Defence Science Journal, Vol. 67, No. 4, July 2017, pp. 375-381, DOI : 10.14429/dsj.67.11451

Simulation of a Diesel Engine with Variable Geometry Turbocharger and Parametric Study of Variable Vane Position on Engine Performance

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ABSTRACT

Modelling of a turbocharger is of interest to the engine designer as the work developed by the turbine can be used to drive a compressor coupled to it. This positively influences charge air density and engine power to weight ratio. Variable geometry turbocharger (VGT) additionally has a controllable nozzle ring which is normally electro-pneumatically actuated. This additional degree of freedom offers efficient matching of the effective turbine area for a wide range of engine mass flow rates. Closing of the nozzle ring (vanes tangential to rotor) result in more turbine work and deliver higher boost pressure but it also increases the back pressure on the engine induced by reduced turbine effective area. This adversely affects the net engine torque as the pumping work required increases. Hence, the optimum vane position for a given engine operating point is to be found through simulations or experimentation. A thermodynamic simulation model of a 2.2l 4 cylinder diesel engine was developed for investigation of different control strategies. Model features map based performance prediction of the VGT. Performance of the engine was simulated for steady state operation and validated with experimentation. The results of the parametric study of VGT's vane position on the engine performance are discussed.

Keywords: Thermodynamic simulation; Variable geometry turbocharger, Diesel engine

NOMENCLATURE

CR DI	Common rail direct injection
VGT	Variable geometry turbocharger
ECU	Engine control unit
EGR	Exhaust gas recirculation
BSFC	Brake specific fuel consumption
BMEP	Brake mean effective pressure
IMEP	Indicated mean effective pressure
PMEP	Pumping mean effective pressure
BTE	Brake thermal efficiency
AFR	Air fuel ratio
IC	Inter cooler
HFM	Hot Film Mass-flow
EPC	Electro-pneumatic Pressure Converter

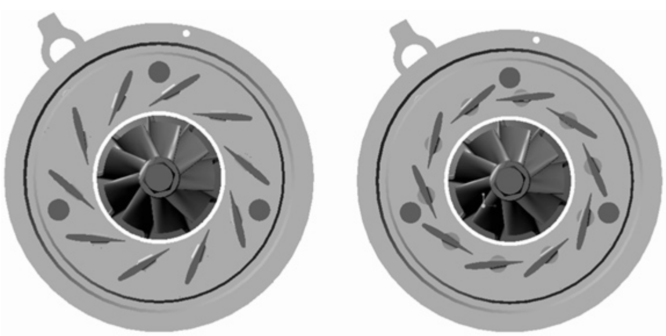
1. INTRODUCTION

Diesel engines will continue to be the prime-movers of future battle tanks owing to their fuel economy, high torque and ease of maintenance. Common rail direct injection (CR DI) technology with electronic engine control is likely to overcome the challenges imposed by extreme weather conditions and other hazards like zero visibility dusty terrains and will replace present mechanical fuel injection systems in the battle tank. Modelling of a turbocharger is of interest to the engine designer as the work developed by the turbine can be used to drive a compressor coupled to it. This positively influences charge air density and engine power to weight ratio. Variable geometry turbocharger (VGT) additionally has a controllable set of nozzles which through a ring is normally electro-pneumatically actuated by the engine control unit (ECU). This additional degree of freedom offers efficient matching of the effective turbine area for a wide range of engine mass flow rates. At the design point, nozzle less turbine is generally 7 per cent less efficient than the turbine with nozzle blades whereas at off design points it performs better than turbine with nozzle blades[1]. Using VGT, this loss in efficiency can be reduced by matching incidence gas flow angle at the turbine rotor entry to the optimum incidence angle thereby reducing incidence loss which is a major loss at off design operation as proven by experimental studies in[2]. Hence, a CR DI engine with electronic control and VGT offers the advantage of closely matched engine-turbocharger coupled operation at all operating points. But multivariable nature of this control problem makes the system complex and makes control strategies and controller design complicated. In the conventional approach, a map of boost pressure as a function of engine speed and throttle

Received : 21 February 2017, Revised : 11 May 2017
Accepted : 18 May 2017, Online published : 03 July 2017

position is generally used for the closed loop control of boost pressure where VGT is electro pneumatically actuated to achieve the reference boost pressure[3]. This map is usually populated by test bed calibration.

Thermodynamic simulation is a very useful tool to arrive at an initial engine design and to define an operational envelope. Commercial simulation software BOOST by M/s AVL is used to simulate the engine performance in this work[4]. A thermodynamic simulation model of a 2.2*l* 4 cylinder diesel engine was developed for investigation of different control strategies. Model features map based performance prediction of the VGT as in[5-7]. Mass flow rate and efficiency maps of different pressure ratios and turbocharger speeds for 5 different vane positions including fully open position (Fig.1. (a)) and minimum flow position (Fig. 1.(b)) were given by the turbocharger supplier. In this approach, turbocharger physics is assumed to be quasi-static with wave-action ignored. The results of simulations and explores simulation as an alternative to engine test bed calibration for arriving at reference boost pressure or VGT position maps discussed.



(a) (b)

Figure 1. VGT open (a) and closed (b) nozzle blade positions.

2. THERMODYNAMIC MODEL

2.1 BOOST Layout

A 1-D thermodynamic model of an engine with specifications as in Table 1, was created in AVL BOOST simulation software with VGT, charge air cooler, air filter connected as shown in Fig. 2. In the model, SB1 and SB2 are used to simulate boundary conditions. CL1, CO1 and TC1

Table 1. Engine specifications

Specifications		
Type		CRDI with VGT
Rated power	[hp]	120
Rated speed	[rpm]	4000
Max torque	[Nm]	280 @(2400-2800rpm)
No of cylinders	[-]	4
Cylinder config.	[-]	I4
Swept volume	[*l*]	2.179
Power density	[hp/l]	55.07
Bore	[mm]	85
Stroke	[mm]	96

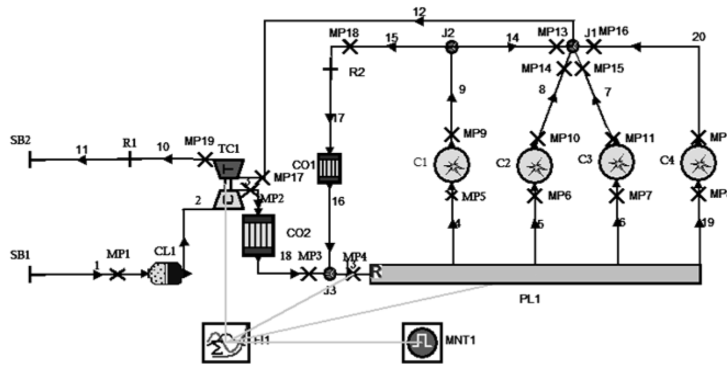


Figure 2. Thermodynamic model in AVL BOOST.

simulate air cleaner, charge cooler and VGT respectively. Intake manifold is modelled as plenum 'PL1' with four intake runners connecting it to the intake ports and cylinder. Cylinders are C1, C2, C3 and C4 with numbering starting from damper end. Exhaust manifold in the engine feature runners with different length and centrally joined. Restriction 'R1' which is added to the tail pipe to simulate any back pressure is set with flow coefficients as 1 since engine setup in test bed does not feature any after-treatment devices or any other devices which could have created back pressure to the engine. A restriction 'R2' is used to control EGR flow rate and is set to 0 as EGR line was disconnected during the experimentation.

2.2 Friction Power

Figure 3 shows friction power obtained from experimental cylinder pressure data with regulated coolant temperature which was used in the model.

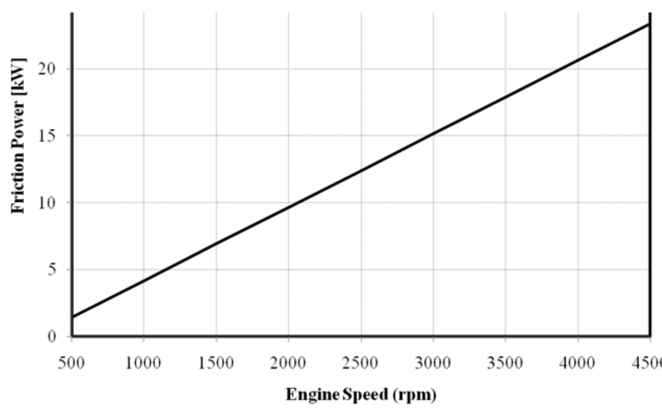


Figure 3. Friction data.

2.3 Combustion

Figure 4 shows combustion data which was provided as normali sed heat release rates calculated from experimental in-cylinder pressure-trace.

2.4 VGT

Performance maps of centrifugal compressor (Fig. 5) and radial inflow turbine (Figs. 6 and 7) were used to model the VGT.

Post processing and correction of the performance parameters with respect to the reference test conditions were carried out prior to use in model as in[8]. Performance maps of the turbine feature characteristics of the turbine for five

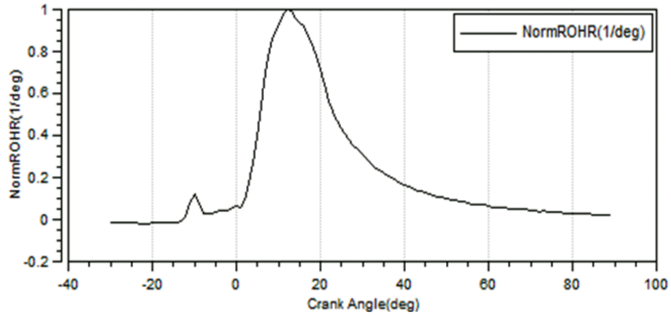


Figure 4. Combustion data as rate of heat release.

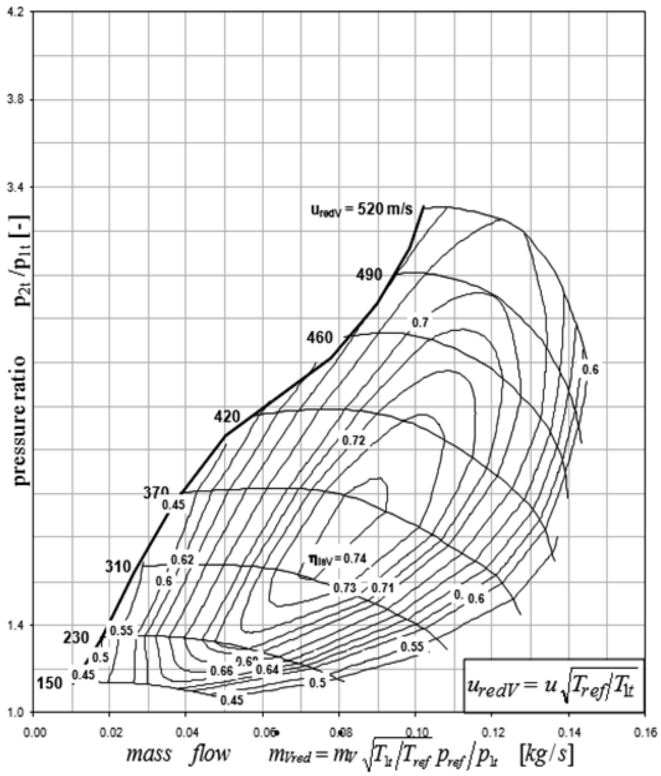


Figure 5. Compressor map.

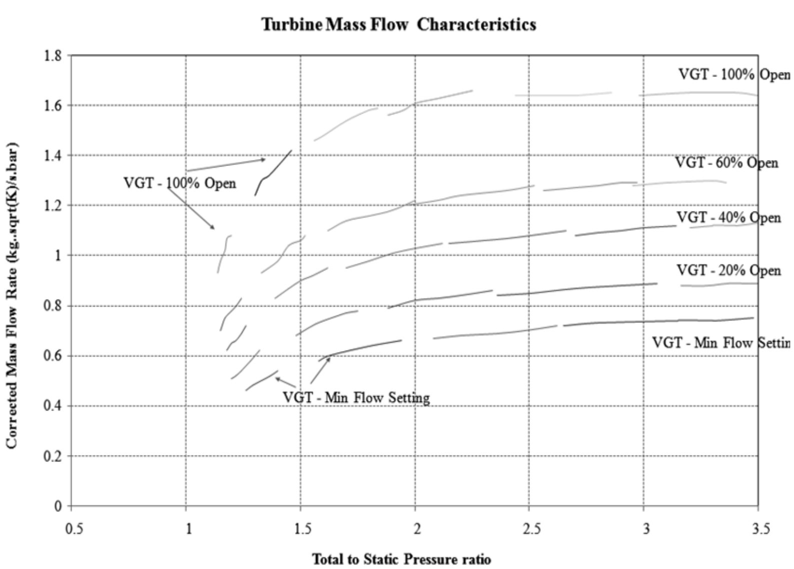


Figure 6. Turbine mass flow characteristics for different VGT positions.

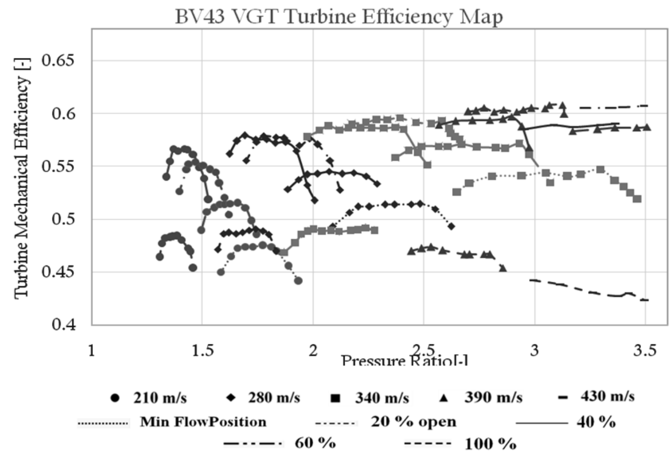


Figure 7. Turbine mechanical efficiency characteristics for different VGT positions.

different VGT positions including fully open as in Fig. 1(a) and fully closed position as in Fig. 1(b). Turbocharger Inertia used is 20 kg. mm^2.

3. INSTRUMENTATION AND EXPERIMENTATION

Engine under consideration was instrumented with thermocouples for temperature measurement and piezo-resistive pressure sensors for pressure measurement as in Fig. 8. Additional parameters acquired were combustion related parameters using piezo-electric pressure transducer, turbocharger rotational speed using an eddy current type sensor and VGT position. Air mass flow rate was measured using an HFM sensor. Experimentation facility with the test engine featuring CRDI and VGT is setup as shown in Fig. 9.

4. VALIDATION OF THE MODEL

4.1 Steady State Performance Parameters

A steady state engine operating point with 1800 rpm, 222 Nm brake torque was selected for validation of the model. Major performance parameters are compared in Table 2. Results indicate that prediction of performance by simulation is well within 5 per cent of measured data from experimentation.

4.2 Pressure Trace Comparison

Pressure trace comparisons also correlate well as shown in Fig. 10 with simulated peak firing pressure and crank angle at peak firing pressure values within 5 per cent of measured values.

4.3 Turbine Inlet Pressure Comparison

Only reliable measurement available for verifying intra-cycle pulsations is the pressure trace in the exhaust manifold. Measuring instantaneous pulsating temperature and mass flow are not feasible with conventional instrumentation. Hence, pressure trace in the exhaust manifold measured from the cooled piezo-resistive pressure transducer was compared with the simulated values. Matching these vales, it is evident that simulated turbine inlet pressure pulsations correlates well with the measured values

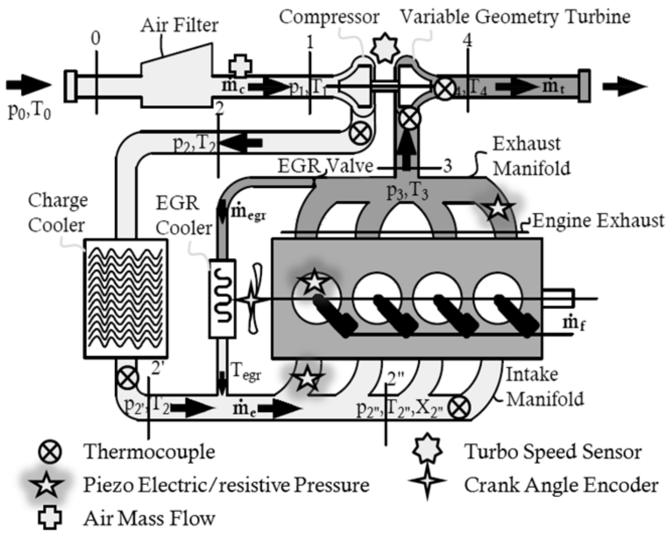


Figure 8. Instrumentation layout.

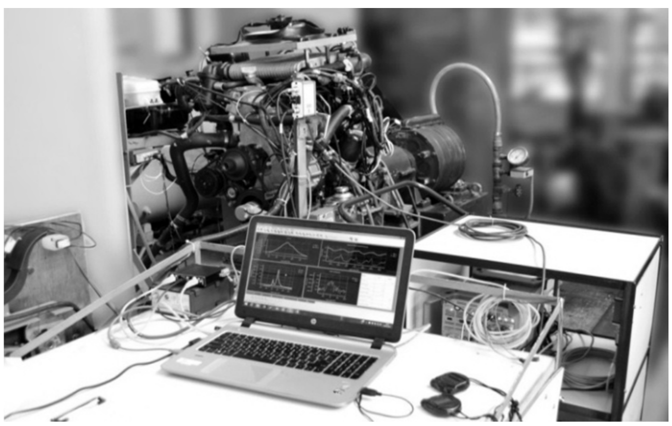


Figure 9. Experimentation setup.

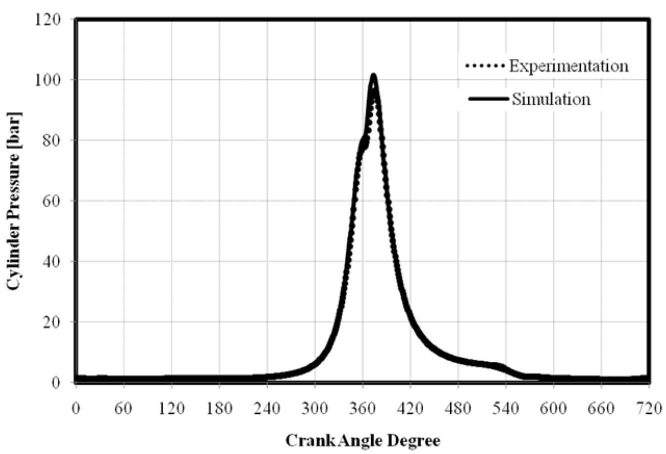


Figure 10. Turbine inlet pressure pulsations comparison.

as shown in Fig. 11. Hence model can be reliably applied to generate boundary conditions for VGT turbine modelling and transient studies.

5. APPLICATION OF THE MODEL: PARAMETRIC STUDY OF VGT VANE POSITION

Validation of the model was carried out with a base VGT position of 60 per cent open. Simulations were carried out

Table 2. Validation of performance parameters

Performance parameter	Experimentation	Simulation	Error
Speed [rpm]	1800	1800	0.00
Torque [Nm]	221.7	221.7	0.01
Power [kW]	41.8	41.8	0.02
BMEP [bar]	12.8	12.8	0.01
BSFC [g/kWhr]	224.0	232.5	-3.81
BTE [per cent]	38.3	36.3	5.25
AFR [-]	17.61	17.56	0.30
Air mass flow rate [g/s]	45.91	47.73	-3.96
Fuel flow rate [g/s]	2.6	2.7	-3.85
Fuelling [mg/stroke]	43.3	45.0	-3.85
Boost pressure [bar]	1.50	1.44	4.00
Turbo speed [rpm]	101473	98235	3.19
Air in [°C]	35	35	0.00
IC in [°C]	88.2	92.5	-4.89
IC out [°C]	64.2	66.0	-2.85
Turbine in [°C]	683.0	685.7	-0.40
Turbine out [°C]	606.0	640.1	-5.63

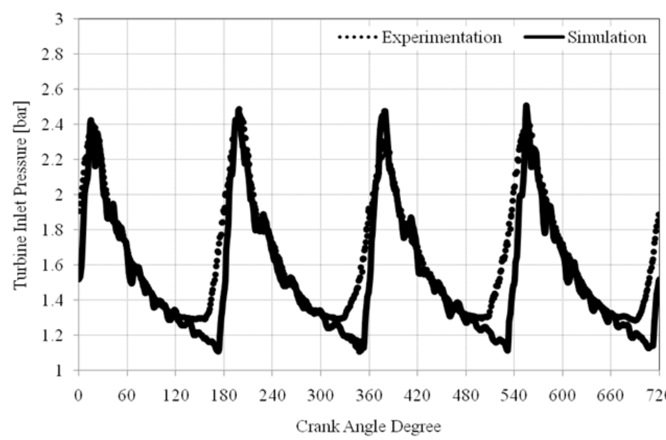


Figure 11. Turbine inlet pressure pulsations comparison.

to study the effect of VGT vane positions with a full sweep from fully open position to fully closed position. VGTP in the graphs indicates VGT position and 0 correspnds to fully closed position and VGTP 1 indicates fully open position (in Figs. 12-18). Other inputs in the model were kept same.

Figure 12 shows that vane position corresponding to 60 per cent open gives a minimum brake specific fuel consumption (BSFC) while extreme vane positions result in an increase in fuel consumption of about 4-5 per cent . Two contradicting physical effects - i.e, closing of vanes producing higher rotational speed (Fig. 13) of the turbocharger thereby resulting in an increase of turbine work (Fig. 14), compressor work (Fig. 15), boost pressure (Fig. 16) and 'indicated mean effective pressure for the high pressure phase of the thermodynamic cycle' (Fig. 17) and also, closing of vane reduces effective turbine flow area (Fig. 18) resulting in higher back pressure (Fig. 19) on

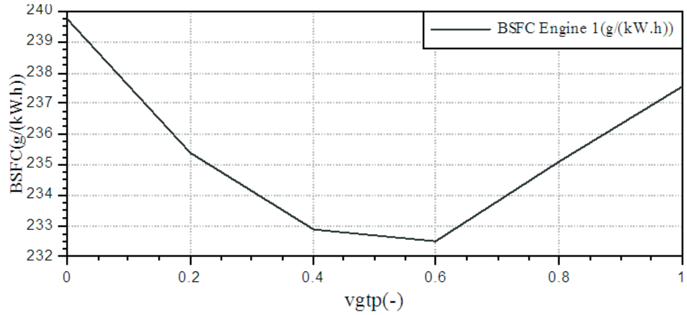


Figure 12. Effect on BSFC.

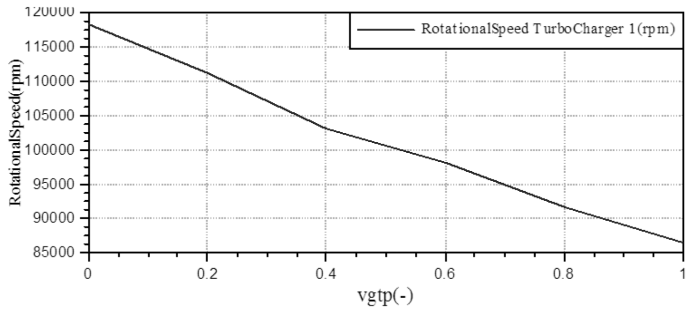


Figure 13. Effect on rotational speed of VGT.

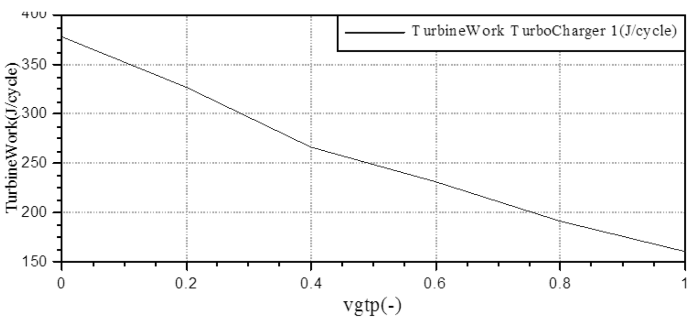


Figure 14. Effect on turbine work.

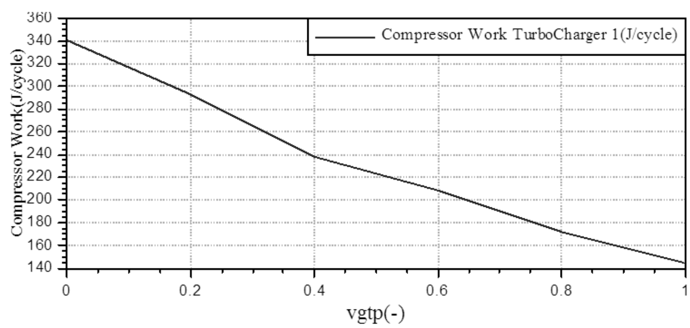


Figure 15. Effect on compressor work

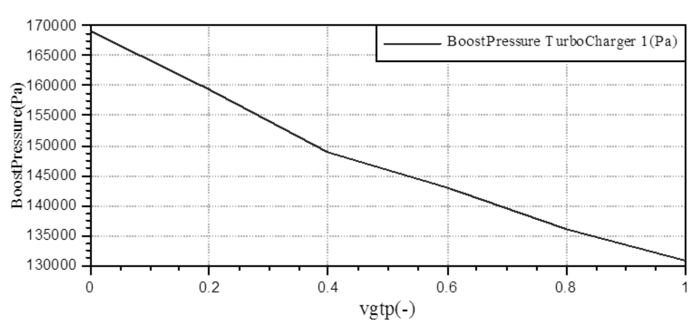


Figure 16 Effect on Boost Pressure.

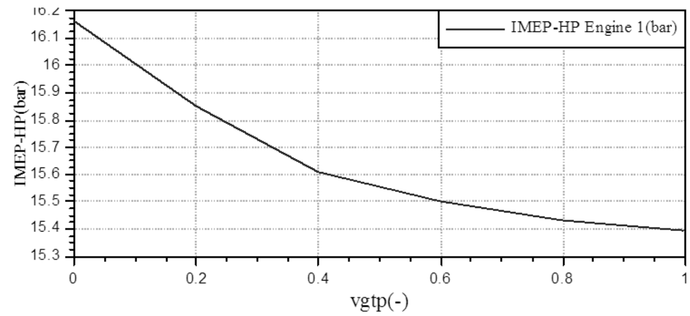


Figure 17. Effect on IMEP-HP.

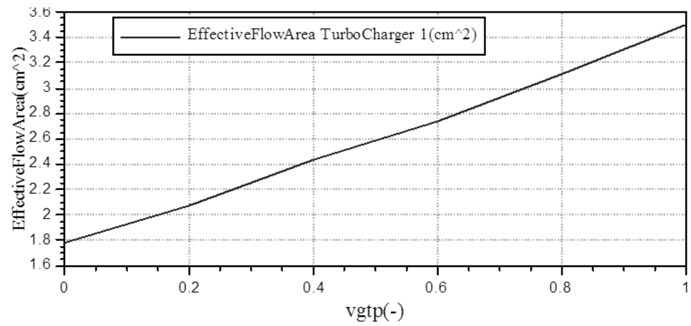


Figure 18. Effect on effective flow area.

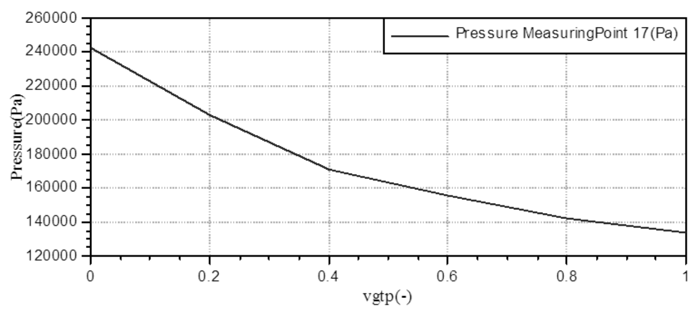


Figure 19. Effect on engine back pressure.

the engine resulting in higher pumping mean effective pressure (Fig. 20) during the gas exchange phase of thermodynamic cycle – explain the BSFC variation from optimum.

Also, turbine isnetropic efficiency (Fig. 21) reduces either side of the optimum value but this effect is not strong enough to reduce the turbine work (Fig. 14) in the region to the left of optimum point, as this is offset by higher rotational speed of the turbocharger.

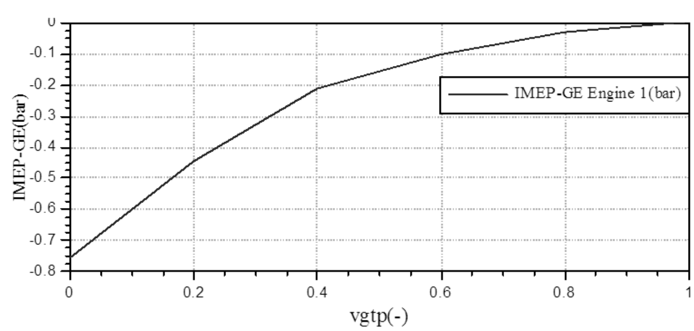


Figure 20. Effect on IMEP –GE.

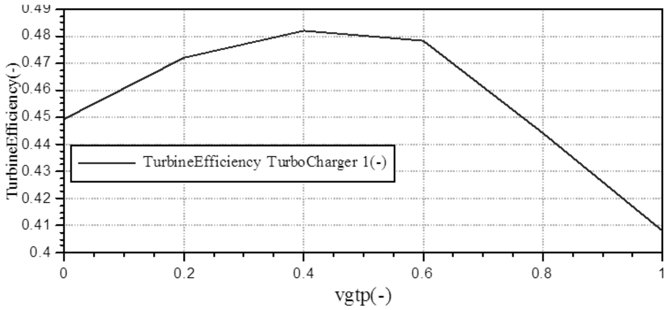


Figure 21. Effect on turbine efficiency.

6. CONCLUSIONS

A thermodynamic model was developed for a 2.2 *l* 4 cylinder engine which was validated with steady state experimentation. Simulation results correlate well within 5 per cent of measured values of performance parameters from experimentation. Model was further applied to study the effect of variable vane position of VGT on engine performance. An optimum position of VGT vane is evident from the results with reference to fuel consumption. Studies shows that closing of the vanes (nozzle ring) result in more turbine work and deliver higher boost pressure but it also increases the back pressure on the engine induced by reduced turbine effective area. This adversely affects the net engine torque as pumping work required increases. Hence, there is a need to find an optimum boost pressure map for the closed loop control of the VGT using ECU and EPC. Thus this study presents simulation as an alternative to engine test bed calibration for arriving at reference boost pressure or VGT position maps.

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ACKNOWLEDGMENTS

Authors wish to thank Dr V. Ramanujachari, Founder Director, RIC, Dr V. Natarajan, Director, RIC and Dr P. Sivakumar, DS and Director, CVRDE for giving permission to publish the work.

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Contribution in the current study, he has carried out the thermodynamic simulations in the study and supported the experimentation and did some of the instrumentation of the engine.

Mr Jensen Samuel J., completed his Masters in Internal Combustion Engineering from College of Engineering Guindy, Anna University Chennai in 2011. Presently pursuing his PhD from Indian Institute of Technology Madras and working on modelling combustion for RT model-based control of Diesel engines during steady and transient operating conditions. Research interests include: IC engine testing, data acquisition, physics-based modelling and control.
Contribution in the current study, he has carried out the instrumentation and experimentation of the engine in the study.

Mr Paul Pramod M., completed his Bachelor's in Mechanical Engineering at Indian Institute of Technology, Madras. Currently pursuing his Master's in Automotive Engineering at RWTH Aachen University, Germany and also working as student worker in FEV Europe GmbH.
Contribution in the current study, he has developed ECU application software for the engine.

Dr A. Ramesh, obtained his PhD (Internal combustion engines) from IIT Madras and post doctoral research work from Ecole Des Mines De Nantes, France. He is an Institute Chair Professor of Mechanical Engineering and also head of the Center for Continuing Education in IIT Madras. He has published over 135 research papers on GDI and HCCI engines, alternative fuels and engine management. He has filed/obtained several patents.
Contribution in the current study, he has arrived at the control logic of the CR DI engine and guided the overall work.

Mr R. Murugesan, is a post graduate engineer and working as Scientist 'F' in Combat Vehicles R&D Establishment, Chennai. He has more than 28 years of experience and worked extensively in design and development of diesel engine and its sub-systems for armoured fighting vehicles and its evaluation. He is currently leading the Centre for Engineering Analysis Division (CEAD) in CVRDE.

Contribution in the current study, he has supported overall work and contributed to generation of the input data for the simulations.

Mr A. Kumarasamy, completed Masters from IITM, Chennai in 1995. Working as Scientist 'G' in CVRDE, Chennai and heading engine division. Developed compact cooling system for combat vehicles. Guided and upgraded power output of an existing engine and evaluated at field conditions. Presently guiding a team for developing engines for 400 – 1500 hp range. His research interests include diesel engine for armoured fighting vehicles (AFVs) for 400 to 1500 hp range, compact cooling system for AFVs, efficient air filtration system for AFVs, advanced & efficient technologies for sub systems of diesel engine and its peripheral systems.

He has initiated this study, reviewed the work and contributed to the integration of the cooling system of the engine under study.