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Feedback Control during Mode Transition for a Marine Dual Fuel Engine ⋆

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Abstract: Mode transition is an important control challenge for dual fuel engines, particularly for marine applications where fuel quality and composition may vary over a wide range. Feedback control is critical for dealing with fuel uncertainties and assuring robust performance. In this paper, a model-based approach is pursued for dual fuel engine mode switch control. A mean-value control-oriented model for a marine dual fuel engine is constructed in MATLAB^{TM}/ Simulink^{TM}environment. This model is used to emulate the mode transition process for shipboard generator set applications. Three different control architectures are examined for feedback control based on engine speed regulation during mode transitions. Based on the metric that reflects engine speed tracking error, it is found that a Multiple Input Single Output (MISO) architecture with feedback corrections applied to both gas fuel and diesel is advantageous versus architectures that apply corrections to only one (either diesel or gas) fuel command.

Keywords: Dual fuel engine, Engine modeling, Mode transition, Feedback control, PI controllers, Performance analysis

1. INTRODUCTION

Dual fuel engines are an increasingly popular choice for marine (Aesoy et al. (2011)) and heavy-duty road vehicle applications. These engines can operate in either diesel mode, gas mode or mixed gas and diesel mode. This flexibility is exploited to adapt to fuel availability and fuel cost changes, as well as to improve engine efficiency and to comply with local emission regulations.

Challenges of operating dual fuel engines include satisfying operating limits/constraints, fuel variability, and control during mode transitions between gas mode and diesel mode. As reported by an engine manufacturer (Wärtsilä (2013)), changing the operating mode from gas to diesel can be done within one second at any load while transferring back takes around two minutes and can be performed only at certain load conditions. Performing robust mode transitions is thus considered as an opportunity for model-based control. However, modeling of dual fuel engines has not been sufficiently well-addressed in existing literature. One specific challenge is the combustion characterization during the dynamic fuel substitution when transitioning from diesel mode to gas mode. Compared with steady state characterization at various natural gas substitution rates, which has been experimentally validated for smaller automotive engines (Papagiannakis et al. (2010) and Maxey et al. (2013)), the modeling task for a large marine engine is more involved given its large size and limited available experimental facilities and data. In addition, non-traditional components, such as the gas supply chain from tank to admission valve, need to be carefully modeled considering their importance for the engine dynamic response.

This paper focuses on developing a dual fuel engine model and demonstrating its use for mode transition control and performance evaluation. The rest of the paper is organized as follows. The dual fuel engine modeling, with a Wärtsilä engine (16V34DF engine) being the target for our case study, is described first in Section 2, followed by model validation in Section 3, which is based on available steady state engine data. In Section 4, a mode transition problem is formulated and a comparative assessment of control architectures is conducted. Section 5 presents conclusions and discusses directions for future work.

2. MEAN VALUE ENGINE MODELING

The 16V34DF engine is a 4-stroke medium speed engine with maximum continuous rated (MCR) power of 8MW. Engine specifications are given in Table 1. When running in gas mode, it complies with the latest IMO Tier 3 emission standard (Wärtsilä (2013)). The ability of running on diesel fuel improves the system reliability and provides operational flexibility. This and other dual fuel engines are used in LNG carriers where boil-off natural gas is available onboard. Fig. 1 represents a schematics of the target engine with the following modifications:

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Table 1. Engine specifications.

Engine type	Wärtsilä 16V34DF	
Numbers of cylinders	16	
Bore	340	mm
Stroke	400	mm
Brake power (MCR)	8000	kW
Engine speed (MCR)	750	rpm

(1) The real engine is equipped with 2 turbochargers, one for each cylinder bank. In our model, these two smaller turbochargers are replaced by a larger one, assuming that the larger turbocharger can achieve the same air supply requirement. The reason for this approach will be discussed in Section 2.1.
(2) Wastegate is assumed to be closed during simulated mode transitions and is not included in the engine schematics in Fig. 1.

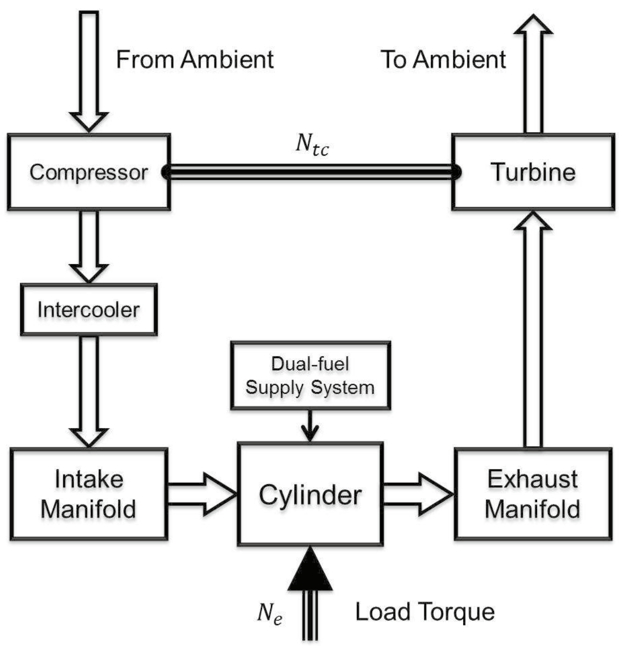

Fig. 1. Marine dual fuel engine system schematics.

The descriptions of our mean value engine model will be separated into two parts. The first part will describe the sub-models for the intake/exhaust manifold, compressor, turbine, intercooler and rotational dynamics of turbocharger and crankshaft. These components are modeled by incorporating the modeling principles for turbocharged internal combustion engines from the literature. The second part will focus on the modeling of cylinder outputs and dual fuel supply system which addresses the unique features of the dual fuel engine.

2.1 Modeling of Conventional Components

The pressure, density and temperature dynamics in the intake and exhaust manifolds are represented by the following equations:

$$\dot{p}_1 = \frac{\gamma R}{V_1}(W_{c1}T_{ic1} - W_{1e}T_1), \tag{1}$$

$$\dot{\rho}_1 = \frac{1}{V_1}(W_{c1} - W_{1e}), \tag{2}$$

$$T_1 = \frac{p_1}{R\rho_1}, \tag{3}$$

$$\dot{p}_2 = \frac{\gamma R}{V_2}(W_{e2}T_{e2} - W_{2t}T_2), \tag{4}$$

$$\dot{\rho}_2 = \frac{1}{V_2}(W_{e2} - W_{2t}), \tag{5}$$

$$T_2 = \frac{p_2}{R\rho_2}, \tag{6}$$

where (p_1, V_1, ρ_1, T_1), (p_2, V_2, ρ_2, T_2) are the pressure, volume, density and temperature in the intake and exhaust manifolds, respectively. W_{c1}, W_{1e}, W_{e2}, W_{2t} [1] are the gas mass flow rates through the compressor, to the engine cylinders, into the exhaust manifold and through the turbine, respectively. T_{ic1}, T_{e2} are the temperatures of the gas flowing into the intake manifold and into the exhaust manifold, respectively. γ is the ratio of specific heats and R is the specific gas constant of air.

The compressor and turbine mass flows, outlet temperatures and isentropic efficiencies can be expressed by the following equations, see Sun et al. (2005):

$$W_{c1} = \phi_c\left(\frac{p_1}{p_{amb}}, N_{tc}, T_{amb}\right), \tag{7}$$

$$T_{cic} = T_{amb}\left[1 + \frac{1}{\eta_c}\left(\left(\frac{p_1}{p_{amb}}\right)^{\frac{\gamma\ 1}{\gamma}} - 1\right)\right], \tag{8}$$

$$\eta_c = \phi_{\eta_c}\left(\frac{p_1}{p_{amb}}, N_{tc}, T_{amb}\right), \tag{9}$$

$$W_{2t} = \frac{p_2}{\sqrt{T_2}}\phi_t\left(\frac{p_{amb}}{p_2}, \frac{N_{tc}}{\sqrt{T_2}}\right), \tag{10}$$

$$T_{2x} = T_2\left[1 - \eta_t\left[1 - \left(\frac{p_{amb}}{p_2}\right)^{\frac{\gamma\ 1}{\gamma}}\right]\right], \tag{11}$$

$$\eta_t = \phi_{\eta_t}\left(\frac{p_{amb}}{p_2}, \frac{N_{tc}}{\sqrt{T_2}}\right). \tag{12}$$

In these equations, N_{tc} is the turbocharger shaft speed. T_{cic}, T_{2x} are temperatures at the outlets of compressor and turbine, respectively. p_{amb}, T_{amb} are constant ambient pressure and temperature. ϕ_c, ϕ_{η_c}, ϕ_t, ϕ_{η_t} represent the regression models for compressor mass flow (W_{c1}), compressor isentropic efficiency (η_c), turbine mass flow (W_{2t}) and turbine isentropic efficiency (η_t). For a specific turbine and compressor combination, the maps derived from experimental data are fitted with analytic functions ϕ_c, ϕ_{η_c}, ϕ_t, ϕ_{η_t}. Detailed information on regression techniques is found in Moraal and Kolmanovsky (1999). In this work, turbine and compressor maps from Karlsen (2012) are used, which are representative of a turbocharger in a 2-stroke low speed diesel engine application. Since this engine has similar output power as 16V34DF engine, we adopted the maps in Karlsen (2012) and used them for our model.

[1] The subscripts for the flow indicate the upstream and downstream locations where: 1 stands for intake, 2 for exhaust, c for compressor, t for turbine and e for cylinder. For example, W_{c1} is the flow from the compressor to intake, while W_{e2} refers to the flow from the engine cylinder to the exhaust.