Modeling transient control of a turbogenerator on a drive cycle

PLEASE CITE THE PUBLISHED VERSION

https://doi.org/10.4271/2022-01-0415

PUBLISHER

SAE International

VERSION

AM (Accepted Manuscript)

PUBLISHER STATEMENT

This paper was accepted for publication in the journal SAE Technical Papers and the definitive published version is available at https://doi.org/10.4271/2022-01-0415.

2022-01-0415

Modeling Transient Control of a Turbogenerator on a Drive Cycle

Author, co-author (Do NOT enter this information. It will be pulled from participant tab in MyTechZone)

Affiliation (Do NOT enter this information. It will be pulled from participant tab in MyTechZone)

Abstract

GTDI engines are becoming more efficient, whether individually or part of a HEV (Hybrid Electric Vehicle) powertrain. For the latter, this efficiency manifests itself as increase in zero emissions vehicle mileage. An ideal device for energy recovery is a turbogenerator (TG), and, when placed downstream the conventional turbine, it has minimal impact on catalyst light-off and can be used as a bolt-on aftermarket device. A Ricardo WAVE model of a representative GTDI engine was adapted to include a TG (Turbogenerator) and TBV (Turbine Bypass Valve) with the TG in a mechanical turbocompounding configuration, calibrated using steady state mapping data. This was integrated into a co-simulation environment with a SISO (Single-Input, Single-Output) dynamic controller developed in SIMULINK for the actuator control (with BMEP, manifold air pressure and TG pressure ratio as the controlled variables). Transient verification with WAVE-RT was conducted on WLTP and NEDC drive cycles, estimating dynamic energy recovery and fuel consumption improvement. Hints are given for a more advanced MIMO (Multiple-Input, Multiple-Output) control system architecture and calibration.

Keywords — GTDI, Turbogenerator, Modeling, SISO/MIMO Control, Co-Simulation, Energy Recovery, HEV.

Introduction

CO_2 targets and more stringent legislation are the reaction to global warming, deterioration in air quality and lack of petroleum resources [1]. In Europe, many OEMs, like Volvo, will produce only BEVs from the timeframe 2025-2030 [2]. US and China are expected to follow. However, full roll-out depends on many factors such as High Voltage battery technology, charging infrastructure as well as customer acceptance.

One common requirement for any type of propulsion system is *efficiency*. In the case of ICE-powered vehicle (PHEV, HEV or with minimal electrification, energy recovery can aid efficiency which can equate to more driven emissions-free kilometers. However, although ICE-equipped vehicles have more and more efficient combustion engines ([3] Nissan claim 50% BTE with ePOWER series), there is still a considerable amount of heat wasted through the exhaust which could be put to better use [4].

Indeed, several turbocharging companies still foresee a need for electrified turbos in HEV powertrains [5].

WHR Systems

The general area of energy recovery on ICEs is called Waste heat Recovery (WHR) and consists of 3 main types, summarized in Figure 1:

- ETC Electric (or Mechanical) Turbocompounding
- ORC Organic Rankine Cycle
- TEG Thermoelectric Generator

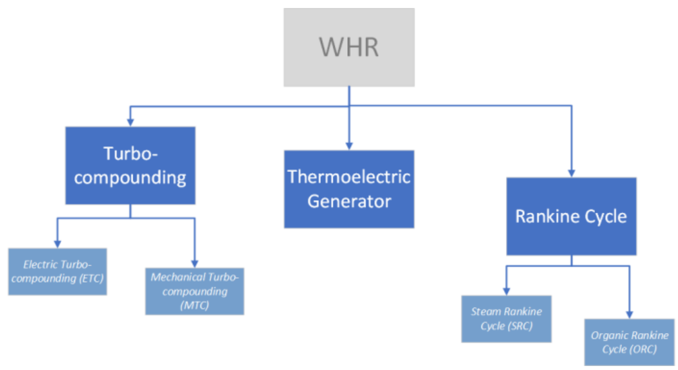

Figure 1: Classification of WHR systems.

Rankine Cycle technologies use waste energy to heat a working fluid (steam or organic) to drive a turbine, while TEGs (Thermoelectric Generators) convert heat directly into electricity, based on the Seebeck effect. [6] reviews the 3 types in detail and concludes that more efficient engine performance (both CO_2 and emissions) will result if OEMs industrialize WHR.

The TG work presented here is a subset of the Turbocompounding classification.

Benchmarking

Turbocompounding solutions presented in [4]; however, there are few details in the literature about controller design or full simulation.

There are 3 benchmark references used, with the result that a downstream TG could produce between 0.5% and 3.9% FC benefit on a highly-loaded US06 drive cycle. This work is summarized in Table 1.

However, a detailed analysis is required. GM [6]: alternator used (not "smart," i.e. charging without engine fueling). No map of power turbine is supplied, plus no details on controller, but TG turbine efficiency assumed 70%

Marelli [8]: strategy B used (alternator enabled at high load only). VVA modifications. Note, although the authors refer to the architecture as ETC, it is actually an electrified turbocharger (or EAT, Electric Assist Turbocharger).

Michigan [9]: assumes the eTG expander has 60% efficiency, use key point analysis. No transient controller was implemented.

The common conclusion is that higher loaded cycles mean more recovered power.

Table 1: Turbogenerator results comparison.

	GM (Y. He et al, 2010)	Marelli (M. de Cesare et al, 2015)	Uni. of Michigan (A. Stefanopoulou et al, 2018)
Engine/Vehicle	1.8L, 125kW	1.4L	Ford Escape 2015
TG Concept	Downstream Turbine	Electric Assist Turbocharger	Downstream Turbine
US06 FC [%]	0.5	N/A	3.9
US06 Power [W]	108.4	N/A	N/A
FTP75 FC [%]	0.1	N/A	0.8
FTP75 Power [W]	18.9	N/A	N/A
NEDC FC [%]	N/A	4.9	N/A
NEDC Power [W]	N/A	N/A	N/A
WLTC FC [%]	N/A	5.3	N/A
WLTC Power [W]	N/A	N/A	N/A

Model-Based Control

In order to demonstrate the attribute impact of the TG technology, the concept of Model-Based Control (or Design) is employed. Generically, development steps include: modeling a plant, designing, analyzing and synthesizing a controller for the plant, simulating the plant and controller together, then finally integrating all these phases by implementing the controller.

This work focusses only on co-simulation, developing the controller in SIMULINK and the plant in Ricardo WAVE [10]. However, the implementation of the system on a real vehicle could be a follow-on project for the work developed here.

[11] is a good overall reference for Model-Based Control (Engine, Transmission, Vehicle), whilst [12] refers to a practical application of the technique in industry.

Overview

This paper presents full transient control of TG device (as previously defined in [4]). The TG system is built up in a 1D modeling environment, then a real-time controller built around it in Ricardo WAVE in a co-simulation environment, after extensive model calibration is performed.

Some drive cycle results are presented and discussed with suggestions for further improvement.

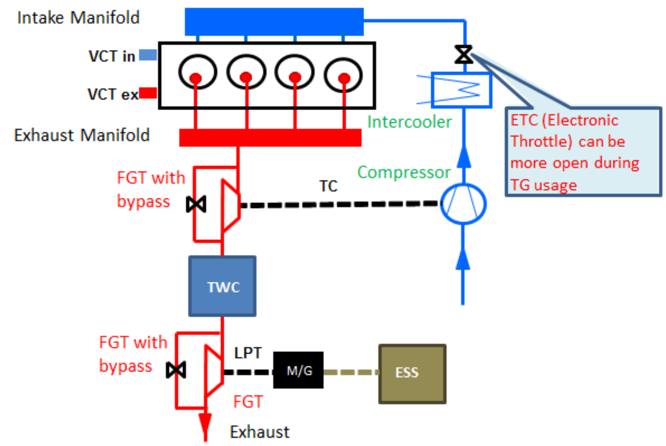

Figure 2: Schematic diagram of the turbogenerator system.

The schematic of the TG system is shown in Figure 2. Due to the peak efficiency occurring at about 1.1-1.15 for higher LPT speeds, the TBV controller was set to control to an LPT PR of 1.15. The generic electrified GTDI model schematic shows the TG connected to a motor/generator (M/G) and an optional Energy Storage System (ESS). The turbochargers are of type FGT (Fixed Geometry) with a bypass valve.

In order to calculate the recovered thermodynamic power harnessed by the TG, equation (1) is used:

$$P_{TG} = c_p \dot{m}_{exh} T_{CAT} \eta_{TG} \left(1 - (P_{dn}/P_{up})^{\frac{\gamma-1}{\gamma}}\right) \quad (1)$$

with the following symbols:

P_{TG} : Thermodynamic power/[W]

c_p : Specific heat capacity at constant pressure/[J/kgK]

$\dot{m}_{exh}$: Exhaust flow/[kg/s]

T_{CAT} : Catalyst outlet temperature/[K]

η_{TG} : Total-to-static efficiency/[0-1]

γ : Ratio of specific heats/[-]

P_{dn}/P_{up} : Downstream/upstream pressure ratio/[-]

1-D Modeling

The 1-D simulation environment was set up in Ricardo WAVE and included the LPT *downstream* the catalyst with a continuously variable bypass. A summary of keypoint analysis is included in [4]. There were 2 controls loops: throttle controlling the BMEP and the WG controlling the intake manifold pressure. The engine has VVT (Variable Valve Timing) and the exhaust valve phasing was used to control the residues to a setpoint, but there was no additional intake/exhaust valve optimisation. The LPT was connected in a mechanical compounding configuration, meaning the recovered thermodynamic power was added directly to the engine with a mechanical efficiency of 90%, with a gearing of 32, and not to an external electrical device (electrical turbocompounding). Due to the peak efficiency occurring at about 1.1-1.15 for higher LPT speeds, the TBV controller was set to control to a LPT PR of 1.15. Knock impact was additionally considered.

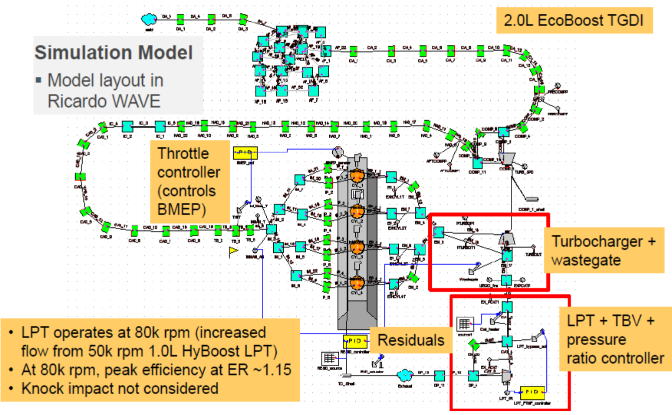

Figure 3: 1D Ricardo WAVE model of 2.0L GTDI engine.

In order to validate the model versus real-world engine mapping data, several parameters were matched, catalyst outlet temperature, BSFC, as well as turbine and compressor conditions. The residuals did not exceed 25%.

Model Calibration

The steady state mapping was used to demonstrate model accuracy.

In order to set up the virtual test rig to be used for system identification, further steady state calibration was required, on the following grid, which is sufficient for both NEDC and WLTP (48 points):

Engine Speed: 1000, 1500, 2000, 2500, 3000, 3500 rpm

Brake Torque: 1, 2, 4, 6, 8, 10, 12, 14 bar BMEP

The calibration tuning handles were residuals (exhaust valve timing), throttle opening, wastegate opening, turbine efficiency and massflow scalings. There are additionally engine block heat flow parameters, for example, which are held constant. The LPT WG was set to fully open, meaning there is no TG power.

Figure 4 illustrates the BSFC and intake manifold pressures for 3000, 2500, 2000 and 1500rpm, with acceptance target maximum 5% error.

Co-Simulation

To set up the RT-WAVE model architecture, the 1D WAVE model from Figure 3 must be modified to include a signal I/O interface, then it is compiled, making available the interface in the SIMULINK modeling environment. The simulation sample time can be modified.

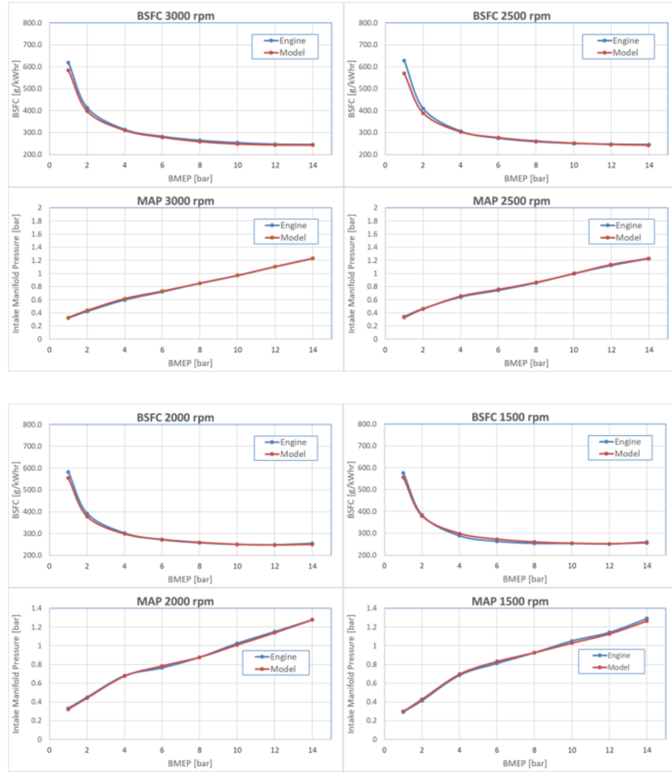

Figure 4: WAVE Model performance (BSFC and intake manifold pressure).

When the steady state calibration is complete, and parameters such as boost pressure setpoint or IVP/EVP intake/exhaust valve phasing are included as speed/BMEP lookup tables, the virtual test Ricardo WAVE model in Figure 5 can be used to perform the drive cycle simulations. Further, system identification, can be performed in order to create fast-running 3x3 sub-models in SIMULINK (as FRMs, Fast-Running Models) which can be used for optimization, for example.

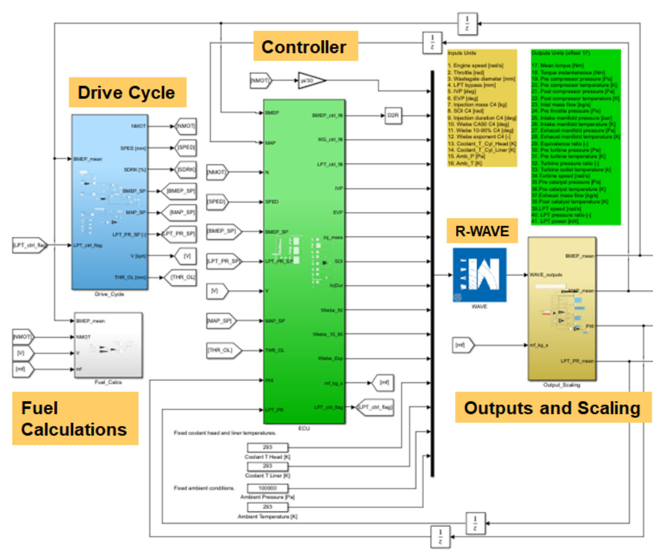

Figure 5: Co-simulation WAVE-RT model.

The controller interface, fuel calculations and drive cycle subsystems are separate blocks, as shown.

An example of a WAVE-RT co-simulation model showing the interfaces is shown in Figure 5. Note, it includes cylinder-by-cylinder inputs for fueling, start of injection, etc.. Outputs, like torque, can be mean-valued or 1D, and further calculations can be made, e.g. corrected flow or speed.

Controller Development

A SISO (Single-Input, Single-Output) controller is developed in the first step. The generic controller is shown in Figure 6 which is used for both BMEP, MAP (Manifold Air Pressure) and LPT pressure ratio control. It has 4 main parts: **feedforward** which has the open-loop settings, as defined by steady state mapping, **closed-loop** which calculates the actuator trim based on transient conditions with PI gains stored as functions of engine speed and BMEP, **selection** flag which decides on closed or open loop operation and **clipping/scaling** which convert the final actuator setpoint to an opening in [deg] from [mm]. Note, the PID block has no derivative/D action and its authority is clipped to meaningful values.

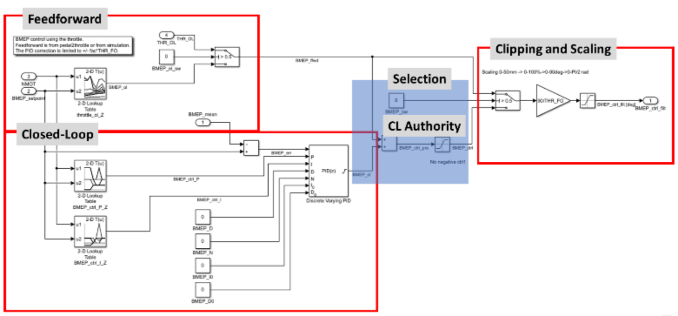

Figure 6: Closed-Loop controller structure.

Simulation Results

Further validation of the model and calibration involved system simulations for NEDC part 2, namely the extra-urban final part of the drive cycle from 800-1180s which is 6.955km in length. This is chosen since the city cycle parts 0-800s are low load meaning there is no possibility for fuel-efficient energy recovery.

For reference, the reported EUDC fuel consumption is 6.4L/100km, but the benchmarked value is 7.74L/100km, hence there is already a wide discrepancy, showing that a pre-conditioning cycle for homologation is important.

A summary of the NEDC simulations is shown in Table 2. It is a good test of both the controller design and calibration. 0.2% fuel consumption benefit can be achieved for LPT fully closed (maximum energy recovery) with an average of 280W recovered power on the cycle, or alternatively, 106.4kJ. Further, Figure 7 to Figure 10 show details of the transient operation for different TG controlled states.

It should be observed that the stated improvements are small on the low-loaded drive cycle. The intent is not to produce a production-level controller calibration, but a performance indication or trend.

Table 2: NEDC part 2 simulation results.

Description	Fuel Consumption [L/100km]/Benefit [%]	Thermodynamic Power [kW]
Baseline	7.25/0.0	0.003 (w/ DFSO recuperation)
LPT Open-Loop Control	7.25/0.03	0.023
LPT Closed-Loop Control	7.25/0	0.060
LPT Fully Closed	7.24/0.212	0.280

Simulation Analysis - Controller

After significant controller calibration, 4 cases were tested, namely:

- Baseline (LPT control valve FO, Fully Open)
- LPT valve FC (Fully Closed)
- LPT valve OL only
- LPT under CL control (setpoint is pressure ratio)

For the baseline case there is good BMEP control whilst the intake manifold pressure control has slightly too high MAP, which is due to the WG opening limitation (10mm, slightly open, in order not to over-boost the engine). For the turbogenerator, although the setpoints are calculated the TG device is fully open (apart from DFSO (Deceleration Fuel Shut-Off) events where it is fully closed when no propulsion is required) which gives about 3W average on-cycle.

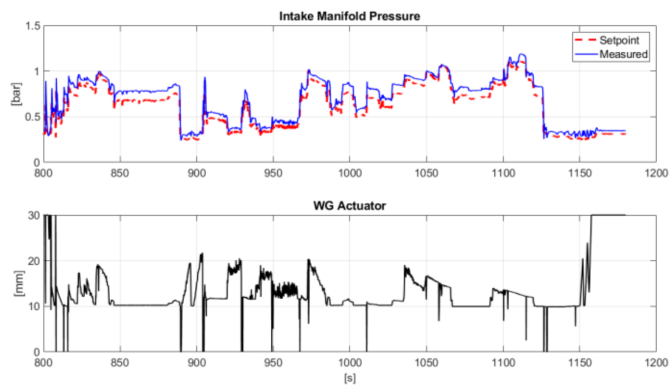

Figure 7: Baseline EUDC: Intake manifold pressure control.

In the TBV FC case (maximum power recovery), the LPT pressure ratio is exceeded all the time and the BMEP setpoint is not reached (for a short period, 1100-1120s) shown in Figure 8.

This case is beneficial for ease of LPT control but for higher loads, a corresponding torque deficiency will reduce driveability, hence some form of CL LPT control is a general requirement.

Figure 9 illustrates the case where the TG actuator runs in open-loop only. There is good BMEP and MAP control but the LPT feedforward control settings are *too open*, resulting in the TG pressure control being *too low* (between 1050s and 1090s).

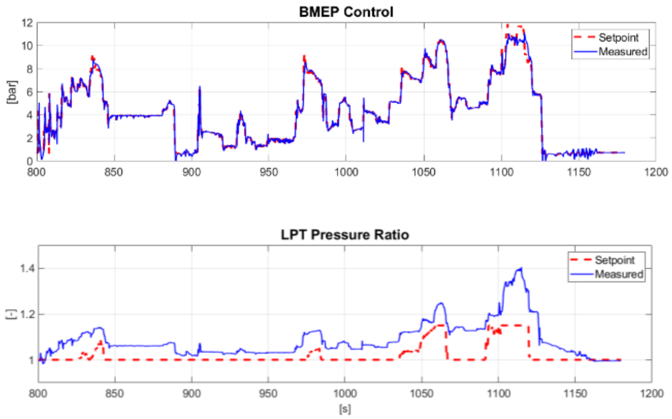

Figure 8: Simulation case with LPT FC. Top BMEP control, bottom significant over-pressure.

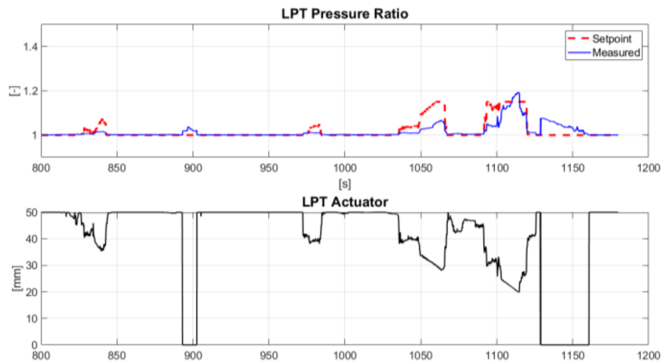

Figure 9: LPT TBV Open-Loop EUDC: turbogenerator control.

Finally, full closed-loop control LPT TBV results are plotted in Figure 10. There is good BMEP, MAP and TG control, whose actuator never goes below about 10mm opening in normal control mode (50mm is fully open).

Note, fully closing the LPT TBV on zero propulsion torque demand is clearly shown between 1130-1155s where there is PR spike.

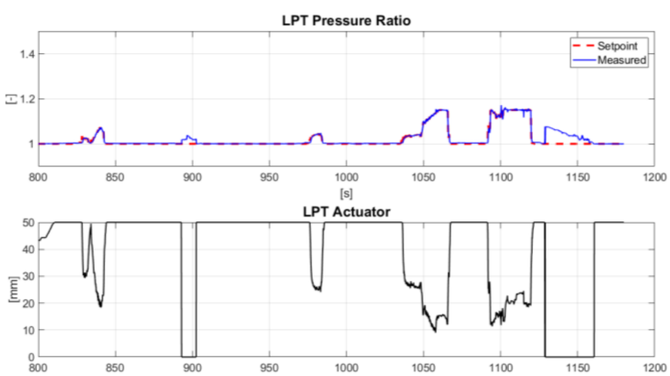

Figure 10: LPT TBV Closed-Loop EUDC: turbogenerator control.

Simulation Analysis - Performance

Further simulation results for the overall system performance can be shown, namely the TG phase plot and the recovered thermodynamic power. The TG speed limit was 80krpm (from the turbine maps), however, in the extreme TG LPT valve FC case, the speed is slightly exceeded in the WAVE simulation (1110-1115s) so both the 2.1kW in Figure 11 and pressure ratio of 1.4 in Figure 12 are too high.

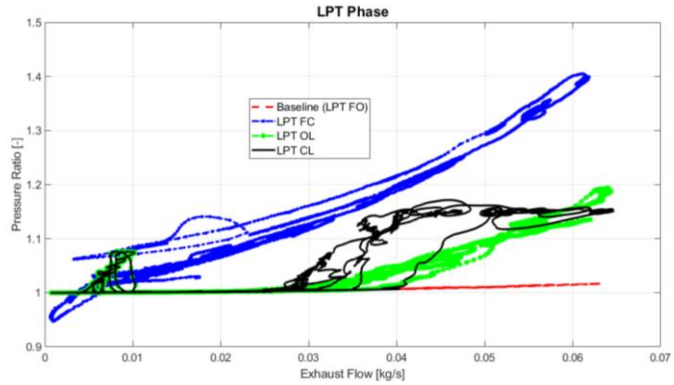

Figure 11: Thermodynamic power recovered on-cycle.

Figure 11 shows the thermodynamic power which, shows the maximum envelope (blue with LPT valve FC), but the CL trace (black) shows a more realistic CL control. It should be noted that the green trace (OL) requires further tuning but generally follows the trend.

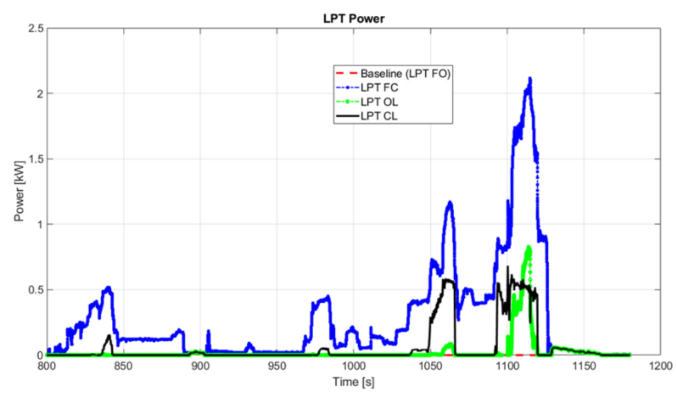

Figure 12: LPT phase diagram on-cycle.

The pseudo phase plot in shown in Figure 12 (complete exhaust flow vs. TG pressure ratio). Both the CL and OL (black and green, respectively) show significant TG activity above an exhaust flow threshold of about 0.03kg/s.

Suggestions for Improvement

Modifications can be made to the controller strategy, as well as the model calibration. The VVT (Variable Valve Timing) calibration was not optimized in this simulation study, but could be included.

More work has been undertaken to create a multivariable control methodology (throttle controlling MAP, WG controlling exhaust pressure, and LPT control valve controlling TG power). This has the advantage of exploiting the interactions in the system, and noting that BMEP is not normally controlled directly in a production controller.

As part of an overall HEV electrification strategy, the TG itself can be directly connected to the vehicle powergrid, specifying an e-machine and power electronics (electric turbocompounding).

Summary/Conclusions

Benchmarking of WHR turbocompounding literature has shown there is a requirement to demonstrate a full transient simulation for a TG concept to fully assess improvement potential. This work has been carried out and summarized in this paper. Building up a SISO controller using MBD was shown, as well as its co-simulation implementation in a 1D engine simulation environment with significant model calibration effort.

SISO control is adequate to demonstrate transient operation but leads to performance limitations in both torque and TG PR setpoint control. This is with non-standard BMEP controller configuration; TG control architecture used the TBV actuator with system output LPT PR.

In summary, maximum benefits are 0.2% FC and 280W over a low-loaded NEDC drive cycle, but further controller refinement (including multivariable architecture) and controller calibration (with VVT optimisation) should be investigated for improved benefits.

References

1. Ehsani, M., Gao, Y., Gay, S., Emadi, A., "Modern Electric, Hybrid Electric, and Fuel Cell Vehicles: Fundamentals, Theory and Design," 2004
2. Volvo Cars Media Relations, "Volvo Cars to focus on range and fast-charging for next generation of fully electric cars," Jun 30, 2021
3. Tsurushima, T., Shiraishi, T., Tsuyuki, M., "Future Internal Combustion Engine Concept Dedicated to NISSAN e-POWER for Sustainable Mobility," 29th Aachen Colloquium Sustainable Mobility 2020
4. Petrovich, S., Ebrahimi, K., Kalantzis, N., and Pezouvanis, A., "MIMO Control of a Turbogenerator for Energy Recovery," SAE Technical Paper 2020-01-0261, 2020, doi:10.4271/2020-01-0261.
5. Garrett E-Turbo Whitepaper, "The Case for Electric Turbo to Optimize Performance, Efficiency in Advanced Hybrid Powertrains," November 20, 2019
6. Noor, A., Puteh, R., Rajoo, S., "Waste Heat Recovery Technologies In Turbocharged Automotive Engine – A Review," Journal of Modern Science and Technology Vol.2 No.1 March 2014. Pp.108-119
7. Wei, W., Zhuge, W., Zhang, Y. and He, Y., "Comparative Study On Electric Turbo-Compounding Systems For Gasoline Engine Exhaust Energy Recovery," Proceedings of ASME Turbo Expo 2010: Power for Land, Sea and Air GT2010, June 14-18, 2010, Glasgow, UK
8. Arsie, I., Cricchio, A., Pianese, C., Ricciardi, V. et al, Modeling Analysis of Waste Heat Recovery via Thermo-Electric Generator and Electric Turbo-Compound for CO2 Reduction in Automotive SI Engines, Energy Procedia 82 (2015) 81 – 88
9. Kiwan, R., Middleton, R., and Stefanopoulou, A., "Thermodynamic and Practical Benefits of Waste Energy Recovery Using an Electric Turbo-Generator Under Different Boosting Methods," SAE Technical Paper 2018-01-0851, 2018, doi:10.4271/2018-01-0851.
10. Ricardo WAVE, http://www.ricardo.com/en-GB/What-we-do/Software/Products/WAVE
11. Eriksson, L., Nielsen, L., "Modeling and Control of Engines and Drivelines."
12. General Motors Developed Two-Mode Hybrid Powertrain With MathWorks Model-Based Design; Cut 24 Months Off Expected Dev Time, 26 October 2009 https://www.greencarcongress.com/2009/10/general-motors-developed-twomode-hybrid-powertrain-with-mathworks-modelbased-design-cut-24-months-of.html

Contact Information

Author: Simon Petrovich
E-Mail: S.Petrovich@lboro.ac.uk

Author: Professor Kambiz Ebrahimi
E-Mail: K.Ebrahimi@lboro.ac.uk

Author: Dr. Byron Mason
E-Mail : B.Mason2@lboro.ac.uk

Author: Dr. Andrew Watson
E-Mail : A.Watson@lboro.ac.uk

Address: Aeronautical and Automotive Engineering, Loughborough University, Stewart Miller Building, LE11 3TU Loughborough, UK.

Acknowledgments

Thank you to Alessandro Romagnoli / Nanyang Technological University / Singapore, Professor Ricardo F Martinez-Botas, Dr Aaron Costall / Imperial College, UK, Aman MI Bin Mamat / Universiti Teknologi MARA / Malaysia for the collaboration.

Definitions/Abbreviations

BMEP: Brake Mean Effective Pressure	**PR:** Pressure Ratio
BSFC: Brake-Specific Fuel Consumption	**RDE:** Real Driving Emissions
BTE: Brake Thermal Efficiency	**RGA:** Relative Gain Array
EBP: Exhaust Back Pressure	**SISO:** Single-Input, Single-Output
FC: Fuel Consumption	**TBV:** Turbine Bypass Valve
GTDI: Gasoline Turbocharged Direct Injection	**TG:** Turbogenerator
LPT: Low-Pressure Turbine	**VVT:** Variable Valve Timing
MBD: Model-Based Design	**WG:** Wastegate
NEDC: New European Drive Cycle	**WHR:** Waste Heat Recovery
MAP: Manifold Air Pressure	**WLTP:** Worldwide Harmonized Test Procedure