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Governor principle for increased safety and economy on vessels with diesel-electric propulsion

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***Abstract*— In this paper, a governor principle for marine diesel-electric power plants with a normal and an emergency mode is proposed. In the normal mode, the governor is tuned such that the variations in the electric frequency are weighed against operational costs such as specific fuel consumption, pollutant emissions and wear-and-tear of the diesel engine due to thermal variations. In emergency mode, the governor disregards the operational costs and attempts to keep the frequency as steady as possible. This leads to larger margins to the under-frequency condition and therefore reduces the risk of blackout. It also allows for a more reliable synchronization of additional generating sets to the electric grid. Because the emergency mode is entered under abnormal conditions only, the overall increase in operational costs resulting from this addition is negligible. The governor is implemented as a receding horizon controller.**

I. INTRODUCTION

Diesel-electric power plants have become the de facto standard primary power source for ships in the offshore industry that are expected to spend a large part of their time in dynamic positioning. This operational mode is important for ships that perform functions such as supplying offshore installations, drilling and oil production, laying pipes, supporting divers or ROV operations, or performing surface-based measurements or exploration. The main benefits of diesel-electric propulsion and thrusters are reduced power consumption under operational conditions typical for certain vessel types, resilience to equipment failures, lighter engines, and less noise [1]. Ensuring fuel-optimal operation during normal conditions as well as continued fulfillment of the operational requirements and safety despite equipment failures introduces challenges for the control system.

A diesel-driven power plant operates optimally when the load is large and close to being constant. The diesel engine may be physically unable to respond to large and sudden load increases due to the relatively slow turbocharger dynamics. Dependent upon the tuning, the governor may also respond slowly to changes in load, resulting in frequency fluctuations. Although small frequency fluctuations are acceptable, the protection relays typically set a limit of ±10% on the maximal frequency deviation from the nominal value. If this limit is exceeded, the generators and some of the consumers will disconnect from the power grid.

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A thrust allocation algorithm that attempts to assist the power management system in reducing the load variations on the power plant by counteracting the load variations from other consumers on the ship has been introduced in [2] and [3], while in [4] the counteracting of the load variations is performed by the local thruster controllers.

A diesel-electric power plant typically consists of several generator sets connected to one or more electric buses, in a configuration that allows the gensets to be connected and disconnected as the operational conditions change. The connection and disconnection of gensets can be performed for example according to a precalculated optimal start-stop table as in [5] or by ensuring through a dynamic simulation that the power plant is at all times capable of continuing operations in a predefined worst case scenario, as in [6]. The operation of the power plant is supervised by the power management system (PMS). Legacy industrial implementations rely on distributed control to ensure a stable frequency, load sharing and fault tolerance on the electric bus. A typical way of implementing this is to use droop, ie every governor has a set point that depends on its current power output.

This mechanism has several drawbacks. A drooped governor relies exclusively on feedback control. A significant improvement can be achieved with use of a feed-forward of the load variations. A feed-forward from the load measurements on the terminals of each generator was tested in [7], and a feed-forward from the load preview information from the PMS was tested in [3].

Large variations in the power output of the diesel engine will increase the specific fuel consumption, wear-and-tear, soot and particle emissions. Unfortunately, on the marine power plants there are large and often rapid variations in the consumed power, and if those variations are not closely matched by the diesel engines, the electric bus frequency will fluctuate. The use of a centralized control system as suggested in this paper allows better balancing between those considerations.

In normal operational conditions it is preferable to accept some frequency variations. However, under circumstances with high blackout risk it will be beneficial to temporarily disregard the operational costs and keep the frequency as steady as physically possible.

Such circumstances could arise during activation of the fast load reduction system, activation of the power constraint in the thrust allocation algorithm, a significant drop in frequency (that should have been avoided, but wasn't), closing of a bus tie that may cause faults to be exposed, or any other

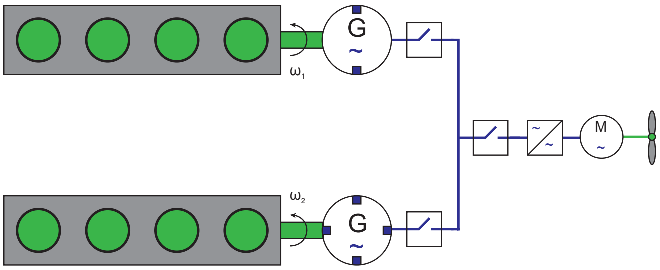

Fig. 1. A simple example of a diesel-electric propulsion system. Two four-cylinder diesel engines each drive its own electric generator. When both circuit breakers are connected, the engine that drives the two-poled generator has to rotate twice as fast as the one that drives the four-poled generator, i.e. $\omega_1 = 2\omega_2$.

signs that the operational safety needs to prioritized.

The main contribution of this paper is an application of a receding horizon control strategy on a marine power plant, allowing a more precise control in presence of predictable load fluctuations, as well as a capability to temporarily disregard the wear-and-tear and economic concerns to aid recovery to safe operational conditions. The biggest challenge with implementing this control strategy was developing a model that is suitable for use in the receding horizon control. The modelling is discussed in Section II. The proposed controller architecture is described in Section III. The results from simulation tests are presented in Section IV.

II. Modeling

High fidelity models of most of the individual components that constitute a diesel-electric power plant are readily available in the literature. The control engineering challenge is thus to find a simple model with sufficient fidelity to predict the behavior of aspects of the controlled system that are relevant on the time scale that is relevant. Although higher fidelity is in general desirable, an overly complex model would impede the engineering task.

A state-space model of a connected bus will be introduced in Subsection II-D, and a model for the torque output from a diesel engine will be introduced in Subsection II-E.

A. Abbreviations and Normalization

Normailization parameters in this paper are as follows:

Description	Abbreviation	Unit
Nominal mechanical angular velocity of engine number i.	$\omega_{r,i}$	rad/s
Nominal electrical angular velocity of the bus.	$\omega_{r,e}$	rad/s
Nominal mechanical power output of engine number i.	$P_{r,i}$	W
Total nominal mechanical power connected to the bus.	$P_r = \sum_i P_{r,i}$	W
Inertia time constant of engine number i.	$H_i = \frac{\frac{1}{2} I_i \omega_r^2}{P_r}$	s

A typical marine power plant runs at 60 Hz, so normally $\omega_{r,e} = 2\pi \cdot 60$. The nominal mechanical angular velocity of each connected engine depends on the number of poles in the generator, such that $\omega_{r,i} = k_i \omega_{r,e}$ with a constant k_i. The inertia time constant is defined as a function of the moment of inertia I_i of the rotating mass in the genset. The parameter H_i is usually available in the specifications of a genset.

The following state variables are defined both in S.I. units and in the normalized units, with the subscript "pu" used to separate the variables expressed in per-unit scale:

Description	Normalization
Mechanical angular frequency of engine i.	$\omega_i = \omega_{pu}\omega_{r,i}$
Electrical angular frequency on the bus	$\omega_e = \omega_{pu}\omega_{r,e}$
Total power generated by the engines	$P = p_{pu} P_r$
Mechanical torque produced by engine i	$\tau_{m,i} = \tau_{pu,m,i}\tau_{r,i}$
Mechanical power produced by engine i	$p_{m,i} = p_{pu,m,i} p_{r,i}$

The reason why ω_{pu} is used as the per-unit rotational velocity for all gensets as well as the electrical frequency will be explained in Subsection II-D.

B. Equations for a single genset

The rotational dynamics of a single genset are described by the swing equation. Its representation in the SI units and the per-unit form are listed in the table below. The difference between the delivered mechanical torque τ_m and the load torque τ_l is represented as $\Delta\tau$ and $\Delta\tau_{pu}$ respectively for SI and per-unit representation.

Description	SI units	Per-unit
Swing equation	$I\dot{\omega} = \Delta\tau$	$2H\dot{\omega}_{pu} = \Delta\tau_{pu}$
Power as function of rotational velocity	$P = \omega\tau$	$p_{pu} = \omega_{pu}\tau_{pu}$

C. Losses

The diesel engines are connected to the loads through the generator and a dynamic electric distribution network that may have many possible configurations and voltage levels. The distribution system is not modeled in this paper, but is assumed to transfer the power from generators to consumers with losses being proportional to the transferred power. This implies that a system with losses is equivalent to a lossless system with a smaller power output, making the losses largely irrelevant to the dynamical behavior of the system.

D. Connected bus

The dynamics of a bus connection are such that if one of the engines for some reason increases its speed relative to the other engines, it would quickly overtake a larger share of the load, forcing it to slow down while the other engines increase their speeds because they would get a lower share of the load.

This dynamic restricts all the gensets connected to a single bus to rotating at the same electrical frequency. For example, if a four-poled and a two-poled genset are connected to the same bus, then the two-poled genset will be forced to spin two times faster than the four-poled genset. This is illustrated in Figure 1.

It is therefore possible to represent the speed of each engine in terms of the electrical frequency as $\omega_i = k_i\omega_e$. In per-unit terms, ${}^{\omega_i}/_{\omega_{i,r}} = {}^{k_i\omega_e}/_{k_i\omega_{e,r}} = \omega_{pu} \forall i$. Thus,

$$I_i\dot{\omega}_i = \Delta\tau_i = {}^{\Delta p_i}/_{\omega_i} \tag{1}$$

$$I_i k_i^2 \dot{\omega}_e \omega_e = \Delta p_i \tag{2}$$

Where $\Delta p_i = \Delta\tau_i\omega_i$ is the difference between the power consumed by the load and power produced by the diesel engine on genset i. Summation over all gensets connected to a bus yields

$$\sum_i \{I_i k_i^2\}\dot{\omega}_e\omega_e = \Delta P \tag{3}$$

$$\sum_i \{I_i k_i^2\}\omega_{r,e}^2\dot{\omega}_{pu}\omega_{pu} = \Delta p_{pu} P_r \tag{4}$$

where Δp_{pu} is the per-unit difference between the load on the bus and the total mechanical power that is generated. Note that since this equation concerns the entire bus, the total rated power on the bus $P_r = \sum_i P_{r,i}$, is used for normalization. Next, defining the inertia time constant for a system of connected gensets as

$$H = \frac{\frac{1}{2}\sum_i I_i\omega_{r,i}^2}{\sum_i P_{r,i}} \tag{5}$$

and then substituting $\omega_{r,i} = k_i\omega_{r,e}$, the equation above can be transformed to

$$H = \frac{\frac{1}{2}\sum_i \left\{I_i k_i^2\right\}\omega_{r,e}^2}{P_r} \tag{6}$$

Inserting this into (4) yields

$$2H\dot{\omega}_{pu}\omega_{pu} = \Delta p_{pu} \tag{7}$$

The electrical torque does not represent any measurable physical quantity, but is defined in analogy to the mechanical torque as $\tau_{pu,e} \triangleq \frac{p_{pu}}{\omega_{pu}}$. This can be used to transform the equation above to

$$2H\dot{\omega}_{pu} = \Delta\tau_{pu,e} \tag{8}$$

which is analogous to the swing equation for a single engine, as defined in Subsection II-B.

Additionally, to calculate the load on individual generators, we can make use of the fact that for each genset, the following holds:

$$2H_i\dot{\omega}_{pu}\omega_{pu} = \Delta p_{pu,i} \tag{9}$$

Since we have that $\Delta p_{pu,i} = p_{pu,m,i} - p_{pu,l,i}$, the load $p_{pu,l,i}$ on each generator can be calculated.

E. Mechanical power from the engine

Many types of engines can be used as a prime mover in a marine power plant, however the type of prime mover dominantly utilized in the maritime industry is the diesel engine.

The model in this text is based on the model presented in [7], which is in turn based on the quasi-steady cycle-mean-value model from [8]. With small variations it is widely available in the literature, such as [9]. The dynamics of a diesel engine are such that as long as there is sufficient pressure in the scavenging receiver it is possible to set the torque delivery of a diesel engine essentially at will by changing the amount of fuel that enters the next cylinder in the firing sequence. The delay before the new torque will be put on the shaft depends on when the next cylinder in the firing sequence will be available for fuel injection, and a short fuel evaporation and ignition delay.

Oxidizer concentration in the scavenging receiver does however set a physical limit on how fast the torque output of an engine can increase. The turbocharger compressor is driven by a turbine, which derives its energy from the exhaust gas of the engine. For a given engine power output at a given engine RPM, the pressure in the scavenging receiver will converge to an equilibrium point. The time it takes to build up the pressure severely limits the dynamic response of the engine. As in [7], we assume that for control purposes it is sufficient to estimate a linear relationship between the engine power output at a given RPM and the equilibrium pressure in the scavenging receiver, and then assume that the pressure will converge to the equilibrium pressure exponentially.

The exact mathematical description of the dependence of the combustion efficiency on the air-to-fuel ratio and the combustion efficiency varies somewhat between different models. What matters for the purposes of this paper is that there is an upper limit on how much the torque can be increased over a very short period of time.

The relevant equations from [7] are restated here

$$\tau_m = \eta_c F_r \tag{10}$$

$$\dot{\omega}_t = -\kappa_1(\omega_t - p_{pu,m}) \tag{11}$$

$$AF = \frac{m_{a,0} + (1 - m_{a,0})\omega_t}{F_r} \cdot AF_n \tag{12}$$

$$\eta_c = \begin{cases} 1 & AF \geq AF_{high} \\ \frac{AF - AF_{low}}{AF_{high} - AF_{low}} & AF_{low} < AF < AF_{high} \\ 0 & AF \leq AF_{low} \end{cases} \tag{13}$$

The new variables are defined in the table below. When they are applied to the individual engines, a subscript index i is used.

Description	Abbreviation
Fuel index of engine. This is the variable that is directly under the control of the governor.	F_r
The combustion efficiency	η_c
Per unit turbocharger velocity	ω_t
The inverse of the time constant of the turbocharger	κ_1
Air-to-fuel ratio	AF
The minimal air-to-fuel ratio needed to achieve full combustion, the minimal air-to-fuel ratio needed to achieve combustion at all, the normal operating point of the engine at full load (disregarding the throttling of the supply air and exhaust gas recirculation)	AF_{high}, AF_{low}, AF_n
Air flow fraction without the turbocharger	$m_{a,0}$

III. Control architecture

The control architecture for a single distribution bus is illustrated in Figure 2. The power management system sets a bus frequency set point, and it provides a prediction of the load over a time horizon T. The task of governing the engines is separated into torque control and frequency control. The local torque controllers need to have a thorough knowledge of the dynamics of the diesel engines that they control. For example, in [10] the engine torque output as a function of the governor current and engine speed is obtained experimentally, and used to tune a PID controller. As long as a sufficient amount of air is provided by the turbocharger, a torque controller is able to change the torque output of an engine very quickly, on the time scale of a fraction of a single crank shaft revolution. For medium or high speed engines which are usually employed in the diesel-electric power plants, this delay is negligible. The advantage of this approach is that the centralized frequency governor does not need to consider the particulars of each engine except for the limitations imposed by the intake air supply.

An even lower level controller sets variables such as combustion timing and throttle openings based on the fuel index. Algorithms for such controllers are available in the litterature, for example [11], [12], [13].

The task of the centralized frequency governor is to maintain a stable bus frequency, a task which has to be balanced against the wear-and-tear of the engines, emissions, soot, and specific fuel consumption. A precise model for those variables would have to be either very complex or ad hoc parameter matching. A study measuring fuel consumption and emissions during transients in load and speed on two full-sized diesel engines that were originally mounted on heavy equipment machines was conducted in [14]. In this work, it is assumed that for a given power output level the economic and environmental costs are generally minimized by reducing the variations in the power output, and the controller implemented in this paper contents itself with this.

$$\min_{\vec{\tau}} \int_0^T \left[Q\tilde{\omega}_{pu}^2(t) + \left\| \frac{d\vec{\tau}}{dt}(t) \right\|_R^2 \right] dt + P\tilde{\omega}_{pu}^2(T) \quad (14)$$

Subject to

$$\text{For each of the gensets, equations (10)-(13)} \quad (15)$$

$$\text{Equation (7)} \quad (16)$$

$$|\tilde{\omega}_{pu}(t)| \leq \tilde{\omega}_{max} \quad (17)$$

$$\tilde{\omega}(0) = \tilde{\omega}_0 \quad (18)$$

$$\omega_t(0) = \omega_{t,0} \quad (19)$$

Here, $\vec{\tau} = \begin{bmatrix} \tau_1 & \tau_2 & \cdots & \tau_N \end{bmatrix}^T$ is the setpoint vector for the local torque governors , $\tilde{\omega}_{pu}(t)$ is the deviation between the desired and the actual frequency on the bus, $\tilde{\omega}_0$ is this deviation measured at the current time at every iteration of the moving receding controller, and $\omega_{t,0}$ is the estimated turbocharger velocity at the current time at every iteration of the receding horizon controller. The variables Q, R and P are weighing scalars and matrices of appropriate dimensions.

IV. Results

This proposed algorithm was tested in simulation on a bus powered by three generator sets, two of them rated at 1 MW, and the third one rated at 2 MW. The inertia time constants for the three generators were 10, 8 and 9 seconds. The load changes are described in Figure 3. Two configurations of the governor were tested. One configuration had a normal weighing of the engine torque variations relative to the bus frequency in the cost function. The other one, and emergency mode ("eager") configuration, had a low cost on the engine torque variations. In the normal configuration, $Q = 1$, $R = I_{3\times3} \cdot 5 \cdot 10^{-3}$, $P = 20$. In the eager configuration the value for R is set to $I_{3\times3} \cdot 10^{-8}$, with all the other parameters remaining the same. A load preview horizon T of one second was used.

The simulation was implemented in Simulink, and the receding horizon controller was implemented in Acado [16].

When a load is increased by a relatively small amount five seconds into the simulations, the normal governor prepares for the load increase by slightly increasing the torque output, and then adjusts it to the new equilibrium over a period of about three seconds. It does essentially the same thing with a larger load increase; however since the diesel engines are physically unable to produce the power that is consumed by the load, a drop in the frequency is inevitable. The governor then pushes the engines as hard as physically possible until the frequency recovers.

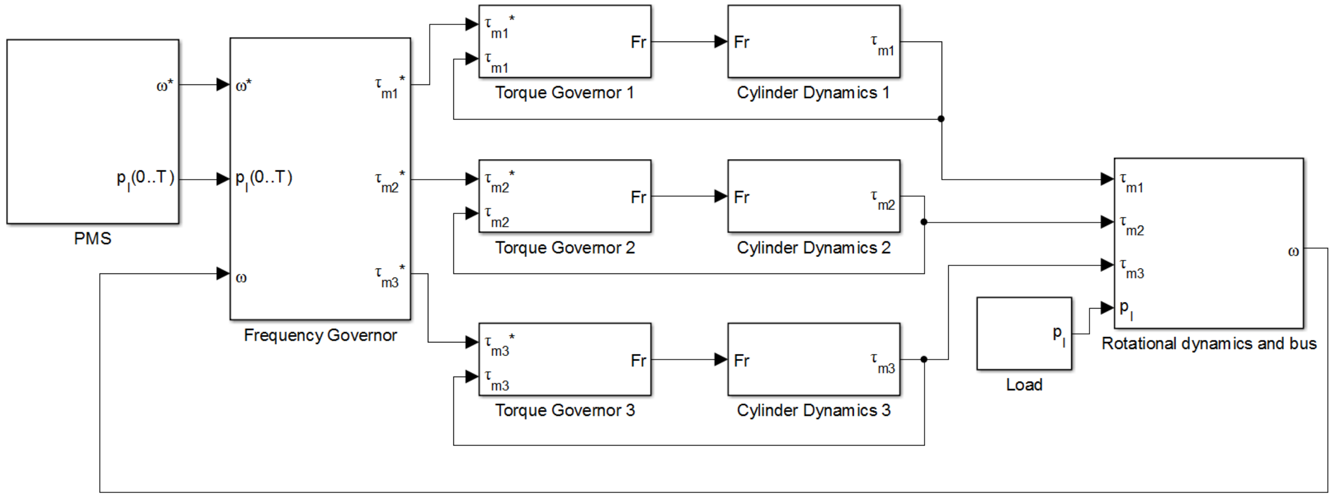

Fig. 2. The proposed control architecture, illustrated on a powerplant with three connected generator sets.

In the eager configuration the governor is more aggressive in stabilizing the frequency, and acts somewhat counterintuitively. Before a load increase, it will first create a dip in the generated torque and thus a very small dip in the bus frequency. Then, just before the inrush load, it will increase the torque to the maximal physically possible value. This maneuver gives a two-fold advantage. Firstly, this means the torque output can be increased more just before the inrush. Without the prior dip, it would result in a larger overfrequency. Secondly, since the preparation for the inrush load involves an initial increase in the frequency above the setpoint, and this cost is proportional to the square of the deviation, this cost can be minimized by minimizing the time spent with a large value of the deviation. The cost of this small dip in frequency in terms of the time integral of the square of the deviation is negligible, while it allows for faster transition of the period with a large overfrequency.

V. Conclusions

The proposed governor is able to increase the power output quite substantially before the load increase hits the bus, both in the normal and in the eager configuration. By doing this it is increasing the energy stored in the rotating machinery and giving the turbocharger a head start to compress more air. The eager configuration provides a larger safety margin to blackout due to underfrequency, at the cost of wearing down the engine. It is therefore best suited for emergency situations. Although only variations in consumer load on the bus were tested, the effect from a sudden disconnection of a generator is expected to be essentially identical.

VI. Future work

The assumptions made in this paper about the diesel engine dynamics and emission model need verification. Experiments on a high fidelity simulator with a physical diesel engine in the simulation loop are currently being conducted by the research group.

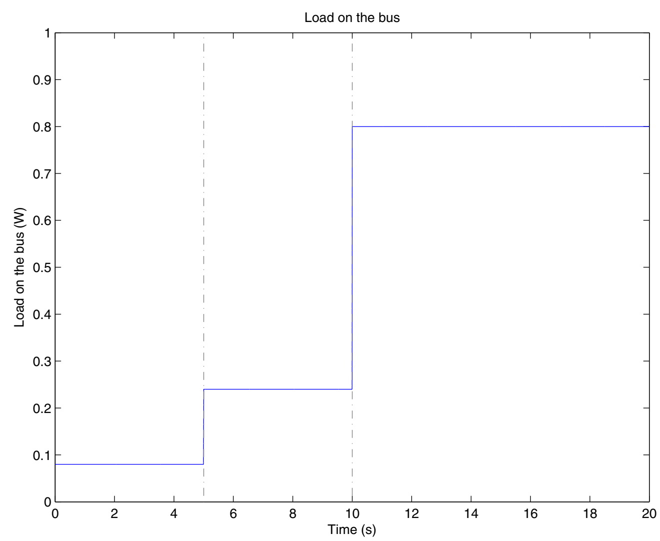

Fig. 3. Load on the bus

VII. Acknowledgments

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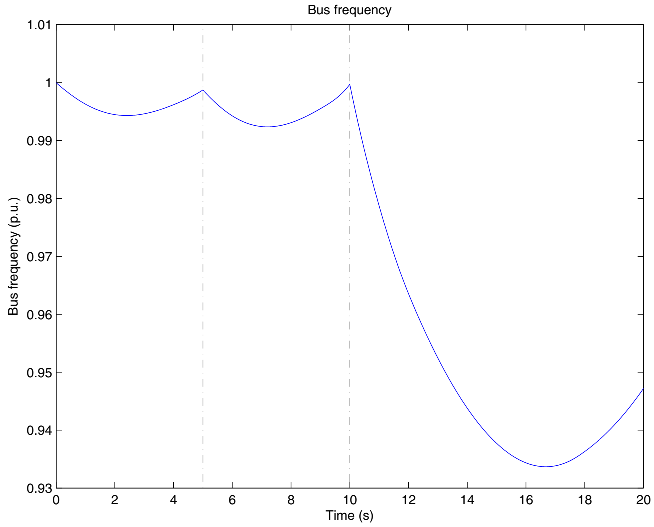

Fig. 4. Bus frequency, normal configuration.

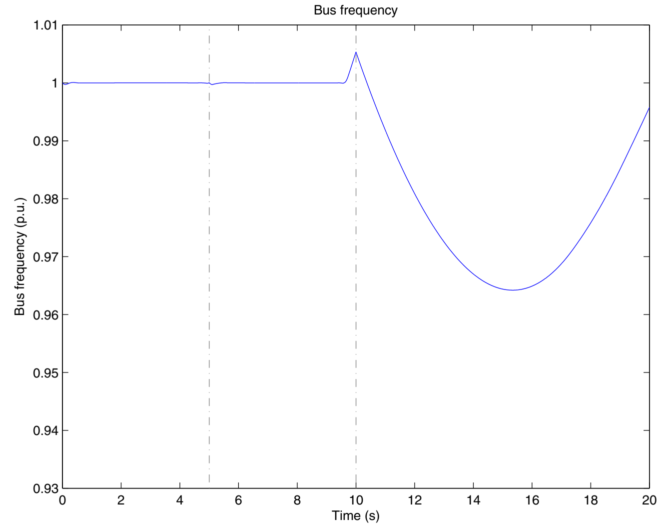

Fig. 6. Bus frequency, eager configuration.

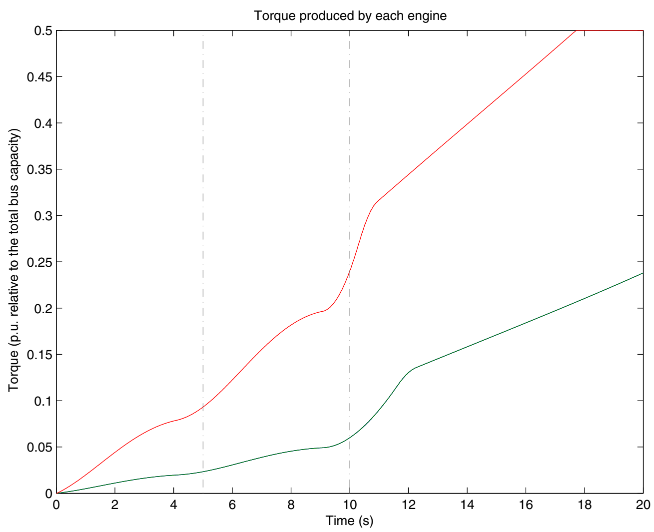

Fig. 5. Torque output of the engines, normal configuration. The two of the smaller engines follow an identical trajectory, displayed in green.

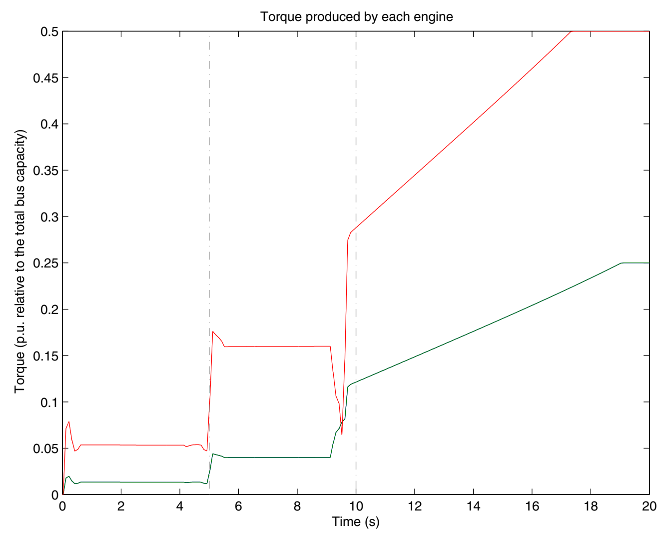

Fig. 7. Torque output of the engines, eager configuration. The two of the smaller engines follow an identical trajectory, displayed in green.

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