehicle}$, vehicle's transmission $W_{k,trans}$ and motor $W_{k,motor}$. The kinetic energy stored in the moving vehicle $W_{k,vehicle}$ could be found as:

$$W_{k,vehicle} = \frac{m \cdot v_{vehicle}^2}{2} \ . \quad (23)$$

To account for the rotational kinetic energy, the total kinetic energy is assumed 1.05 times the linear kinetic energy (for Tesla Roadster) [11]. The inertial momentum of the motor J_{motor} is given in the data sheet and the kinetic energy could be found as follows:

$$W_{k,motor} = \frac{J_{motor} \cdot \varpi_{motor}^2}{2} \ . \quad (24)$$

The linear movement $v_{vehicle}$ speed could be recalculated to the motor's angular speed ω_{motor}:

$$v_{vehicle} = r \cdot \varpi_{motor} \ . \quad (25)$$

From (22)-(25) it follows:

$$J_{eq} = 1.05 \cdot m \cdot r^2 \cdot \left(\frac{1}{3.6^2}\right) . \quad (26)$$

According to data from Table 1, dynamic load coefficients for the imitation of Tesla Roadster road load related to the traction motor shaft are $C_1 = 1.171 \cdot 10^{-6}$; $C_2 = 5.785$; $C_3 = 0.447$.

Obtained coefficients were applied to the MATLAB/Simulink model and the test bench. MATLAB/Simulink model of the dynamic load presents a speed feedback that has an effect on the input torque. The MATLAB/Simulink subsystem for the dynamic load simulation is presented in Fig. 7. The figure shows the calculation of the dynamic load torque reference by using the input speed signal and the coefficients (19)-(20).

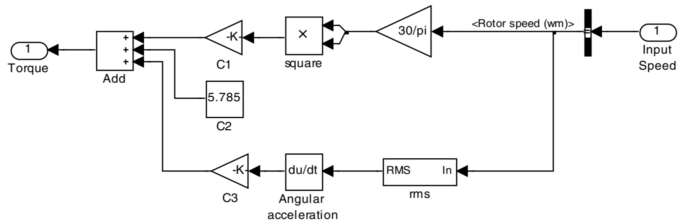

Fig. 7. MATLAB/Simulink subsystem for dynamic load realisation

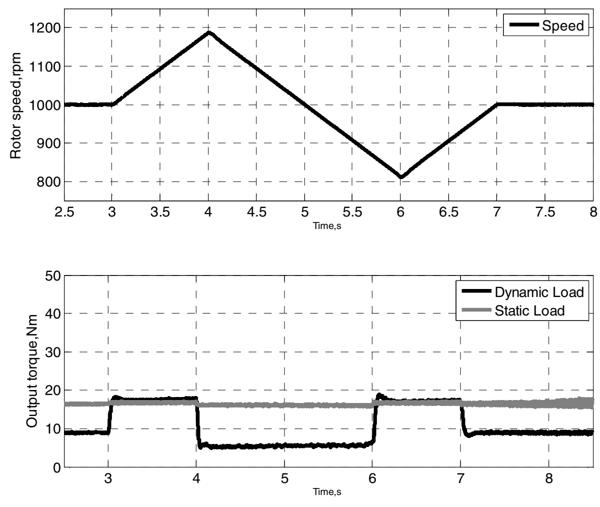

Fig. 8. Torque response on the speed triangle reference under different control modes MATLAB/Simulink model

While the experimental test bench has two frequency converters that contain a built-in controller for programming speed and torque reference signals, an adaptive program for the dynamic load was created. The ABB ACS frequency converter allows measurements of drive speed and acceleration, whereas these two signals could be used as input signals for the adaptive program. The output signal of the adaptive program is an input reference torque signal for the loading drive.

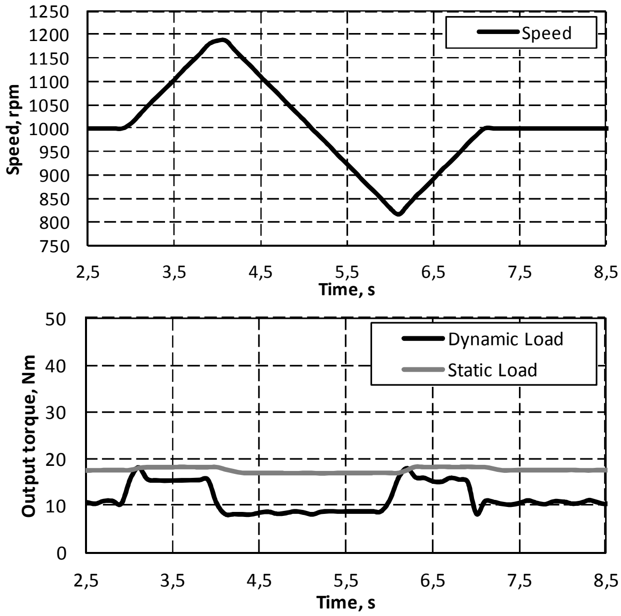

Fig. 9. Torque response on the speed triangle reference under different control modes test bench experiments

Speed and torque traces presented in Fig. 8 and Fig. 9 show the torque response of the MATLAB/Simulink model and the test bench experiments under the testing sample of the speed triangle reference applied on 1000 rpm. As can be seen from the traces, the experimental motor drive load under the dynamic load is higher under acceleration and lower under deceleration, which means that it depends on the reference speed that is very similar to a real vehicle road load.

V. Conclusion

To be able to estimate the processes of the BEV propulsion drive, it is very important to use a proper model of the propulsion drive load. The model of a BEV load is very complex, as it contains many different components, and to prevent mistakes, each component needs to be modelled accurately. In this paper a road load model, scaled to the propulsion motor of Tesla Roadster, is presented. It was found that the load of the propulsion motor increases with the velocity, with acceleration and with the road angle. Forces influencing the propulsion system of the BEV can be reduced to the propulsion motor shaft. It means that the propulsion drive of the BEV can be tested without complicated mechanical transmission and load forces, and the torque could be replaced with the torque of the propulsion motor load.

A dynamic load close to reality is proposed. The methodology for coefficient determination is presented and verified with the MATLAB/Simulink model and the test bench. Simulation and experimental results show that additional forces, occurring during the acceleration and deceleration of the vehicle, are taken into account while the results observed are more realistic. The developed methodology can be recommended to adjust the electrical drives for different kinds of testing equipment, including the synchronous, reluctance, induction, and direct current machines. Experimental validation of the approach described has demonstrated broad possibilities for the steady-state and transient modes of vehicle quality evaluation. It suits for recommendations to be made with regard to the tuning of the drive regulators, control looping, sensor allocation, and feedback arrangements.

Acknowledgment

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