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17–19 May 2018, Zadar, Croatia

Road and Rail Infrastructure V

Stjepan Lakušić – EDITOR

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A CALCULATION OF ELECTRIC DRIVES REQUIREMENTS IN A HYBRID SWEEPER

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Abstract

The article presents electric drives calculation of the new hybrid sweeper. Due to the increasing restrictions of CO_2 emission it is desirable to have zero emission sweeper operating in city centers. Outside of city centers, where regulations are not strict, there is an opportunity to start the auxiliary engine and charge the battery pack. Considering range, gradeability, maximum vehicle speed and gross vehicle weight, it is difficult to achieve the same performances as a conventional sweeper. Currently, the energy density of diesel fuel is more than 50 times higher than the energy density of batteries. An additional challenge for engineers is inadequate charging infrastructure, hence electric vehicles still need subsequent time for charging. In addition, hybrid sweeper requires more frequent charging than the average personal vehicle due to additional drives mounted on the sweeper. A special characteristic of the considered hybrid sweeper is its relatively high maximum speed and high continuous torque demand along with gross vehicle weight of 4500 kg. The article deals with appropriate utilization of the main electric drive regarding given requirements. The sweeper model is developed by considering main forces that act on the vehicle including rolling resistance, aerodynamic drag, incline force and inertia. By using vehicle model and randomly generated road grades, the simulation of sweeper has been performed. Likewise, the simulation of a centrifugal fan drive is developed. Finally, the intermittence of hydraulic drives is used to define requirements of an electrohydraulic drive.

Keywords: Hybrid electric vehicle, road sweeper, traction motor, electric motor drive, centrifugal fan drive, intermittence.

1 Introduction

The European Union is strongly focused on reducing gas emissions by 2050. As the transport sector significantly contributes to gas emission, electric motors are being increasingly used in electric cars [1]. Some cities have already announced that diesel engines will be banned in their city centers. Sweepers are not going to be an exception of the directive. They are even more affected because they operate in pedestrian zones where regulations are strict.
The main challenge in the process of sweeper electrification is to find electric motors of approximately equal size and equal mechanical performance as hydraulic drives. The hydraulic system provides a possibility to change transmission ratio continuously, but their drawback is low efficiency [2]. To maintain approximately the same dimensions of electrical motor, it is desired to use motors with high torque and power density [3].

2 Main electric drive

2.1 Main electric drive design in accordance to required parameters

It is common practice to simplify a vehicle model by representing it with a material point [4]. Therefore, the sweeper is represented by the following motion equation:

$$m\frac{dv}{dt}=F_t-c_{rr}mg\cos(\alpha)-mg\sin(\alpha)-\frac{1}{2}\rho_{air}Ac_dv^2 \tag{1}$$

where m is the vehicle mass, F_t traction force, c_{rr} rolling resistance coefficient, c_d aerodynamic coefficient, A vehicle front surface, α road gradient, g gravitational acceleration, ρ_{air} air density and v is the vehicle speed. A traction force is generated from the main electric drive. If the gearbox is placed between electric drive and the wheel, traction force can be calculated in the following manner:

$$F_t=\frac{M_m}{r_{wheel}}i_{gbx}\eta_{gbx} \tag{2}$$

where i_{gbx} is the gearbox ratio, r_{wheel} is wheel radius, M_m is motor torque and η_{gbx} is gearbox efficiency. As the sweeper uses existing gearbox with predefined ratio and wheels with known radius (Table 1), the required traction force can be achieved only by choosing an appropriate electric motor with enough shaft torque.

Table 1 Sweeper parameters

Parameter	Value
mass, kg	4500
c_{rr}	0.317
c_d	0.65
ρ_{air}, kg/m³	1.2
A, m²	2.5
r_{wheel}, m	0.317
i_{gbx}	10
v_{max}, km/h	50

The main requirement for the sweeper is to have a decent gradeability. The vehicle should be able to overcome road grade of 15 % continuously and 30 % periodically. The calculation of gradeability is performed by using Eq. (3). The equation doesn't consider acceleration of the sweeper because it is driving at low speed (<12 km/h) during the cleaning process.

$$\alpha=2\operatorname{atan}\left(\frac{mg-\sqrt{-F_t\left(F_t+2F_{ad}\right)-F_{ad}^2+m^2g^2\left(1+c_{rr}^2\right)}}{F_t-F_{ad}+mgc_{rr}}\right) \tag{3}$$

The required motor torque for different road grades is shown in Fig. 1. It is important to note that Fig. 1 is calculated considering constant speed of 5 km/h.

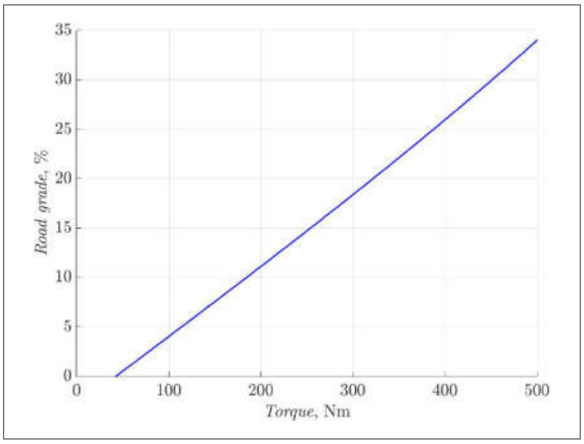

Figure 1 Sweeper gradeability

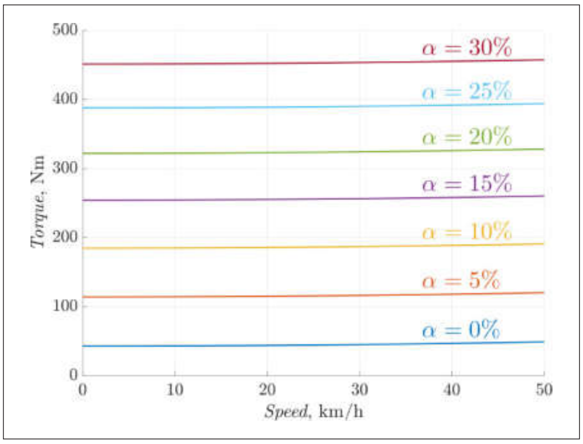

Figure 2 Required motor torque

The analysis of sweeper performances is shown in Fig. 2. Torque vs speed curves are given by considering various road angles. As expected, higher road grade leads to higher motor torque demand. This analysis was performed considering constant acceleration of 1 m/s^2. After the market research has been performed and all constraints were taken into consideration, the motor with a continuous torque of 250 Nm and peak torque of 500 Nm has been chosen. The chosen motor has axial flux configuration and low axial length due to the limited space available for motor installation. The available motor torque is enough to drive the sweeper on the incline of more than 14 % continuously and more than 30 % intermittently which meets set requirements.

2.2 Load simulation

The aim of sweeper control system is to show how different loads (road grades) affect motor torque. A road grade curve is obtained by using random number generator while the total simulation time is divided in six-time slots and each time slot is paired with one road grade group. Time slots are not equal, their length is chosen according to the distribution shown in Table 2.

Table 2 Road grade distribution

Road grade	Length
0 – 5 %	35 % of time
5 – 10 %	25 % of time
10 – 15 %	20 % of time
15 – 20 %	10 % of time
20 – 25 %	8 % of time
25 – 30 %	2 % of time

It is worth to note that this profile is not expected in a real condition because there is no negative slope but only positive ones. In this case it is justified to use this road grade profile because the aim is to test main electric drive performances. If the total sweeper consumption was tested, the more realistic terrain profile would be generated.
Fig. 3 shows sweeper speed response while Fig. 4 shows motor torque response. The road grade curve is almost the same as torque curve. The results indicate that even for this very demanding road profile a motor torque limit of 500 Nm is not reached. To test real limits of the electric motor a thermal calculation needs to be performed.

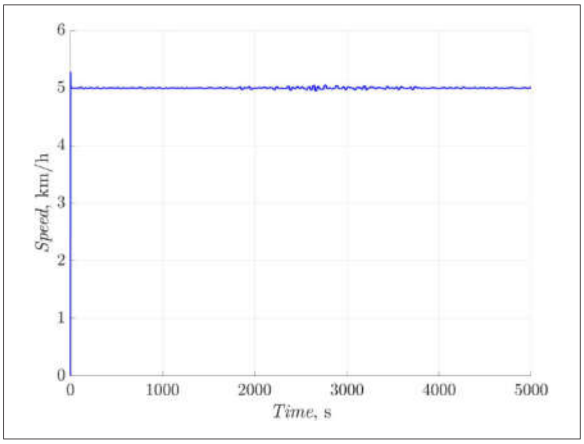

Figure 3 Sweeper speed response

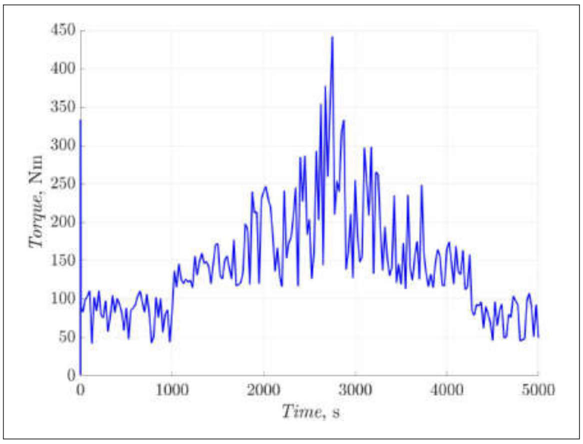

Figure 4 Motor torque response

3 Centrifugal fan drive

3.1 Mechanical characteristics of centrifugal fans

The main function of a road sweeper is cleaning litter, dust, gravel and leaves from the roads. As an actuator of the vacuuming function, a radial type centrifugal fan with direct motor drive has been selected. The motor needs to be chosen carefully regarding torque intermittence and efficiency, as it is the most notable energy consumer of the sweeper for longer time periods. To evaluate the performance of a specific motor which drives the fan, the mechanical characteristics of a fan needs to be considered. The fundamental curve describing fan performance at a constant speed is the total static pressure increase – flow curve Δp_f (Q). An operating point of a fan is defined by an intersection of the total pressure increase curve (Δp_f (Q)) and the system curve (Δp_s (Q)). A system curve describes the load on a fan that a specific system has. That load is the pressure drop (Δp_s) which the fan needs to overcome to have a specific volume flow (Q). Viscous friction of the air causes load torque of the fan to increase quadratically with the angular speed [5].

$$M_t = k \cdot \omega^2 \tag{4}$$

As the suction canal of the sweeper opens and closes (dampening), and as the sweeper sweeps the dirt and litter, the system curve of the fan changes.

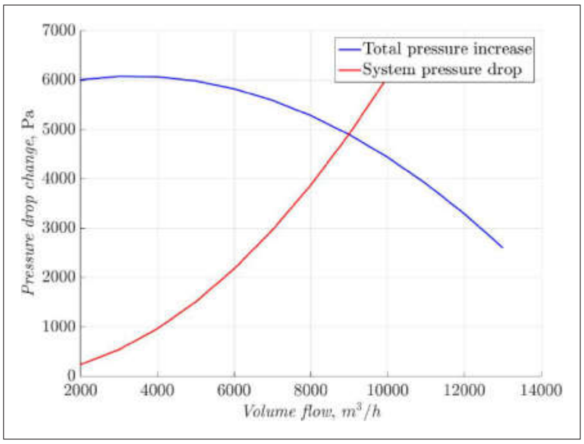

Figure 5 Characteristic curves

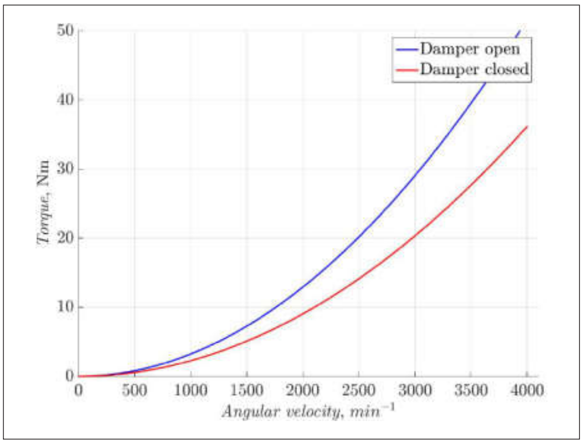

Figure 6 Torque (load) curves

Dampening increases pressure drop, which moves the static operating point higher up the pressure increase curve, and therefore decreases the flow rate Q. Because the output power of a radial centrifugal fan increases monotonically with Δp and Q $\left(\frac{dP(\Delta p,Q)}{dQ}>0\right)$ up to the rated flow, the decreased flow will decrease the output power of the fan, which consequently causes the required shaft torque to decrease [6]. This change could essentially be observed as a decrease in ventilation factor k. It is known that the required shaft power decreases to 70 % of maximum power when system dampening (closing of the suction canal) occurs. Knowing this, applying Eq. (4) and knowing that the output power of the undamped system at 3500 min^{-1} is 14.5 kW, the two torque characteristics can be constructed (Fig 6.). Ventilation factors k for undamped (k_1) and damped (k_2) fan are then calculated as follows:

$$k_1=\frac{P\cdot 30^3}{n^3\cdot\pi^3}=2.94\cdot 10^{-4}\,\frac{\mathrm{Nm}\cdot s^2}{\mathrm{rad}^2},\quad k_2=\frac{0.7\cdot P\cdot 30^3}{n^3\cdot\pi^3}=2.06\cdot 10^{-4}\,\frac{\mathrm{Nm}\cdot s^2}{\mathrm{rad}^2} \tag{5}$$

3.2 Centrifugal fan dynamics

To meet thermal limits, the rated motor torque needs to be at least equal to root mean square (RMS) of the load torque. The RMS torque is calculated as:

$$M_{RMS}=\sqrt{\frac{\sum_{i=1}^{n}T_i M_i^2}{\sum_{i=1}^{n}T_i}} \tag{6}$$

If the acceleration time of a motor is not negligible in comparison to total load sequence period, it needs to be included in calculation of RMS torque.

$$J_{tot}\frac{d\omega}{dt}=M_M-M_t \tag{7}$$

From Eq. (7), the acceleration time could be calculated by solving the integral:

$$\Delta t_a=J_{tot}\cdot\int_{\omega_1}^{\omega_2}\frac{d\omega}{M_M-k_1\omega^2}=J_{tot}\cdot\left(\frac{\operatorname{artanh}\left(\sqrt{\frac{k_1}{M_M}}\cdot\omega_2\right)}{\sqrt{M_M\cdot k_1}}-\frac{\operatorname{artanh}\left(\sqrt{\frac{k_1}{M_M}}\cdot\omega_1\right)}{\sqrt{M_M\cdot k_1}}\right) \tag{8}$$

It is assumed that during the acceleration, the torque is constant, saturated by the inner controller and that the fan damper is fully open (k_1). Other friction forces are ignored. The fan accelerates from ω_1 to ω_2. The total moment of inertia needed to complete the calculation of Eq. (8) is estimated by decomposing the rotating parts of a fan to the sum of the moments of inertia of plain geometric shapes:

$$J_{tot}=J_M+\frac{\rho\pi}{2}\left(d\cdot r_{out}^4+l_1\left(r_{out}^4-r_{in,1}^4\right)+l_2\left(r_{out}^4-r_{in,2}^4\right)+l_3\left(r_{out}^4-r_{in,3}^4\right)\right) \tag{9}$$

Where J_M is the moment of inertia of a motor, ρ is the density of the steel, r_{out} is the outer radius of rotating blades, d is the thickness of the steel sheet, l_i are the lengths of the plain segments of the fan, and $r_{in,i}$ are the inner diameters of the ring section. The total moment of inertia is estimated to be around 1 kgm^2. The maximum allowed angular speed of a motor given the intermittence needs to be determined. It is assumed that the motor is controlled in closed loop and that the speed reference of the fan is set by the vehicle driver. Given the acceleration time, the assumption that the system damper is opened (i.e. centrifugal fan is at its nominal shaft power) for 20 % of the total cycle time, while the remaining time the damper is closed to reduce the total torque (to 70 % of maximum fan shaft power), it can be shown from Eq. (6) and Eq. (4) that the maximum allowed angular speed regarding the rated motor torque (M_{rm}) is:

$$n_{MAX} = \frac{n_r}{\sqrt{\frac{P_r \cdot 30}{n_r \cdot \pi} \cdot \sqrt{\frac{1}{T_{tot}}\left(T_1\left(\frac{M1}{M_r}\right)^2 + T_2\left(\frac{M2}{M_r}\right)^2 + \ldots + T_n\left(\frac{Mn}{M_r}\right)^2\right)}}} \cdot \sqrt{M_{rm}} = C \cdot \sqrt{M_{rm}} \tag{10}$$

If the torque during acceleration is limited to 150 % of the rated fan torque, and it is established that acceleration lasts 2 % of the total period (using Eq. (8)), then if the rated motor torque is 30 Nm, maximum allowable speed setpoint is 3424 min^{-1} . Eq. (10) is applicable if the motor is not operating in field weakening region. However, if the maximum calculated allowable speed is above corner speed (n_s), the Eq. (11) must be applied, and the maximum rotational speed is recalculated as:

$$n_{MAX} = \sqrt[3]{C^2 \cdot M_{rm} \cdot n_s} \tag{11}$$

3.3 Centrifugal fan control system

It is essential that the centrifugal fan operates in a closed loop control system with rotational speed as its reference. Closed loop compensates for the torque change as the fan is dampened and undampened but also compensates for the non-linear mechanical characteristics of a system as seen in Fig. 6. The control system consists of the PI controller with anti-windup. The plant used is an equivalent model of the motor and electrical subsystem with its own inner current control loop, reduced to one dominant time constant. The centrifugal fan is modeled according to Eq. (7), with ventilation factors k changing between the ones observed in Eq. (5) within the intermittence time periods specified before. The torque limit is set according to maximum motor characteristics. RMS torque calculated for the fan running at 3300 min^{-1} is 27.85 Nm. The nonlinearity of the drive is not visible in responses for a single reference speed. It is however reflected as a relatively small change of closed loop gain.

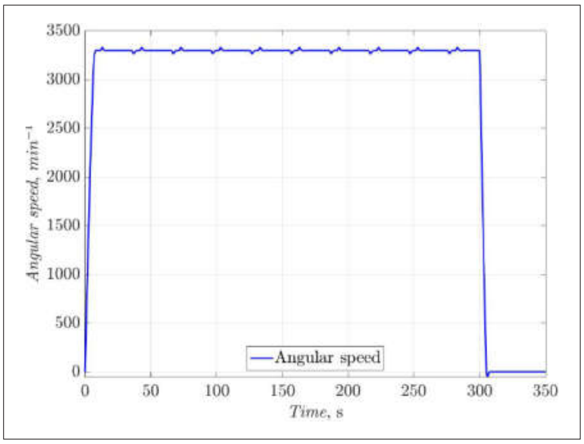

Figure 7 Speed response

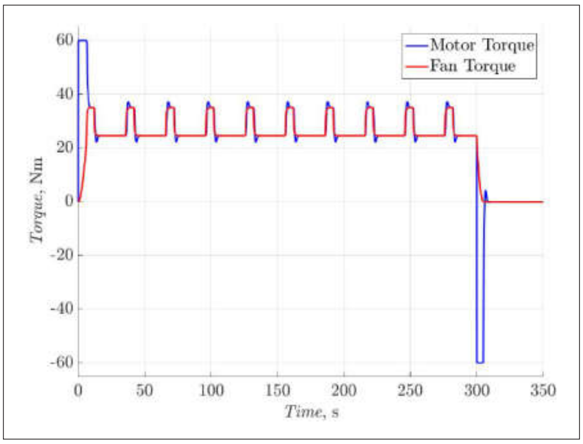

Figure 8 Torque response

4 Electrohydraulic drive

4.1 Electrohydraulic drive power calculation

To calculate required power of an electric motor which will drive a hydraulic pump, an RMS power of a hydraulic subsystem needs to be calculated. The subsystem consists of a main pump, hydraulic motors which run continuously and hydraulic motors which run intermittently. Motors which require continuous power while the sweeper is operating (in cleaning mode) include the drives of brushes, pumps and steering system. The RMS power of such drives is equal to their continuous power. Concurrently, some drives run intermittently and scenario in which intermittent drives require most RMS power needs to be considered. RMS power of such drives could be calculated analogous to calculating the RMS torque as shown in Eq. (6).

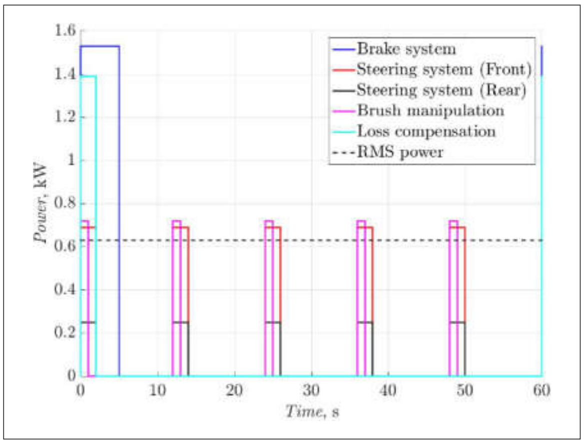

Figure 9 Peak powers of intermittent hydraulic drives

RMS power of intermittent hydraulic drives shown in Fig. 9 is $P_{RMS,I}$ = 0.63 kW, while their peak power is $P_{MAX,I}$ = 4.59 kW. The power of continuous hydraulic drives is $P_{RMS,C}$ = 7.74 kW. Maximum possible total RMS power from which the motor is selected, is calculated as:

$$P_{RMS} = \sum_{i=1}^{m} \sqrt{\frac{\sum_{j=1}^{n} T_j P_j^2}{\sum_{j=1}^{n} T_j}} \tag{12}$$

where total RMS power is calculated as the sum of all RMS powers for all individual hydraulic drives (marked as i), calculated for every discrete time interval j with its appropriate instantaneous power. This concludes that the total rated power of electrohydraulic actuator needs to be P_{RMS} = 8.37 kW. The motor is required to operate at the rated rotational speed of a main hydraulic pump. It is also worth noting that the similar calculation of intermittence also needs to be done for the rated pressure of the hydraulic pump according to the needs of hydraulic motors.

5 Conclusion

The calculation of main electric drive parameters has been performed by using a simple vehicle model. A computer simulation of main electric drive with randomly generated road grades approved that electric motor operates within defined torque limits. A thermal calculation should be implemented to verify that a motor operates within allowed thermal limits. Likewise, centrifugal fan drive requirements have been calculated and tested using closed loop control with speed reference. Finally, a calculation of electrohydraulic drive has been done by taking into consideration the intermittence of each hydraulic drive. The calculation considers peak and root mean square requirements.

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