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MECHATRONIC SYSTEM DESIGN - A HYDRAULIC-BASED ENGINE COOLING SYSTEM DESIGN AND REFINEMENT OF A TECHNICAL ELECTIVE MECHATRONICS COURSE

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MECHATRONIC SYSTEM DESIGN - A HYDRAULIC-BASED ENGINE COOLING SYSTEM DESIGN AND REFINEMENT OF A TECHNICAL ELECTIVE MECHATRONICS COURSE

A Thesis
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The Graduate School of
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In Partial Fulfillment
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Master of Science
Mechanical Engineering

By
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Accepted by:
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ABSTRACT

The improvement of consumer products and industrial processes, in terms of functionality and reliability, has recently focused on the integration of sensors and real time controllers with attached actuators into the given physical system. The likelihood of long-term market penetration of smart devices has placed an emphasis on preparing engineering graduates for technology leadership roles in the workforce. This thesis examines mechatronic systems in two manners. First, an intelligent automotive internal combustion engine cooling system is studied for ground vehicles using hydraulic actuators which offer the opportunity for greater versatility and performance. Second, improvements to a technical elective mechatronics course at Clemson University in the Department of Mechanical Engineering have been completed to offer a better educational experience for both undergraduate and graduate students.

Traditional and modern internal combustion engine cooling systems typically use a mechanical wax based thermostat along with a number of mechanical and/or electric actuators to remove the excessive heat of combustion from the engine block. The cooling system's main objective is to maintain the engine temperature within a prescribed range which optimizes engine performance and promotes mechanical longevity. However, the cooling system adds to parasitic engine losses and vehicle weight, so a mechatronic based smart thermal management system has been designed to explore the higher power density and controllability of hydraulic actuators. In this research project, the experimental data has been initially gathered using a 4.6L gasoline engine with a mechanical wax based thermostat valve, engine driven coolant pump, and a hydraulic motor driven radiator fan

with classical feedback control. A series of mathematical models for the hydraulic, electric, and thermal automotive subsystems have been developed to estimate the engine, coolant, and radiator temperatures as well as the overall system performance for various operating conditions.

The experimental test platform features a medium duty eight cylinder internal combustion engine, stand-alone radiator, engine dynamometer, smart cooling system components, high speed data acquisition system, and real-time control algorithm with associated sensors. Specifically, J-type and K-type thermocouples measure the engine block, coolant, and radiator core temperatures at various locations. A multiplexer switches these input signals at predetermined intervals to accommodate the large number of temperature probes. Further, optical sensors measure the engine and radiator fan speeds, and pressure sensors record the hydraulic line pressures. A hydraulic direction control valve was used to adjust the speed of the radiator fan. The experimentally recorded engine data was compared with the numerical simulation results to estimate the engine's thermal behavior for warm up and idle conditions. The findings demonstrated that the proposed experimental model and mathematical models successfully controlled the engine temperature within $\pm 1.5^{\circ}\mathrm{K}$. In the future, the mathematical models can be used for linear quadratic regulator and Lyapunov-based nonlinear controllers after further refinement and the addition of state variables for the engine thermal management system.

To implement such a mechatronic-based cooling system, engineers must have a fundamental understanding of system dynamics, control theory, instrumentation, and system integration concepts. Given the growing industrial demand for graduates with

diverse engineering knowledge, a mechatronic systems course has been designed in the Department of Mechanical Engineering at Clemson University. This mechatronics course, ME 417/617, has been designed to introduce both engineering and personal skills. The students, who would successfully complete the course, will be able to join global work teams designing smart products. The course uses various teaching paradigms such as classroom activities, laboratory experiments, team based design projects, and plant tours to introduce the concepts and offer hands-on experience. As part of a continuous improvement process, the course has been evaluated using assessment methods such as pre- and post-tests, qualitative measures, and advisory panel observations.

Over a four course offering period (2008-2011), the pre- and post-tests reflect improvements in the students' personal growth (7.0%), team building (12.8%), mechanics/engineering (25.4%), and human factor (17%) skills. The qualitative assessment was completed using student feedback regarding the course content. Most of the students reported that they liked the course and its "hands-on" experimental approach. An advisory panel, consisting of industry experts, course instructors, and faculty analyzed the progress of students and evaluated the course materials. The advisory panel's recommendations established the direction for continuous improvements to successfully teach the concepts of mechatronics and better meet the student needs. Going forwards, the mechatronic systems course will serve an important role in preparing graduates for future endeavors.

DEDICATION

I dedicate this thesis to my family.

ACKNOWLEDGMENTS

I want to thank my father, Bhaskarrao Patil, and mother, Ranjana Patil, who always encouraged me to pursue my academic studies and supported me through the difficult times. I wish to thank my sister, Supriya Patil, and my many friends for their extended moral support. I would also like to express my gratitude to my advisor, Dr. John Wagner, who provided me his valuable guidance to complete my graduate studies at Clemson University and offered me an opportunity to start my research career. I am grateful to Dr. Darren Dawson and Dr. Todd Schweisinger for serving on my research advisory committee.

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NOMENCLATURE

Symbol	**Units**	**Description**
a	mm	Solenoid contact length
A	m^2	Area under consideration
A_e	m^2	Surface area of engine
A_{ein}	m^2	Surface area of the engine in contact with the coolant inside the engine block
A_{fluid}	m^2	Surface area of fluid under consideration
$A_{in,hose}$	m^2	Inside area of hose at radiator inlet
$A_{in,tube}$	m^2	Inside area of coolant carrying radiator tubes
$A_{inlet,tube}$	m^2	Inside metal surface area of coolant carrying radiator tubes
$A_{outlet,tube}$	m^2	Outside metal surface area of coolant carrying radiator tubes
A_{rad}	m^2	Radiator front area
A_{rd}	m^2	Radiation area under consideration
$A_{r,fin}$	m^2	Surface area of radiator fins
B_m	Ns/mm	Hydraulic motor damping
b_{val}	Ns/mm	Hydraulic valve spool damping
C	kJ/°K	Thermal capacitance
C_C	kJ/°K	Coolant thermal capacity
C_d	-	Hydraulic motor damping coefficient
C_e	kJ/°K	Engine thermal capacity

C_{im}	mm^5/Ns	Internal motor leakage coefficient
$C_{\max}$	kW/°K	Maximum heat capacity rate
$C_{\min}$	kW/°K	Minimum heat capacity rate
C_{pa}	kJ/Kg°K	Air specific heat
C_{pc}	kJ/Kg°K	Coolant specific heat
$C_{p,fluid}$	kJ/Kg°K	Specific heat of fluid under consideration
$C_{p,node}$	kJ/Kg°K	Thermal capacity of node under consideration
c_r	-	Heat capacity rate ratio
C_r	kJ/°K	Radiator thermal capacity
C_{rin}	kJ/°K	Thermal capacity of coolant at radiator inlet
C_{rout}	kJ/°K	Thermal capacity of coolant at radiator outlet
D_h	m	Hydraulic diameter
D_m	cm^3/rad	Hydraulic motor displacement
e	°K	Temperature tracking error
e_{T_e}	°K	Steady state error between experimental and simulated engine temperature
$e_{T_{rout}}$	°K	Steady state error between experimental and simulated coolant temperature at radiator outlet
F_s	N	Force generated by solenoid coil
F_{ss}	N	Steady state fluid force on the solenoid
F_{tr}	N	Transient fluid force on the solenoid
h_{ein}	W/m^2°K	Convection coefficient of coolant inside engine block

h_{fluid}	W/m^2°K	Convection coefficient of fluid under consideration
h_∞	W/m^2°K	Convection coefficient of ambient air
$\overline{h}_L$	W/m^2°K	Average heat transfer coefficient over flat plate
i	A	Valve input current
J	kg.cm^2	Hydraulic fan or pump and motor inertia
k	W/m°K	Thermal conductivity
k_{al}	W/m°K	Thermal conductivity of aluminum
k_C	W/m°K	Thermal conductivity of engine coolant
k_{cd}	W/m°K	Thermal conductivity of metal under consideration
k_{val}	N/mm	Hydraulic valve spring constant
L	m	Length
L_{coil}	H	Control valve coil internal inductance
L_{cd}	mm	Length of conduction area under consideration
L_d	mm	Damping length
l_g	mm	Solenoid valve reluctance gap
$L_{hose,inlet}$	mm	Length of hose at radiator inlet
$L_{hose,outlet}$	mm	Length of hose at radiator outlet
L_{tube}	mm	Length of coolant carrying radiator tubes
$\dot{m}_{a_{fan}}$	kg/s	Fan air mass flow rate
$\dot{m}_c$	kg/s	Pump coolant mass flow rate
$\dot{m}_{fluid}$	kg/s	Mass flow rate of fluid under consideration

m_{node}	kg	Lumped mass of node under consideration
m_s	kg	Hydraulic valve spool mass
M	-	Auxiliary electric motor driving hydraulic pump
N_t	-	Number of turns in solenoid coil
NTU	-	Number of transfer units
Nu	-	Nusselt number
P_A	MPa	Hydraulic motor supply pressure
P_B	MPa	Hydraulic motor return pressure
P_L	MPa	Hydraulic motor load pressure
Pr	-	Prandtl number
P_s	MPa	Hydraulic supply pressure
P_T	MPa	Tank return hydraulic pressure
P_{sys}	kW	Power consumed by hydraulic motor operated actuators
ΔP	MPa	Pressure difference between hydraulic motor operated actuator inlet and outlet lines
q	kW	Heat transfer rate
Q_{in}	kW	Heat input
Q_L	LPM	Hydraulic motor load flow
$q_{\max}$	kW	Maximum heat transfer rate
Q_o	kW	Uncontrollable radiator heat losses
R	W/°K	Heat transfer resistance
R_{coil}	Ω	Control valve coil internal resistance

Re	-	Reynold’s number
T	Sec.	Time duration for test
T_C	°K	Engine coolant temperature
$T_{c,i}$	°K	Cold fluid inlet temperature
$T_{c,o}$	°K	Cold fluid outlet temperature
$T_{h,i}$	°K	Hot fluid inlet temperature
$T_{h,o}$	°K	Hot fluid outlet temperature
T_{High}	°K	Higher limit of temperature for thermostat model
T_i	°K	Temperature of i^{th} component
T_∞	°K	Ambient environment temperature
T_e	°K	Engine temperature
T_{e_L}	%	Engine load
T_g	N.cm	Hydraulic motor generated torque
T_j	°K	Temperature of j^{th} component
T_L	N.cm	Hydraulic motor load torque
T_{LF}	°K	Temperature at left-front engine cylinder
T_{Low}	°K	Lower limit of temperature for thermostat model
T_{LR}	°K	Temperature at left-rear engine cylinder
T_r	°K	Radiator temperature
T_{RF}	°K	Temperature at right-front engine cylinder
T_{rin}	°K	Coolant temperature at radiator inlet

T_{rout}	°K	Coolant temperature at radiator outlet
T_{RR}	°K	Temperature at right-rear engine cylinder
t_{tube}	mm	Thickness of coolant carrying radiator tubes
U	W/m^2°K	Overall heat transfer coefficient
V	m/s	Velocity
V_s	v	Supply input control voltage
V_a	m/s	Air velocity
V_{speed}	kph	Vehicle speed
V_f	v	Control input voltage applied to valve for radiator fan
V_t	cm^3	Volume of compressed fluid
w	cm^2/cm	Orifice area gradient
x	m	Control valve spool displacement
ε_e	-	Emissivity of engine
ε_{rd}	-	Emissivity of component under consideration for radiation
β	MPa	Bulk modulus of hydraulic fluid
σ	W/ m^2°K^4	Stefan-Boltzmann constant
η_{hm}	-	Hydro-mechanical efficiency
ε	-	Effectiveness of radiator
μ	Kg/sm	Viscosity
μ_s	H/mm	Solenoid armature permeability
μ_o	H/mm	Solenoid armature permeability

θ	rad	Hydraulic fluid jet angle
ρ	kg/m^3	Fluid density
ρ_a	kg/m^3	Air density
ρ_c	kg/m^3	Engine coolant density
ϕ	-	Thermostat valve opening percentage
ω	rad/sec	Actuator speed
ω_e	rad/sec	Engine speed
ω_{fan}	rad/sec	Radiator fan speed
$\dot{\omega}$	rad/sec^2	Actuator acceleration
ψ	°K	$T_{High} - T_{Low}$
υ	m^2/s	Kinematic viscosity
α	m^2/s	Thermal diffusivity

CHAPTER 1

INTRODUCTION

The word ‘mechatronics’ has been created by combining the two phrases ‘mechanical’ and ‘electronics’ as first prepared in the early 1970’s (Comerford, 1994). Over the past four decades, the definition of mechatronics has fundamentally changed to include more technical concepts. Mechatronic systems can now be defined as the integration of mechanical, electrical and electronics, industrial, computer, and controls engineering disciplines, as well as people skills for the design of smart products and processes. Intelligent systems are used throughout everyday society including consumer products, aerospace vehicles, medical devices, healthcare equipment, energy production, and manufacturing processes to name a few. Ordinary products are now designed with digital technology to monitor and control the application for greater performance and reliability. For instance, a hybrid ground vehicle is a mechatronic system which uses an internal combustion engine (ICE) and an electric motor to propel the vehicle. The mechatronic system controls these actuators to improve the fuel economy and reduce the exhaust gas emissions. The contributing disciplines in mechatronics and some of the representative application areas have been shown in Fig. 1.1.

The advantages of a mechatronic system include improved performance, reliability, and energy consumption often leading to better system quality. One such type of mechatronic system is the mobile, or stationary, robot which helps to reduce human error in repetitive factory work tasks including assembling products through “pick and place” operations, welding parts together, measuring dimensions, and transporting various objects from one plant location to another. Today, robotic systems are widely used around the world to produce better quality products at reduced costs as they lower labor needs and offer lower tolerances on product assembly.

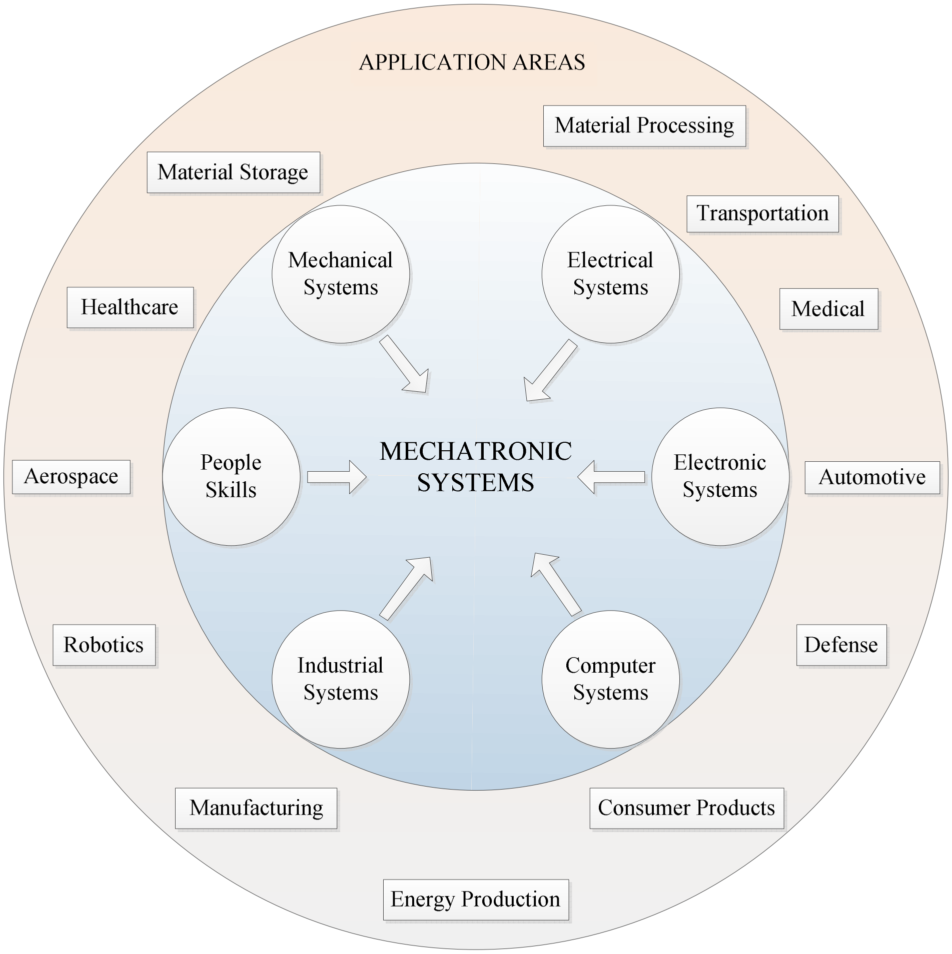

Fig. 1.1: The contributing concepts in mechatronic systems and some representative application areas

To design a mechatronic system, one should understand and have knowledge of various science and engineering concepts, as well as people skills to collaborate on the designs. Each engineering and science discipline has their own core competency, and students typically enrolled in a college program focus in-depth on the related core topics.

At Clemson University, a technical elective mechatronic systems course, ME417/617, has been created which teaches key engineering and mechatronic concepts to design smart products and systems. Along with the classroom activity, a series of laboratory experiments offer students hands-on experience with sensors, actuators, control systems, data acquisition, and electronics. One such mechatronic product, a hydraulic based internal combustion engine thermal management system, has been investigated in the research laboratory to realize advantages over traditional automobile cooling systems which rely only on mechanical elements.

1.1 Hydraulic Based Engine Cooling System

Traditional engine cooling systems use a mechanical water pump, single radiator fan driven by the engine, and a mechanical thermostat to maintain the engine temperature within 10ºC to 20ºC (Melzer *et al.*, 1999). In this configuration, the coolant pump and radiator fan generally operate at speeds proportional to the engine speed regardless of the actual thermal load. The operational premise is that higher engine speeds denote greater heat transfer loads. Further, the cooling system is designed for maximum heat removal at extreme ambient temperature conditions such as desert environments. A significant disadvantage of a conventional cooling system is the overcooling of the engine fluid. For instance, when the engine warms up, it is not necessary to operate the radiator fan since the engine coolant may not have reached its highest desired temperature. By operating the radiator fan somewhat independent of the engine speed, the accessory power consumption can be reduced to increase the vehicle’s overall performance. Next, if the radiator fan excessively chills the coolant by running at a high speed, then the thermostat valve may not be able to open fully (100%) while maintaining desired set point temperature. This condition can also be called as cold start of the engine where fuel efficiency and emissions of the engine are poor (Lehner *et al.*, 2001).

Lastly, the operation of a mechanical thermostat has some inherent disadvantages which may lead to the engine block temperature exhibiting oscillating behavior (the question begs the issue regarding the temperature range of these variations). The transient response of the thermostat is dependent on the component's internal wax properties and the nonlinear valve opening profile. In some respects, the engine temperature fluctuations reflect over-heating and over-cooling in a limited neighborhood of the target temperature. Sharp variations of 10ºC to 13ºC in the engine block temperature can adversely impact the pistons, piston rings, cylinder gaskets, and other structural materials as well as affect the exhaust gas emissions due to the combustion process.

In the past decade, lot of research on automotive engine cooling system has been completed by using servo-motor driven coolant pumps, radiator fans, and thermostats to improve fuel efficiency, thermal efficiency, and exhaust emissions of engines (Cortona *et al.*, 2000, Eberth *et al.*, 2004, Page *et al.*, 2005). These systems have their own advantages and aims to fulfill. But for engines with larger displacement, such as the ones used for heavy duty trucks, a hydraulic based cooling system may offer more advantages than the electric actuators based system. Hydraulic actuators have higher power density values (i.e., power delivery per unit weight of actuator) than the electric actuators. For example, typical standalone pump assembly of hydraulic motor and electric motor can have power density of 3500 W/kg and 500 W/kg respectively (Kluger *et al*, 2007). This particular characteristic of hydraulic actuators has been utilized in developing the hydraulic based engine cooling system. It includes the hydraulic motor operated radiator fan, engine driven coolant pump, and a mechanical wax based thermostat. Some of the advantages of using a hydraulic based engine cooling system are listed below.

- The hydraulic motor operated actuators can be placed anywhere in the engine compartment since hydraulic hoses drive these devices.
- Due to the higher power density, hydraulic actuators can be used for heavy duty engines.
- Instead of using multiple electric motors, a single large hydraulic actuator can be introduced into the engine thermal management system.
- A higher operating life cycle can be realized when compared to electric actuators for harsher environmental conditions.

Some of the disadvantages of using a hydraulic based cooling system include the following items.

- If an actuator fails, then the whole system might stop working due to little or no redundancy.
- Hydraulic fluid drives these actuators, so there is a chance of hydraulic fluid leakage.
- The power source to drive the actuators will be an engine-crankshaft driven hydraulic pump putting the accessory load on the engine.
- An accumulator, or a storage tank, will be required which adds to the weight of the cooling system.

To design such a hydraulic based automotive thermal management system, the engineer should have some familiarity with mechatronic systems. Although this application is but one example, engineers need to recognize the growing trend of mechatronics technology in different industries. To ensure that entry level engineers can apply the relevant concepts and principles to design smart products and processes, educational institutions need to offer mechatronics courses. Hence, an undergraduate/graduate technical elective course on mechatronic systems has been designed at Clemson University which is explained in the next section.

1.2 Mechatronic Systems Course and Assessment

The mechatronic course in the Department of Mechanical Engineering teaches concepts from disciplines such as electrical, electronics, industrial, mechanical engineering, computer science, controls, and robotics. The classroom topics covered include electrical/electronic circuits, actuators, sensors, data acquisition, PLC (programmable logic controller) and robot programming, as well as hydraulics, pneumatics, and thermal systems. As the growth in mechatronics continues, important ideas from human factors and the human-machine interface have gained greater attention during the classroom discussions. Apart from these topics, people skills (e.g., collaborative learning, project management, team building, leadership, business ethics, etc.) are taught that differentiate this course from other engineering classes. These skills are essential for students, especially engineering students, to enter into a competitive industrial work environment. They will be able to join a diverse team for completing different multi-disciplinary tasks and interact with a host of individuals who might have different work-backgrounds.

The three-credit hour course also offers hands-on experiences with various mechatronic systems through laboratory experiments. These mechatronic systems include sensors, actuators, data acquisition systems, robots, and material handling systems. The laboratory features a variety of experiments that are developed by the student teams for their mechatronic design projects. While working on laboratory experiments, students refer to the laboratory manual which has the information about all the experiments and specific procedures to follow. The manual also has thought provoking exercises at the end of each experiment for students to summarize the learning of those particular experiments. Some of the experiments include making of an electronic dice circuit and rotation counter circuit on bread-boards; programming of Allen Bradley PLCs by using RS Logix 500 ladder logic software; programming of a Staubli robotic arm; and controlling pneumatic actuators and material handling conveyors with a vision system. Students gain hands-on skills, understand problem solving methods, and work with industry equipment while completing such laboratory experiments. Further, the laboratory also establishes a sound foundation for students to complete a semester long collaborative team activity of designing mechatronic systems.

The assessment of the mechatronic systems course is performed at the start, middle, and end of the semester to measure the students' knowledge gain through lectures, laboratory experiments, home work assignments, and design project. The assessment tools include pre- and post-questionnaires, industrial advisory panel review and feedback, observations by the instructor and laboratory teaching assistant, and student feedback.

The assessment results are reviewed and considered to revise and improve the classroom as well as the laboratory activities for students.

1.3 Organization of Thesis

The thesis has been organized as follows. Chapter 2 describes a hydraulic based engine cooling system and presents experimental and simulation results. Chapter 3 presents the mechatronic systems course designed for undergraduate/graduate engineering students along with the course assessment results. Chapter 4 concludes the thesis with future research challenges and recommendations. The Matlab/Simulink simulation algorithms and complete experimental results for the hydraulic based engine cooling system are contained in the Appendices.

CHAPTER 2

A HYDRAULIC ACTUATED ENGINE THERMAL MANAGEMENT SYSTEM – EXPERIMENATION AND SIMULATION

A nonlinear mathematical model for a hydraulic-based engine cooling system has been derived using experimental testing which utilizes classical controller to maintain the engine temperature at a set point value. The traditional engine cooling strategies involve engine driven coolant pump and radiator fan, and a mechanical wax based thermostat; or engine driven coolant pump, electric motor driven radiator fan, and a thermostat to control the engine temperature over wide range of values. Instead of these strategies, a system of operating a hydraulic motor driven radiator fan, an engine driven coolant pump and a mechanical thermostat is proposed. The most important advantage in the usage of hydraulic actuators is their higher power density values than mechanical or electric actuators of same size and weight. They are flexible enough to place anywhere around the engine. This strategy allows engine temperature control over a narrow range and under adverse operating conditions. The applications of proposed strategy can be on off-highway engines, heavy duty construction vehicles, stand-alone power generator engines and some of the hybrid vehicles utilizing hydraulically stored energy for traction and/or operation of engine accessories. The numerical results for this proposed strategy are validated by comparing with the experimentally found engine test results. The proposed mathematical model can be used to design advanced engine thermal system management strategy of controlling both, radiator fan and coolant pump, driven by hydraulic motor along with electric valve instead of mechanical wax based thermostat. This advanced strategy can further reduce engine accessory losses.

2.1 Introduction

Traditionally engine manufacturers have used mechanical coolant pumps, and radiator fans as well as thermostat valves to manage the engine temperature within a wide range of values. But this strategy involved lot of mechanical losses, large engine warm up time, loss of engine power to coolant pump and radiator fan when they need not be used, which leads to higher BSFC (Brake Specific Fuel Consumption), and higher tailpipe emissions as suggested by Wambsganss (1999) and Lyu *et al.* (2007). The parasitic losses like energy consumption by coolant pump, radiator fan, heater, and compressor need to be reduced in order to get lower BSFC of an engine operating under different conditions. Considering this fact, most of the engine manufactures now have started using combinations of continuously varying electric coolant pump, radiator fan and thermo-valve to efficiently manage engine thermal system and meet other cooling requirements. Geels *et al.* (2003) have mentioned that by using electrical cooling system it was possible to get 5% reduction in engine fuel consumption, 10% reduction in CO and 20% reduction in HC tailpipe emissions.

Page *et al.* (2005) has shown fuel economy improvements between 5-20% under steady-state operation to investigate confined space condition by avoiding ram air. An advanced thermal management module has been built behind the cab of a military truck. Multiple numbers of electric radial fans, electric pump, mixing tank, diverter valve, electronic control valve, heat exchangers, etc are used to maintain the engine temperature, engine oil temperature and truck cabin temperature. They have discussed about the different engine accessory power consumptions and possible accessory power consumptions that could be saved by using their engine thermal system management strategy. Also, Redfield *et al.* (2006) have developed an engine accessory electrification system powered by a fuel cell auxiliary unit to operate the engine cooling pump, valves, eight simultaneously operating radiator cooling electric fans, water spray system, A/C compressor and condenser, etc. Authors have demonstrated that the engine temperature variations are kept within ± 3°C by utilizing radiator fans as the last option for engine cooling. The engine thermal system management strategies suggest minimizing the parasitic losses by using multiple actuators like coolant pump, radiator cooling fan, mixing tank, control valve, etc and sometimes in multiple numbers.

Installation of different types of actuators and sometimes in different numbers for engine temperature control may not be suitable for the engines that are to be used in earth moving vehicles, army vehicles, for marine applications, AC or DC generator sets, and off-highway engine applications where engines operate under adverse environment conditions and engine cooling requirements are high. For such applications actuators like electric coolant pump, electric radiator fan, smart valve or combinations of these might become insufficient to maintain engine temperature within narrow range due to their lower power densities as mentioned by Kluger and Harris (2007). It might be necessary to use such electric equipments in multiple numbers in order to control engine temperature. But as the number of actuators increase, the management system becomes more complex, bulky and difficult for equipment maintenances. It becomes necessary to design advanced engine thermal management system to control dynamic behaviors of various actuators and their combinations to maintain engine temperature within narrow temperature variation range. Under such conditions, it is beneficial to use actuators having higher power density and which can be installed in minimum numbers. This requirement can be satisfied by installing hydraulic motors and pumps which have higher power densities and also have higher energy transfer efficiency than electric motors and pumps. For example, hydraulic motor/pump has power density of 3500 W/kg whereas electric motor/pump has power density of 500 W/kg as suggested by Kluger and Harris (2007) for the same application.

Some of the different research works that have been done in engine thermal management system are presented below. Luptowski *et al.* (2005) have developed enhanced engine thermal simulation which predicts the effects of different cooling systems, accessory loads, etc on engine performance by linking with GT-POWER software for engine/cycle analysis. Cortona and Onder (2000) designed the engine thermal system model with relevant equations and presented model validation results by conducting testing on small supercharged two cylinder engine. They have presented the results in energy consumption reduction that they could achieve by using their control strategy. Lehner *et al.* (2001) have designed a model based engine thermal management strategy using variable speed coolant pump, position controlled thermostat and a radiator fan for heavy duty diesel engine. As one of the part of control strategy, they have used PID controllers for feedback control of engine temperature. They too have successfully achieved power consumption reduction.

Wagner *et al.* (2003) designed lumped parameter model to describe engine thermal management system using heater, smart thermostat valve, radiator and variable speed electric pump. The operations of smart thermostat and electric water pump are controlled using model-free PI control architecture and a table lookup approach with relevant equations. They have presented the numerical results to show that set point temperature tracking disturbances are 0.2°C for maximum steady-state errors and overshoot of 1.7°C is possible while controlling the engine temperature. Eberth *et al.* (2004) have designed a dynamic mathematical model to analyze various thermal management architectures. They have used a factory configuration engine, smart valve, variable flow pump and radiator fan. They have compared various combinations of these actuators using AMESim simulation model to show which combination gives lower power consumption results and better engine set point temperature tracking results. They have used PID controllers for tracking the engine temperature. They suggest that combination of smart valve, variable flow coolant pump and electric radiator fan gives best overall numerical results.

Salah *et al.* (2010) proposed use of nonlinear control architecture which will track temperatures of different cooling systems in engine. They have shown that by using robust controller for controlling thermostat valve, and electric radiator fan, both of which will be continuously varying, it is possible to accurately track engine and transmission temperatures using steam heated engine block, heat exchanger, radiator, and different sensors. Bruckner *et al.* (2006) have presented model predictive control using electrical coolant pumps to regulate cylinder head temperature. They have presented resulting control problem as an optimal problem with its cost function and the plant model in terms of state space representation. Frick *et al.* (2008) have investigated a thermal management system by using a hydraulic cooling system which involves hydraulically controlled radiator fan as well as coolant pump. Their experimental study uses electric emersion heaters to emulate automotive engine. By using the servo-solenoid proportional control valves and a hydraulic cooling system, they have shown that PID controllers have successfully regulated the engine coolant temperature.

The unique features of proposed paper are design of the simulation model of engine thermal management system which contains mathematical models for thermal systems, hydraulic systems as well as electrical systems, estimation of engine and coolant temperature values, estimation of thermostat valve opening as well as the estimation of speed of hydraulic motor operated radiator fan to cool down the radiator. These mathematical expressions are presented in such a way that they can be presented in state space representation form and can be used with the controllers such as LQR (Linear Quadratic Regulator), Lyapunov-based nonlinear controllers. Such controllers can reduce the cost of operations of different actuators and control the engine temperature. Hence the experimental results for a hydraulic motor operated radiator fan, an engine driven coolant pump and a mechanical thermostat are presented to validate the simulation results.

2.2 Cooling System Configuration

The proposed engine thermal system management uses a hydraulic motor operated radiator fan, an engine driven coolant pump and a mechanical thermostat. This configuration is used to control the engine temperature. The experimental data gathered can be used to development the simulation model for the same configuration of a hydraulic motor driven radiator fan, an engine driven coolant pump and a mechanical thermostat. As shown in the Fig. 2.1, the engine coolant passes through two coolant passage circuits.

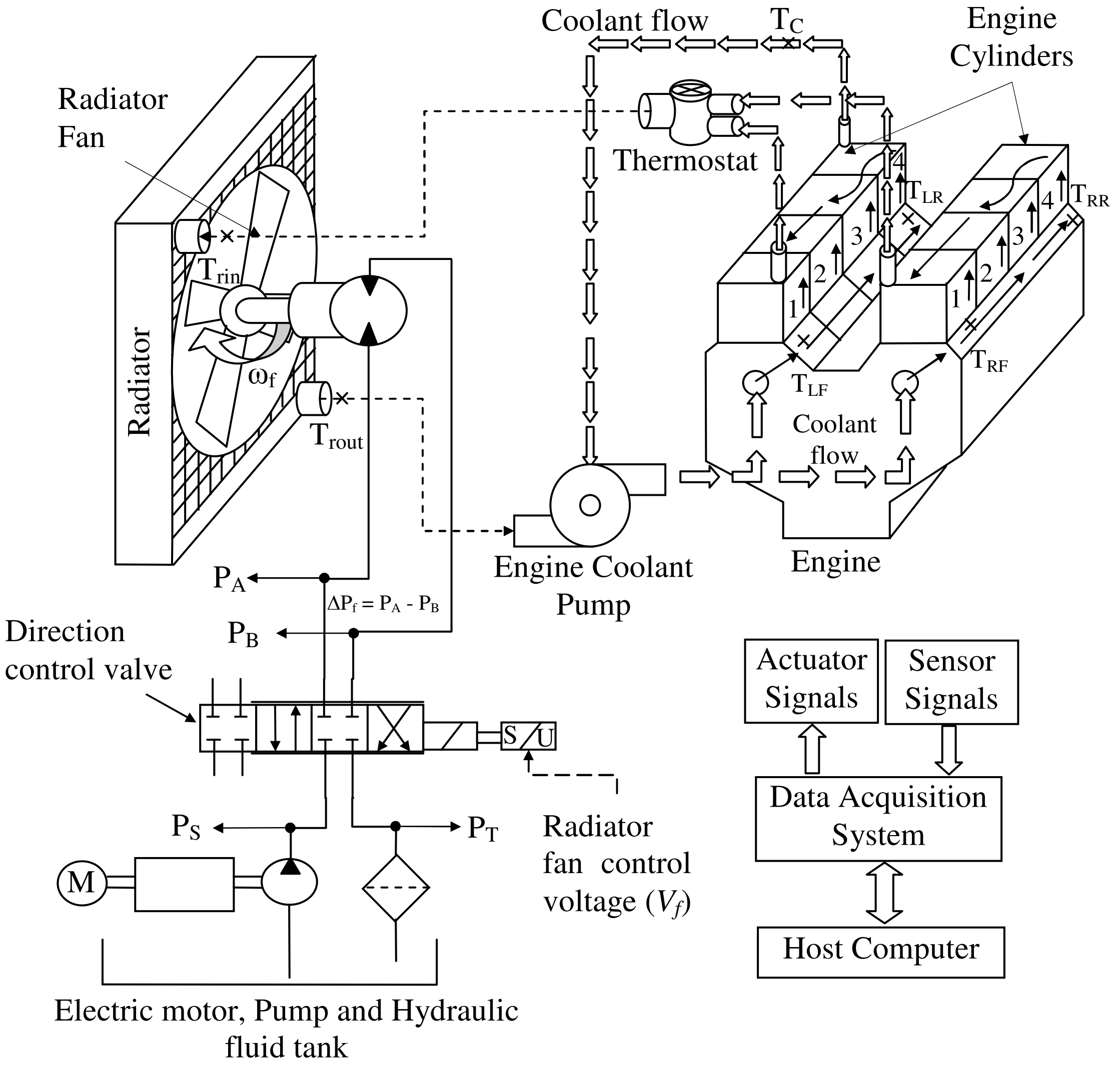

Fig. 2.1: Proposed engine thermal management system configuration

The first one is used to circulate coolant within engine itself when engine starts to warm up and coolant is not required to be cooled. Second coolant circuit passage is used to exit hot coolant, when engine has warmed up, to enter through the top side of radiator which acts as a heat exchanger. Radiator looses heat to ambient environment when radiator fan starts to run and cools the hot engine coolant entering through its top side. At the bottom side of the radiator, the cold (when coolant temperatures at the top and bottom side of radiator are compared) engine coolant again enters the engine block to decrease the engine temperature and maintain the engine temperature at set point temperature tracking value. The data acquisition system is used to collect data from different sensors, and send control input signals to actuators. The control signal is generated by controller which processes the incoming signals according to control algorithm written in host computer and generates the control signal. The control signals are amplified through data acquisition system in order to operate the actuators.

2.3 Mathematical Models for Engine Thermal Management System

The mathematical models that are used for the design of an engine thermal management system are presented in following sections (Frick *et al*, 2006, 2008 and Salah *et al*, 2009). A lumped parameter approach will be applied to realize the governing differential equations.

Engine and Radiator Thermal System Dynamics

To remove the excessive heat out of the engine, i.e. engine block, hot coolant passes to the radiator where radiator loses heat to the ambient surrounding. Considering this process and the aim of maintaining the engine temperature at constant value, the following heat balance equations can be considered.

$$C_e \dot{T}_e = Q_{in} - C_{pc} \dot{m}_c \phi (T_e - T_r) \tag{2.1}$$

$$C_r \dot{T}_r = -Q_o + C_{pc} \dot{m}_c \phi (T_e - T_r) - \varepsilon C_{pa} \dot{m}_{a_{fan}} (T_e - T_\infty) \tag{2.2}$$

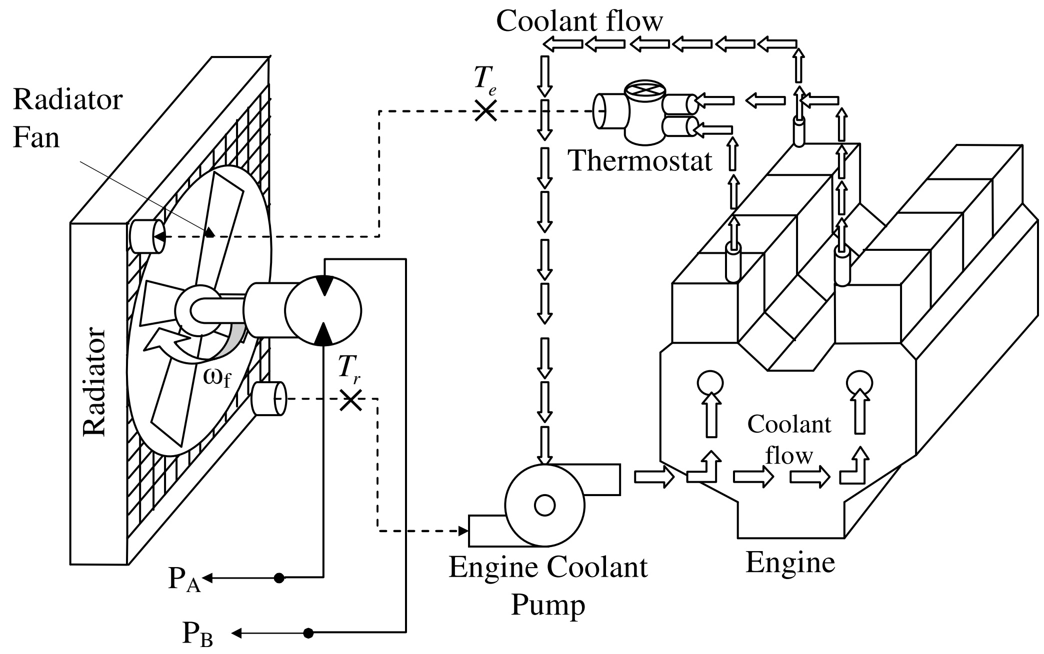

Fig. 2.2: Temperature nodes, T_e and T_r considered for the heat balance equations

The Fig. 2.2 shows the temperature nodes used in the equations 2.1 and 2.2. The system variables used in the above equations, $Q_{in}(t), Q_o(t), \dot{m}_c = f(\omega_e)$, and $\dot{m}_{a_{fan}} = f(\omega_{fan}, V_{speed})$ represent the rate of heat used to warm up the engine, rate of heat lost at the radiator due to ram air flow (considering vehicle speed), engine coolant pump flow rate and air mass flow rate through the radiator, respectively. The term Q_{in} will be adding heat in the system and Q_o is an uncontrolled heat that will be removed from the system. The $\dot{m}_{a_{fan}}$ has been calculated empirically (Frick *et al*, 2008).

Engine Thermostat Dynamics

For the proposed configuration (refer to Fig. 2.1), the engine thermostat has a wax based material which expands or contracts non-linearly with the temperature of coolant inside the engine block. It opens up widely when the engine coolant temperature reaches certain threshold value and varies as per the coolant temperature thereafter. This particular property of thermostat is called as hysteresis. For proposed configuration, following assumptions are made.

A.1: The thermostat opens linearly with the engine temperature and has time constant of approximately 30 seconds (Guzzella and Onder, 2004).

A.2: The thermostat time constant varies with the engine speed, engine and coolant temperatures.

A.3: It is assumed that at all time, there is at least 10% internal coolant flow within the engine block and rest might be sent to radiator for cooling down the hot coolant.

Following conditions define the control action performed by thermostat valve. The temperature governing the control action of thermostat valve is T_e . The thermostat operating temperatures are defined as,

$$\phi = \begin{Bmatrix} 0; & T_e \leq T_{Low} \\ \dfrac{\psi}{(T_{High} - T_{Low})}(T_e - T_{Low}); & T_{Low} < T_e < T_{High} \\ \psi; & T_e \geq T_{High} \end{Bmatrix} \tag{2.3}$$

Heat Exchanger Analysis Using Effectiveness-NTU Method

The Effectiveness-NTU method has been used for the heat exchanger analysis since the inlet temperatures of both the fluids, i.e. hot engine coolant temperature at radiator inlet and ambient air entering the radiator, are known. If the inlet and outlet temperatures of the coolant as well as air would have been known, the log mean temperature difference (LMTD) model could have been used for the analysis of radiator effectiveness (Incropera and DeWitt, 2002). The number of transfer units (NTU) calculated by this model provides the effectiveness of the heat exchanger, i.e. radiator. For a counterflow heat exchangers like radiator, the effectiveness, ε, is defined as the ratio of the practical heat transfer rate to the maximum theoretically possible heat transfer rate.

$$\varepsilon = \frac{q}{q_{\max}} \tag{2.4}$$

$$q_{\max} = C_{\min}(T_{h,i} - T_{c,i}) \tag{2.5}$$

where $C_{\min}$ will be equal to the smallest value of heat capacity rates of the engine coolant or ambient air. Also $q = C_{pc}\dot{m}_c(T_{h,i} - T_{h,o})$ or $q = C_{pa}\dot{m}_{a_{fan}}(T_{c,o} - T_{c,i})$ is the heat transfer rate within the engine coolant, or ambient air entering and leaving the radiator respectively. For counterflow and both fluids unmixed condition, the effectives of radiator, ε, can also be described as

$$\varepsilon = 1 - \exp\left[\left(\frac{1}{c_r}\right)(NTU)^{0.22}\{\exp[-c_r(NTU)^{0.78}] - 1\}\right] \tag{2.6}$$

where $NTU = \dfrac{UA}{C_{\min}}$ and $c_r = \dfrac{C_{\min}}{C_{\max}}$. The term UA is known as the overall heat transfer coefficient. If the effectiveness-NTU for the radiator is known, the equation (2.4) could be used to calculate the temperature of either the engine coolant or the ambient air leaving the radiator.

The overall heat transfer coefficient for the radiator on air side, $U_a A_a$, depends on the speed of hydraulically operated radiator fan and ram air. Under the assumption of laminar flow of air over the flat plate, the Nusselt Number can be calculated to find out the convective heat transfer value (Frick *et al.*, 2006).

$$\bar{N}u_L \equiv \frac{\bar{h}_L L}{k} = 0.664\,\mathrm{Re}_L^{1/2}\,\mathrm{Pr}^{1/3} \tag{2.7}$$

For air velocities of $0 < V_a < 15$ m/s and at the trailing edges of radiator tubes and fins, it was found that Reynolds number was around 40,000 which suggests that air flow can be considered as a laminar flow. The Reynolds number and Prandtl numbers are calculated by $\mathrm{Re}_L = \frac{VL}{\upsilon}$, and $\mathrm{Pr} = \frac{\upsilon}{\alpha}$ respectively.

Similarly, the overall coefficient for the radiator on coolant side, $U_c A_c$, can be calculated by computing the Nusselt Number for turbulent flow as

$$Nu_D \equiv \frac{hD_h}{k} = 0.027\,\mathrm{Re}_D^{4/5}\,\mathrm{Pr}^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14} \tag{2.8}$$

and for $0.7 \le \mathrm{Pr} \le 16{,}700$, $\mathrm{Re}_D \ge 10{,}000$, and $\frac{L}{D_h} \ge 10$. These values of Prandtl and Reynolds number were satisfactory for the 34.3 cm length of the radiator tubes. Hence using these Nusselt Number values, the overall heat transfer coefficient for the radiator could be calculated. The heat transfer values calculated from this analysis could be used to calculate the temperature of the coolant and the air leaving the radiator.

Using Effectiveness-NTU method, and for the condition when coolant flow starts to flow towards the radiator and steady state conditions of coolant and air flow are reached, the effectiveness of radiator, ε, can be calculated as

$$\varepsilon = \frac{q}{q_{\max}} = \frac{C_{pc}\dot{m}_c \phi (T_e - T_r)}{C_{pa}\dot{m}_{a_{fan}} (T_e - T_\infty)} \tag{2.9}$$

Hydraulic Motor Driven Radiator Fan

A single auxiliary electric motor and hydraulic pump can provide the necessary hydraulic power required to operate a radiator fan. As shown in the Fig. 2.3, an electronically operated servo-solenoid valves control the flow of hydraulic fluid going to radiator fan. The control voltage, V_s, is applied to solenoid coils of valves to generate a mechanical force which proportionally moves the spool shaft to open hydraulic fluid paths for rotating the radiator fan. The mechanical force $F_s(t)$ and solenoid current $i(t)$ can be related by using equations given below (Merritt, 1967).

$$\frac{di}{dt} = \frac{1}{L_{coil}}(V_s - iR_{coil}) \tag{2.10}$$

$$F_s = \left(\frac{N_t^2 a \mu_o}{4 l_g}\right) i^2 \tag{2.11}$$

As shown in Fig. 2.3, the magnitudes of transient and steady-state forces acting on the spool of valve are given as

$$F_{tr}^{1,2} = [L_d C_d w \sqrt{2\rho (P_{SB} - P_{AT})}]\dot{x} \tag{2.12}$$

$$F_{ss}^{1,2} = [2 C_d w \cos(\theta)(P_{SB} - P_{AT})]x \tag{2.13}$$

where $P_{SB} = P_S \text{ or } P_B$, and $P_{AT} = P_A \text{ or } P_T$. F_{ss}^1 is the steady state force when fluid exits the main chamber and enters the port-A, and F_{ss}^2 is the steady state force when fluid exits the port-B and enters main chamber. Transient forces are generated when the spool of valve is displaced to the left side. Hence F_{tr}^1 is the transient force generated because of acceleration of fluid in the main chamber, and between the port-A and port-B. F_{tr}^2 is the transient force generated because of the acceleration of fluid in the main chamber, and right of the port-B.

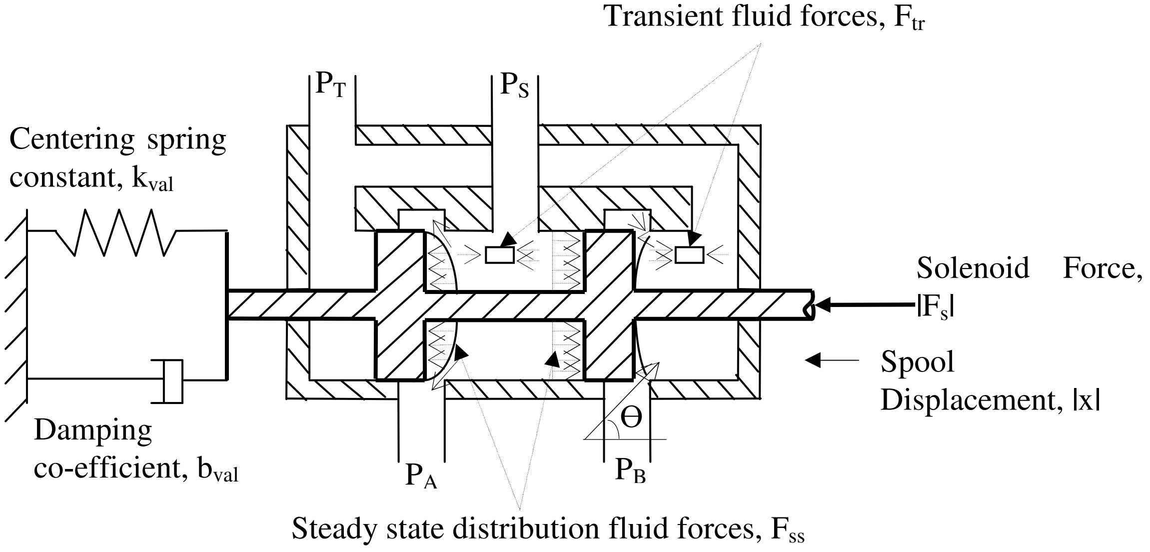

Fig. 2.3: Solenoid and fluid forces acting on the spool in the hydraulic valve

The internal spool displacement of hydraulic valve can be expressed as

$$\ddot{x} = \frac{1}{m_s}[F_s + (F_{ss}^2 - F_{ss}^1) + (F_{tr}^2 - F_{tr}^1) - k_{val}x - b_{val}\dot{x}] \tag{2.14}$$

Position of valve spool $x(t)$, determines the hydraulic fluid flow $Q_L(t)$, and the corresponding hydraulic load pressure generated $P_L(t)$ for radiator fan, which can be given by following equations.

$$Q_L = D_m \omega_{fan} + C_{im}P_L + \frac{V_t}{2\beta}\dot{P}_L = \left(C_d w \sqrt{\frac{(P_S - P_L)}{\rho}} \right) x \tag{2.15}$$

$$\dot{P}_L = \frac{2\beta C_d w}{V_t}\sqrt{\frac{(P_S - P_L)}{\rho}}x - \frac{2\beta C_{im}}{V_t}P_L - \frac{2\beta D_m}{V_t}\omega_{fan} \tag{2.16}$$

With the assumption of zero tank return pressure in hydraulic line, $P_T = 0$, and considering a hydro-mechanical power transformation efficiency of 98%, the motor shaft acceleration $\dot{\omega}_{fan}(t)$ can be given by

$$\dot{\omega}_{fan} = \frac{1}{J}(T_g - B_m \omega_{fan} - T_L) \tag{2.17}$$

where $T_g \triangleq D_m P_L$ and $T_L \triangleq$ constant load torque.

An expression for $\dot{\omega}_{fan}(t)$ can be obtained from Eq. (2.14) and Eq. (2.15) as

$$\dot{\omega}_{fan} = \left(\frac{D_m{}^2 + B_m C_{im}}{JD_m} \right) P_L + \left(\frac{B_m V_t}{2JD_m \beta} \right) \dot{P}_L - \left(\frac{B_m}{JD_m} C_d w \sqrt{\frac{P_s - P_L}{\rho}} \right) x - \frac{T_L}{J} \tag{2.18}$$

The power consumption of the hydraulic motor operated radiator fan can be expressed as

$$P_{sys} = \frac{1}{T} \int_{t_0}^{T} P_L(\tau) Q_L(\tau) d\tau \tag{2.19}$$

where P_L and Q_L are the load pressure and flow of hydraulic motor operating the radiator fan respectively.

2.3 Experimental Setup and Control System

The engine thermal management system experimental setup uses Ford V8 4.6L engine (Peak power: 173 kW at 4500 rpm, Peak torque: 407 Nm at 3000 rpm) mounted with Superflow dynamometer setup, 6.8L radiator, radiator fan, engine driven coolant pump and mechanical thermostat for proposed configuration. For the future configuration i.e. hydraulic motor driven coolant pump and radiator fan, and electric valve, a centrifugal pedestal mount pump which can deliver 220 L/min of coolant can be used with hydraulic motor of 6.36 cm3/rev displacement capacity. The hydraulic motor operating radiator fan has 11.65 cm3/rev displacement capacity. 5.6kW Baldor electric motor is used to drive Bosch hydraulic pump to operate hydraulic motor used for radiator fan and coolant pump. For data acquisition and control signal generation, dSPACE 1104 controller board is used. The board has ATD (Analog to Digital signal), DTA (Digital to Analog signal) and I/O channels that are used to collect data from different sensors and generate control signal for the valve from 0 to 10V. This control signal is given to hydraulic servo-solenoid proportional control valves (Bosch NG6) which uses Bosch PL6 amplifier for displacement of valve spool. The data acquisition system is connected to real-time control algorithm through Matlab/Simulink and Control Desk softwares.

The J-type and K-type thermocouples are used to record various temperatures of engine, coolant, radiator and ambient environment. The temperature signals are passed through Omega OM5-LTC signal conditioner and amplifier to generate voltage signal proportional to temperature values. A digital multiplexer circuit is used to record multiple temperature signals through single ATD channel due to large number of temperature signals. The positions of some of the thermocouples are shown in Fig. 2.1 by $T_{rin}, T_{rout}, T_C, T_{LF}, T_{LR}, T_{RF}$, and T_{RR}. The engine, radiator fan and coolant pump speed can be measured by Monarch Instruments ROS-W optical sensors. The hydraulic line pressures P_A, P_B, P_S and P_T as shown in Fig. 2.1 are measured by Honeywell (Sensotec A-5) pressure transducers and hydraulic supply pressure dial gauge. A turbine flow meter (AW TR-1110) records the coolant flow rate passing through radiator for cooling purpose. The overall experimental setup is shown in Fig. 2.4.

To control the engine temperature within a required temperature range, PID controller has been used for both experimental and simulation engine tests. The controller takes in error signal $e(t)=T_e-T_{e\_ref}$ and generates control signal i.e. radiator fan control voltage (V_f) for proposed system configuration to adjust the fan speed. The radiator fan cools the coolant inside the radiator to maintain the engine temperature within the required range. This classical controller is preferred because it is necessary to develop the mathematical models of various systems in the simulation environment and further these mathematical models needs to be used with advanced controllers which will include cost functions for operations of various actuators. Cost functions will make the engine thermal management system more efficient. Cost functions will also help to operate actuators with minimum cost of operation and without affecting the functional requirement of maintaining the engine temperature within required temperature range. The controllers such as Linear Quadratic Regulator (LQR) for time varying systems could be used along with the simulation mathematical models to include the cost functions. By optimizing the engine cooling system, when engine driven hydraulic motors are used to operate radiator fan and coolant pump, it is possible to achieve even more optimized engine performance results. Hence for the future research, the next configuration of engine thermal management system will involve hydraulic motor operated coolant pump, a hydraulic motor operated radiator fan and an electric valve to achieve the best possible engine performance results.

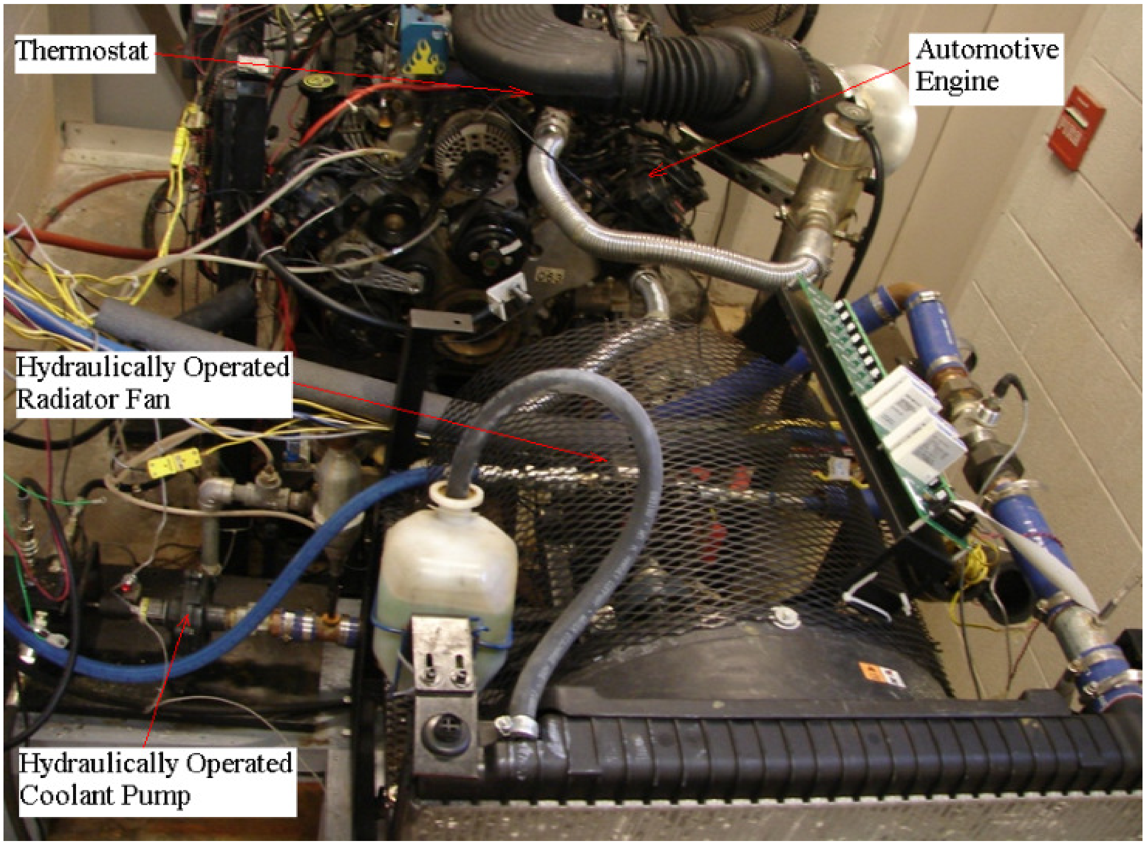

Fig. 2.4: Engine thermal system management experimental setup

2.5 Experimental and Numerical Results

For proposed engine thermal management strategy, the experimental engine testing has been completed to analyze the dynamic behavior of the thermal system when hydraulic based radiator fan has been used. The experimental results will show the temperatures of engine and coolant at various locations along with the speeds of engine and radiator fan. The different engine operating conditions are mentioned in the Table A.2. The control algorithm used to perform the engine testing has been presented in the Appendix C. The set point engine temperature tracking value was set at T_{e_ref}= 88°C for engine test nos. 1 and 6-10, and T_{e_ref}= 89°C for engine test nos. 2-5. The Table A.1 lists the values of gains used in the PID controller to conduct these experimental tests. To explain the various events happening over the engine testing period, the engine testing period has been divided into three stages. The Stage I shows the engine warm up, thermostat opening and coolant flow towards the radiator events. The Stage II presents the events when thermostat is continuously open, the coolant is flowing through the radiator continuously, and radiator fan starts to draw ambient air through radiator to cool down the hot coolant inside the radiator tubes. Finally, the Stage III illustrates the engine and radiator condition where both of them reach their highest desired temperature values, and heat is continuously dissipated to ambient air by operating hydraulic based radiator fan or by ram air burst. These stages have been explained briefly in Table A.3.

The Fig. 2.5 shows the experimental engine test no. 1 with graphs of temperatures of engine and coolant at various locations. The average engine temperature has been used to calculate the tracking error between the set point temperature value and average engine temperature. As the engine starts, the temperatures of engine at various locations start to rise from the room temperature. The engine testing period considers the warm up, idle and high speed run conditions of engine to show the various events happening due to the dynamic responses of engine thermal system.

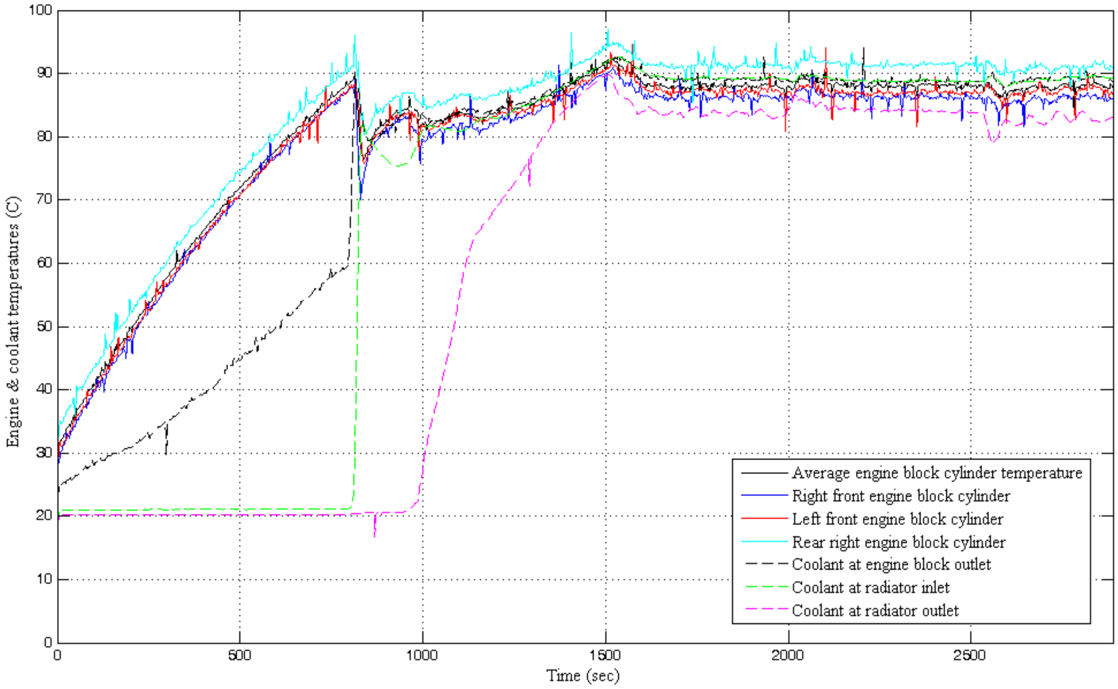

Fig. 2.5: Temperatures of engine and coolant for experimental engine test no. 1

The Stage I of the engine test no. 1, as shown in Fig. 2.6, shows that temperatures of engine blocks and coolant at engine outlet starts to rise. Since thermostat is closed at this condition, the temperatures of coolant at radiator inlet and outlet remain at the room temperature. After approximately 800 seconds the thermostat starts to open at average engine temperature of 88°C. As the hot coolant starts to come out of the engine, the cold coolant inside the radiator starts to enter the engine block. This action leads to sudden drop in the temperatures of engine blocks and average engine temperature. The thermostat controls the coolant going towards the radiator as the engine temperature and hot coolant inside the engine block starts to vary. The thermostat has a wax based material which expands as the coolant temperature rises. It shows a particular characteristic behavior towards the change in temperature which is called as hysteresis. Because of this property of thermostat, there is variation in the control of engine temperature. Till this point, the hydraulic motor operated radiator fan doesn't start. As the coolant starts to circulate towards the radiator, the coolant temperature at radiator inlet rises sharply. The coolant at radiator outlet takes longer time to show first rise in its temperature because of the time taken by the hot coolant at radiator inlet to travel through the radiator tubes and then reach at the outlet of radiator. This time delay is approximately of 165 seconds.

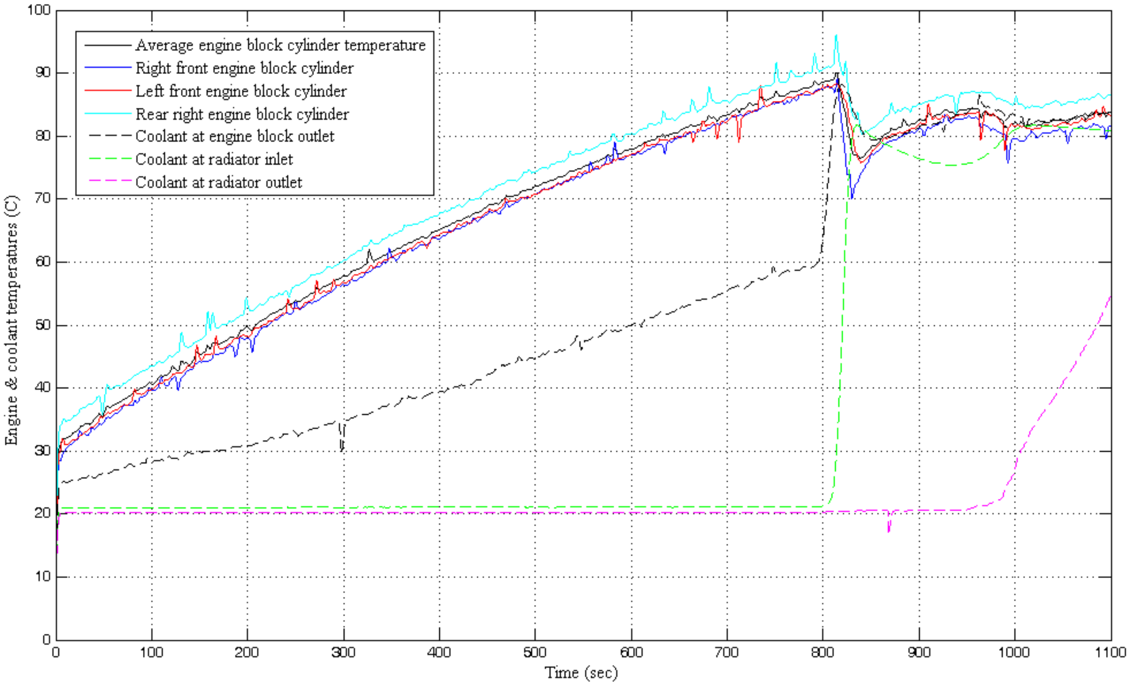

Fig. 2.6: Engine test cycle, Stage I for test no. 1

The Stage II of engine test, as shown in Fig. 2.7, shows that as the hot coolant coming out of the engine starts to circulate continuously through the radiator, the temperatures of coolant at radiator inlet and outlet approaches the engine temperature. At this stage, the thermostat opens completely and temperatures of engine and radiator achieve their maximum temperature under given testing conditions.

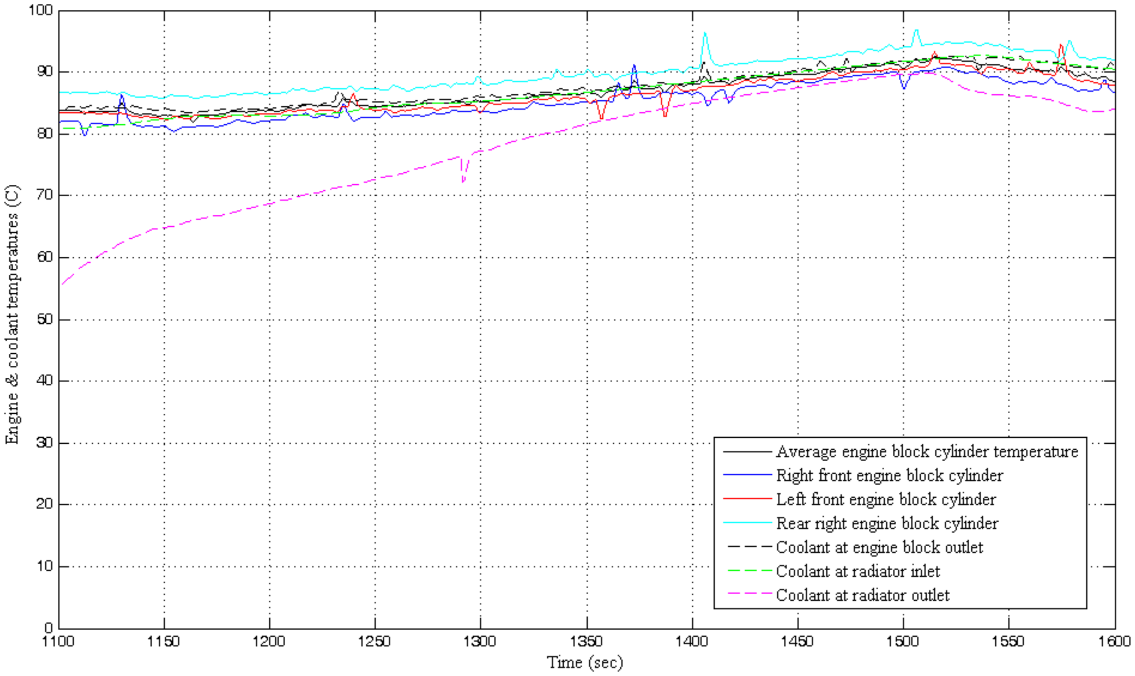

Fig. 2.7: Engine test cycle, Stage II for test no. 1

Fig. 2.8 shows the Stage III of the engine test period when all the thermal components of the systems have warmed up completely and hydraulic motor operated radiator fan has been able to maintain the engine temperature around the set point tracking temperature of engine. It has been possible to maintain the engine temperature around the set point value even though there is sharp increase in the engine speed. At such condition, the time delays for the coolant circulating from engine to radiator and again from radiator to engine reduce. This condition of the system can be used as the steady state condition of the thermal system to evaluate the control system's performance. For engine test no. 1, PID controller shows continuous control over the engine temperature. But if only P-type controller is used, the engine temperature shows oscillating behavior. This oscillating behavior is presented in the engine test no. 6 to 10. It is not desired to have such oscillating temperature of engine because it causes more wear and tear of the engine components.

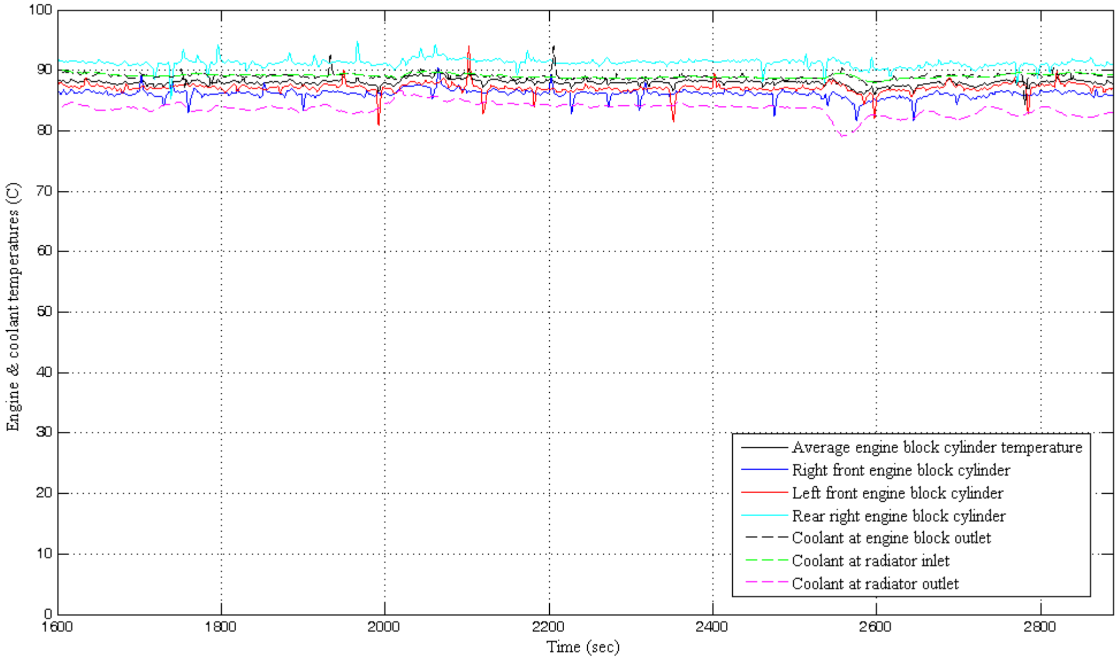

Fig. 2.8: Engine test cycle, Stage III for test no. 1

The corresponding engine and hydraulic motor operated radiator fan speed graphs are shown in the Fig. 2.9. The engine runs on idle condition at around 900 rpm and on high speed condition after 2000 seconds. It is observed that as the engine speed increases, the hydraulic motor operated radiator fan also shows increase in its speed. At such condition, the control action performed by radiator fan produces better results than the engine coolant flow rate control. Although for this proposed configuration, it is not possible to control the engine coolant flow rate, this observation will be useful for the future research where coolant pump will be controlled by the hydraulic motor.

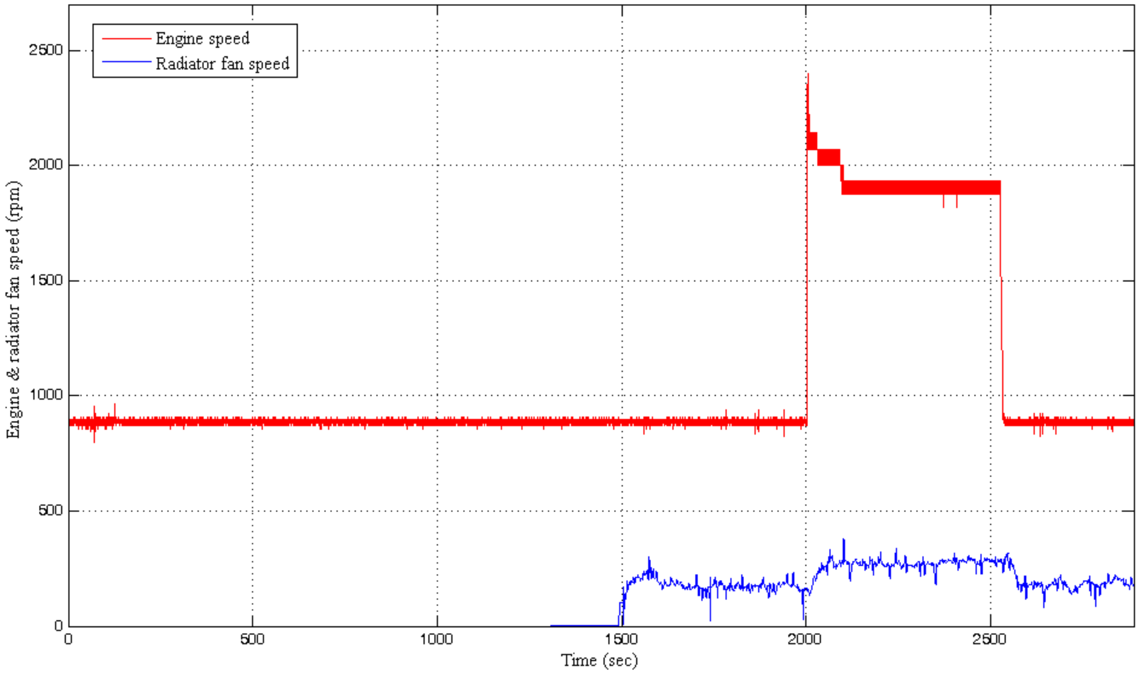

Fig. 2.9: Engine and radiator fan speed for test no. 1

Table 2.1: Steady state errors for the set point engine temperature tracking with PID controller

Engine Test No.	Event time (sec)	ω_e (RPM)	$\lvert e_{T_e} \rvert$ (°K)
1	2,323	1,935	0.25
2	2,717	2,308	0.32
3	2,694	2,143	0.44
4	2,570	1,935	0.28
5	2,485	1,875	0.3
6	2,279	2,124	1.91
7	2,855	2,159	1.89
8	2,332	2,069	1.46
9	2,233	2,308	1.76
10	2,128	2,000	1.45

Similar to engine test no. 1, the other engine test results have been presented in the Appendix B. The steady state errors for different engine tests are listed in the Table 2.1 to analyze the performance of two control strategies employed for the experimental engine testing. It is clearly visible that strategy with PID control provides better results than just with P-type control.

Based on the experimental data and mathematical models presented under the Mathematical Models for Engine Thermal Management System (Section 2.3), a simulation model has been developed in Matlab/Simulink software to estimate the temperatures of the coolant at engine outlet and at radiator outlet. It has been assumed that coolant at radiator inlet has same temperature as the engine temperature and coolant at radiator outlet has same temperature as the radiator. Considering these assumptions, the comparison of estimated temperatures of engine and radiator with the experimental data of temperature of engine and coolant at radiator outlet for the engine test no. 1 has been shown in Fig. 2.10 to Fig. 2.14 along with other simulation results. The values of various parameters and control algorithm for this simulation model are documented in the Appendix E and Appendix F respectively. Table 2.2 lists some of the important values of parameters which are used for this simulation model.

The simulation model takes experimental data of engine speed as an input and calculates the estimated heat input to the engine thermal management system. The simulation model uses PID controller for feedback loop control. Since the first four experimental tests use the same classical controller, the simulation model results are only compared with the first four experimental results. The rest of the simulation results are presented in the Appendix D.

Table 2.2: Engine thermal management system simulation parameters

Symbol	**Value**	**Units**	**Symbol**	**Value**	**Units**
a	13.7	mm	N_t	1600	-
A_{rad}	0.38	m^2	P_s	6.89	MPa
B_m	0.082	Ns/mm	P_T	0	MPa
b_{val}	7	Ns/mm	T_{High}	368.15	°K
C_e	54	kJ/°K	T_L	6000	Nmm
C_d	0.63	-	T_{Low}	358.15	°K
C_{im}	250	mm^5/Ns	T_∞	292.75	°K
C_{pc}	2.36	kJ/Kg°K	V_t	36870	cm^3
C_{pa}	1.01	kJ/Kg°K	w	3.62	cm^2/cm
C_r	44	kJ/°K	β	689.5	MPa
D_m	1.85	cm^3/rad	η_{hm}	0.98	-
J	0.0029	kg.cm^2	ε	0.1	-
k_{val}	52.53	N/mm	μ_o	4.9e-8	H/mm
L_{coil}	0.02	H	θ	1.2	rad
L_d	12.7	mm	ρ	899.8	kg/m^3
l_g	0.99	mm	ρ_a	1.18	kg/m^3
R_{coil}	4.5	Ω	ρ_c	997	kg/m^3
m_s	4.5	kg	ψ	0.9	-

As shown in Fig. 2.10, the simulated engine temperature has been compared with the experimental average engine temperature of engine test no. 1. At steady state, i.e. when engine is running at high speed, the time delays between circulation of coolant through the engine and towards the radiator decreases, the thermostat is widely open, and hydraulically operated radiator fan is continuously working to remove excessive heat out of the system, the error between the simulated engine and experimental engine temperature has been found to be approximately $|e_{T_e}| = 1.3\ ^\circ\mathrm{K}$. The simulated engine temperature has similar thermal system response as the experimental engine temperature.

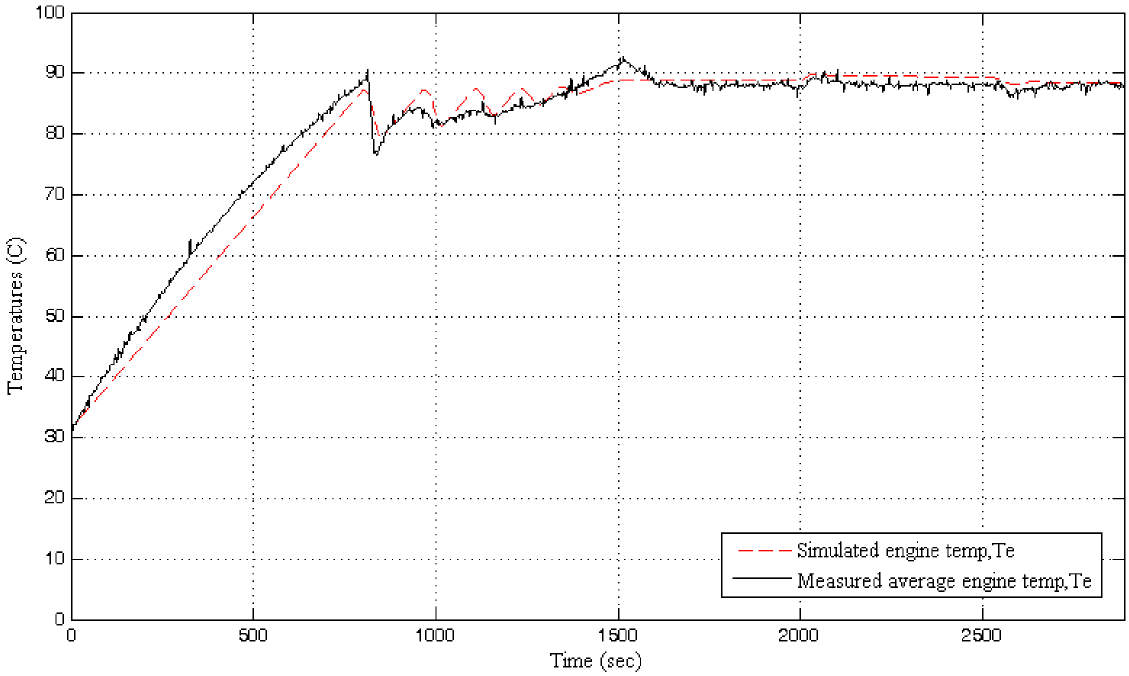

Fig. 2.10: The comparison of simulated and measured engine temperatures for engine test no. 1

Fig. 2.11 presents the experimental data and simulated results for the engine coolant at various locations. At steady state condition, the error between the simulated and experimental coolant temperature at radiator outlet is found to be approximately $|e_{T_{rout}}| = 0.7\,^\circ\mathrm{K}$. The simulated coolant temperature at radiator outlet doesn't exactly behave as observed in the experimental data during the engine warm up stage. The mathematical models need to take into account the hot coolant transportation time delay from radiator inlet to the radiator outlet when thermostat opens for the first time during the engine testing period. The mathematical models also need to take into account the hysteresis property of engine thermostat. The current simulation model assumes that the engine thermostat opens proportional to T_e. The estimated thermostat valve opening percentage has been presented in the Fig. 2.12. The current thermostat model also assumes that thermostat can divert maximum of 90% of engine coolant pump delivery towards the radiator.

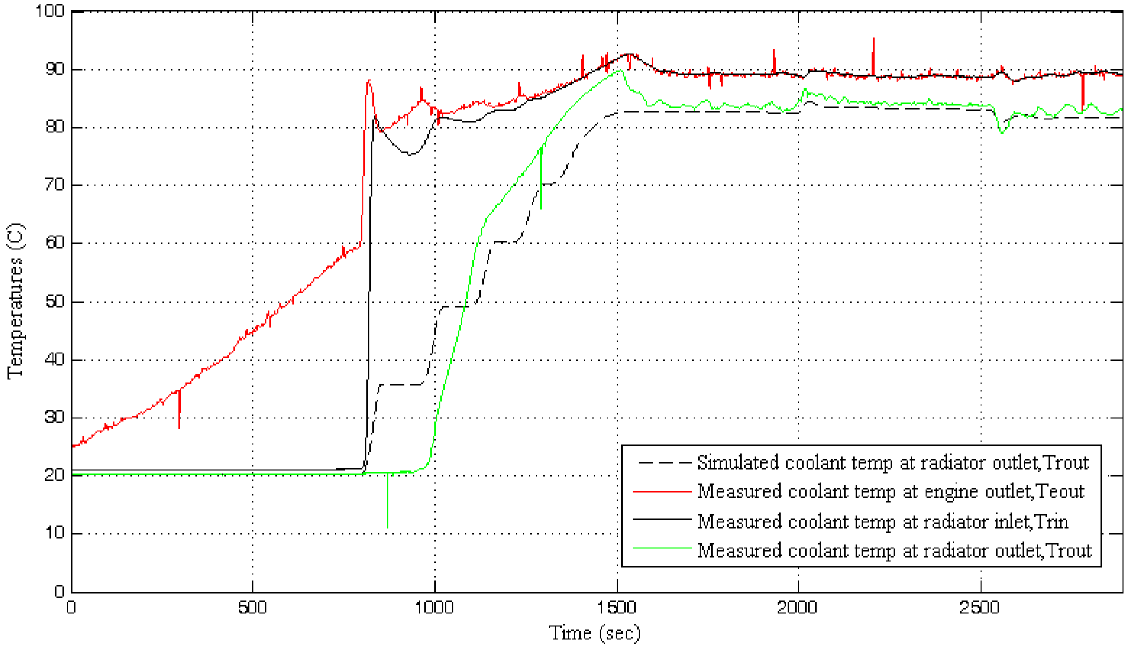

Fig. 2.11: The comparison of simulated and experimental coolant temperatures for engine test no. 1

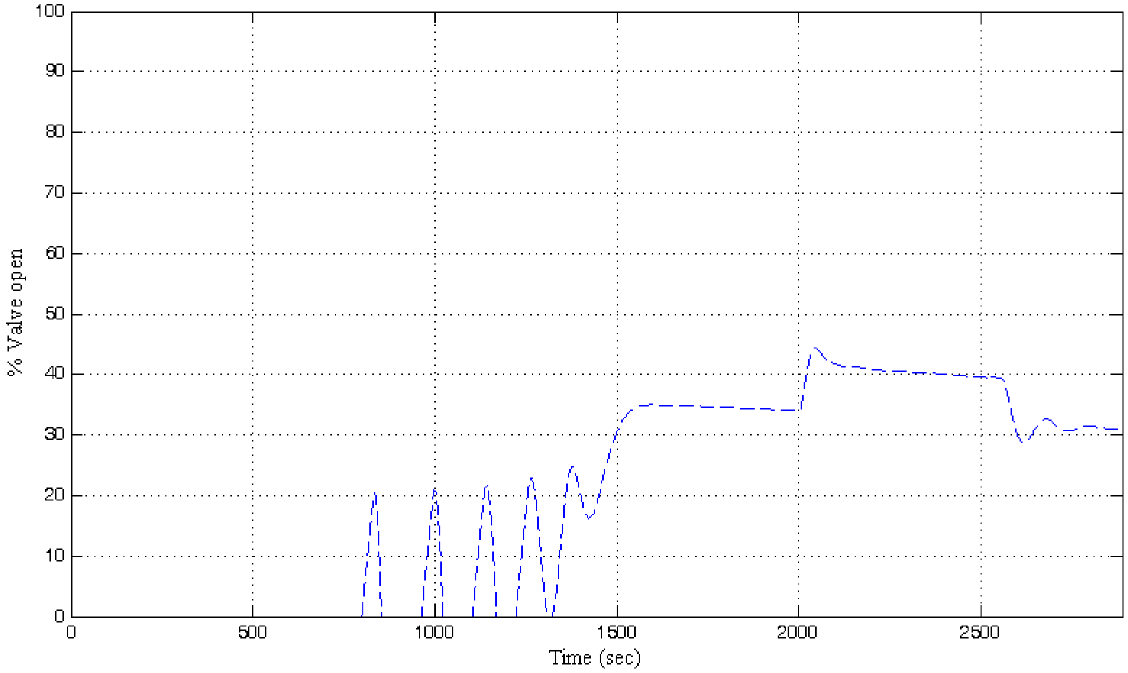

Fig. 2.12: The estimated engine thermostat valve opening for engine test no. 1

The Fig. 2.13 shows the experimental engine speed and comparison of estimated hydraulically operated radiator fan speed with the experimentally recorded hydraulic motor driven radiator fan speed. The simulation results showed the similar dynamic response as the experimentally recorded radiator fan speed response. Although the steady state results of simulated radiator fan speed exactly match with the experimental one, the mathematical model estimating the radiator fan speed still need to be improved.

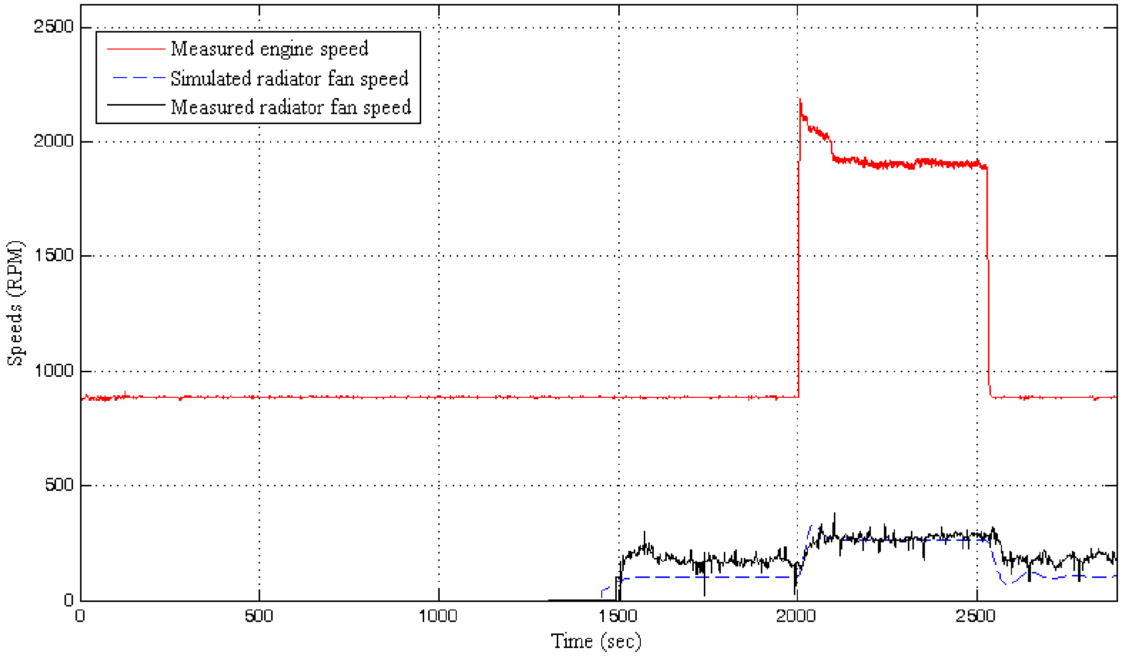

Fig. 2.13: Measured engine speed, and comparison of simulated and measured radiator fan speed for engine test no. 1

The estimated heat input to the engine thermal management system, and the engine temperature tracking errors for simulated and experimental tests have been shown in the Fig. 2.14. It can be seen that the errors tend to become zero as the engine testing period continues. The zero error tracking of the control system has been insured by the integrator in the PID controller. It can be observed that there is a sharp increase in the heat input to the system, i.e. Q_{in} rises from 4.18 kW to 12.67 kW, as the engine speed increases from 890 rpm to 2,185 rpm. Because of this heat input into the system, the hydraulic motor operated radiator fan also operates at higher speed in both the simulated and experimental tests to dissipate excessive heat out of the system.

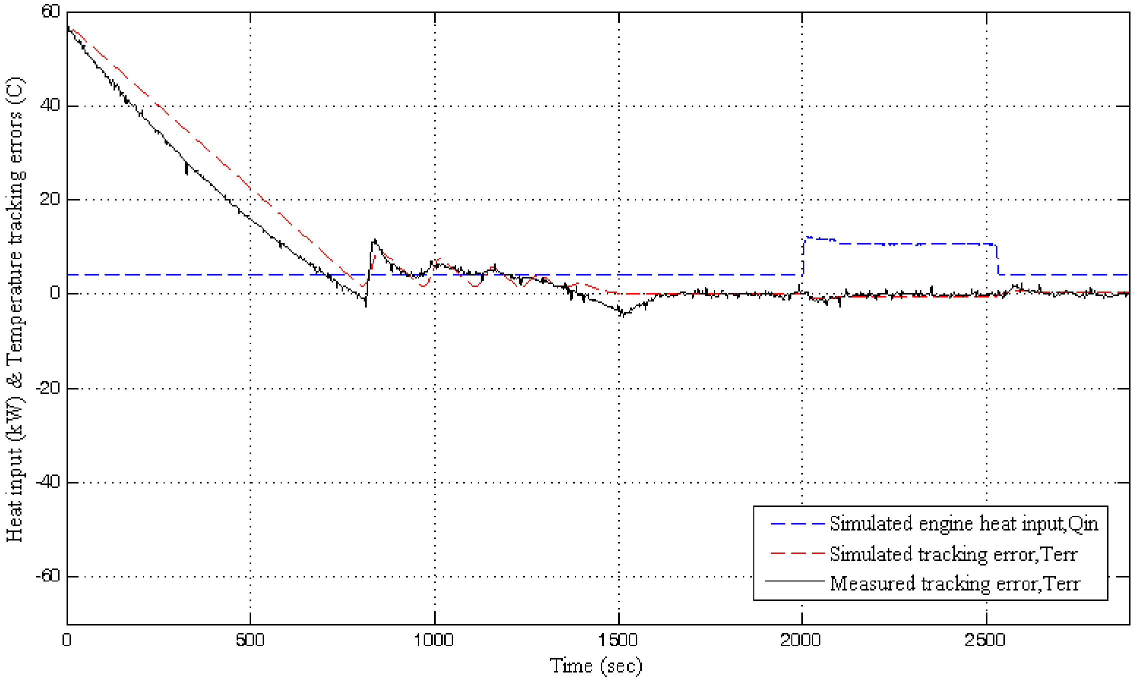

Fig. 2.14: Simulated engine heat input to warm up the engine, and simulated/measured reference temperature tracking errors for engine test no. 1

Table 2.3 summarizes the simulation results for the rest of the engine tests along with some other simulation parameter results. It can be seen that steady state error for coolant at radiator outlet is less than the steady state error for the engine temperature. This could be because of the time delays between circulations of coolant from the radiator outlet to the engine and radiator acting as the only source of excessive heat dissipation. The table also lists the estimated values of heat input to the engine thermal system, Q_{in}, and power consumed by the hydraulic motor operated radiator fan, P_{sys}.

Table 2.3: Summary of temperature errors of engine and coolant at radiator outlet when experimental results are compared with simulation test results at steady state conditions

Test No.	ω_e (RPM)	$\vert e_{T_e} \vert$ (°K)	$\vert e_{T_{rout}} \vert$ (°K)	Q_{in} (kW)	P_{sys} (kW)
1	1,890	1.22	0.68	10.67	0.553
2	2,243	1.16	0.38	13.1	0.720
3	2,100	1.46	0.62	12.14	0.663
4	1,930	1.23	0.11	10.88	0.528

The alternative approach to refine the earlier explained mathematical models for engine thermal system management has been presented in Appendix G. It provides the estimation for the temperatures of engine block, radiator block, coolant inside the engine block, and coolant at radiator inlet and outlet along with other estimated simulation parameters.

2.6 Summary

The engine thermal system management using a hydraulic motor operated radiator fan to maintain engine temperature at a set point temperature value has been proposed as one strategy for engine cooling systems. The classical control approach is utilized for maintaining the engine temperature at set point value by controlling the speed of a hydraulic motor operated radiator fan. The mathematical model for proposed configuration is presented with simulation results for comparison with the experimental results. The model has been able to show the dynamic responses of the engine thermal system with steady state engine temperature errors of $\pm 1.5°K$. The model can be improved by considering the coolant mass transport along with the convection heat transfer effects and the variable transport delays between the thermal systems. The proposed model could be used with the controllers such as Linear Quadratic Regulator

(LQR) or Lyapunov-based nonlinear controllers which will control electric valve, hydraulic motor operated radiator fan and coolant pump to maintain engine temperature within certain range and also reduce the cost of operations of various actuators.

CHAPTER 3

A MULTI-DISCIPLINARY MECHATRONICS COURSE WITH ASSESSMENT – INTEGRATING THEORY AND APPLICATION THROUGH LABORATORY ACTIVITIES

The mechatronics course for undergraduate and graduate level engineering students, a technical elective offered by the Department of Mechanical Engineering at Clemson University, promotes the exploration of mechatronic systems integration concepts. The holistic course activities include studying fundamental knowledge from mechanical, electrical, computer, industrial, and robotics engineering which is re-enforced through hands-on laboratory experiments and semester long projects. The design projects foster collaborative team work activities, leadership, and project management skills as well as offer the opportunity for in-depth experience with sensors, actuators, and material handling systems. The course assessment, which establishes a basis for continuous improvement, considers student performances, their written feedback on qualitative surveys, and feedback offered by an advisory panel composed of industry experts. The assessment results show that the mechatronics course successfully offers students an unique learning environment which is truly practical and helpful in integrating theory with real world applications.

3.1 Introduction

Mechatronics is the integration of mechanical, electrical, computer, industrial, and robotics engineering concepts in the design of smart products and processes. As the size and cost of digital hardware and sensing technology decreases, more mechatronic systems are being used in industries such as aerospace, defense, health care, material handling, and transportation as well as consumer products including kitchen and laundry appliances, garden/lawn care, and entertainment. To design mechatronic systems and smart products, engineering students must acquire necessary skills and practical experience. Specifically, they should be able to apply electronic circuits, sensors, actuators, microprocessors, control theory, and systems integration so that diverse technologies can be combined together to realize a functional product. A multi-disciplinary mechatronic course for undergraduate/ graduate students at Clemson University has been developed to address the needs of engineering students and industrial companies. This course integrates fundamental concepts with hands-on experiences during laboratory activities and design projects.

Students generally learn and retain more knowledge when they experience or practice what they have learned (Nilson, 2003). The design of a multi-disciplinary mechatronics course with laboratory component is well suited for this learning approach since students receive extensive opportunities to practice and explore concepts. They participate in dynamic team interactions to apply knowledge gained from past courses and investigate real ideas to solve assigned problems. Diong *et al.* (2004) described a similar approach used at the University of Texas at El Paso for a mechatronics course. The assessment analysis and results indicated that the hands-on mechatronic projects had improved student learning in control systems. Ramasubramanian *et al.* (2003) reported on a graduate level multi-disciplinary course in mechatronics at North Carolina State University for electrical and mechanical engineering students. Smaili and Chehade (2005) discussed the efforts taken by the American University of Beirut to offer a mechatronics course which emphasized just-in-time learning, projects, learning-by-doing, and minimal lecturing. Guerra-Zubiaga *et al.* (2010) developed a senior level mechatronic course at ITESM of Tecnológico de Monterrey where students used design methodology concepts to realize design requirements of a selected manufacturing company. Kurfess (2001) presented the challenges and lessons learned while integrating a new mechatronics course into a large second-year design course at Georgia Tech. He reported on the different devices used for the course project and their relative costs. Yavuz and Mistikoglu (2009) described a study to determine whether to create a separate mechatronics department at Mustafa Kemal University. An interesting aspect of the article was the discussion of the approaches by global universities to offer mechatronic courses. Gupta *et al.* (2003) presented a mechatronics syllabus designed for undergraduate students at Malaviya Regional Engineering College which consisted of eight semesters. They documented the necessary laboratory equipment, commercial software, and other requirements needed for the course.

Krishnan *et al.* (2006) designed two mechatronics courses at the University of Detroit Mercy entitled “Modeling & Simulation of Mechatronic System,” and “Sensors & Actuators for Mechatronic Systems.” Rogers *et al.* (2009) at the United States Military Academy offered a mechatronic course to solve open-ended problems in interdisciplinary fields and provided course assessment results. Grimheden (2007) described a mechatronics course designed at the Royal Institute of Technology which involved international collaboration projects with universities from Australia, Europe, Japan and the United States. Uelschen *et al.* (2011) described an introductory course on software engineering for undergraduate mechatronic students which focused on goal-orientation and pragmatic problem solving at the University of Applied Sciences Osnabrück. Solis *et al.* (2009) presented an introductory mechatronic course for undergraduate students using robotic systems at Waseda University. The authors adopted a Project Based Learning (PBL) model to introduce the laboratories and undertake an inverted pendulum-based robotics competition. Finally, Rojko *et al.* (2010) conducted a mechatronics E-course for both traditional students and industry professionals with classical and remote laboratory experiences using an adaptable learning approach. Overall, these academic efforts indicated a growing need of mechatronic courses to prepare students to work in multi-disciplinary areas, and embraced rapidly changing industrial environments. However, the offerings at these institutions did not necessarily provide students with industrial material handling equipment and project management applications within a laboratory setting to meet the needs of the manufacturing industry.

The mechatronics course at Clemson University has been offered since 2001 and covers the traditional areas of mechanical, electrical, computer, and industrial engineering. The unique features of this course are the hand-on experiences with Programmable Logic Controller (PLC) programming for stand-alone and networked applications, an industrial Staubli robotic arm featuring sensor feedback, and material handling (conveyor) systems. It also includes the use of bread-boards for electronic circuits, as well as various electrical machines, sensors, actuators, and data acquisition systems common to the workplace. Apart from this, the course includes people skills such as business ethics, leadership, team building, collaboration, and human factors. To understand the relevant materials, the students meet twice per week in a classroom and have an accompanying weekly laboratory session to experience mechatronic systems. From fall 2008 to spring 2011, this course had been offered four times. Enrollment data showed that the majority of students were mechanical engineering majors. The evaluation results for this period indicate that the course has received very positive responses from students (refer to Section 3.3). As part of a continuous improvement process, an industry advisory panel has been formed to work with the teaching faculty in analyzing the progress of the overall course activities.

This article describes a mechatronics course offered at Clemson University and the accompanying assessment process. Some of the key course features include integrated classroom and laboratory teaching, design projects, and emphasis on people skills. The manuscript is organized as follows. Section 3.2 reviews the classroom, laboratory, team design project activities, and industrial plant tours which establish the basis for learning; Section 3.3 contains the assessment methods, assessment data, and accompanying discussion about the results which reflect the successful development of the course; and the summary is presented in Section 3.4.

3.2 Student Learning Methods in the Mechatronics Course

The student learning strategies emphasize hands-on laboratory experiences using current technology, design projects, and collaborative classroom activities. The laboratory experiments and team based design projects require students to integrate sensors, actuators, and computer control into electro-mechanical systems. The classroom teaching efforts incorporate these technical concepts with people and business skills in a peer setting. Students learn and practice those lessons in both the classroom and the laboratory assignments. The application of mechatronic systems are best illustrated by industrial plant visits to companies located within a 50 mile radius of the university. Collectively, the classroom activities, experiments, projects, and plant tours are designed to emphasize systems integration, a team approach, and to showcase practical applications. These methods will be explained in the following subsections.

Classroom Activities

The classroom activities encourage independent student readings, in-class discussions, and laboratory explorations. The short lessons and accompanying discussions focus on various topics within mechanical, electrical, computer, and industrial engineering, plus systems integration as listed in Fig. 3.1. A special aspect of the course is the emphasis on people skills including collaborative learning, project management, team building, leadership development, ethics, procurement, and writing design specifications. Students are assigned to multi-disciplinary teams that collaborate towards completing design projects. One of the course objectives is to organize students of different backgrounds together for learning a common platform, namely mechatronic systems.

Mechanical Engineering
- Actuators
- Hydraulic Systems
- Mechanical Systems
- Pneumatic Systems
- Sensors
- Thermal Systems

Electrical Engineering
- Amplifiers
- Circuits
- Data Acquisition
- Electric Power
- Electronics
- Electric Motors

Controls Engineering
- Block diagram
- Control Systems
- Robotic Systems
- State Space
- Transient Response

People Skills
- Collaborative Learning
- Project Management
- Team Building
- Leadership
- Ethics
- Procurement & Specifications

Computer Engineering
- Digital Logic
- Matlab / Simulink
- PLC Algorithms
- Robot Arm Commands

Industrial Engineering
- Human Factors
- Human / Machine Interface
- Safety
- Workers

Systems Integration
- System Design
- Case Study of Integrated Material Handling System

Fig. 3.1: The various mechatronic system classroom topics covered during a semester

The classroom activities also involve solving in-class examples which allow students to practice recently learned course material. These examples help to develop a collaborative approach towards problem solving and team building. Students learn to respect and share ideas reflecting different points of view. Weekly assignments on course material are given to students for an in-depth understanding of subject areas such as state space representation, use of operational amplifiers, hydraulic and pneumatic circuit design, data acquisition, derivation of transfer functions for electro-mechanical systems, etc. Homework assignments include problems based on the conceptual design and PLC programming for mechatronic systems (e.g. automatic car wash, bank ATM machines, railway crossing systems, etc). During classroom sessions, different mechatronic devices such as electronically controlled hydraulic and pneumatic valves, photo-electric switches, proximity sensors, accelerometers, and electronic fuel injectors are inspected by students to view the practical applications of mechatronic devices. Students are also required to demonstrate continual progress on their design projects by presenting activities related to various sensors, actuators, project planning, cost estimates, and team accomplishments.

Laboratory Experiments

The laboratory experiments have been designed and created by students enrolled in past course offerings to offer hands-on experiences of electrical, hydraulic, mechanical, and pneumatic systems. The laboratory is scheduled for three hours weekly for student teams of 3 to 4 individuals per station. A laboratory manual (Wagner, 2011) which describes the laboratory experiments is provided to guide students when conducting the experiments and to focus their attention on the learning objectives. Fig. 3.2 provides a list of the experimental topics covered during the laboratory sessions. A variety of different software packages such as LabVIEW, Matlab/Simulink, RS Logix 500, and Solid Works are used for these investigations. Students learn to integrate different sensors, actuators, hardware, and software into the experiments which offer challenging hands-on experiences. These endeavors prepare students to better serve industry needs.

No.	Experiment Name
1	Electronic Dice Circuit
2	Rotation Counter Circuit
3	Introduction to Ladder Logic
4	Allen Bradley PLCs & RSLogix500
5	Traffic Light Experiment
6	Introduction to Staubli Robot Arm
7	Staubli Robot Arm & integrated PLCs
8	Control of Pneumatic Actuator
9	Conveyor Material Handling System
10	Torsional & Swinging Pendulums

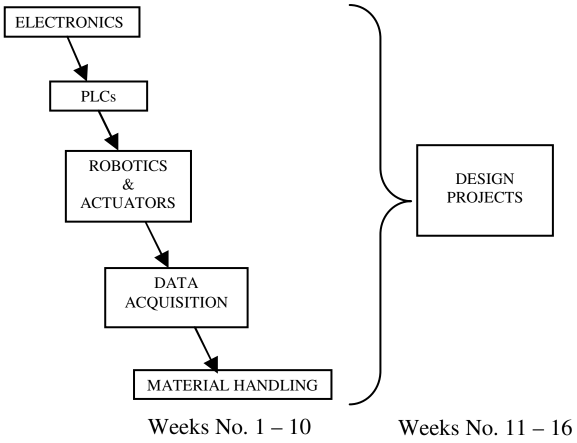

Fig. 3.2: Mechatronics laboratory activities and associated time frame

As part of the laboratory experiments, the teams are required to integrate two PLCs with a Staubli robot to complete process cycles. Students create algorithms for the robot to perform “pick and place” operations to assemble a connecting rod, piston, and wrist pin on a fixture. After this activity, the assembly must be transported on a conveyor system from one location to another with the help of proximity sensors. To coordinate the robot movements, the students are required to store robot arm positions using the teaching pendent. These stored positions are sequentially retrieved in the robot program. Once students successfully complete this task, they are requested to integrate the conveyor system controlled by two networked PLCs. The first PLC coordinates information with the Staubli robot while the second PLC collects conveyor operational data including the part color, barcode number on the storage box, and storage box progression along the conveyor. Since the two PLCs are networked together, they share this information to perform different operations according to the loaded ladder logic written in the PLC program. In the next laboratory session, students have to combine earlier explained laboratory session activities to integrate the robot arm movements with PLC program commands. It has been observed that the students enjoy working on robot programming and coordinating it with the PLCs to complete different processes. This enthusiasm was helpful to promote engagement for student learning, persistence, and success. Students also suggested increasing their laboratory session duration time so that they could undertake more experiments.

Team Based Design Projects

Team based design projects have been introduced to encourage students to synthesize the classroom and laboratory concepts throughout the semester by focusing on a single comprehensive engineering challenge in the design of a mechatronic system. Students need to apply the knowledge they gained in the classroom and laboratory to complete their design projects as shown in Fig. 3.2. The design projects require critical thinking by students while working on collaborative design tasks, project planning, team management, and material procurement. The students also need to complete documentation for their project which is often neglected in the workplace. Further, Clemson University is committed to “writing across the curriculum” to improve students’ technical communication skills. To develop leadership skills among students, team leaders are selected by each team to guide their efforts. The team leader has the responsibility of coordinating the different tasks for the project, communicating with the course instructor and laboratory teaching assistant, and ensuring completion of the project within the given time period. Weekly meetings of the team leaders with the instructor are necessary to complete the projects within a semester. Every team is required to evaluate different sensors, actuators, and electronic devices that may need to be purchased. They subsequently submit a procurement request to the instructor. Progress update presentations are given in class during the semester. The following list of projects show the contributions made by the student design projects to the mechatronics laboratory experiments (Bassily *et al.*, 2007 and Trey *et al.*, 2009). Some of the student design projects are shown in Fig. 3.3.

a) Conveyor System Design: Students designed a modular conveyor system with individual smart rollers and assorted sensors to operate under networked PLCs control.

b) Hydraulic and Pneumatic System Integration: Using National Instruments hardware and software for data acquisition, students integrated hydraulic and pneumatic components together to perform assigned tasks.

c) Library of Electronic Circuits: Different types of small electronic circuits were developed using bread boards. Some of these electronic circuits, including the electronic dice and rotation counter, are mentioned in Fig. 3.2.

d) Staubli Robot Programming: Students developed programs for the robot to pick and place objects and transport them on the conveyor system. They developed programs to coordinate the Staubli robot with the PLCs to start and stop the conveyors when required.

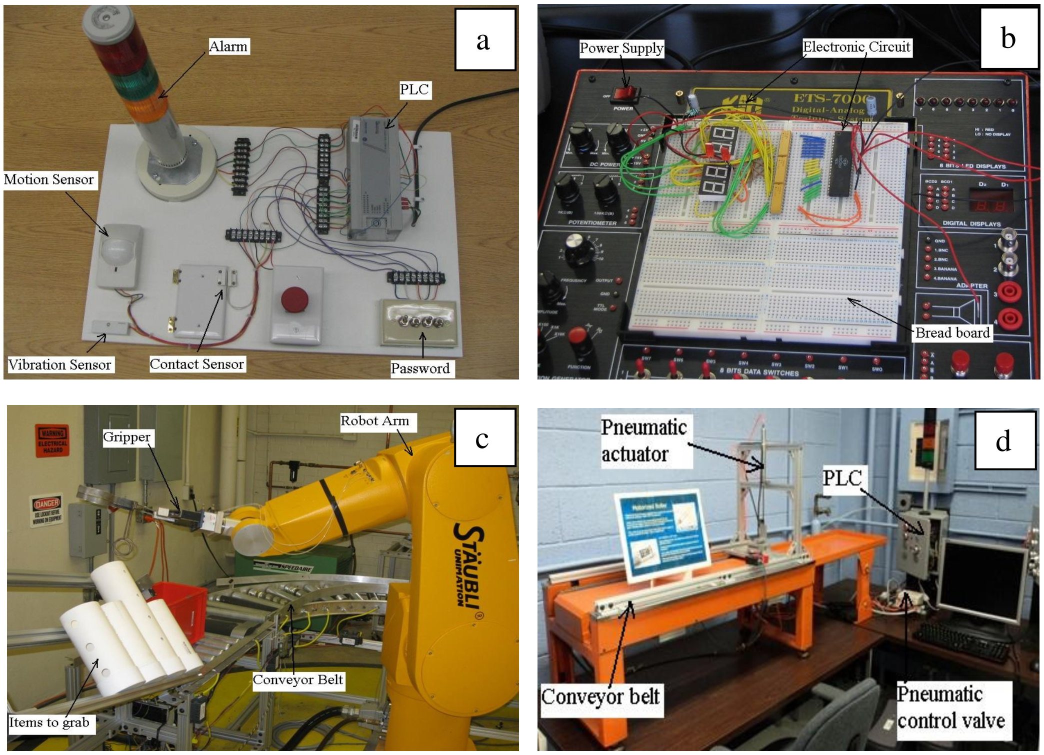

Fig. 3.3: Assortment of past and present mechatronics laboratory experiments – (a) security system with the PLC, (b) bread board electronic circuit with timing chip and digital display, (c) Staubli robot arm with part pick and place operation, and (d) bench top conveyor belt system with pneumatic actuation

Plant Tours Showcasing Manufacturing Technologies

The plant tour is an important aspect of this mechatronic course and provides the students an opportunity to view and understand mechatronic system applications in manufacturing environments. The students view different sensors, actuators, robotic assemblies, PLC controlled systems, product assembly lines, automated part storage and transport systems, testing facilities, etc. Further, they can directly observe the applications of human factors, and human-machine interactions. For many students who haven't toured plants before, these trips offer them motivation to consider working in the mechatronics field. Instructors and students have toured industrial companies such as BMW Assembly Plant (Greer, SC), Michelin Tire Plant (Sandy Springs, SC), Bad Creek

Pumped Storage Station (Salem, SC), Duke Oconee Nuclear Power Plant (Seneca, SC), Santee Cooper (Abbeyville, SC), and Advanced Automation (Greenville, SC). Students learn the importance of project planning, leadership skills, integration of different mechatronic systems, clear communication between project team members, and necessity of multi-disciplinary study.

3.3 Course Evaluation Using Assessment Data

The assessment for this mechatronic course has been performed in three manners to evaluate student learning and course structure. The first assessment method is called Pre-Test and Post-Test where data gathered before implementation of an activity (starting of the course) and after implementation of an activity (at the end of the course) are compared to determine how the outcome has changed. The second method is Qualitative Assessment where the opinion of a person (i.e., student) who just performed an activity is recorded to evaluate the effect caused by the task. The last assessment method is collection of feedback from a Technical Advisory Panel which contains industry experts and faculty members to evaluate the progress of the mechatronic course. Together, the assessment data allows analysis so that necessary actions can be taken to improve the course.

Pre and Post-Course Test Results

The pre/post-test is designed to assess the knowledge gained from classroom activities and assignments. It has twenty one items assessing mechatronic systems, collaborative learning, and team building methodologies. Results from the questions are presented on a 5 point scale from 1- indicating not correct to 5 - indicating completely accurate. Table 3.1 shows the four learning goals. Personal growth targets individual knowledge gained by the student, team building focuses on team performance, mechanics/engineering targets specific engineering disciplines like controls, electronics, and mechanics for student's knowledge gain, and human factors focuses on the industrial work perspective gained by students. The results in Table 3.1 show that there has been growth in the performance of students over each semester. The pre-test scores indicate that students enrolling in the course were deficient in the four learning goals. The post-test scores indicate that students performed well in the mechatronics course after going through the classroom and laboratory activities. The standard deviations (SD) for the post-tests are observed to be low, which indicates that most students have gained knowledge through the mechatronic course over the period of time.

Table 3.1: Pre-test and post-test means and standard deviations (SD) for four semesters

Learning Goal		Personal Growth	Team Building	Mechanics/ Engineerin	Human Factors
Fall 2008	Pre-Test Mean (SD)	3.23 (0.50)	4.34 (0.41)	2.89 (0.85)	3.38 (1.71)
	Post-Test Mean (SD)	3.60 (0.52)	5.00 (0.00)	4.61 (0.23)	4.92 (0.28)
Fall 2009	Pre-Test Mean (SD)	2.99 (0.42)	3.88 (0.75)	2.38 (0.72)	4.05 (1.62)
	Post-Test Mean (SD)	3.57 (0.46)	4.73 (0.36)	3.78 (0.52)	4.68 (0.95)
Spring 2010	Pre-Test Mean (SD)	3.18 (0.52)	4.03 (0.77)	2.56 (0.79)	3.74 (1.63)
	Post-Test Mean (SD)	3.39 (0.51)	4.68 (0.44)	3.64 (0.51)	4.52 (1.2)
Spring 2011	Pre-test Mean (SD)	3.52 (0.48)	4.03 (0.51)	2.66 (0.66)	4.56 (1.28)
	Post-test Mean (SD)	3.77 (0.38)	4.44 (0.48)	3.52 (0.53)	5.00 (0.00)

To validate the pre-test and post-test statistics, a within subjects or repeated measures approach has been selected. This research project calculated the ‘F’ scores as the ratio of two variances which were calculated in Table 3.1 for four parameters (learning goals). The respective ‘F’ scores are listed in Table 3.2; the data show that the student’s knowledge of the course material had improved. The largest increase in knowledge was observed for Mechanics/Engineering. There was a significant increase ($p<0.05$) in the knowledge of students for each of the four learning goals when the post-test results are compared with the pre-test results.

Table 3.2: Quantitative student learning data for four semesters with ‘F’ scores and accompanying ‘p’ levels

Learning Goal	Personal Growth	Team Building	Mechanics/ Engineering	Human Factors
	F (p)	F (p)	F (p)	F (p)
Fall 2008	8.35 (0.014)	33.62 (0.000)	65.23 (0.000)	11.82 (0.005)
Fall 2009	24.77 (0.000)	29.50 (0.000)	140.61 (0.000)	3.40 (0.080)
Spring 2010	6.98 (0.015)	14.68 (0.001)	48.18 (0.000)	7.39 (0.013)
Spring 2011	8.54 (0.002)	15.79 (0.001)	55.82 (0.000)	3.27 (0.083)

Qualitative Assessment by Students

The qualitative assessment of the course was completed by assessing student feedback to supplement the quantitative assessment results. In the qualitative assessment, students were asked about what they liked in the course, the instruction methods, and their recommendations for the future offerings of the mechatronic systems course. Similar to the previous assessment, the qualitative assessment was completed near the beginning of the semester and at the conclusion of the course to evaluate student perceptions regarding the mechatronic course as a whole. Table 3.3 lists the student likes, dislikes, and recommendations for the pre- and post-qualitative assessments. Students generally liked the hands-on approach to learning and suggested adding extra sample problems in the notes to help them solve the home work problems. However, several students disliked the workload or difficulty level of the course.

Table 3.3: Qualitative student comments

Assessment	Student Likes	Student Dislikes	Suggested
Pre	• Class notes are clear and informative • Instructor has a good understanding of the material. • Use of pictures/videos of related topics • Hands-on approach and interaction is helpful • Discussion of real life applications • Potential for plant/factory visits • Instructor enthusiasm for the material • Course merges different areas of engineering • Interactive • Course keeps attention and focus is well-structured	• Material goes too fast • Complexity of the some material • Homework difficulty • Not all topics received adequate attention • Not enough examples in class • Class time too short • Projects are intimidating and extensive, like a capstone project	• Cover less material in more depth • Spend more time on the notes • Slower communication • More in-depth talk about the homework • Work through more examples • Stress the important topics

Post	• Lab goes well with class • Instructor has a lot of energy which makes the class exciting • Hands-on application of the systems in the lab and field trip • Good class notes • Real-life examples • Good communication skills • Interesting discussions • In-class problems • "Show and Tell" with mechatronic components • Appropriate material level of background • Availability of instructor • Teaching style and willingness to help • Homework assesses knowledge/ understanding	• Homework does not assess knowledge – not connected to the class material • Project is time-consuming • More examples • Difficulty of the homework • Pages in the notes aren't numbered • A lot of information to learn • More instruction on what to expect on tests	

3.3 Technical Advisory Panel Observations

The technical advisory panel (TAP) consists of the NSF grant investigators, external industry experts, and selected faculty members. TAP assesses the overall learning objectives of the mechatronic course, the progress of students, academic course material improvements, actual applications of student projects, future laboratory equipment requirements, possible industry equipment donations for academic purpose, software license requirements, etc. As part of the TAP assessment process, students present their completed projects to the TAP to demonstrate their knowledge and their approach to achieve the team objectives for their projects. Along with the student presentations, faculty members present the pre- and post-assessment data, progress made by students, and difficulties of students and faculty in delivering the classroom materials and laboratory experiments. Some of the most challenging aspects of the mechatronics laboratory include software license renewals, new software and hardware procurements, proper maintenance of equipment, and industry sponsors. TAP suggestions included attention to practical issues in the workplace while completing the projects, and improvements in the laboratory. The observations and comments suggested by the TAP are presented in Table 3.4. Apart from the assessment by the TAP, the faculty also consults with industry experts to further resolve laboratory issues.

Table 3.4: Comments from the Technical Advisory Panel (TAP)

Laboratory Design Project	**Miscellaneous Feedback**
• Suggested improvements to HMI (human-machine interface) – process information, status bits, sensor status, number of parts processed, number of parts rejected, operating time, some robot information displayed	• Vision system with dice, ProE software to implement, simulation studies, robot works • "Done a good job, lot going on in a semester. Real challenge to find things to suggest." • "Team and project important aspects" (of this mechatronics course) • Recognized improvements to these courses at Clemson University and Greenville Technical College during the grant's four-year time period
• Write on text terminal in V++, network resource (Allen Bradley HMI; Factory Talk); Ethernet connection for PC version of Factory Talk; programming client, RSView 32, HMI platform - RSLogix 5000	• Realistic design projects are offered which would be encountered in industry • Researchers have taken panel's suggestions and comments to improve the courses • Go ahead for NSF Phase II proposal to partner with regional/national schools and companies

The mechatronics course has undergone continuous improvements based on the TAP suggestions as well as the analysis of the assessment results and student feedback. The course has benefitted by including new engineering topics in the course syllabus, through interactions with students about the in-class problems, and introducing new student projects to upgrade the laboratory experiments and student's laboratory manual.

3.4 Summary

The prevalence of mechatronics system design in manufacturing systems, consumer products, and a host of other engineered items have increased the need for engineering schools to offer mechatronic courses. The mechatronics course at Clemson University builds upon best practices for class room instruction, laboratory experiments designed by the students, and semester long projects to synthesize the mechatronic concepts. In this paper, the learning activities and assessment methods used in the mechatronics course have been presented and discussed. As part of the teaching methods, the classroom activities focus on fundamental engineering concepts while the laboratory tasks offer hands-on experiences with sensors, actuators, and different mechatronic systems. Semester long design projects prepare students to acquire critical people skills such as leadership, project management, and collaborative approaches while designing mechatronic systems. The industrial plant tours offer students first-hand insight into manufacturing facilities. The analysis of the course assessment data and feedback from the students plus technical advisory panel show that the mechatronic course development has been successful.

Acknowledgment**:** The authors would like to thank the National Science Foundation

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CHAPTER 4

CONCLUSION AND RECOMMENDATIONS

As technology advances, new possibilities exist for mechatronic devices in everyday applications. Mechatronic systems offer smart operation with greater convenience, cost efficiency, and reliability. As expected, mechatronic systems integrate the benefits of using mechanical, electronic, electrical, and computer disciplines into a single entity. One such mechatronic system, a hydraulic based engine cooling system, has been studied with comparison to the traditional method. In this study, the system utilizes a hydraulic motor to operate the radiator fan to take the advantage of higher power density values of hydraulic actuators. The system has been investigated to effectively maintain the engine temperature within $\pm 1.5°K$. The identified advantages over a traditional or an electric actuators based engine cooling system include reduction in the engine temperature variations from 12 ºC to 3ºC range, flexibility of placement around the engine, and minimization of energy consumption by engine accessories.

Along with this system, a technical elective ME417/617: Mechatronic Systems course, which has been offered by the Mechanical Engineering Department at Clemson University is described along with the improvements made in the course. To better equip the undergraduate and graduate engineering students for their future collaborative industry work environment, this course offers them the necessary hands-on experience of collaborative work environment coming through mechatronic systems. The course teaches students concepts from different engineering disciplines such as mechanical, electrical, computer, control and industrial engineering. Specially, the course allows students to gain hands-on experience of various sensors, actuators, data acquisition systems, and digital circuits through laboratory experiments. The course also requires students to design a mechatronic system using collaborative approach and team efforts. The assessment results for the course are also presented and analyzed to improve the course.

Overall, the thesis offers contributions in the modeling of a hydraulic based engine cooling system, and the refinement of a mechatronic system course which are described in the following section.

4.1 Contributions to Engine Cooling System and Mechatronics Course

The thesis offers the mathematical models of the electrical, hydraulic, thermal and mechanical systems which have been investigated to design a mechatronic system for a hydraulic based engine cooling system. The models provide a numerical solution to estimate the engine, and coolant temperatures for the engine operation time. The proposed hydraulic based engine cooling system also attempts to maintain the desired engine temperature by effectively utilizing the hydraulic motor operated radiator fan to remove excessive heat out of the engine. The simulation results along with the experimentally gathered results under the laboratory settings are compared for the proposed engine cooling system configuration. The steady state engine temperature errors of $\pm 0.5°K$ are observed for experimental engine testing when PID controller is used. These results are used to develop the mathematical models for hydraulic based engine cooling system in a simulation environment.

Apart from these results, the model has been able to estimate the state variables of the system, namely, temperatures of engine and coolant, current for valve controlling the hydraulic actuator, displacement and acceleration of valve spool, load pressure and rotation speed of hydraulic actuator. Also the model provides the approximate energy consumption by hydraulic actuators while maintaining the engine temperature. The model can be used for designing automotive engine cooling system where both the radiator fan and coolant pump are operated by hydraulic motors to achieve even better control for maintaining engine temperature and reducing accessory power consumption of the engine. While moving forward the simulated model can be further refined by considering the mass transport of the coolant and convective heat transfer between the various subsystems of the engine thermal system.

Along with the design and study of a hydraulic based engine cooling system, the thesis describes the various teaching and course assessment methods in a mechatronic systems course. The assessment results of the course show that students have demonstrated improvements in the personal growth (7.0%), team building (12.8%), mechanics/engineering (25.4%), and human factors (17%). These assessment outcomes show that the mechatronics course has offered students beneficial skills and knowledge required to successfully work in the industrial work environment. Also, the assessment and feedback by an industrial advisory panel ensured that industry needs are considered while developing the course and students will be successful in industrial work environment.

Some of the recommendations for future research related to hydraulic based engine cooling system and the mechatronic system course are listed in the next section.

4.2 Recommendations for Future Research

The recommendations to pursue future research on hydraulic based engine cooling systems and an academic mechatronics course are listed below.

Hydraulic Based Engine Cooling System

a) The mathematical equations presented in the thesis can be expressed in a nonlinear state space format which can be used with the controllers such as Linear Quadratic Regulator (LQR), and Lyapunov-based nonlinear controllers. They can optimize the system performance by reducing the costs of operations of different actuators.

b) The system can be integrated into a vehicle for field testing to analyze the actual system performance when both the radiator fan and coolant pump are driven by hydraulic motors directly coupled or decoupled to engine crankshaft.

c) The mechanical wax based thermostat could be replaced with an electric valve to further examine the performance of smart engine cooling systems.

d) The system can be evaluated for failure modes which are inherent with the hydraulic systems.

e) The hydraulic fluid could be replaced by a “green” fluid with redesigned valves, hydraulic pump, and hydraulic motors.

f) To get better temperature readings of coolant inside the engine block, a thermocouple could be placed inside the engine block’s coolant path.

Mechatronics Course and Assessment

a) To enhance student diversity, the enrollment of students from electronics, electrical, computer science, and industrial engineering could be increased through recruitment efforts.

b) To create a greater collaborative environment, industry sponsored projects of designing mechatronic devices and systems could be offered to the students.

c) More plant visits and tours which cover different types of industry plants can be arranged.

d) The laboratory experiments should be rotated and upgraded each semester to cater to the needs of diverse student backgrounds.

e) Different take-home experiments could be assigned to students which offer hands-on experience with sensors, actuators, etc.

f) Professional experts from various industries could be invited as guest lecturers who can share their industry experiences and address the future prospects of mechatronics in the industries.

APPENDICES

APPENDIX A: EXPERIMENTAL TESTING CONDITIONS

Appendix A lists the controller gains and engine testing cycle events for the different experimental tests as summarized in Tables A.1 – A.3.

Table A.1: PID controller gain values for experimental engine testing

Engine Test No.	**K_P**	**K_I**	**K_D**
1	0.706571	0.007939	0.277865
2	0.706571	0.007939	0.277865
3	0.706571	0.007939	0.277865
4	0.706571	0.007939	0.277865
5	0.706571	0.007939	0.277865
6	0.23817	0	0
7	0.23817	0	0
8	0.23817	0	0
9	0.23817	0	0
10	0.23817	0	0

Table A.2: Engine conditions for engine tests

Engine Test No.	**Event Starting Time, t (sec)**	**Engine Speed, ω_e (RPM)**	**Engine Testing Condition**		
			Idle	**Light Load**	***Ram Air Burst**
1	t_1=0	883	x	-	-
	t_2=2,003	1,935	-	x	-
	t_3=2,545	883	x	-	-
2	t_1=0	885	x	-	-
	t_2=2,323	2,308	-	x	-
	t_3=3,037	885	x	-	-
3	t_1=0	882	x	-	-
	t_2=2,081	2,222	-	x	-
4	t_1=0	885	x	-	-
	t_2=2,165	1,935	-	x	-
	t_3=2,263	885	x	-	-
	t_4=2,276	2,000	-	x	-
	t_5=2,689	885	x	-	-
5	t_1=0	896	x	-	-
	t_2=1,996	1,935	-	x	-
	t_3=2,538	896	x	-	-
	t_4=2,910	2,500	x	-	x
	t_5=3,003	2,500	-	x	x
	t_6=3,606	2,500	-	x	-

	t_7=3,617	896	x	-	-
6	t_1=0	888	x	-	-
	t_2=1,964	2,348	-	x	-
	t_3=2,331	888	x	-	-
	t_4=2,869	2,145	-	x	-
	t_5=3,050	888	x	-	-
7	t_1=0	889	x	-	-
	t_2=1,837	2,179	-	x	-
	t_3=2,025	889	x	-	-
	t_4=2,535	2,284	-	x	-
	t_5=2,938	889	x	-	-
8	t_1=0	895	x	-	-
	t_2=1262	2,500	-	x	-
	t_3=1342	895	x	-	-
	t_4=1743	2,308	-	x	-
	t_5=1848	895	x	-	-
	t_6=2041	2,222	-	x	-
	t_7=2383	895	x	-	-
	t_8=3085	2,000	-	x	-
	t_9=3300	895	x	-	-
9	t_1=0;	896	x	-	-
	t_2=2066	2,400	-	x	-
	t_3=2396	896	x	-	-
10	t_1=0;	870	x	-	-
	t_2=1666;	2,000	-	x	-
	t_3=2210;	870	x	-	-
	t_4=2330	2,000	-	x	-
	t_5=2473	870	x	-	-

*Ram Air Burst speed = 11.67 m/s.

For above mentioned engine tests, the various events happening over the testing time are shown with the zoomed in parts of the experimental testing graphs using stages shown in the following table.

Table A.3: Different engine test cycle events

Stage	**Engine Cycle Events**
I	Warm up
	Thermostat opening
	Coolant starts to flow through radiator
II	Temperature of coolant at radiator outlet rises
	Coolant flows continuously through radiator
	Temperature of coolant at radiator outlet goes near engine temperature
III	Engine, radiator, and coolant warms up completely
	Heat is dissipated to ambient by operating radiator fan or ram air burst on radiator to maintain the engine temperature

APPENDIX B: EXPERIMENTAL TEST RESULTS

The appendix B lists the experimental testing results (Fig. B.1 – Fig. B.50) for all the engine tests and conditions mentioned in Appendix A.

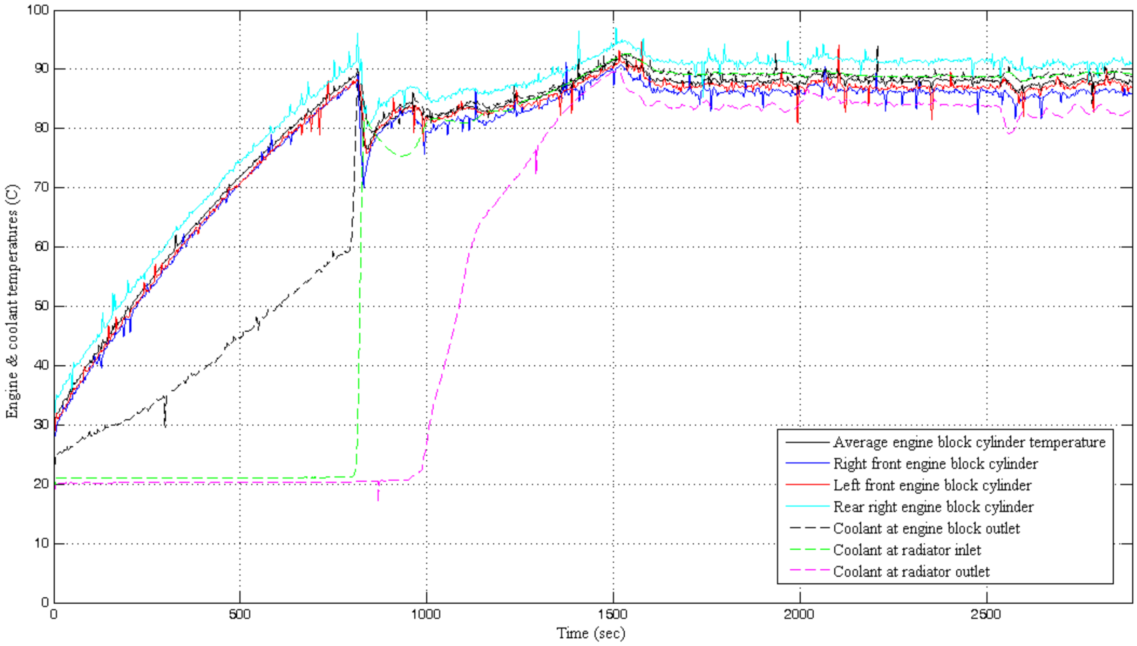


Fig. B.1: Temperatures of engine and coolant for experimental engine test no. 1

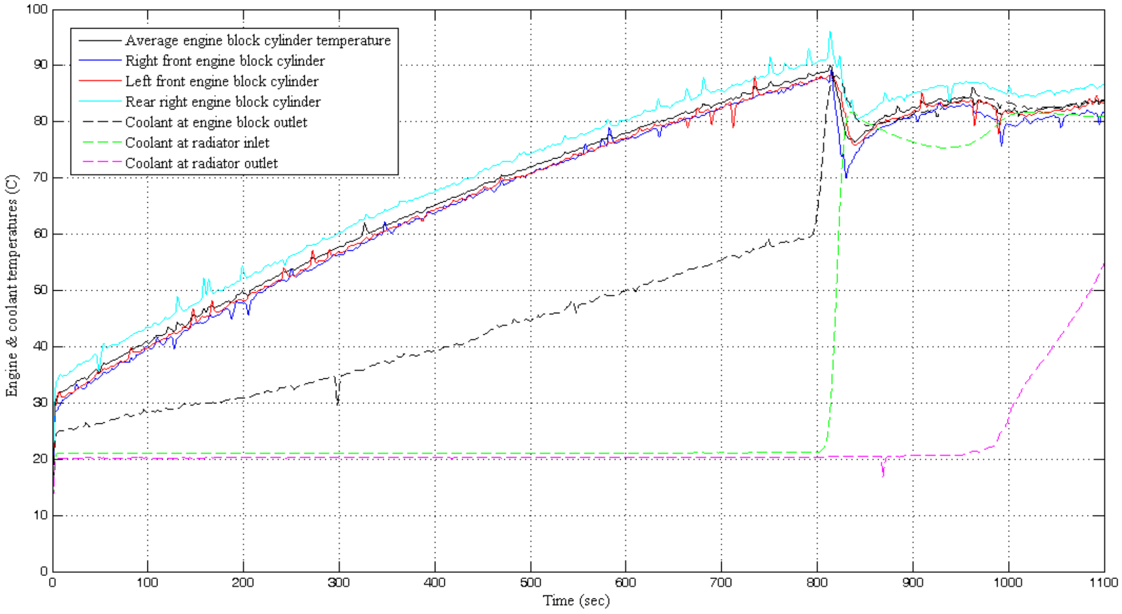


Fig. B.2: Engine test cycle, Stage I for test no. 1

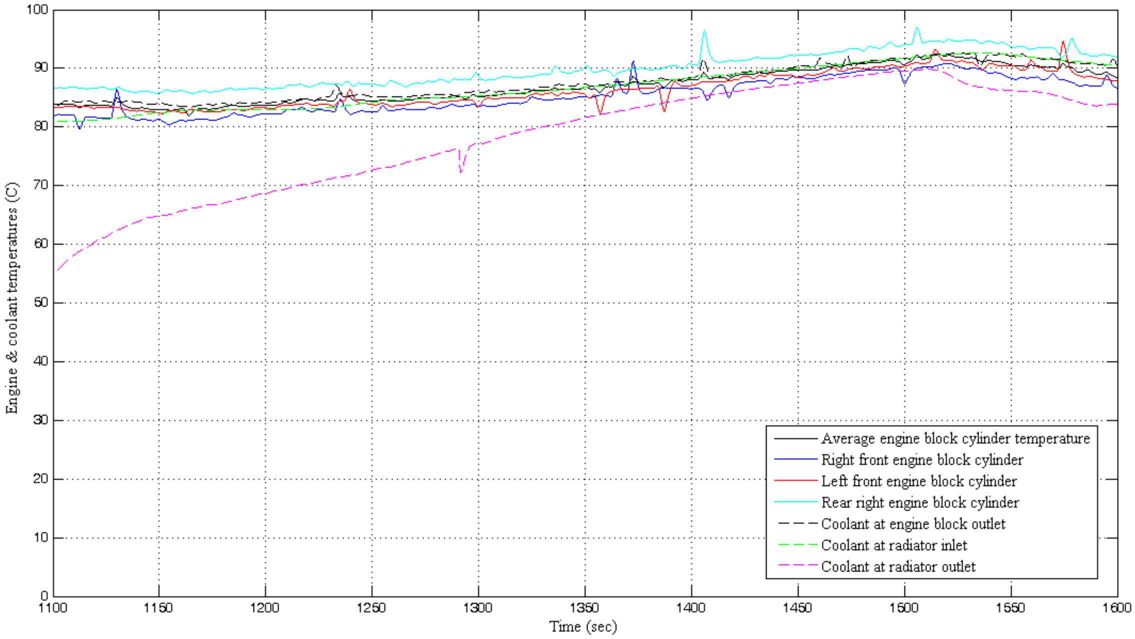


Fig. B.3: Engine test cycle, Stage II for test no. 1

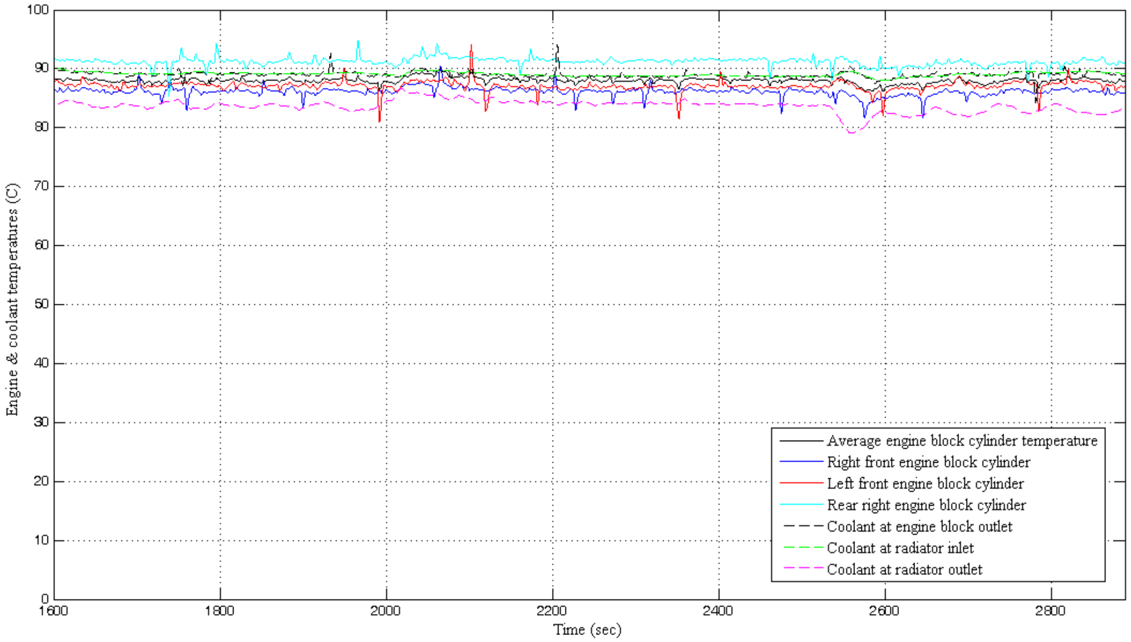


Fig. B.4: Engine test cycle, Stage III for test no. 1

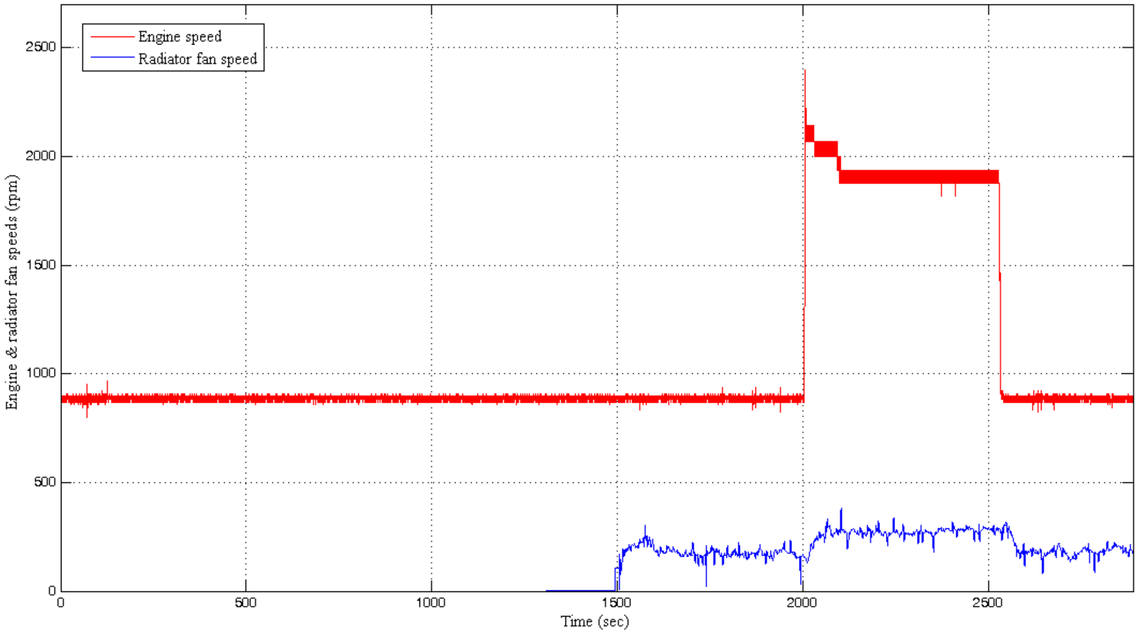


Fig. B.5: Engine and radiator fan speed for test no. 1

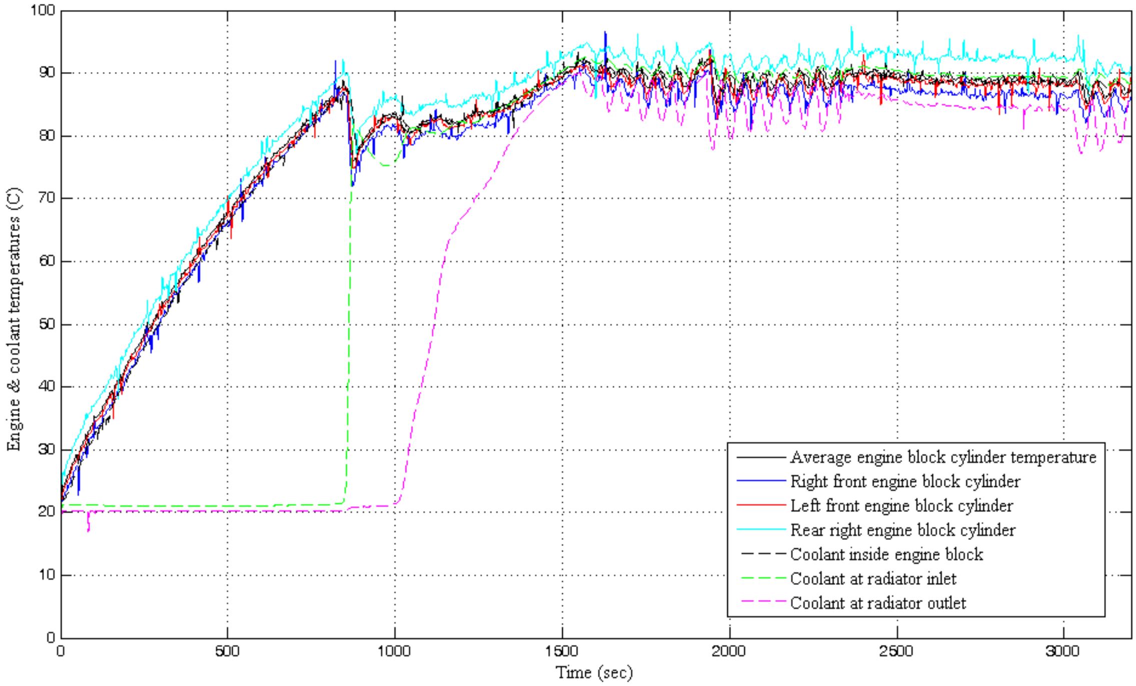


Fig. B.6: Temperatures of engine and coolant for experimental engine test no. 2

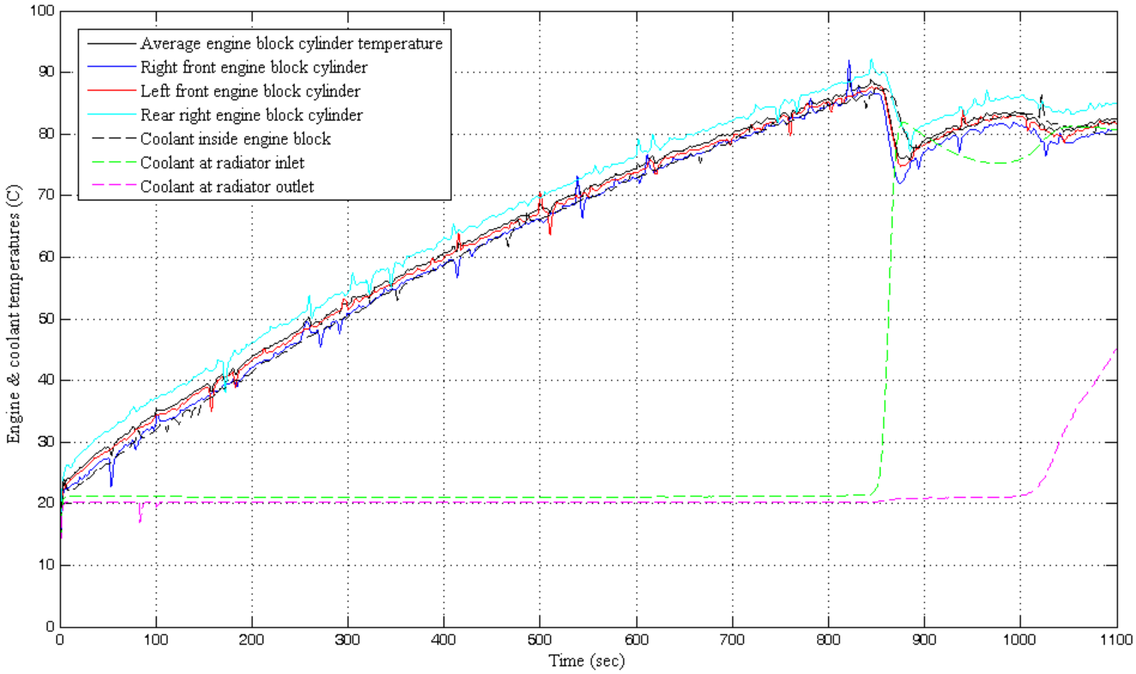


Fig. B.7: Engine test cycle, Stage I for test no. 2

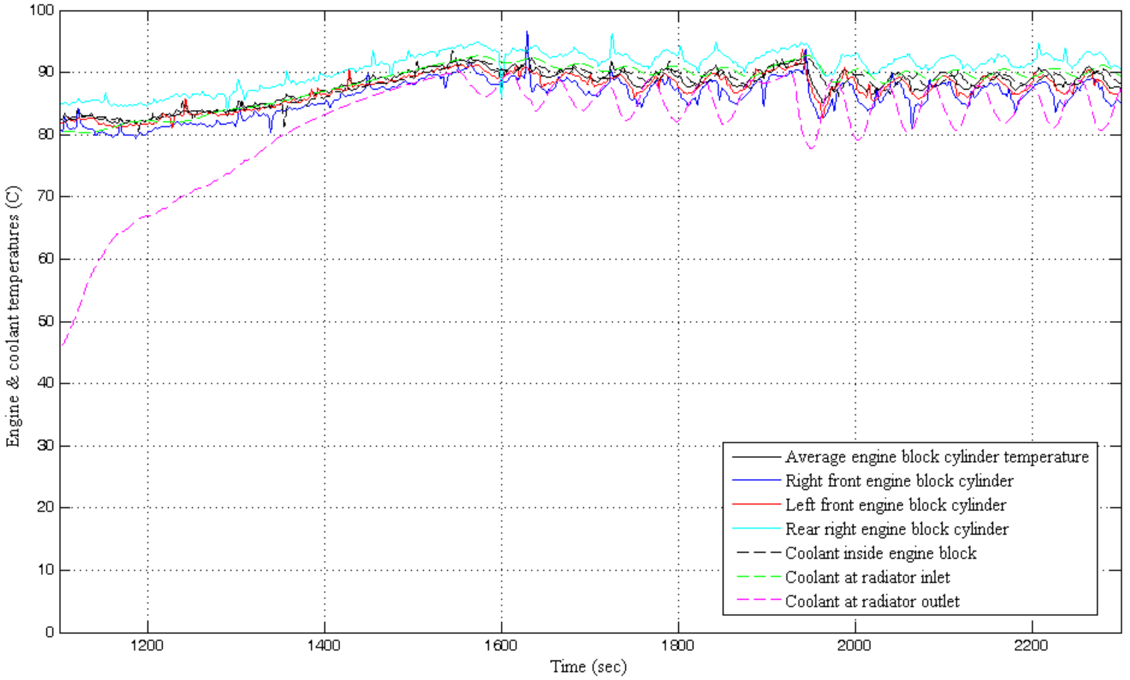


Fig. B.8: Engine test cycle, Stage II for test no. 2

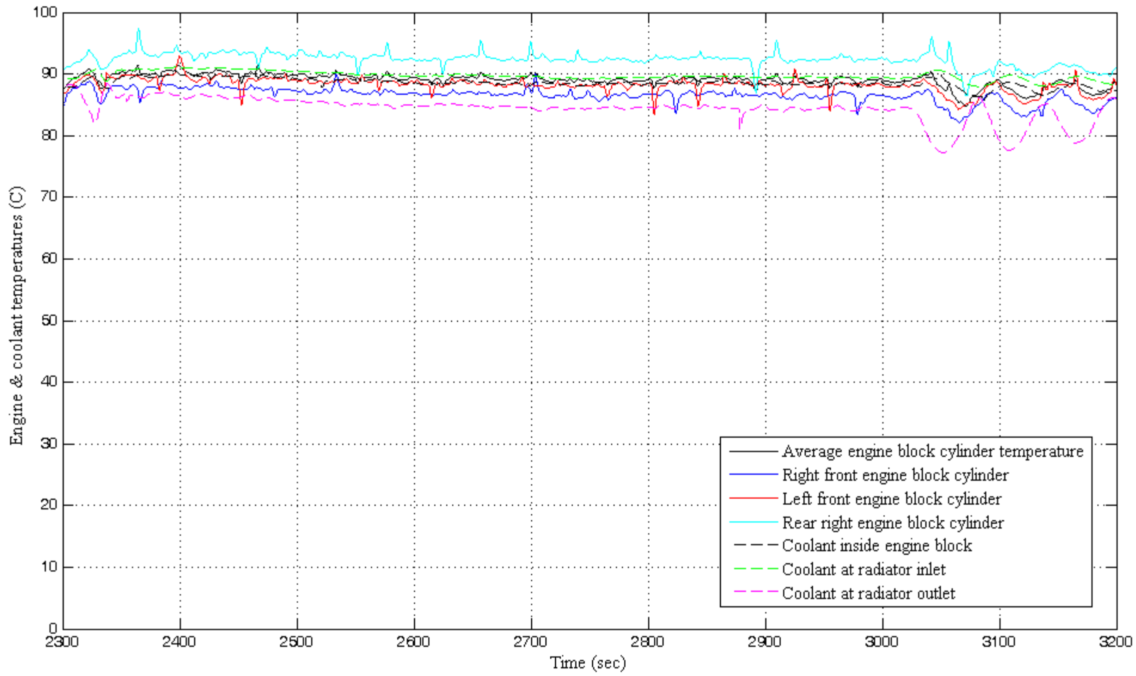


Fig. B.9: Engine test cycle, Stage III for test no. 2

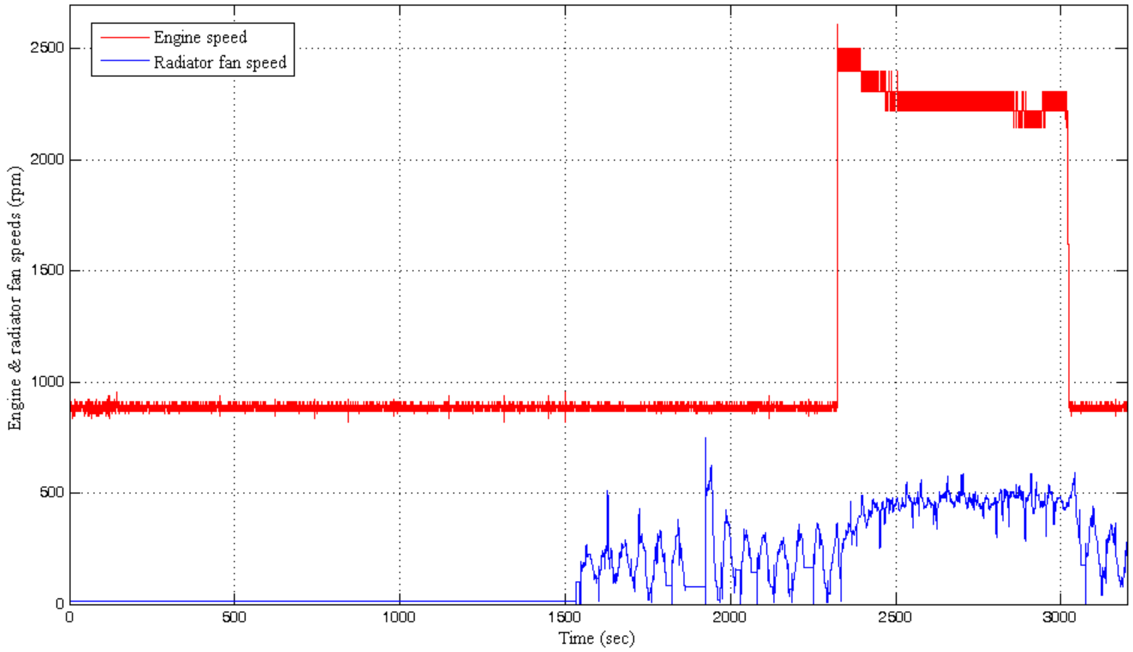


Fig. B.10: Engine and radiator fan speed for test no. 2

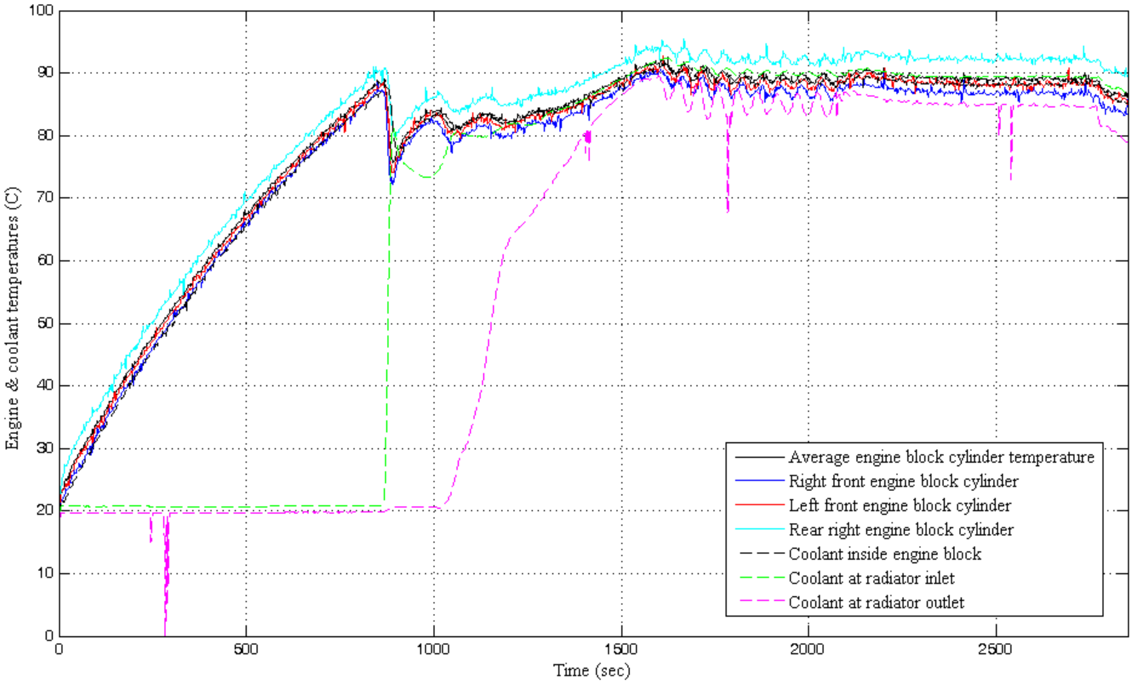


Fig. B.11: Temperatures of engine and coolant for experimental engine test no. 3

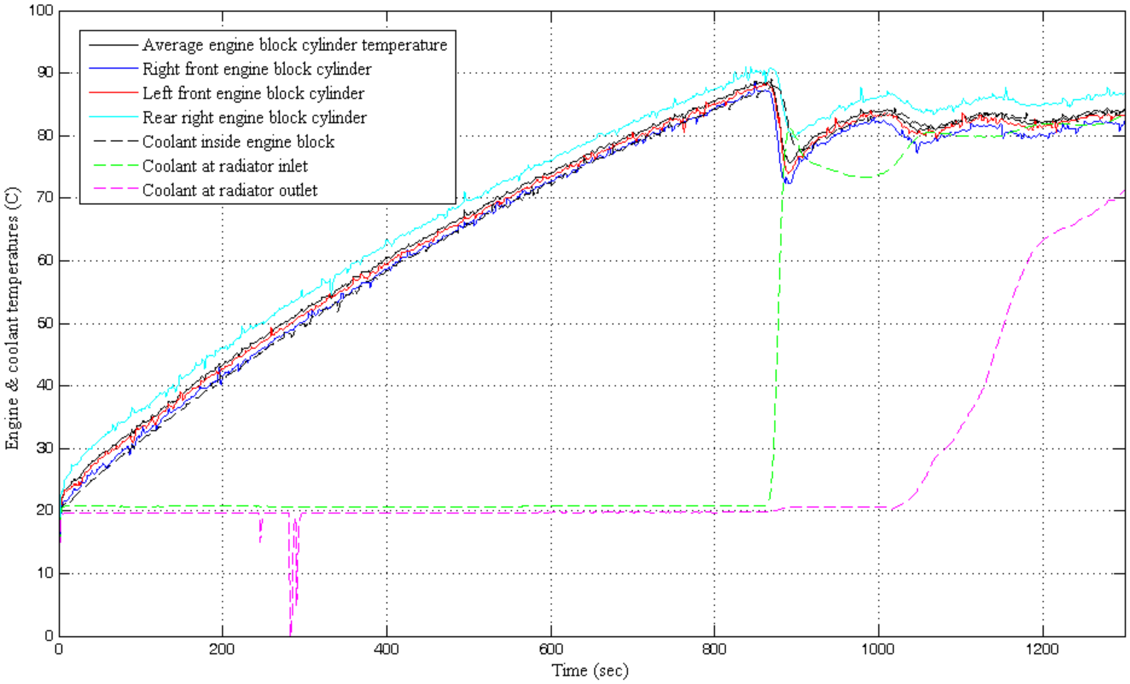


Fig. B.12: Engine test cycle, Stage I for test no. 3

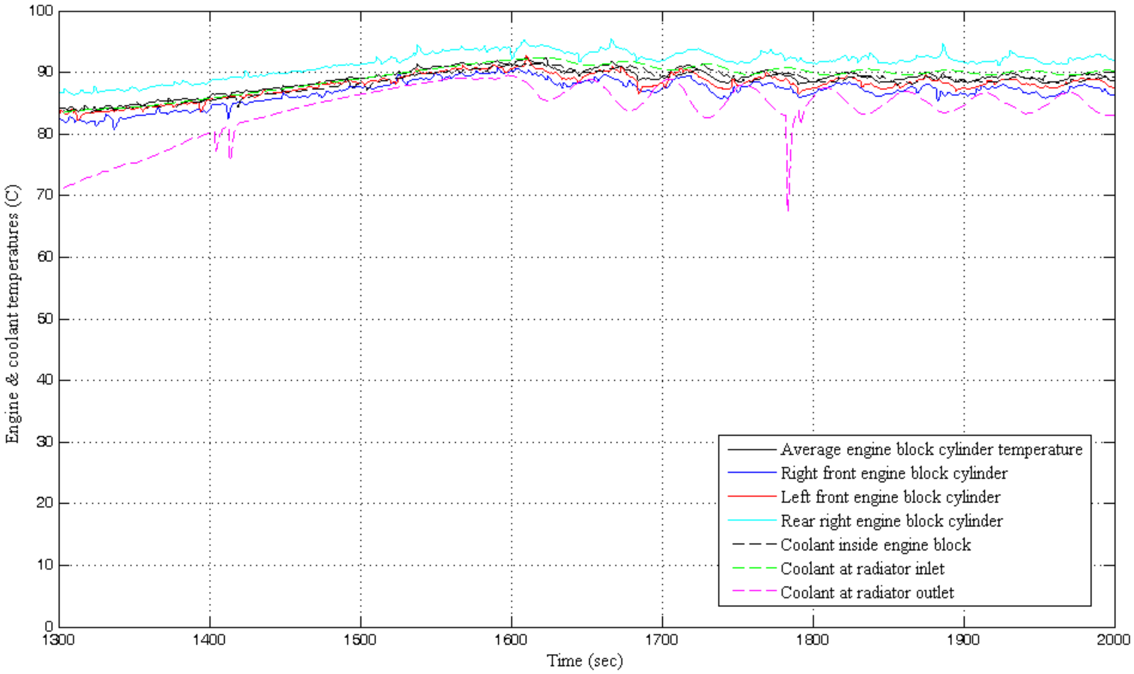


Fig. B.13: Engine test cycle, Stage II for test no. 3

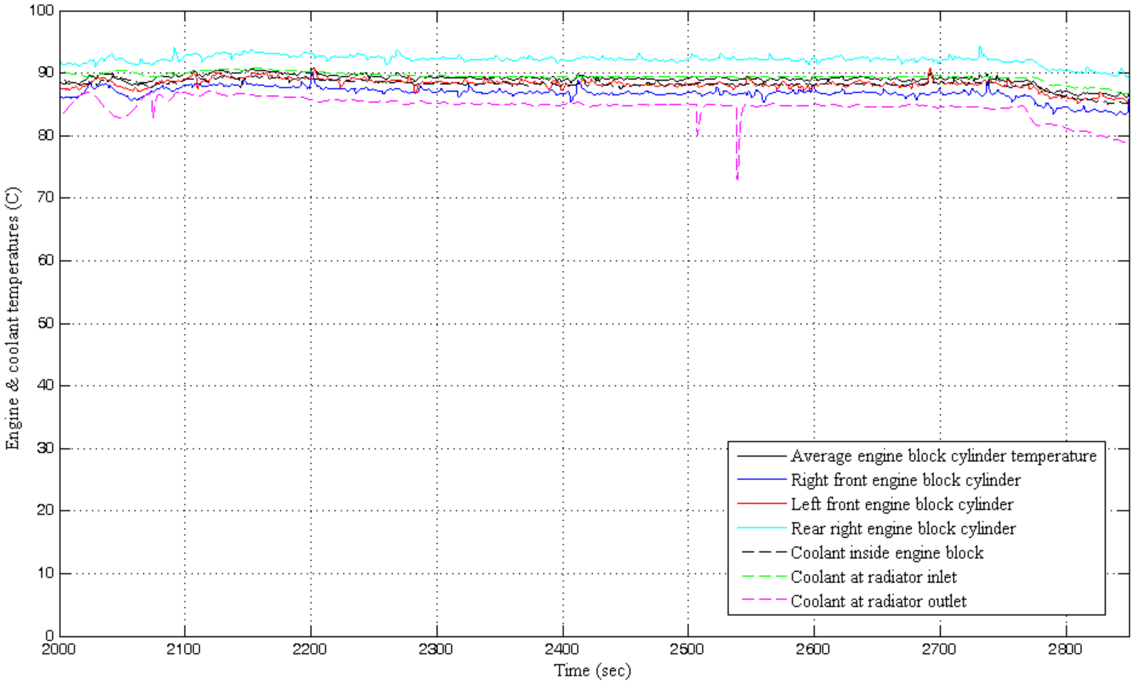


Fig. B.14: Engine test cycle, Stage III for test no. 3

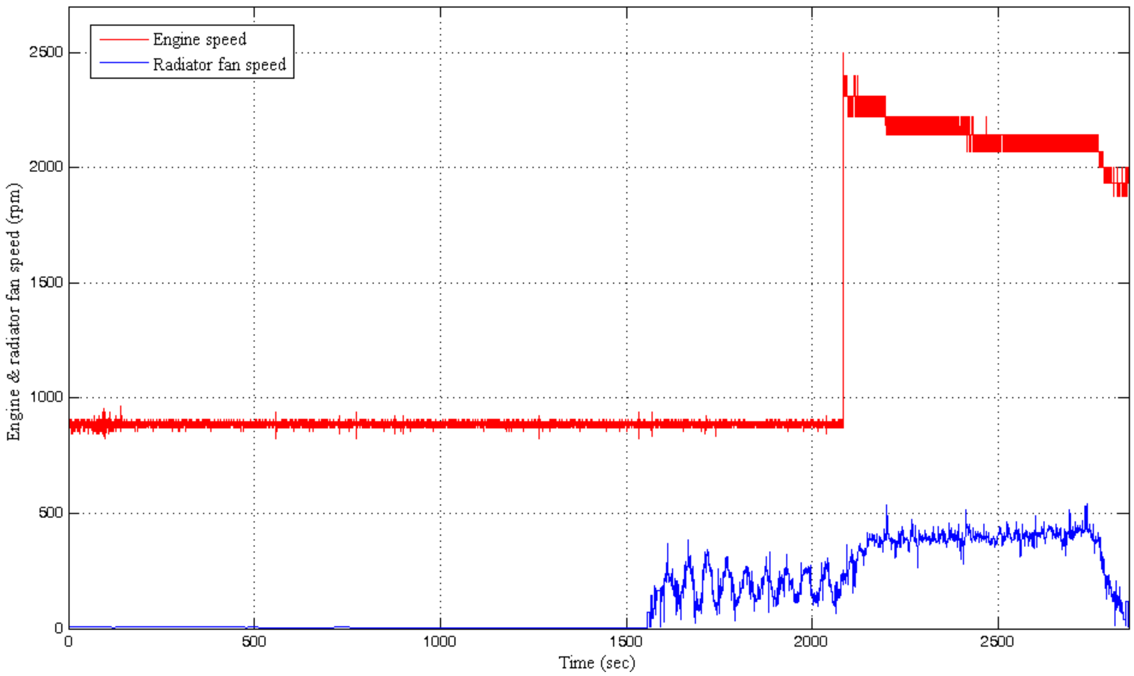


Fig. B.15: Engine and radiator fan speed for test no. 3

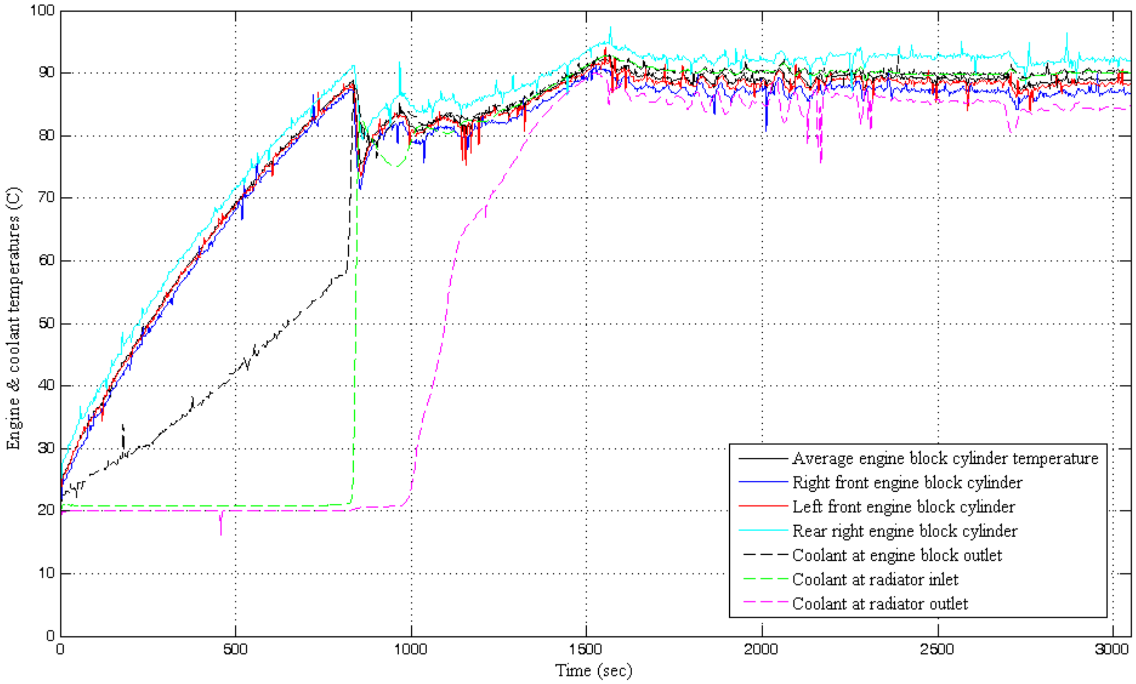


Fig. B.16: Temperatures of engine and coolant for experimental engine test no. 4

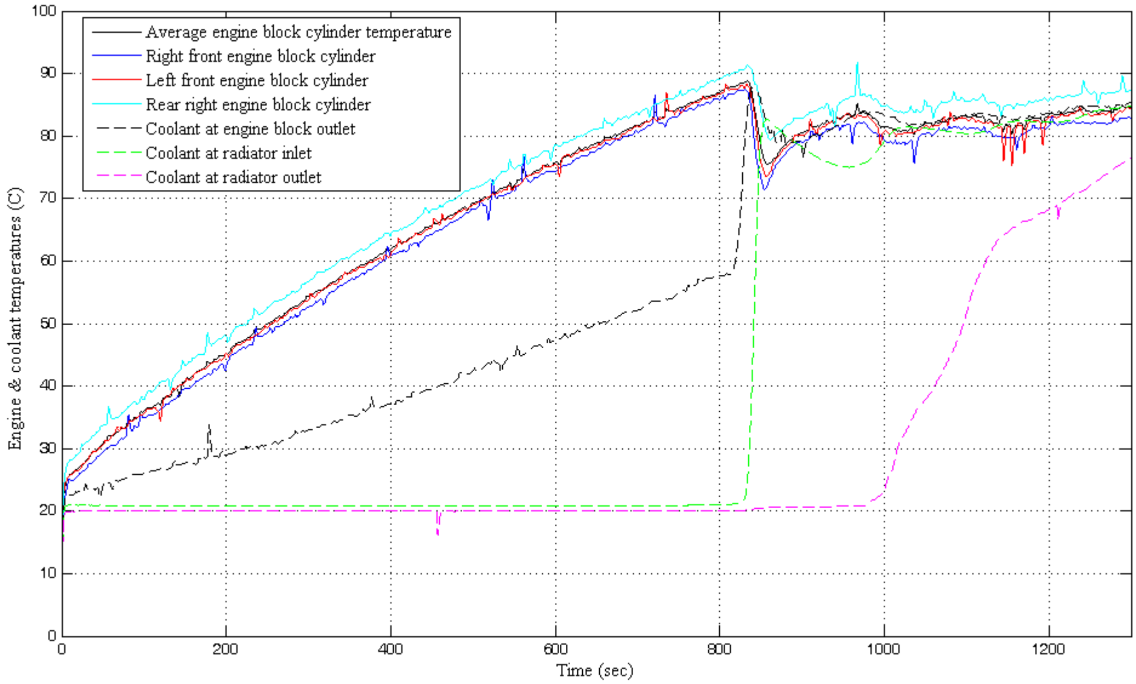


Fig. B.17: Engine test cycle, Stage I for test no. 4

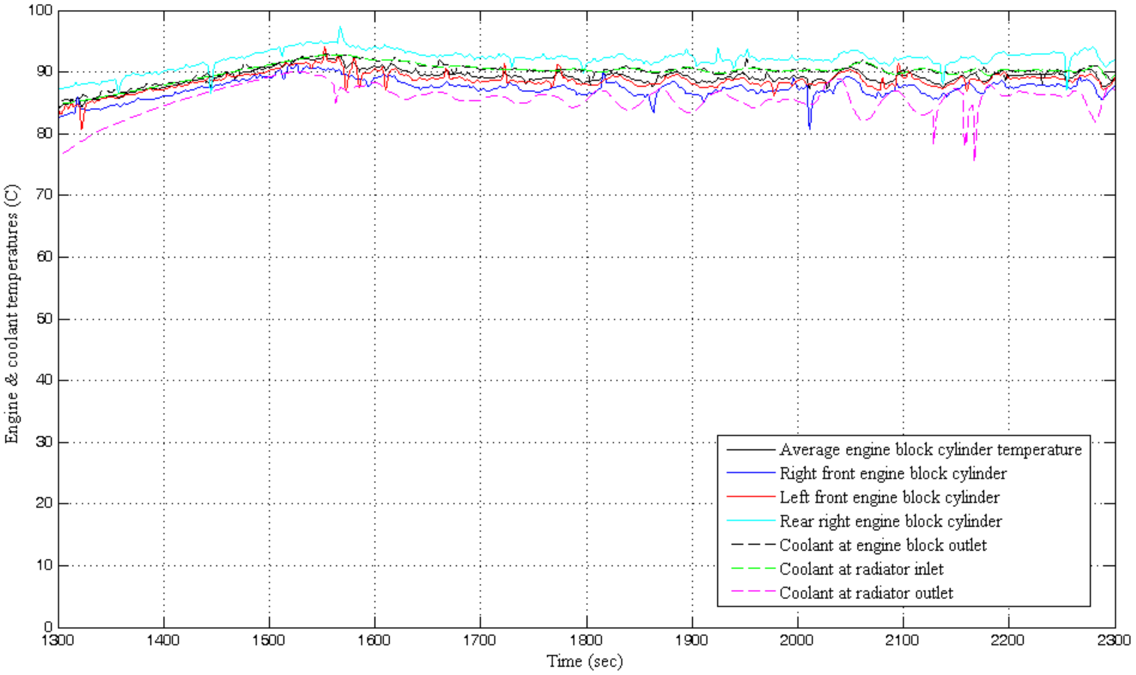


Fig. B.18: Engine test cycle, Stage II for test no. 4

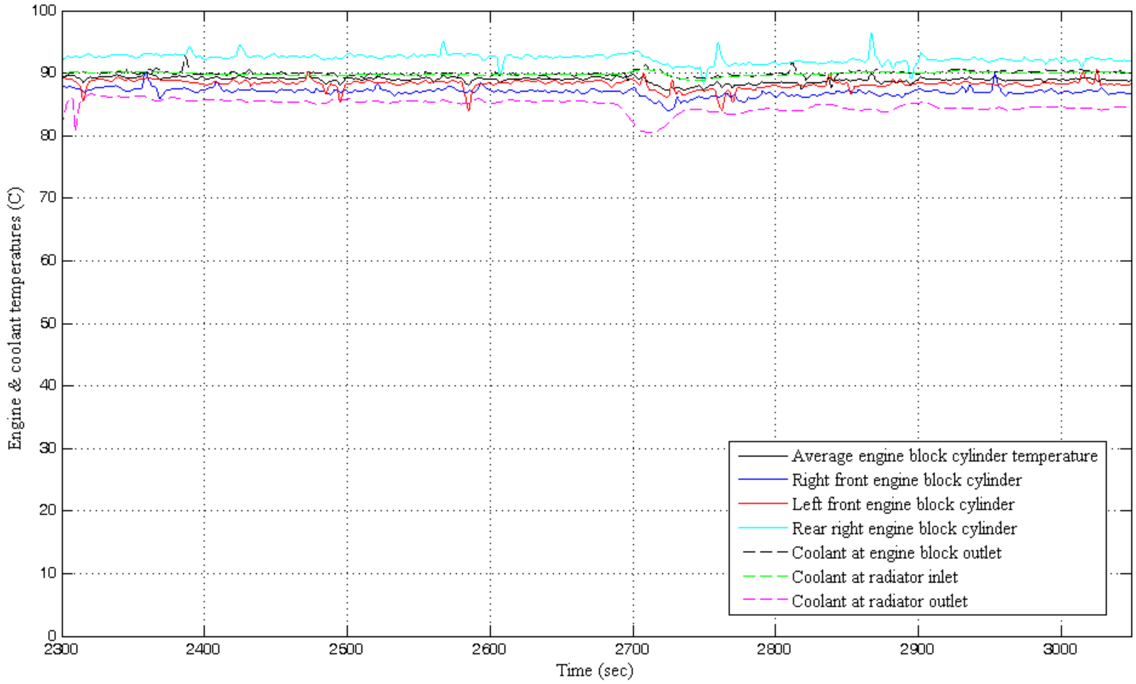


Fig. B.19: Engine test cycle, Stage III for test no. 4

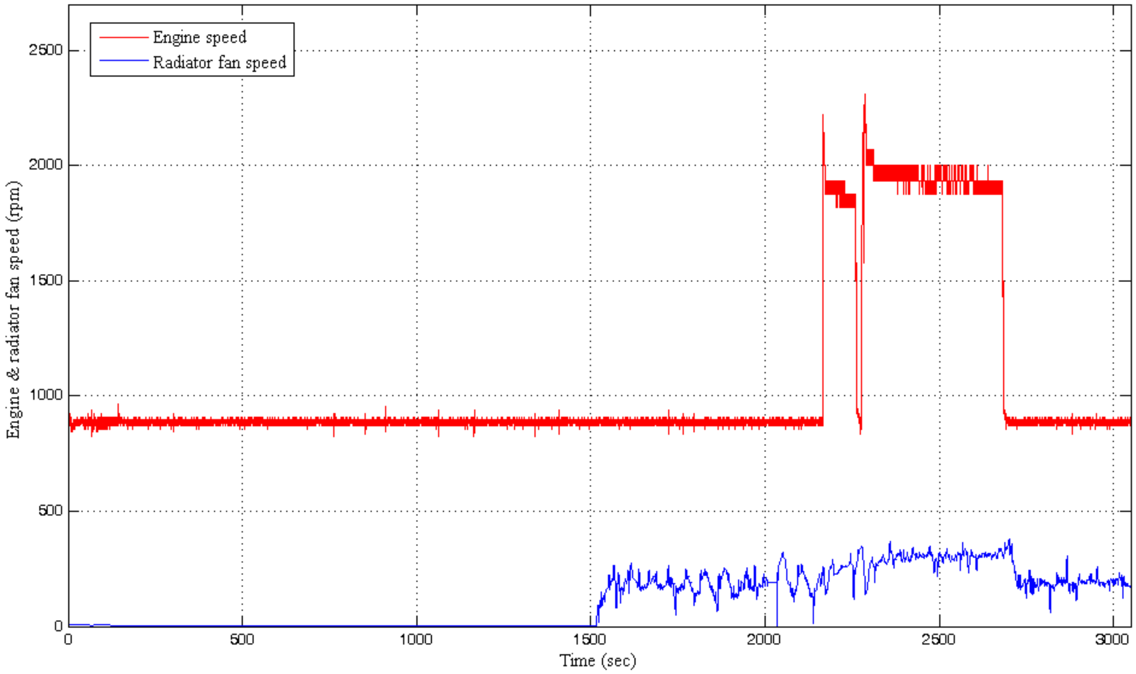


Fig. B.20: Engine and radiator fan speed for test no. 4

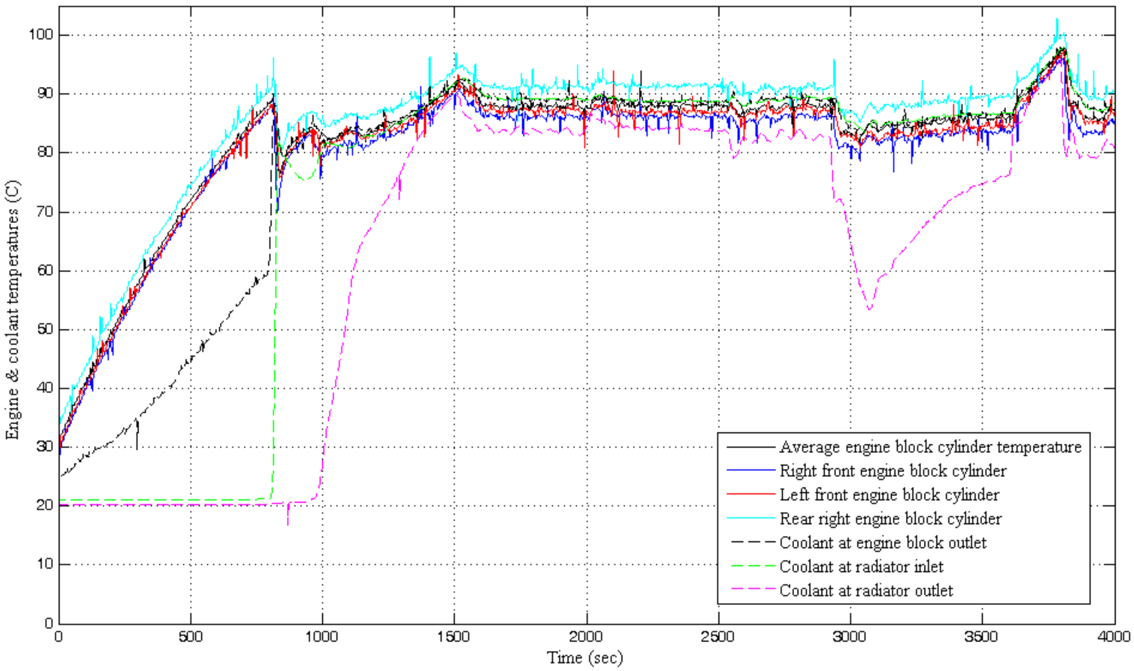


Fig. B.21: Temperatures of engine and coolant for experimental engine test no. 5

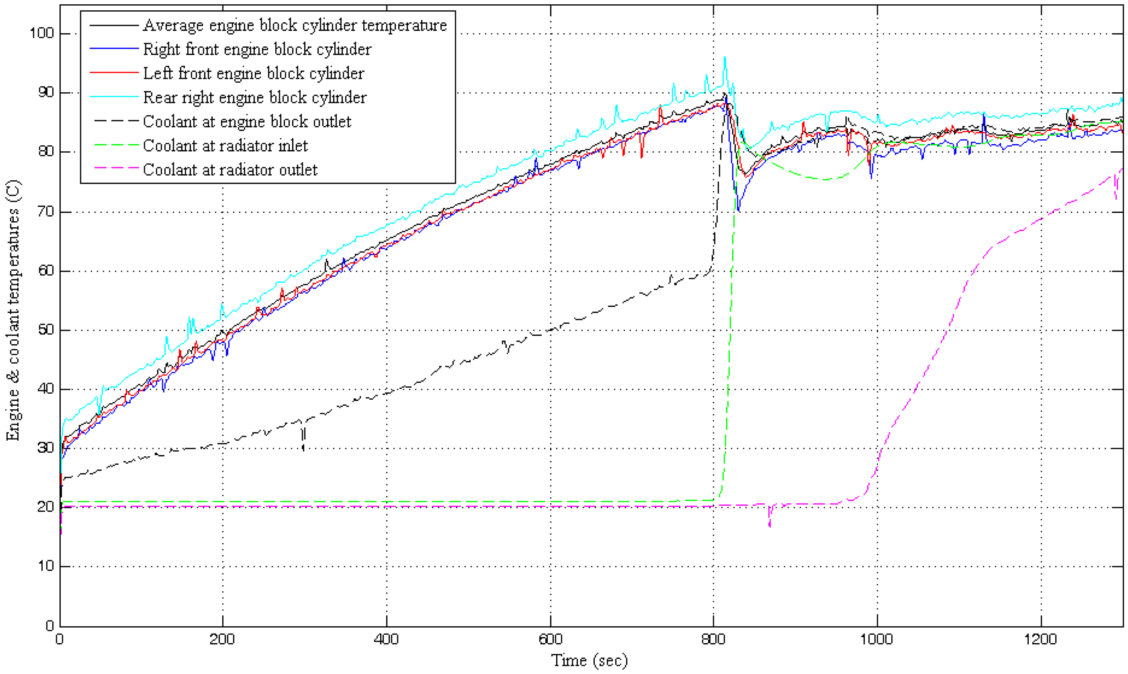


Fig. B.22: Engine test cycle, Stage I for test no. 5

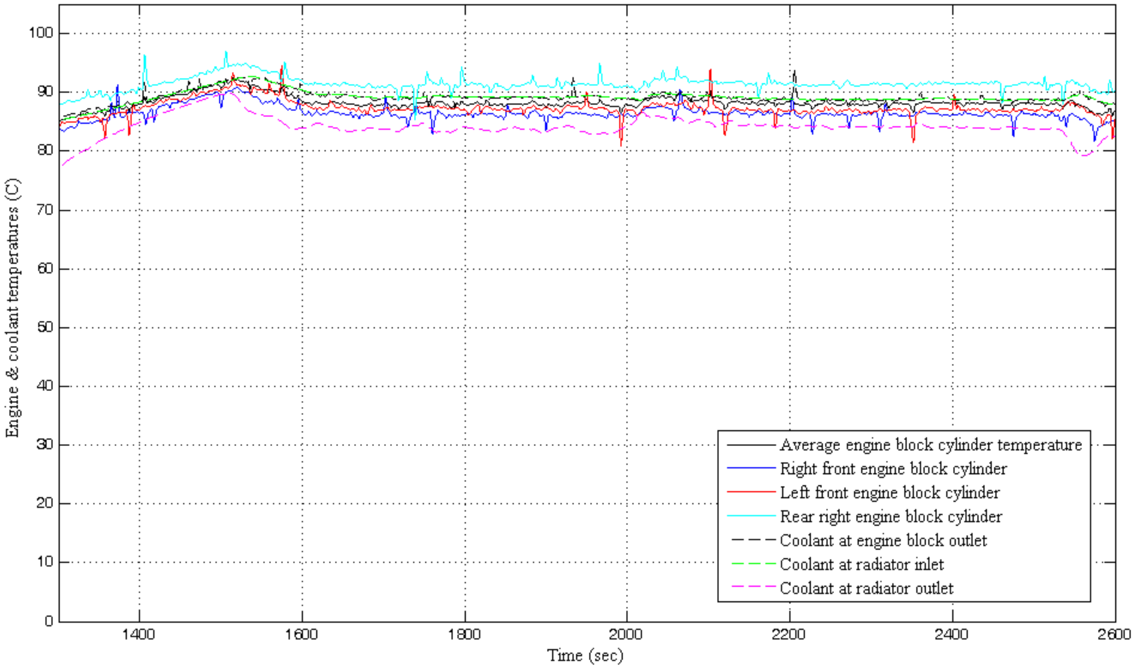


Fig. B.23: Engine test cycle, Stage II for test no. 5

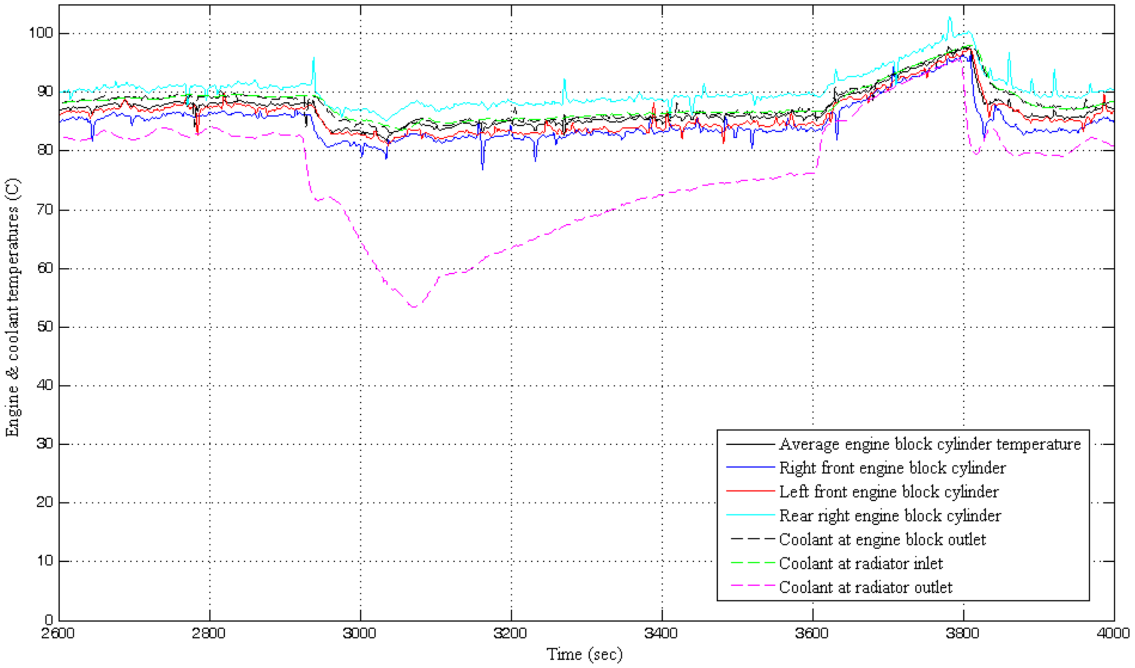


Fig. B.24: Engine test cycle, Stage III for test no. 5

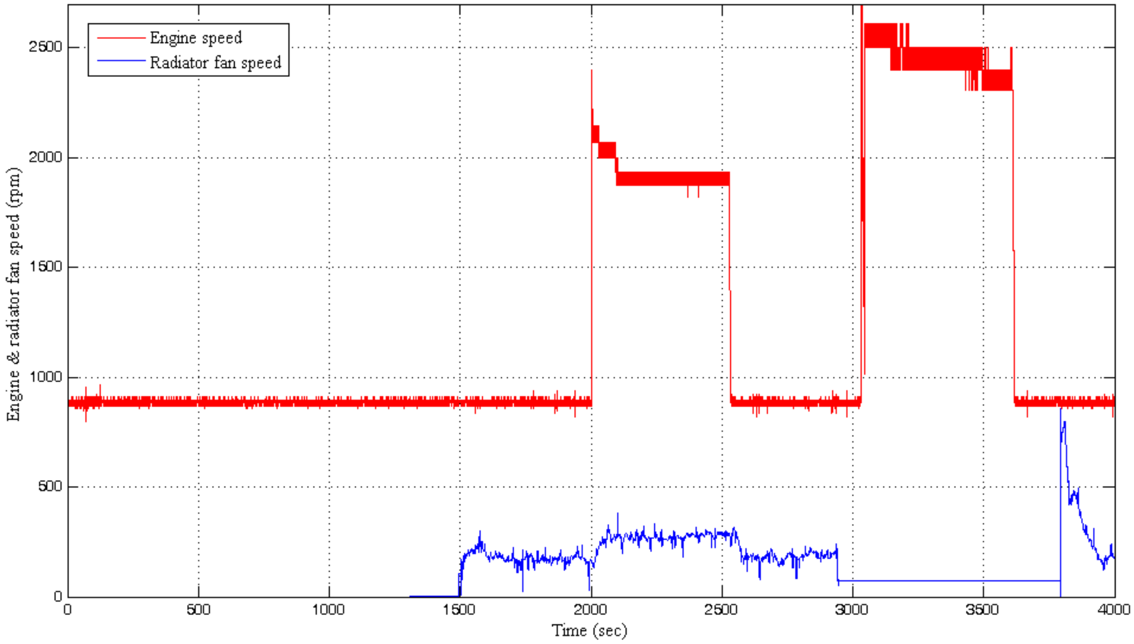


Fig. B.25: Engine and radiator fan speed for test no. 5

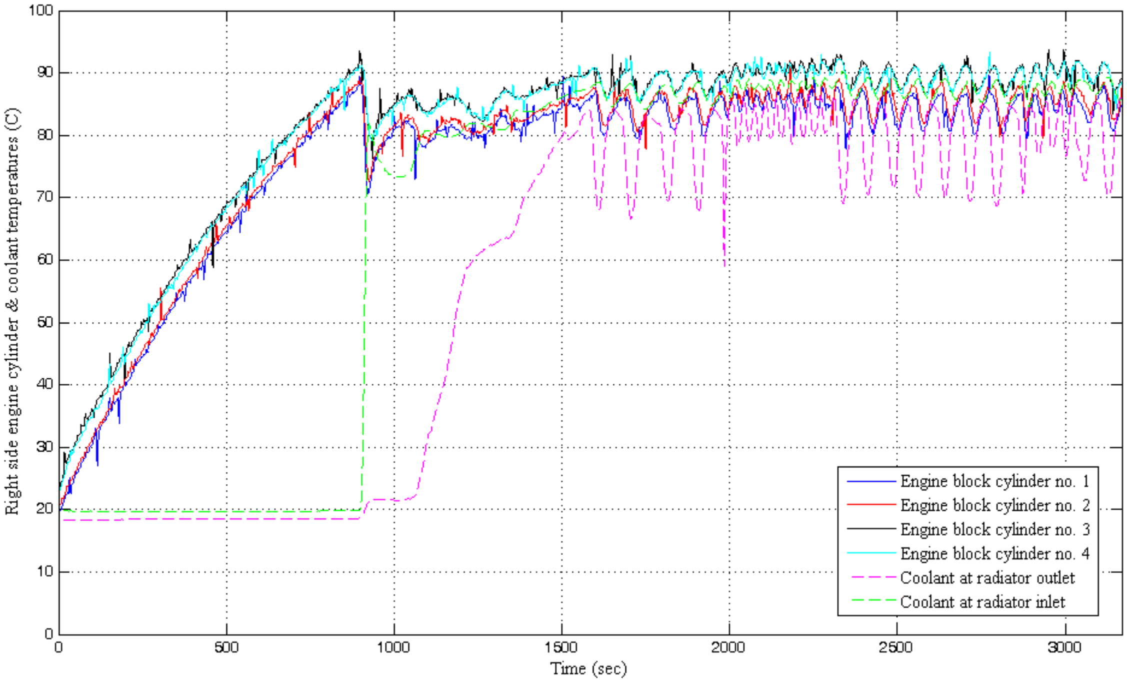


Fig. B.26: Temperatures of engine and coolant for experimental engine test no. 6

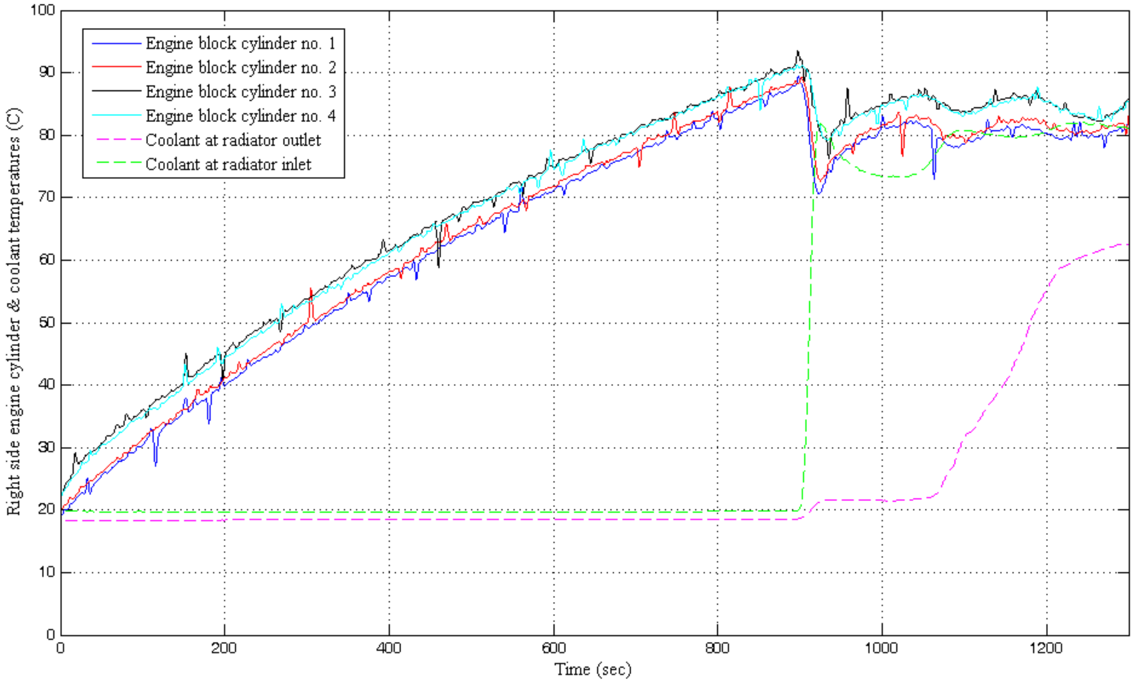


Fig. B.27: Engine test cycle, Stage I for test no. 6

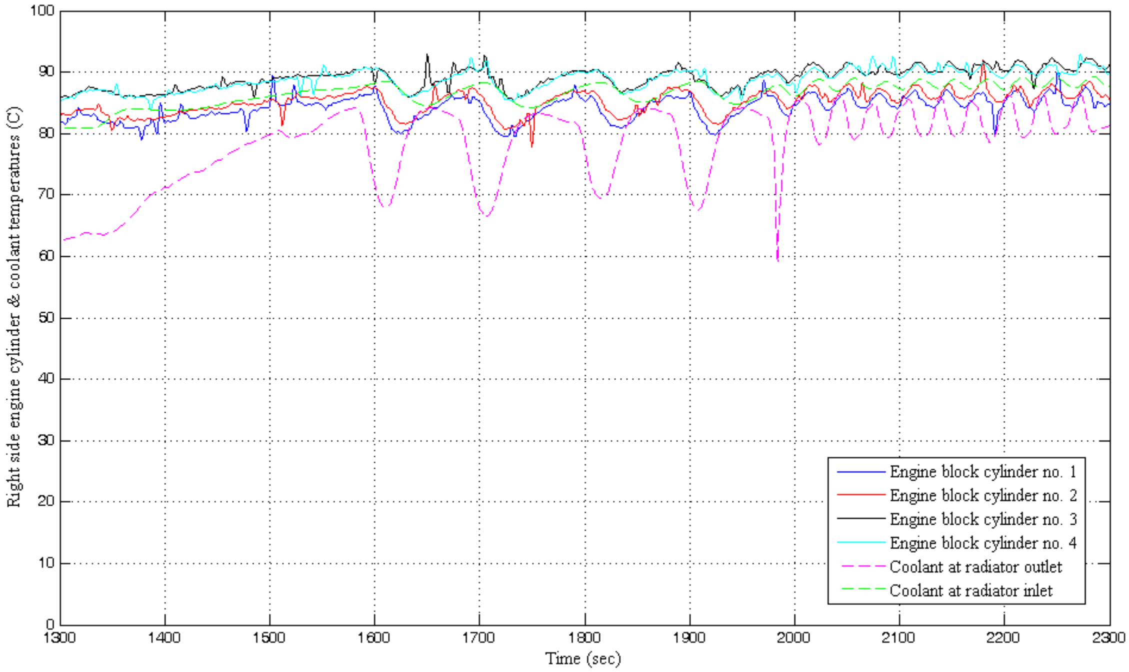


Fig. B.28: Engine test cycle, Stage II for test no. 6

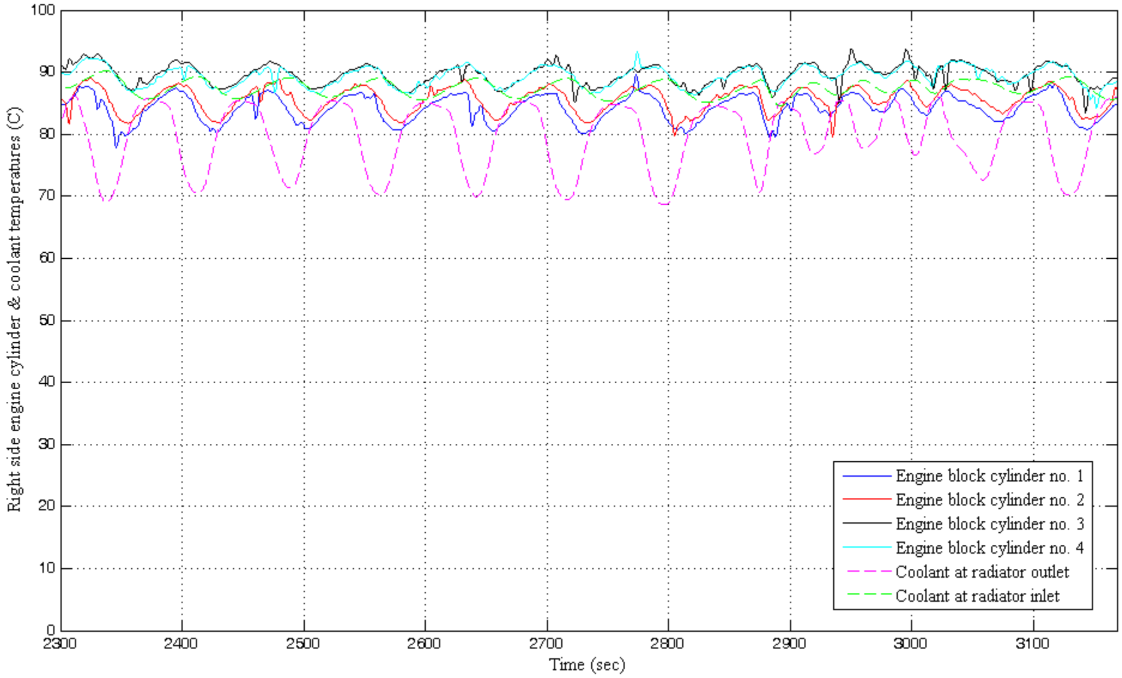


Fig. B.29: Engine test cycle, Stage III for test no. 6

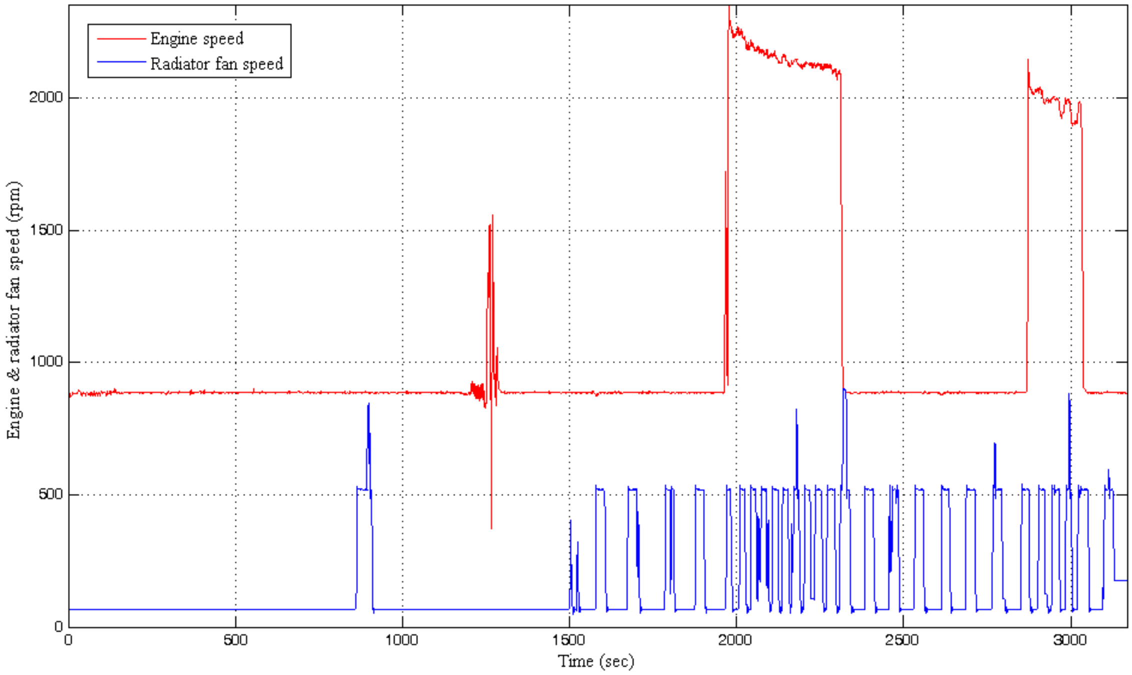


Fig. B.30: Engine and radiator fan speed for test no. 6

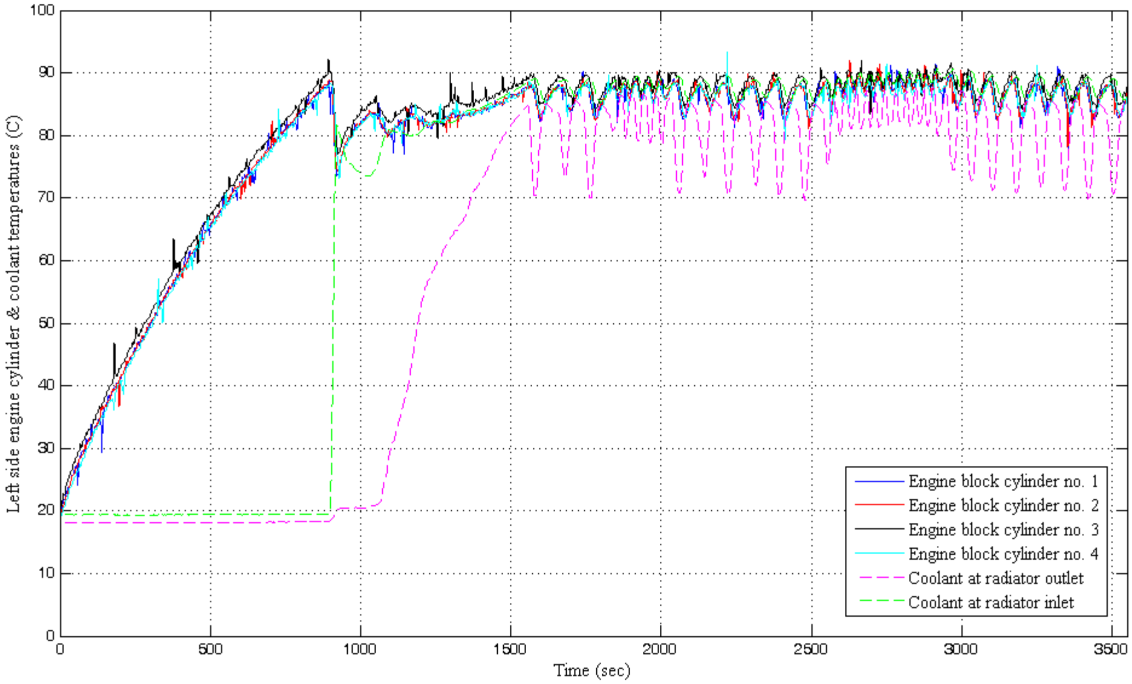


Fig. B.31: Temperatures of engine and coolant for experimental engine test no. 7

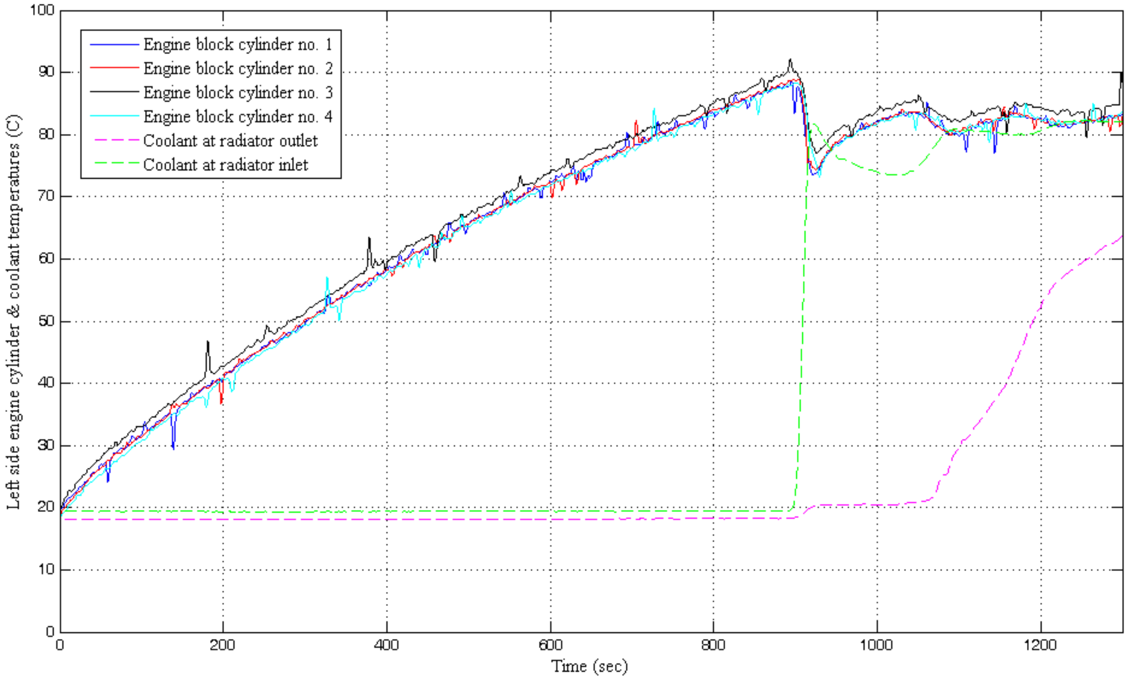


Fig. B.32: Engine test cycle, Stage I for test no. 7

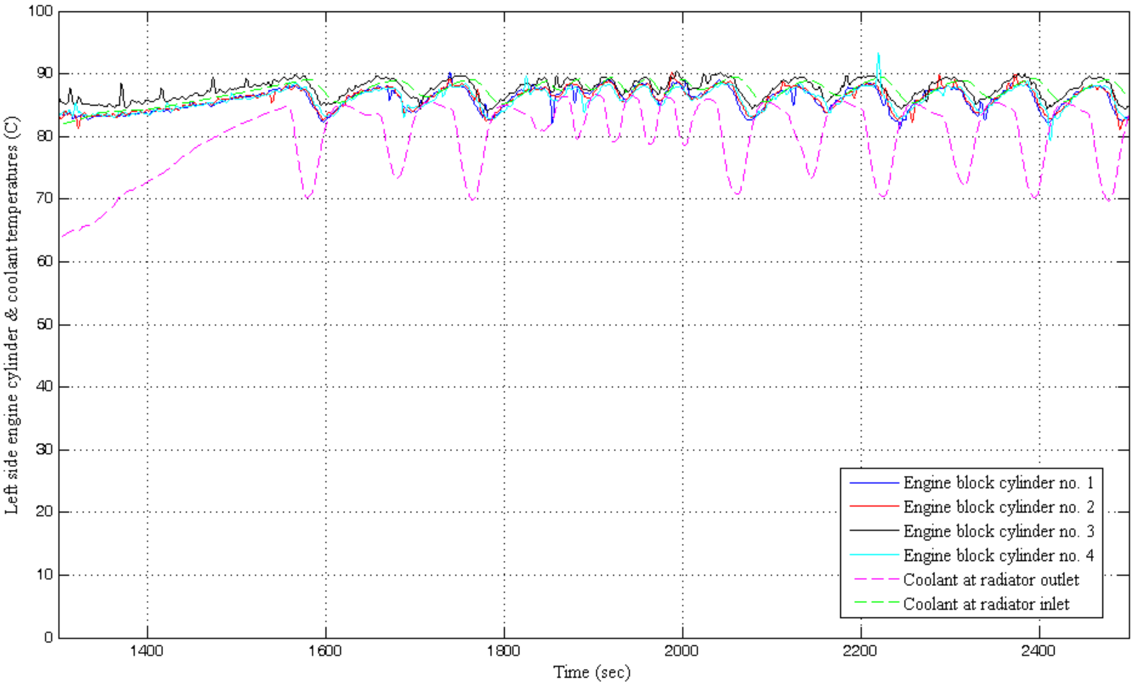


Fig. B.33: Engine test cycle, Stage II for test no. 7

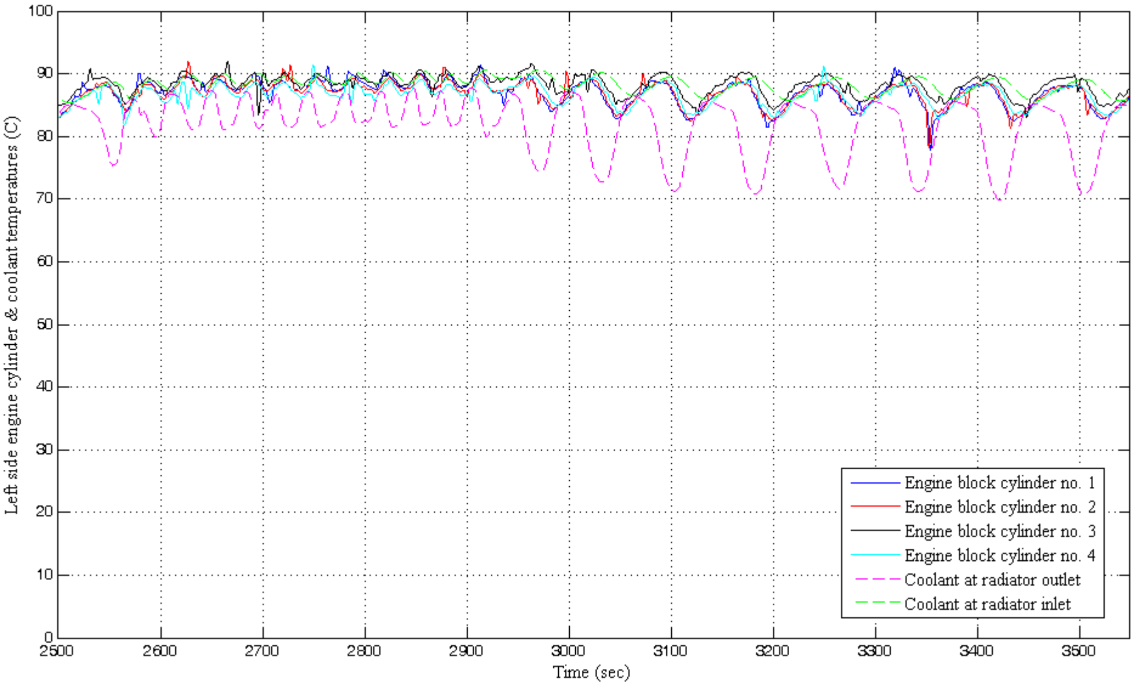


Fig. B.34: Engine test cycle, Stage III for test no. 7

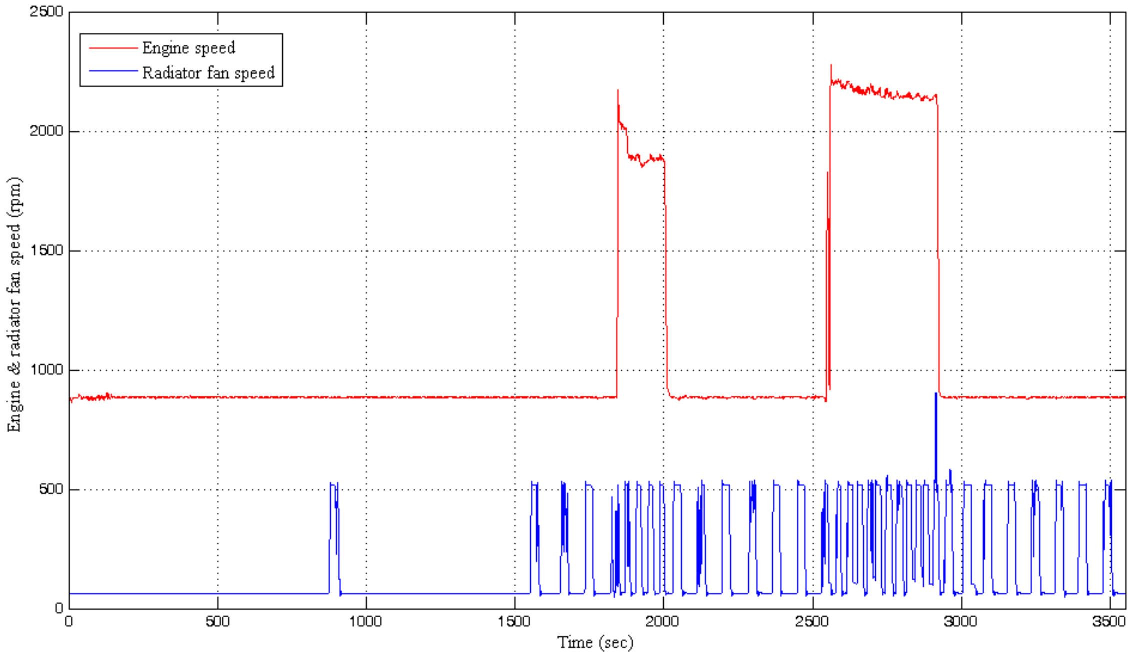


Fig. B.35: Engine and radiator fan speed for test no. 7

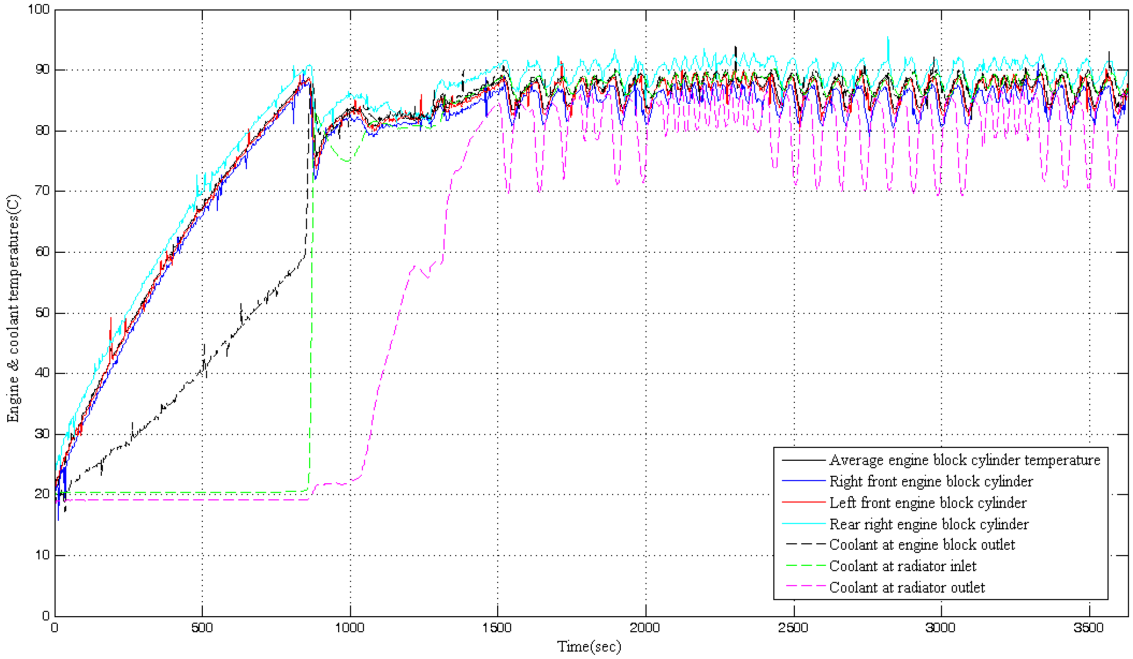


Fig. B.36: Temperatures of engine and coolant for experimental engine test no. 8

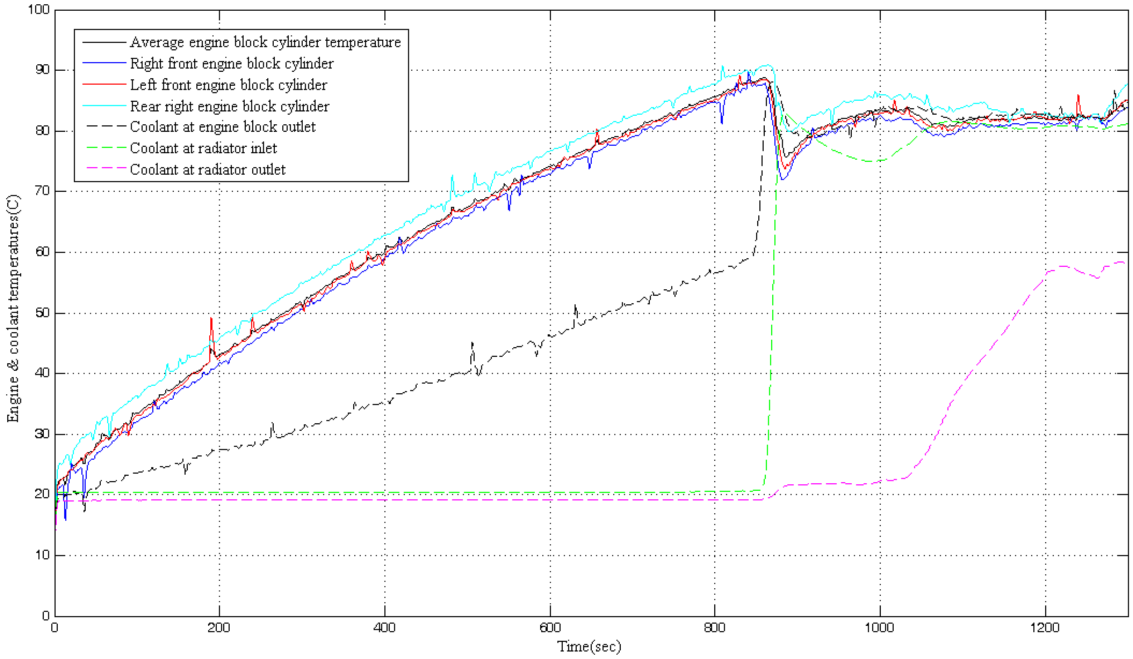


Fig. B.37: Engine test cycle, Stage I for test no. 8

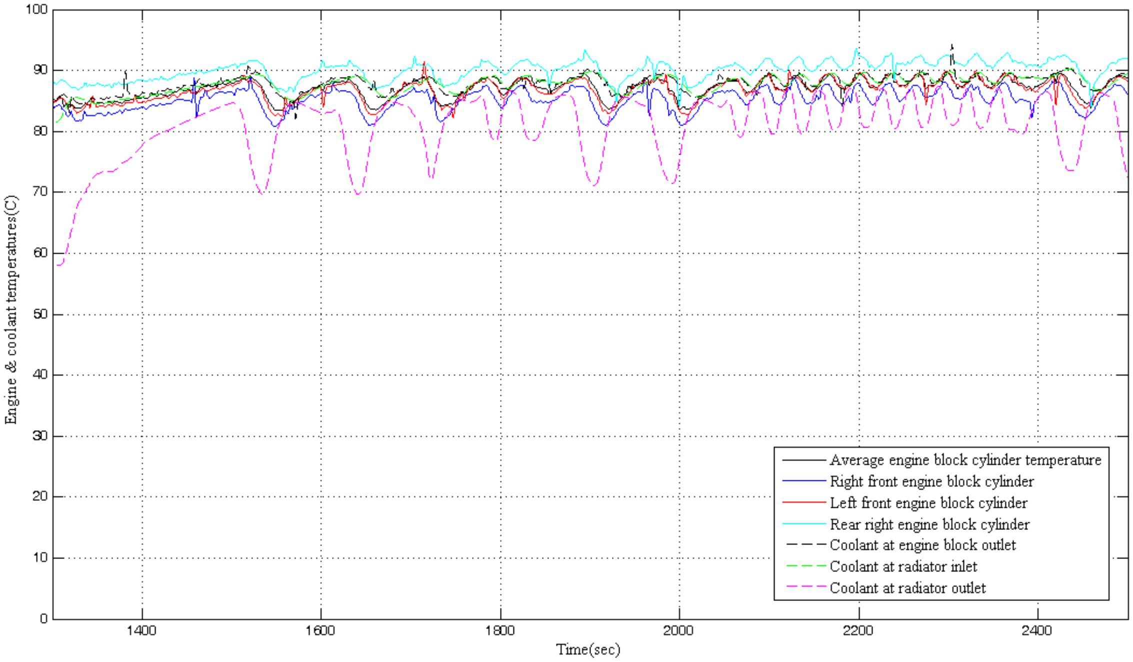


Fig. B.38: Engine test cycle, Stage II for test no. 8

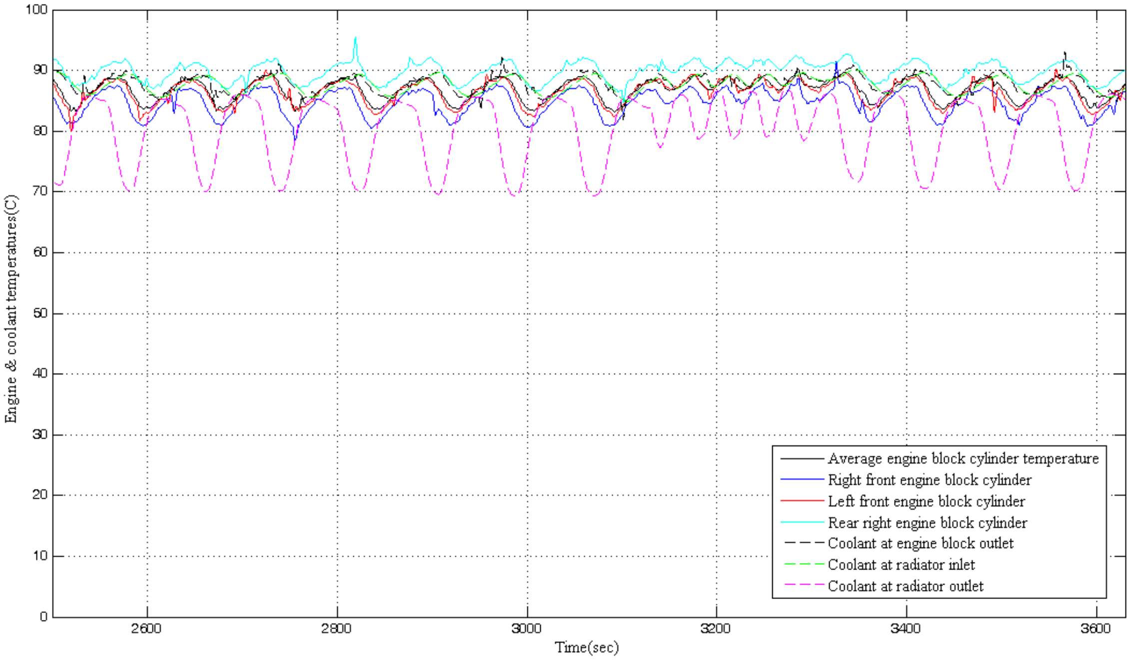


Fig. B.39: Engine test cycle, Stage III for test no. 8

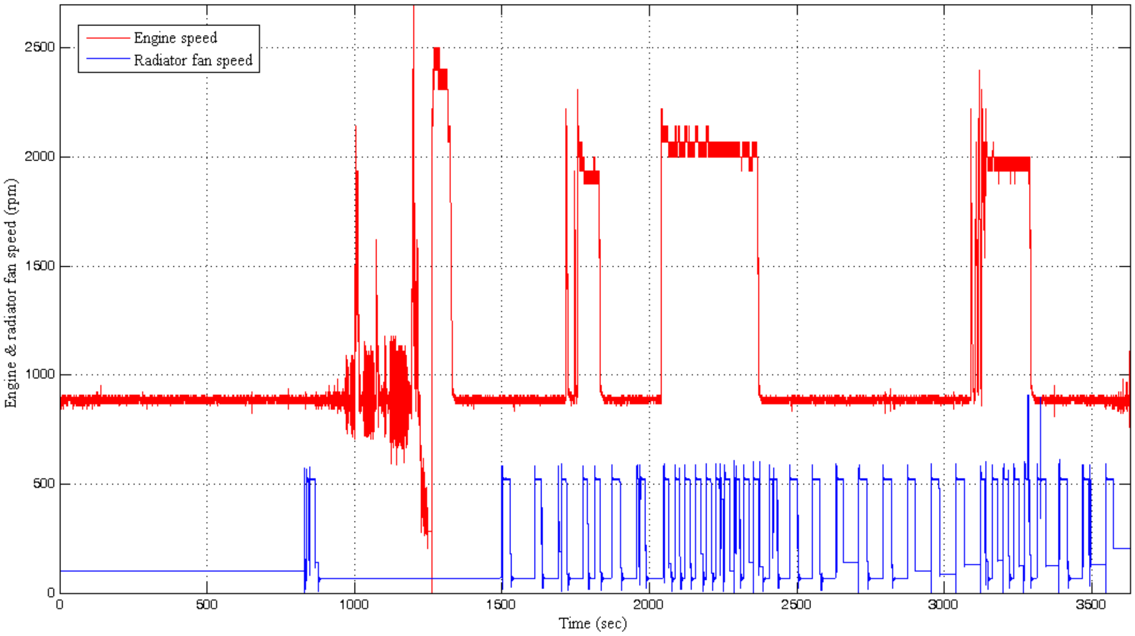


Fig. B.40: Engine and radiator fan speed for test no. 8

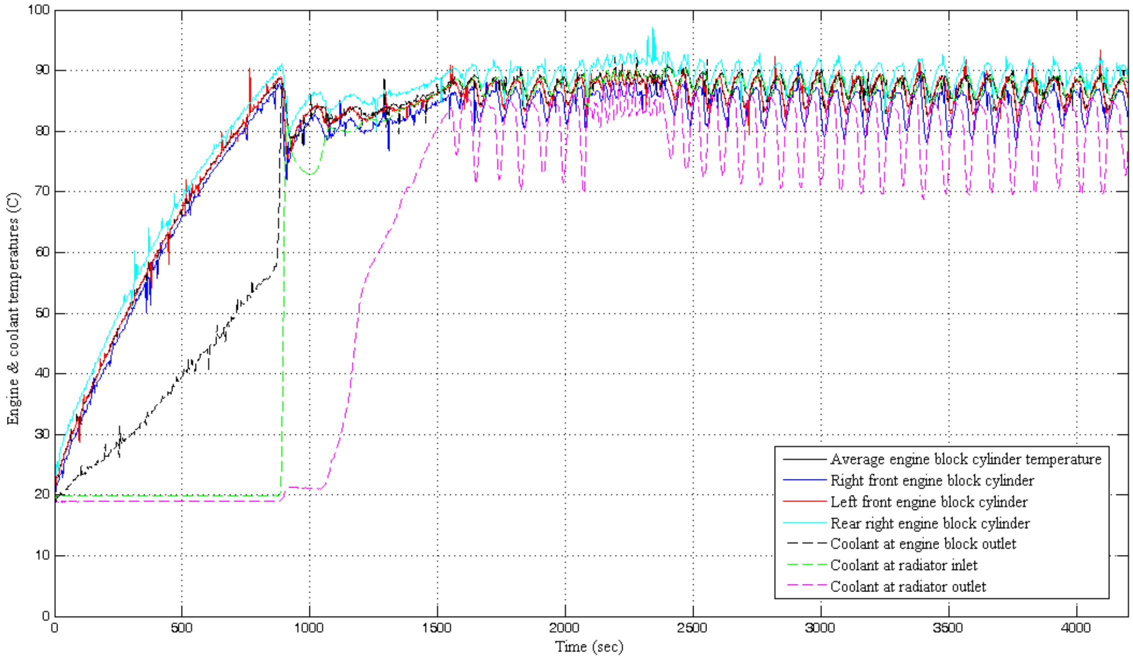


Fig. B.41: Temperatures of engine and coolant for experimental engine test no. 9

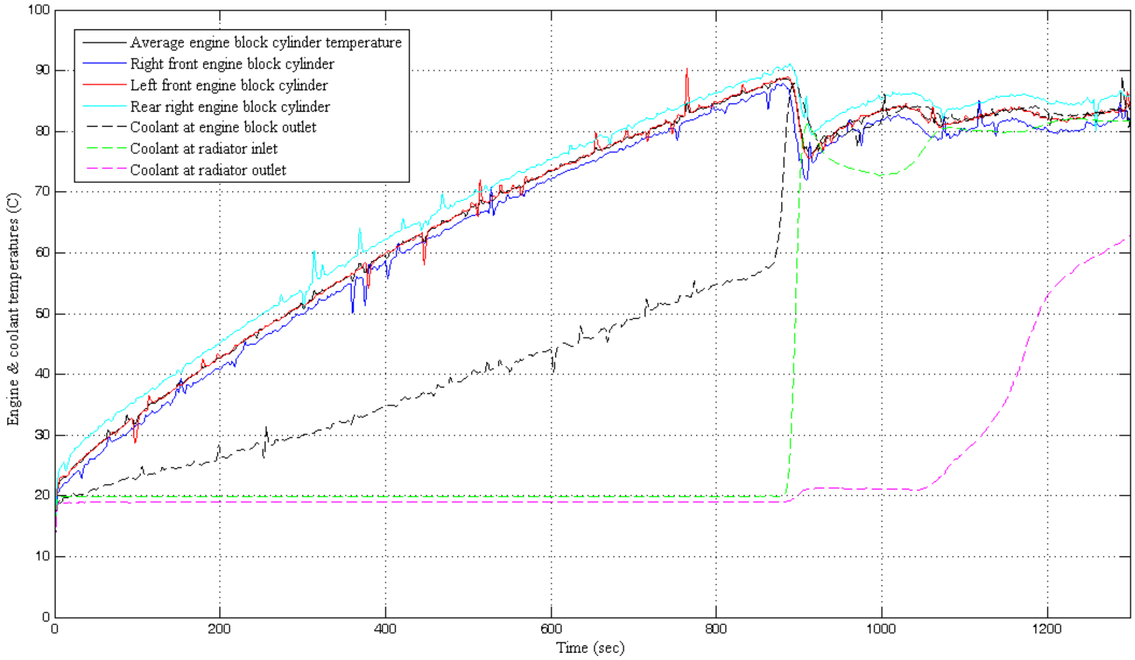


Fig. B.42: Engine test cycle, Stage I for test no. 9

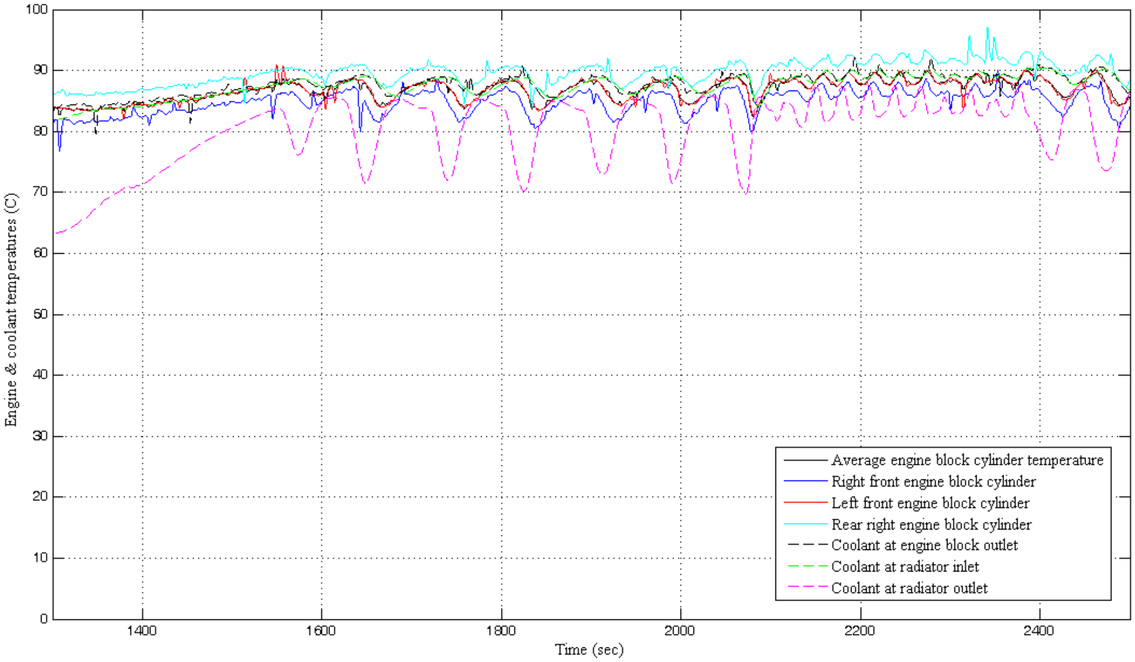


Fig. B.43: Engine test cycle, Stage II for test no. 9

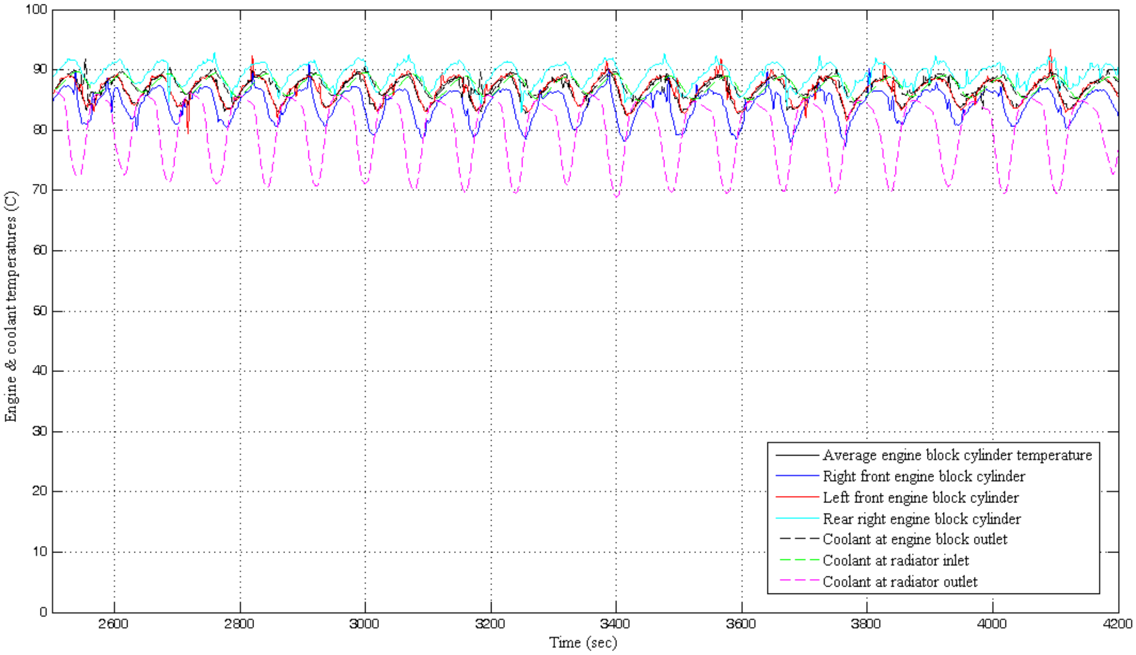


Fig. B.44: Engine test cycle, Stage III for test no. 9

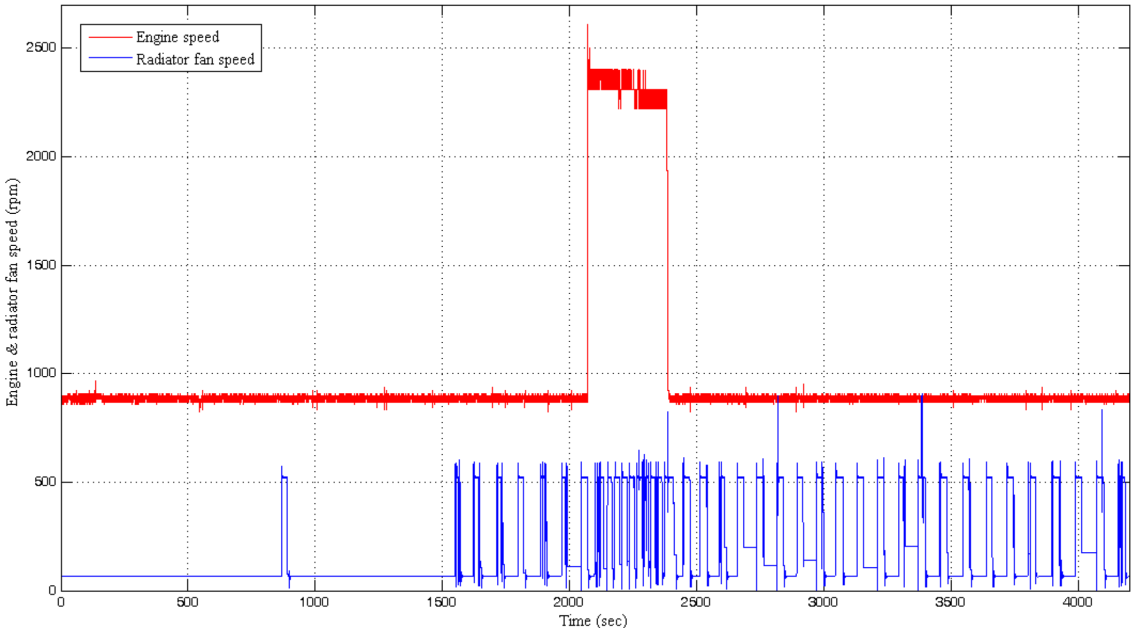


Fig. B.45: Engine and radiator fan speed for test no. 9

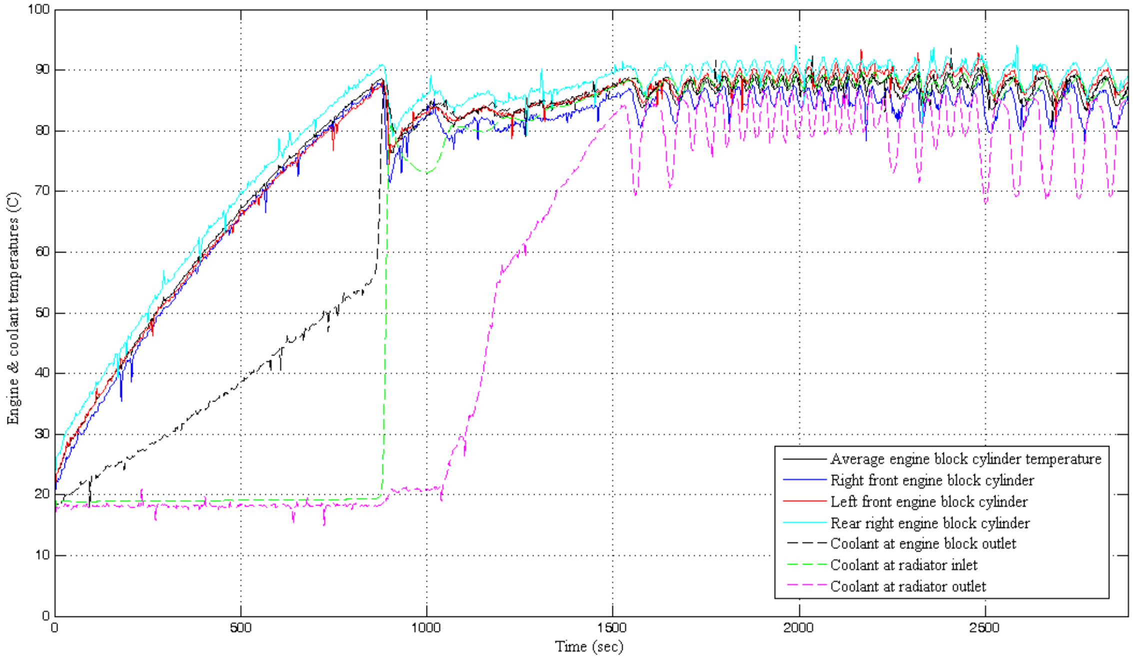


Fig. B.46: Temperatures of engine and coolant for experimental engine test no. 10

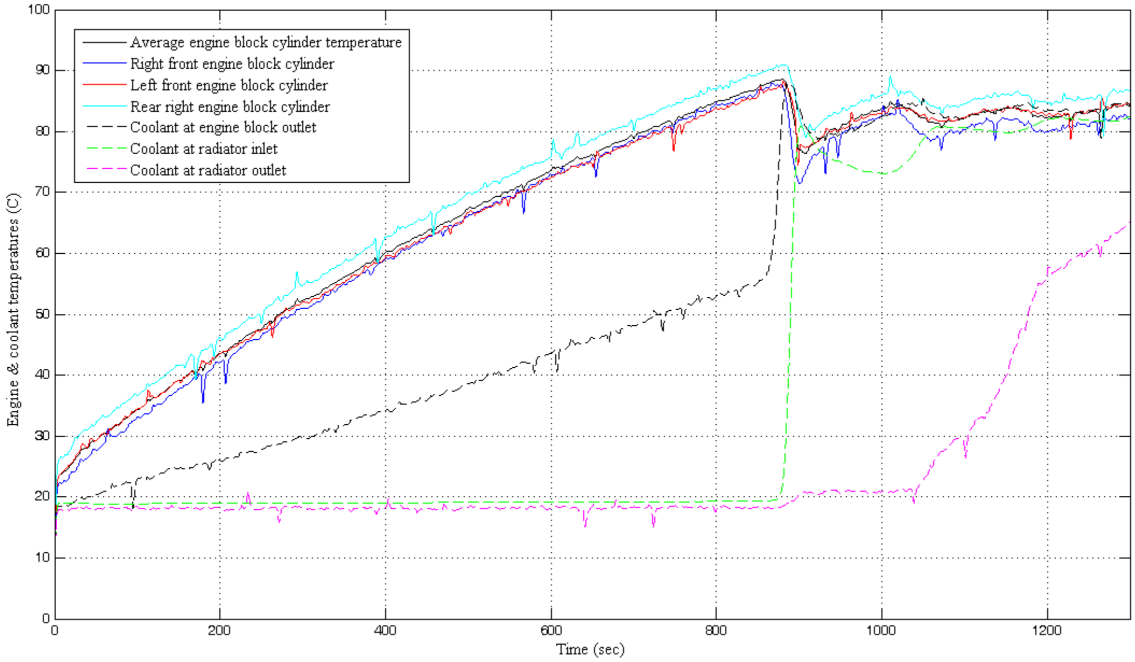


Fig. B.47: Engine test cycle, Stage I for test no. 10

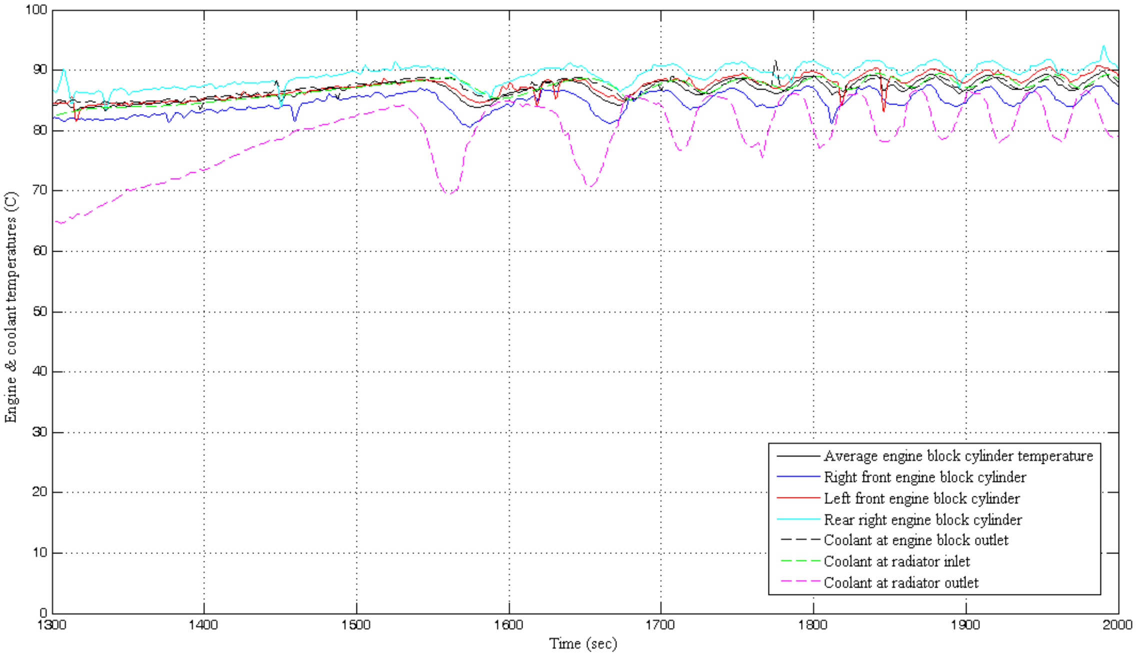


Fig. B.48: Engine test cycle, Stage II for test no. 10

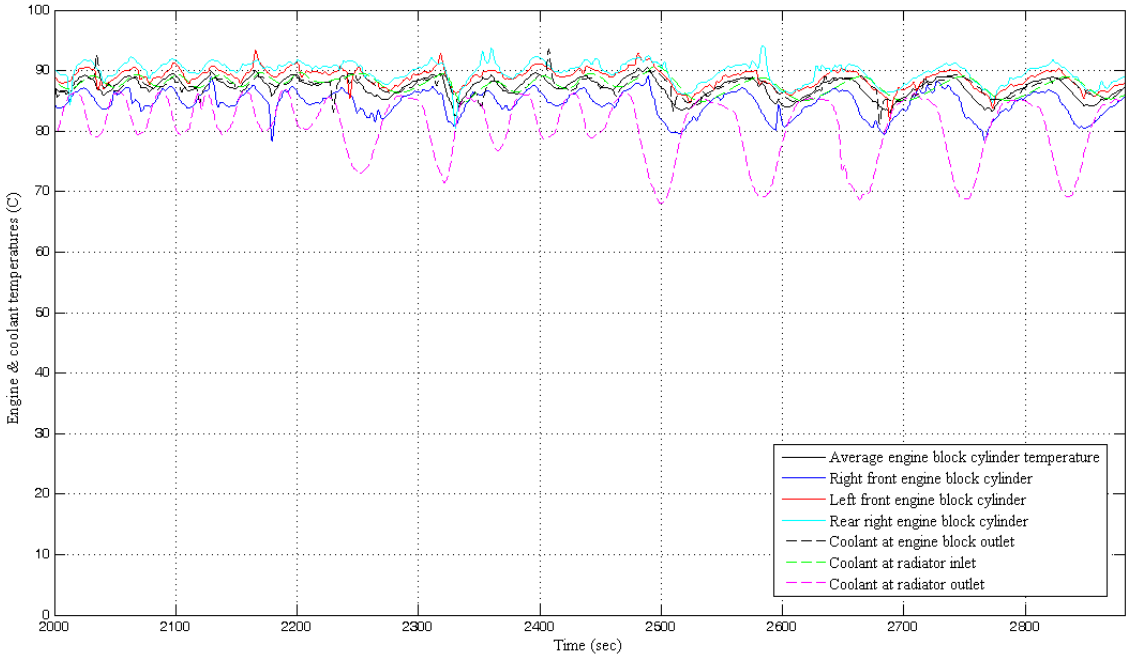


Fig. B.49: Engine test cycle, Stage III for test no. 10

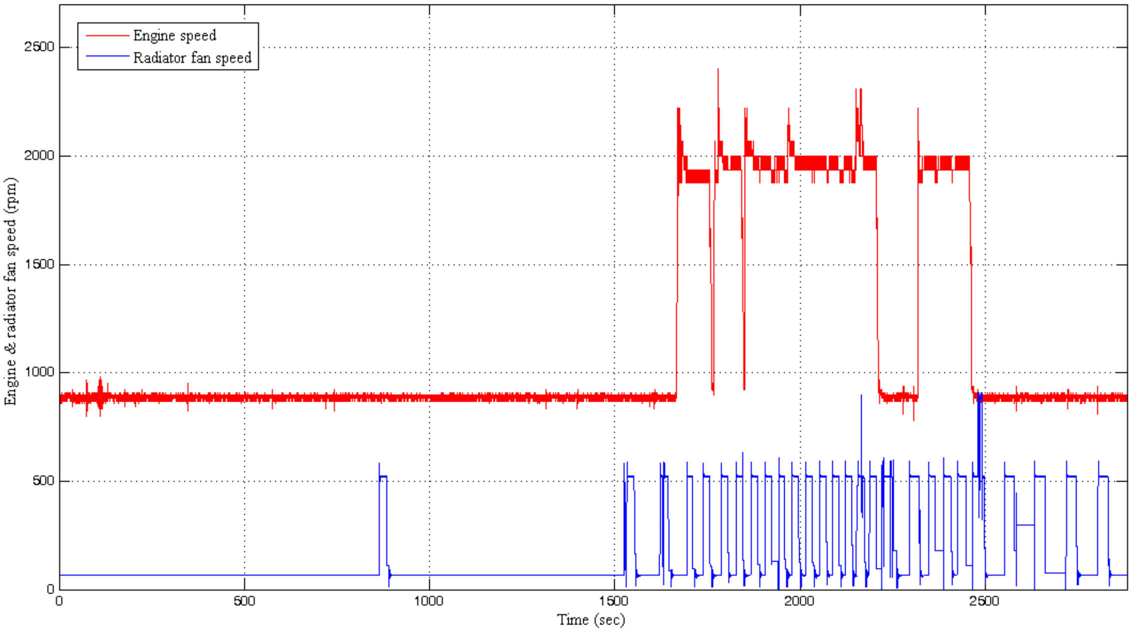


Fig. B.50: Engine and radiator fan speed for test no. 10

APPENDIX C: EXPERIMENTAL ENGINE TEST ALGORITHM

Appendix C shows the control algorithm in Matlab/Simulink used for experimental testing of hydraulic based engine cooling system (Fig. C.1 – Fig. C.17).

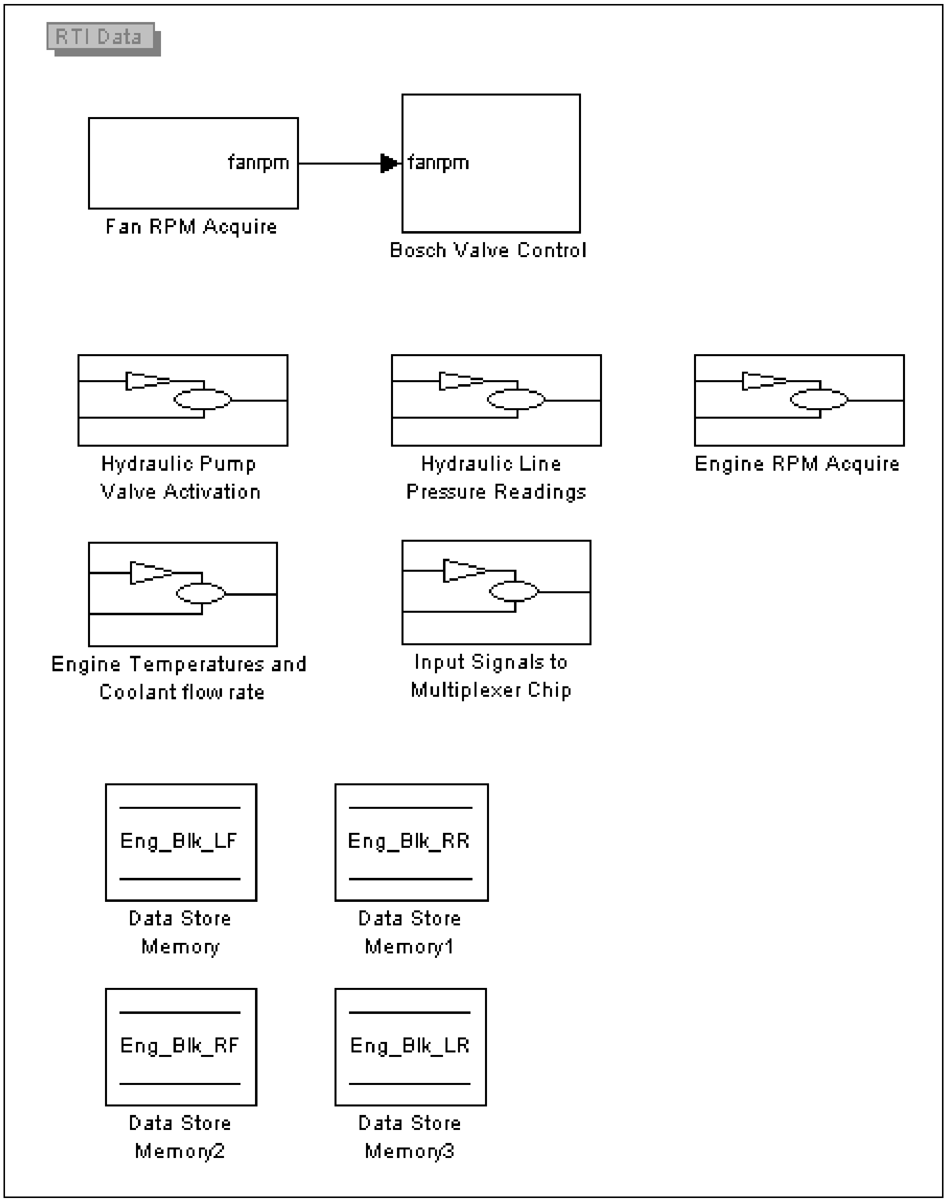


Fig. C.1: Control algorithm for experimental setup of hydraulic based engine cooling system, 1st level

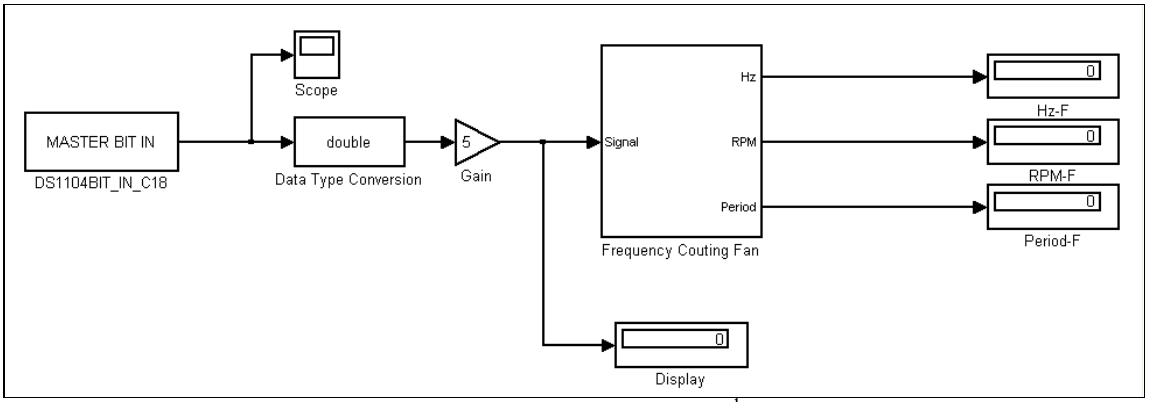


Fig. C.2: Fan RPM acquire, 2nd level

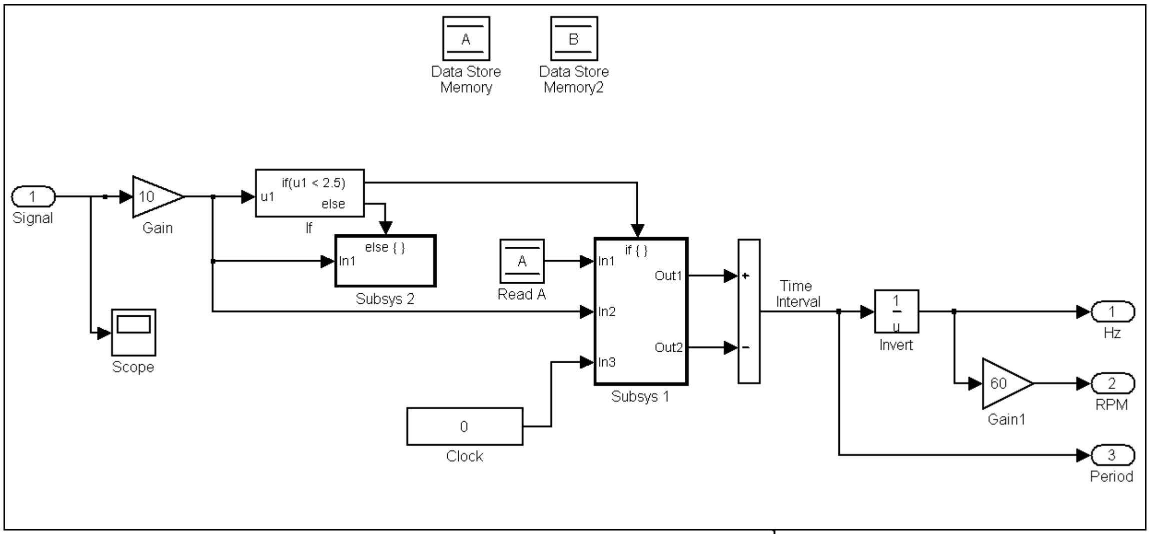


Fig. C.3: Frequency counting fan, 3rd level

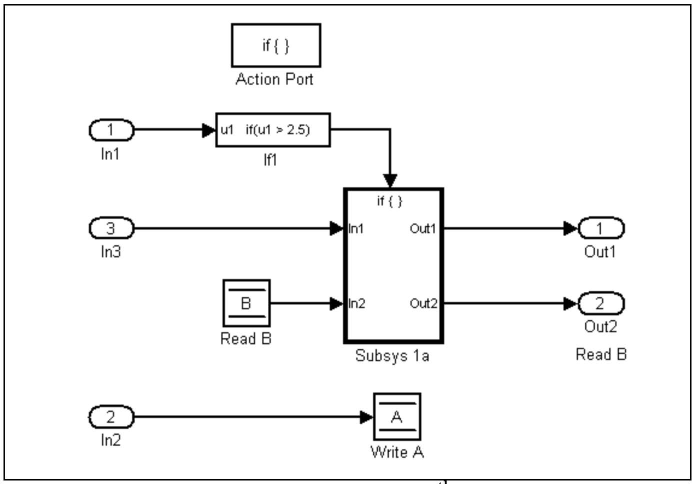


Fig. C.4: Subsys 1, 4th level

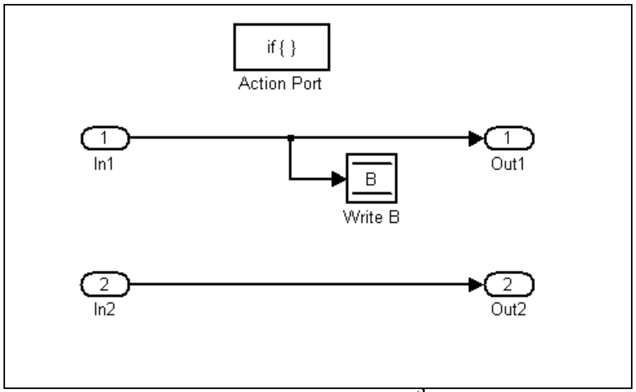


Fig. C.5: Subsys 1a, 5th level

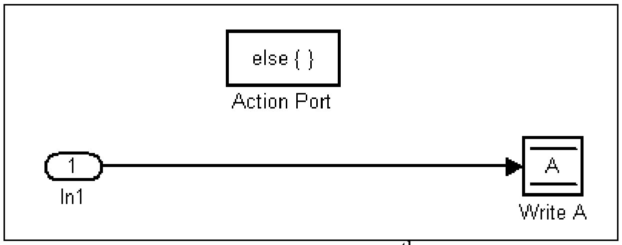


Fig. C.6: Subsys 2, 4th level

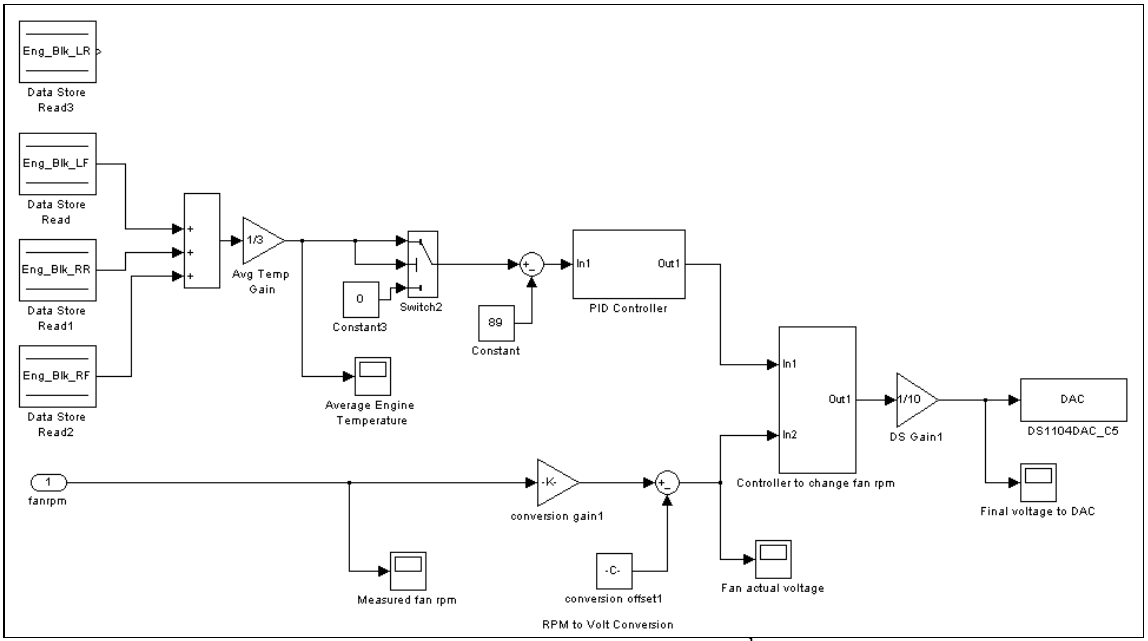


Fig. C.7: Bosch valve control, 2nd level

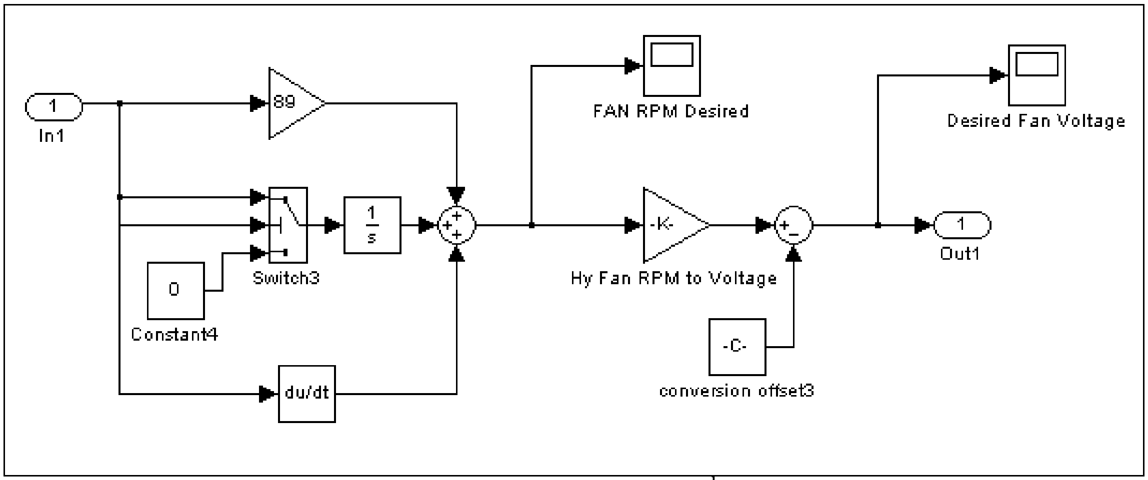


Fig. C.8: PID controller, 3rd level

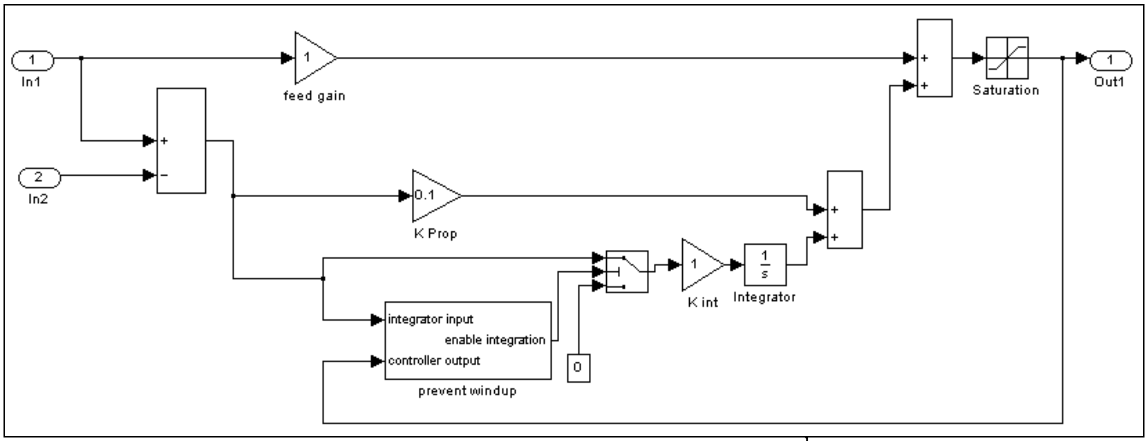


Fig. C.9: Controller to change fan rpm, 3rd level

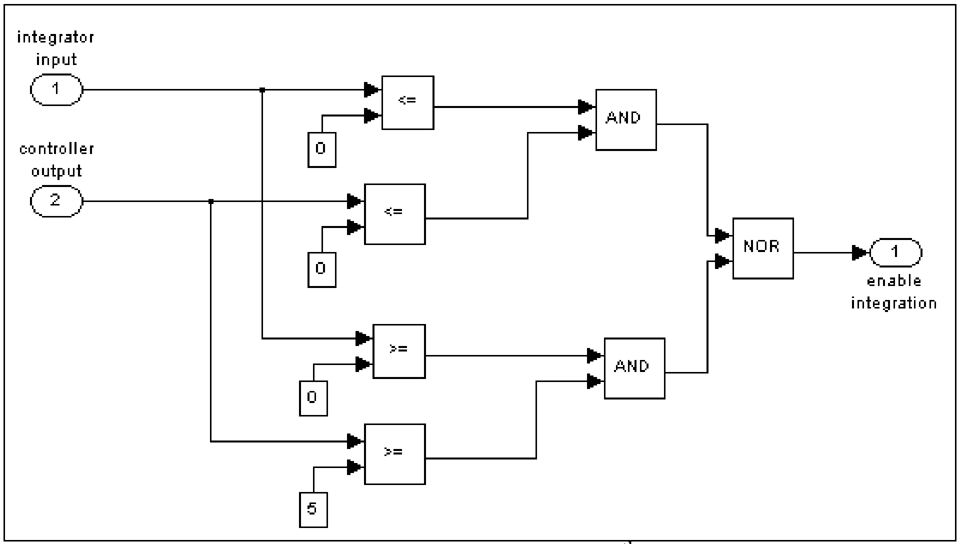


Fig. C.10: Prevent windup, 4th level

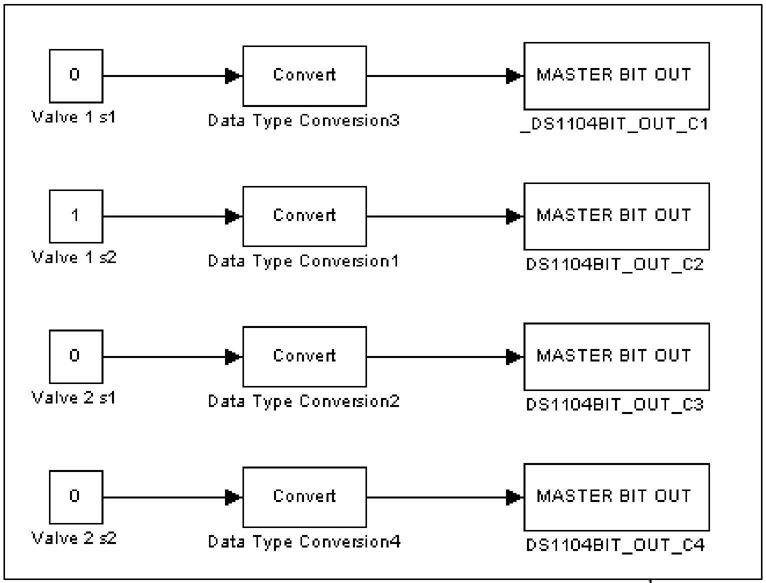


Fig. C.11: Hydraulic pump valve activation, 2^{nd} level

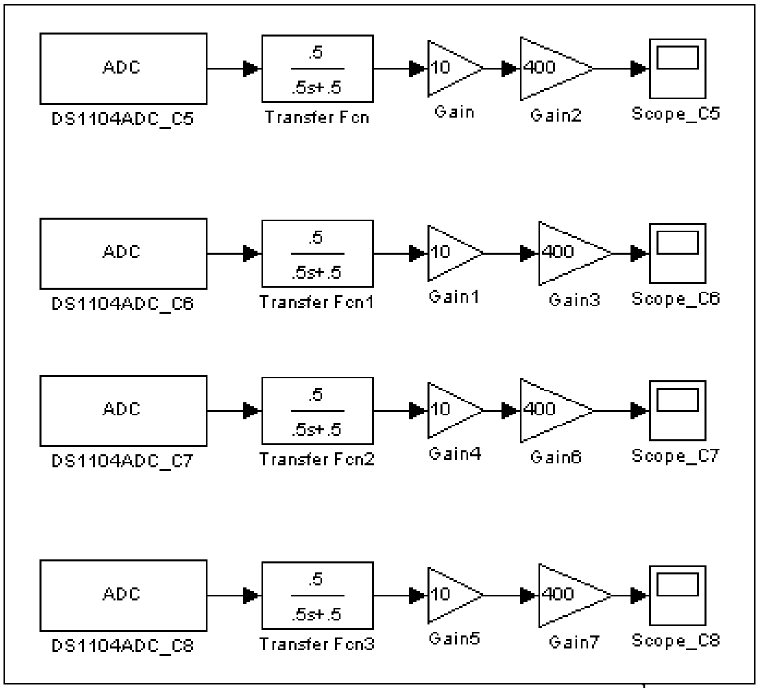


Fig. C.12: Hydraulic line pressure readings, 2^{nd} level

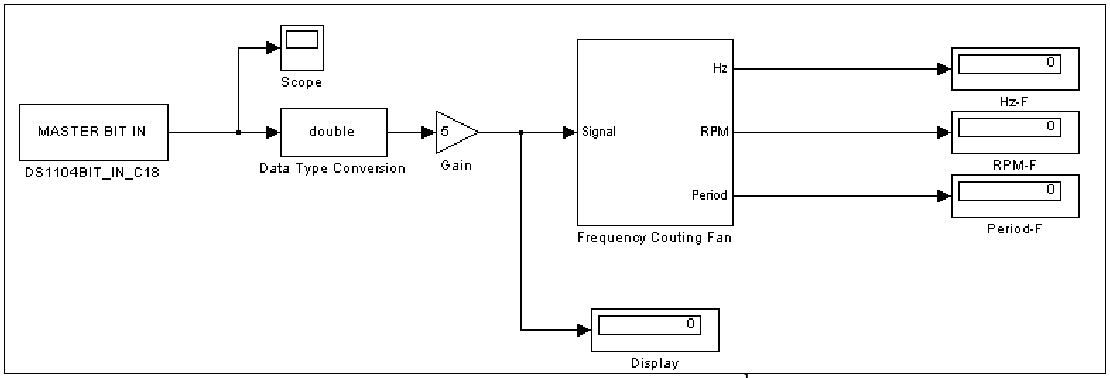


Fig. C.13: Engine RPM acquire, 2nd level

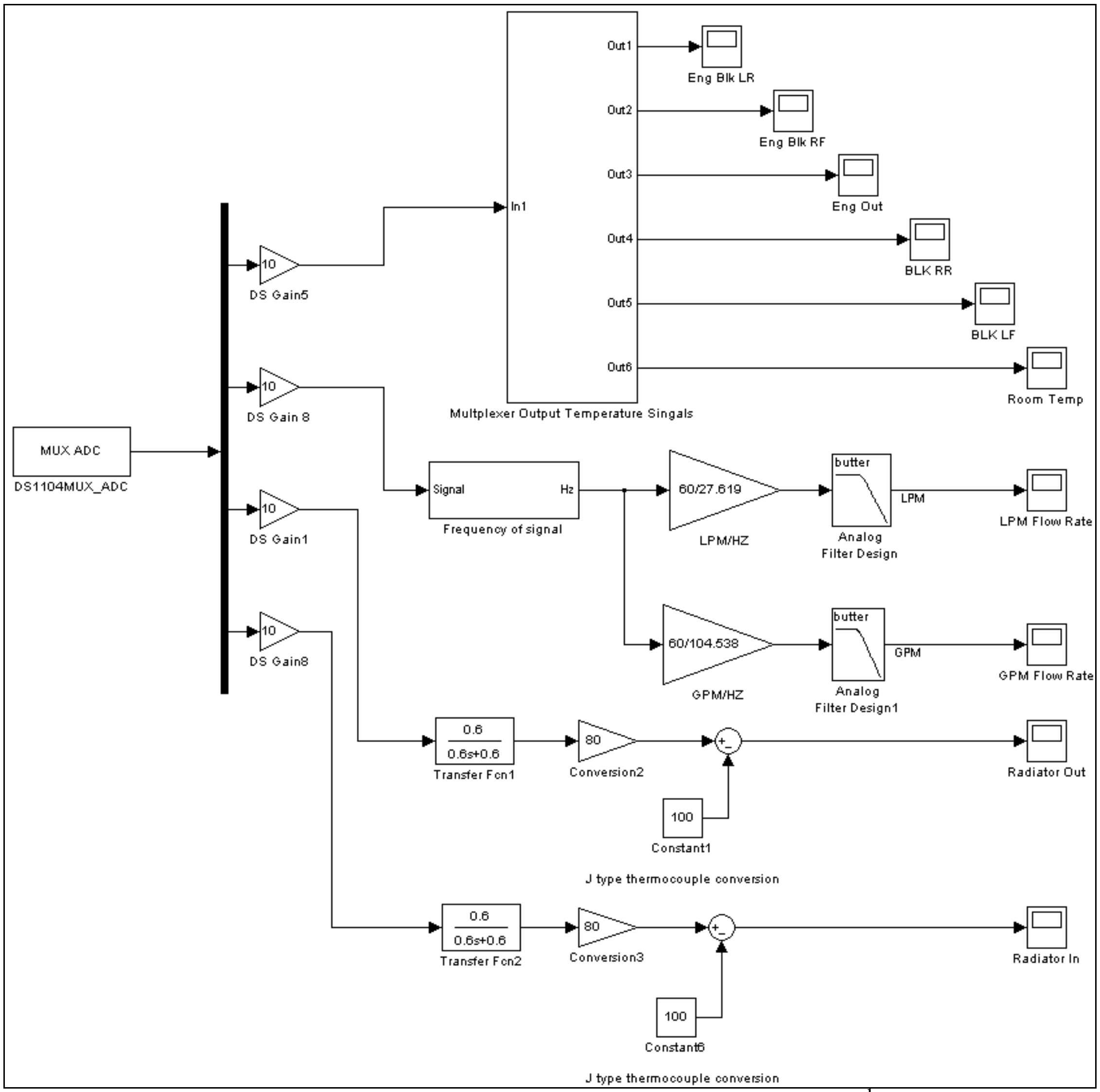


Fig. C.14: Engine temperatures and coolant flow rate, 2nd level

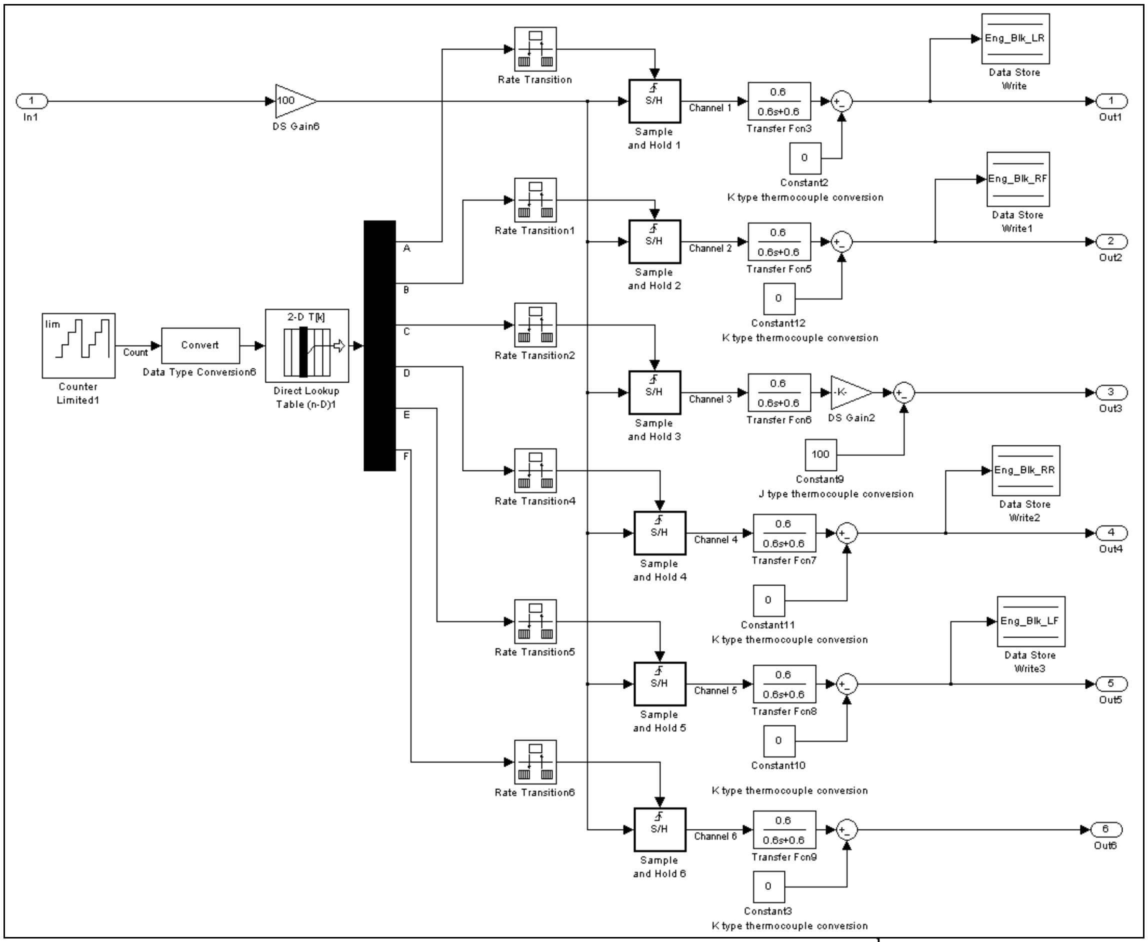


Fig. C.15: Multiplexer output temperature signals, 3rd level

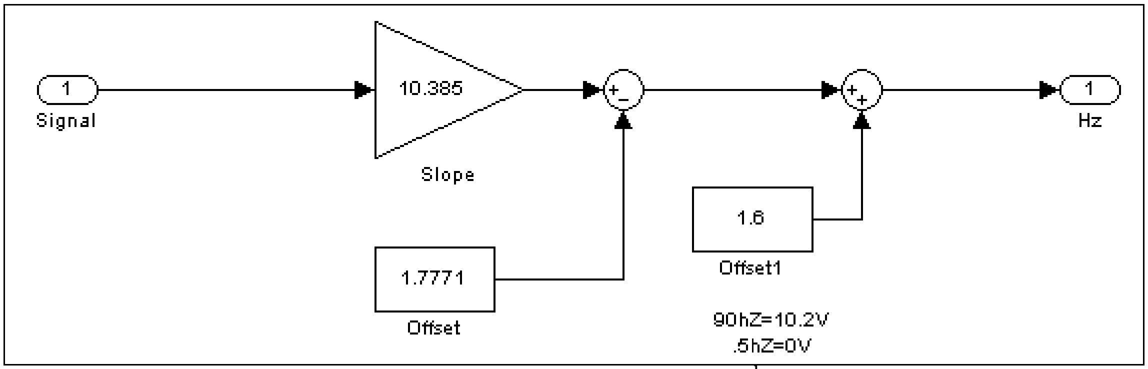


Fig. C.16: Frequency of signal, 3rd level

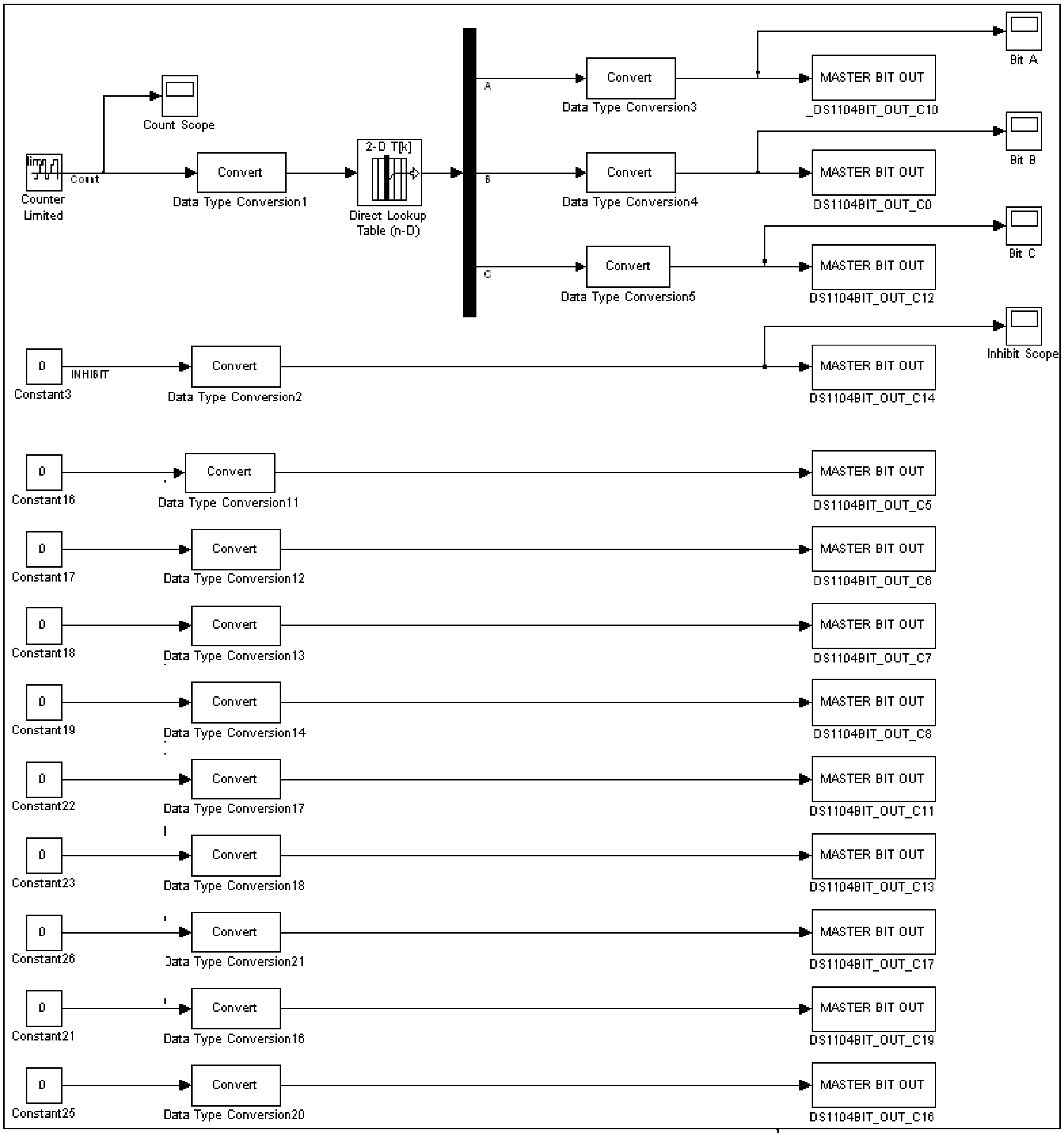


Fig. C.17: Input signals to multiplexer chip, 2nd level

APPENDIX D: EXPERIMENTAL AND SIMULATED TEST RESULTS

The Appendix D presents the experimental and simulated engine test results (Fig. D.1 – Fig. D.20) for validating the mathematical models presented in the Section 2.3.

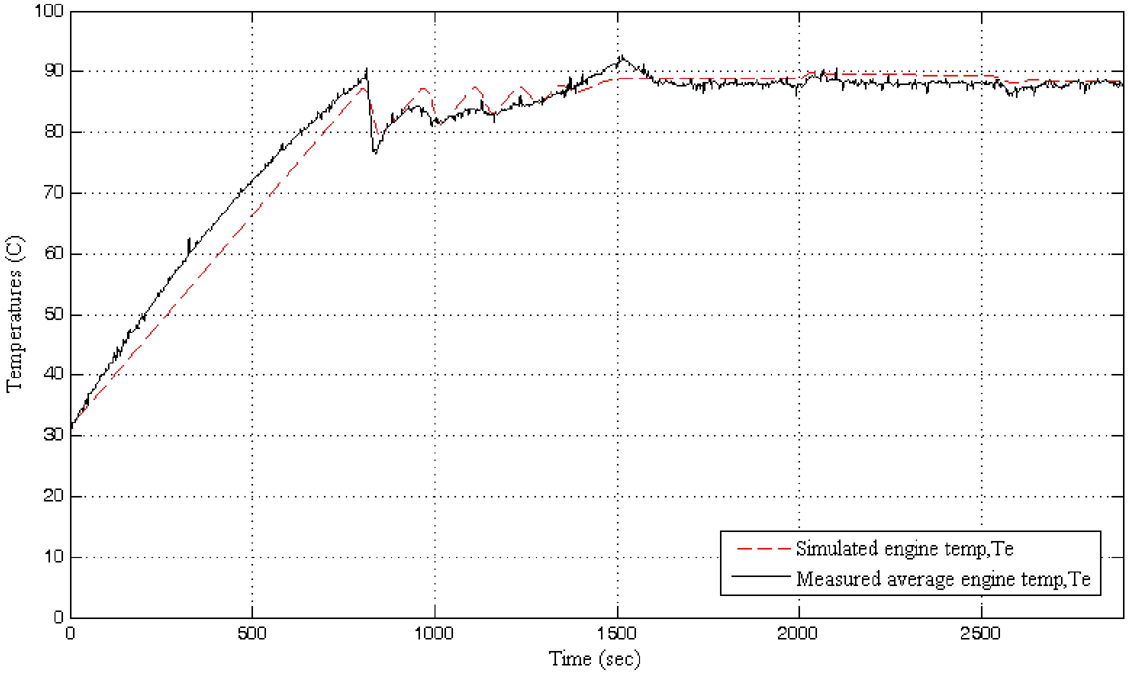


Fig. D.1: Simulated engine and coolant temperature in comparison with experimental average engine temperature for engine test no. 1

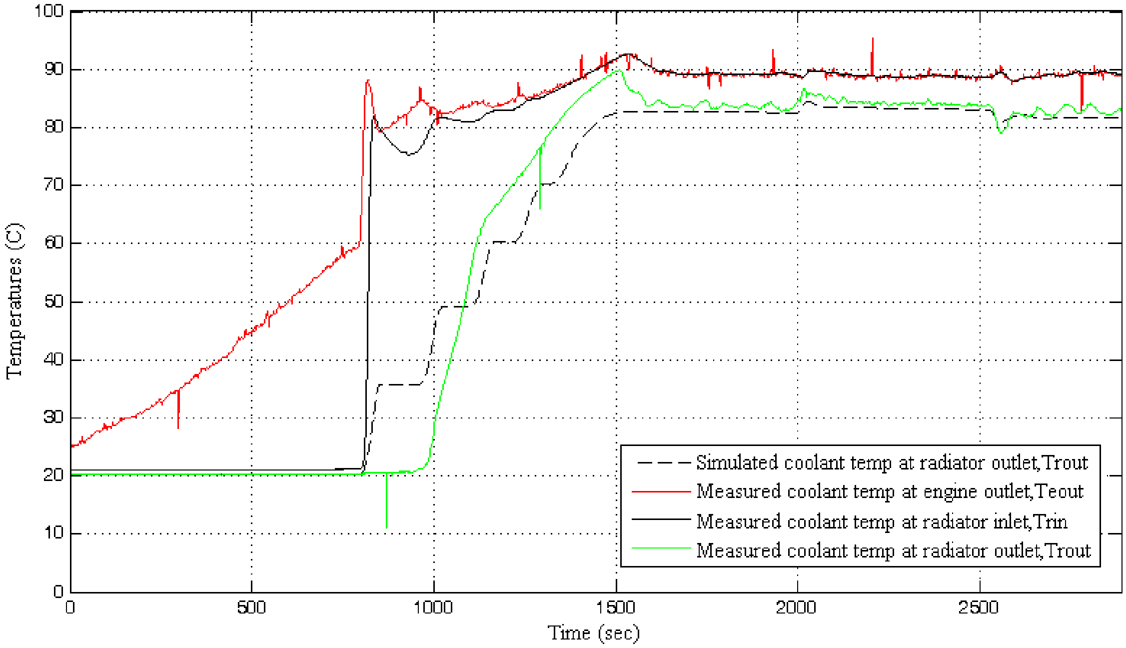


Fig. D.2: Simulated coolant temperatures at various locations in comparison with the experimental coolant temperatures at engine outlet, radiator inlet and radiator outlet for engine test no. 1

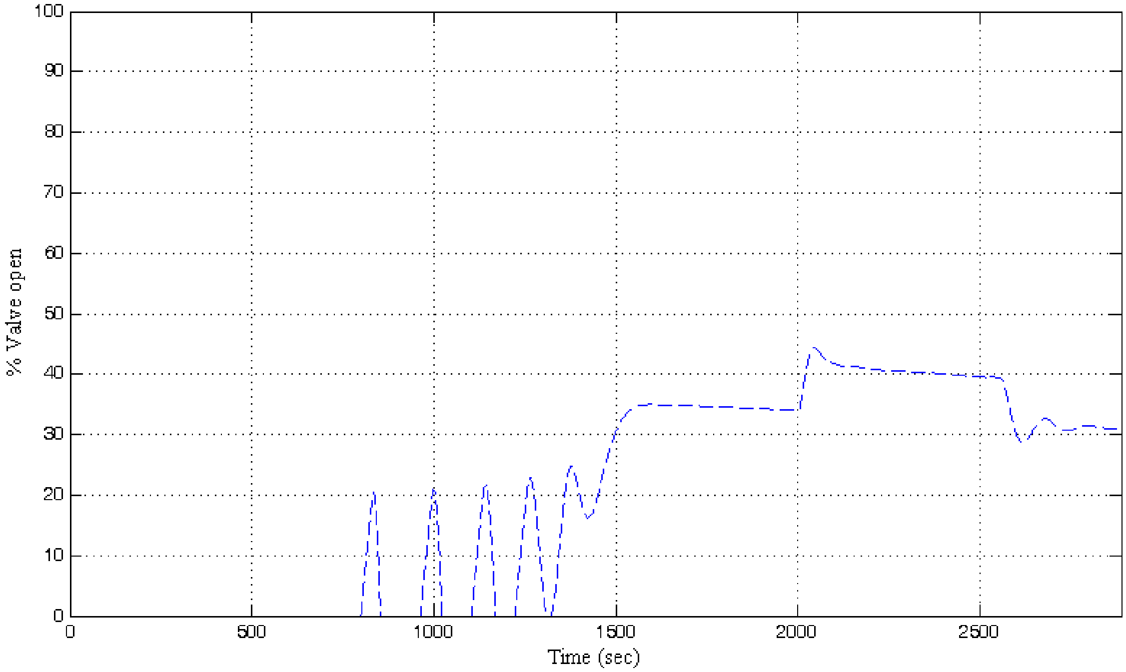


Fig. D.3: Simulated valve opening for engine test no. 1

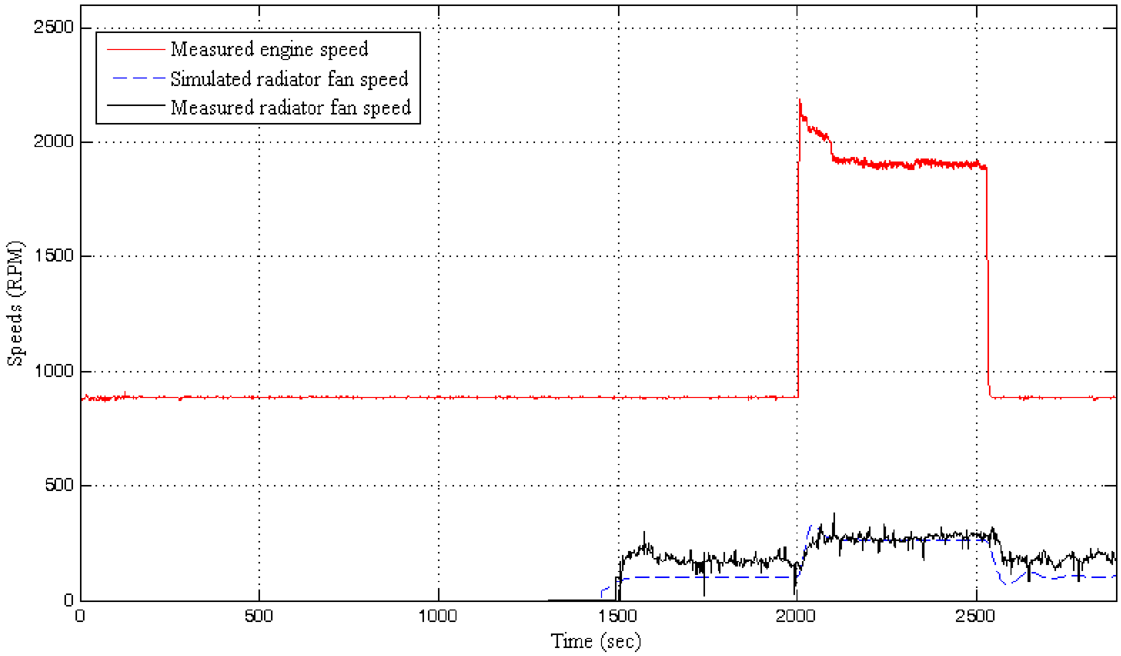


Fig. D.4: Simulated radiator fan speed in comparison with the experimental radiator fan speed and engine speed for engine test no. 1

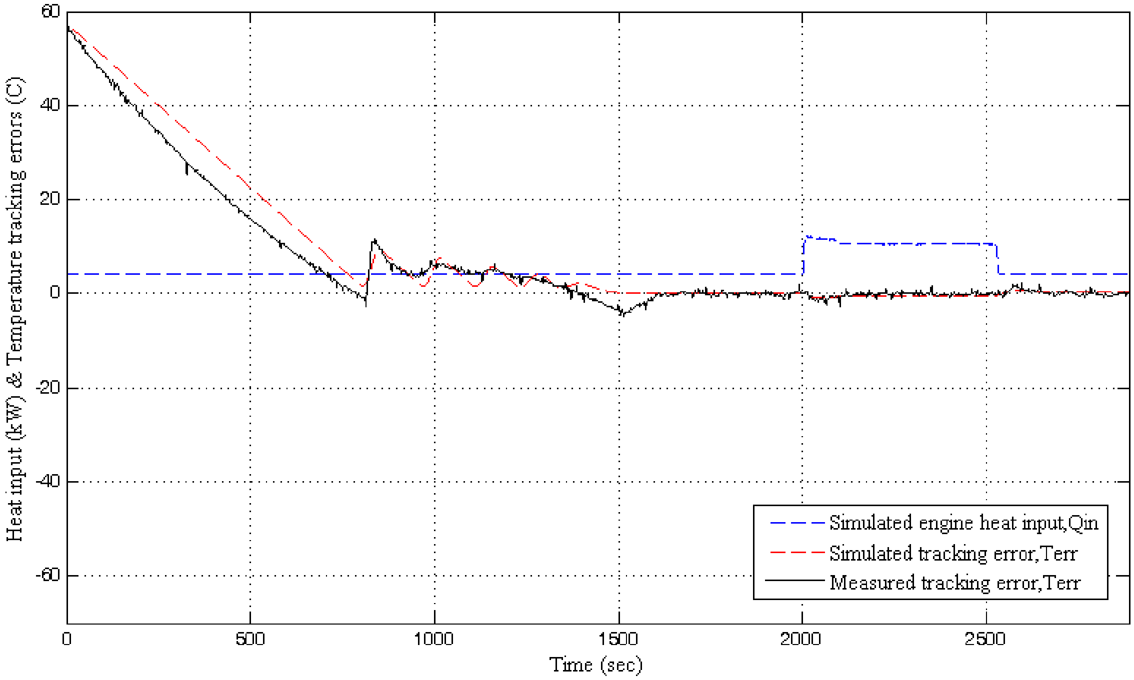


Fig. D.5: Estimated engine heat input and temperature tracking errors for simulated and experimental engine test no. 1

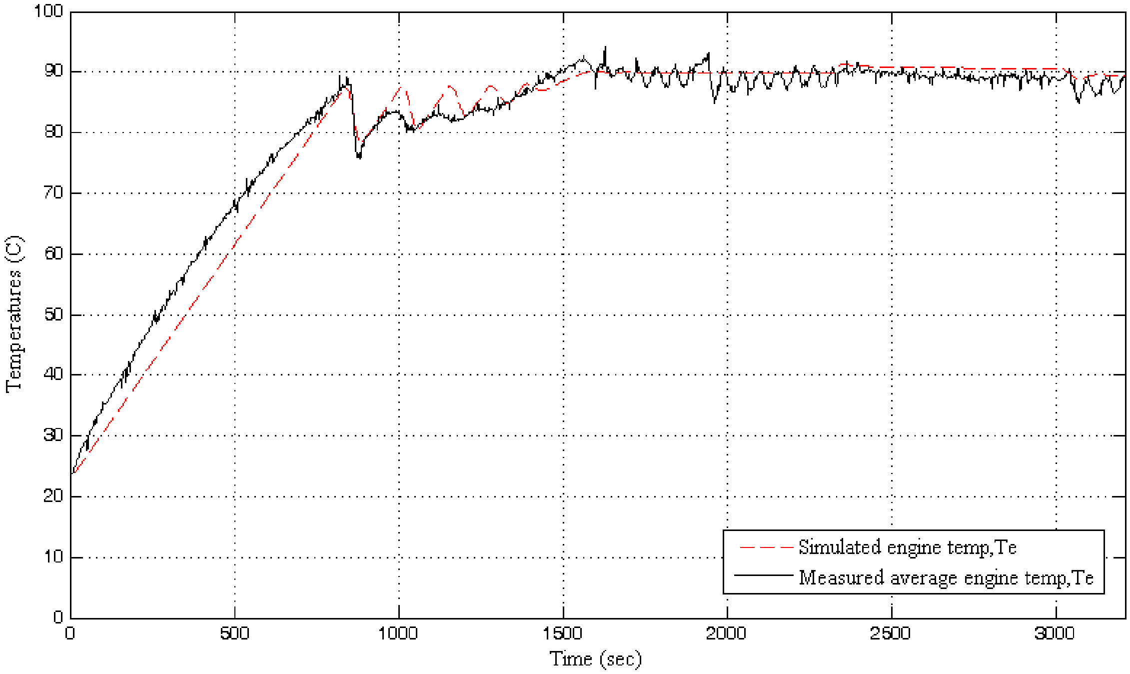


Fig. D.6: Simulated engine and coolant temperature in comparison with experimental average engine temperature for engine test no. 2

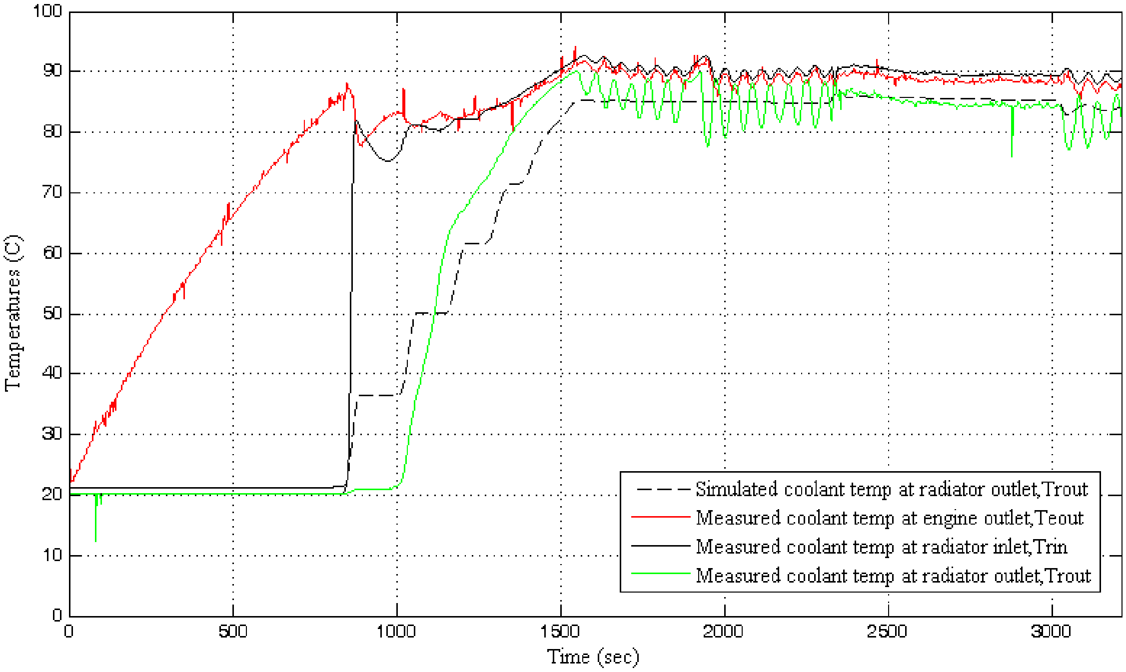


Fig. D.7: Simulated coolant temperatures at various locations in comparison with the experimental coolant temperatures at engine outlet, radiator inlet and radiator outlet for engine test no. 2

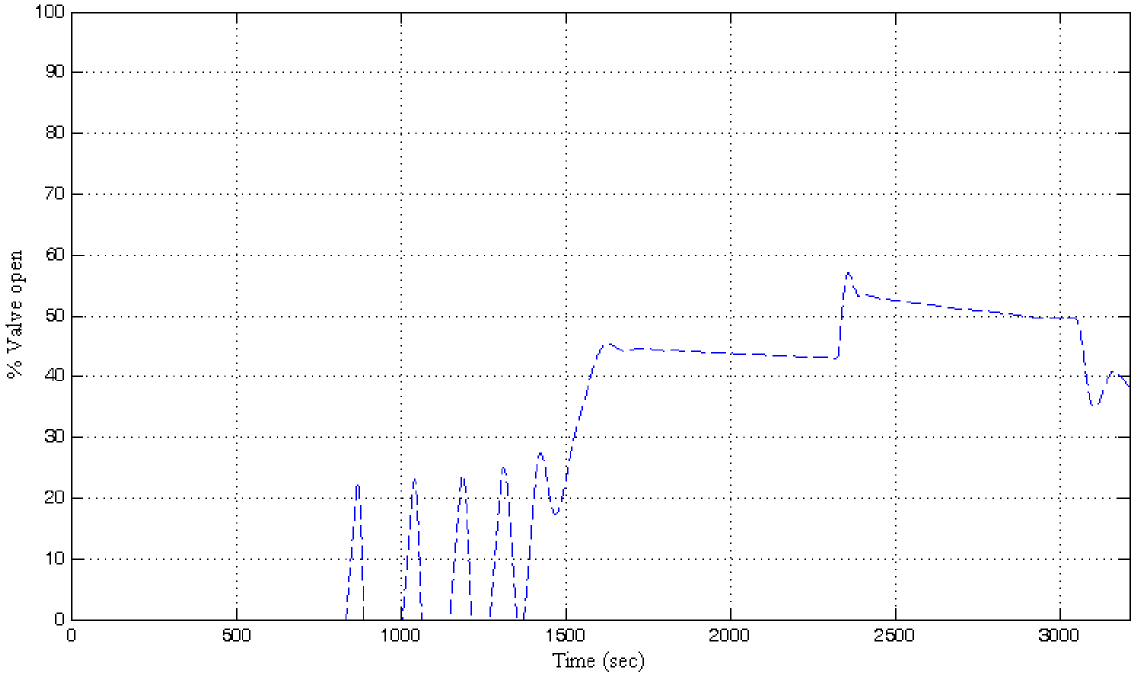


Fig. D.8: Simulated valve opening for engine test no. 2

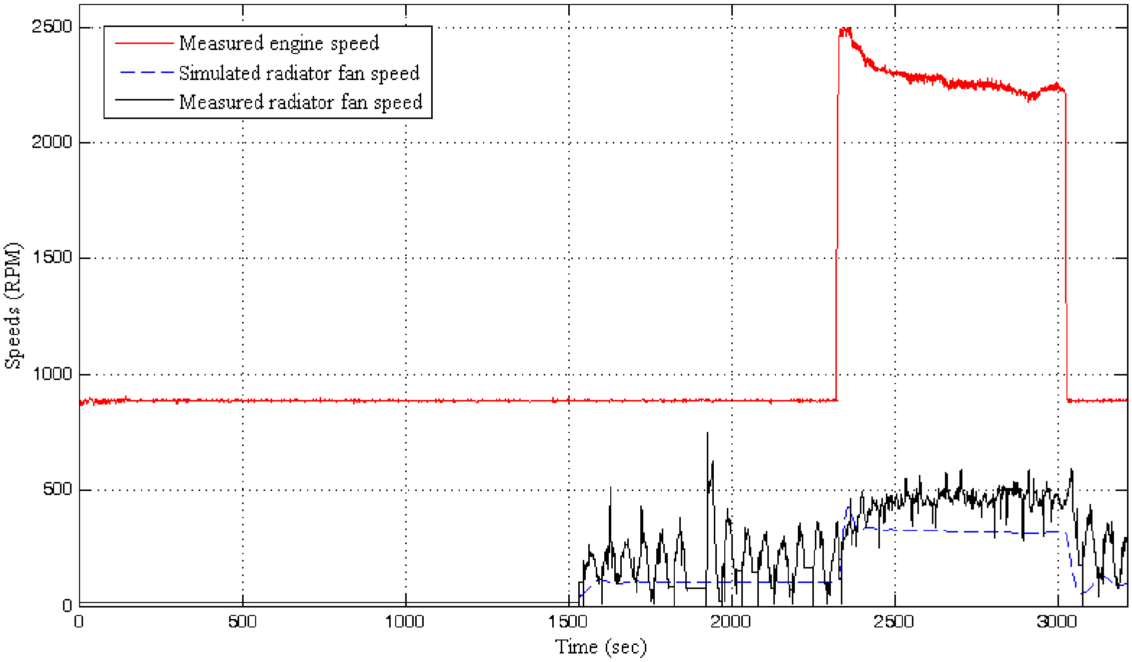


Fig. D.9: Simulated radiator fan speed in comparison with the experimental radiator fan speed and engine speed for engine test no. 2

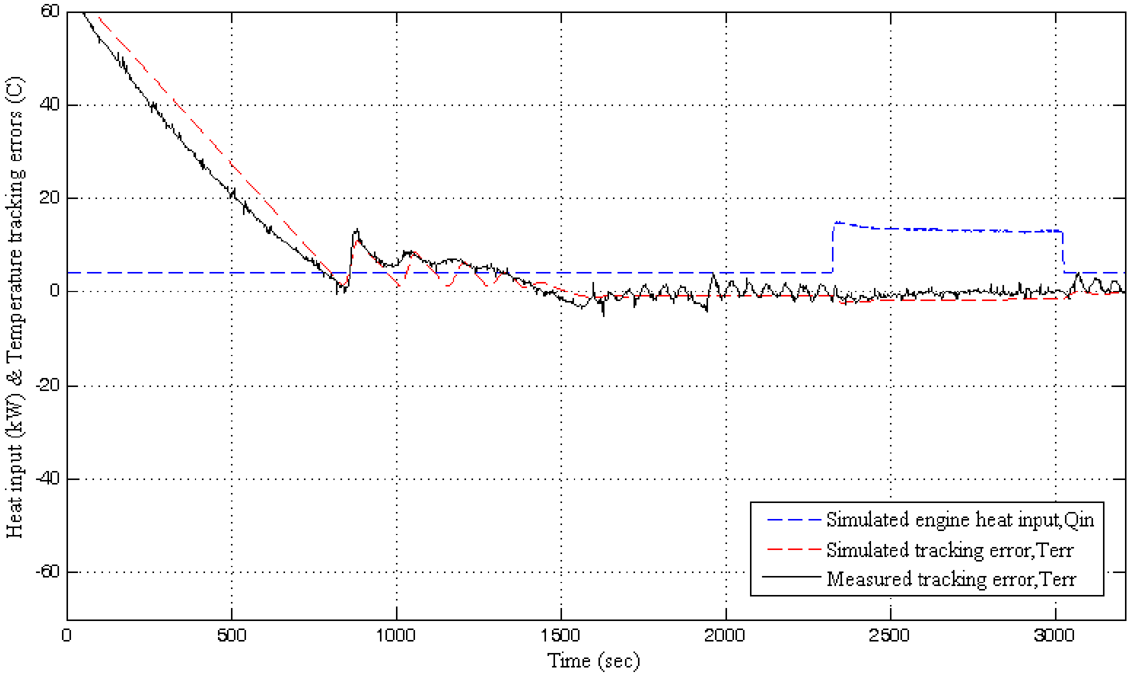


Fig. D.10: Estimated engine heat input and temperature tracking errors for simulated and experimental engine test no. 2

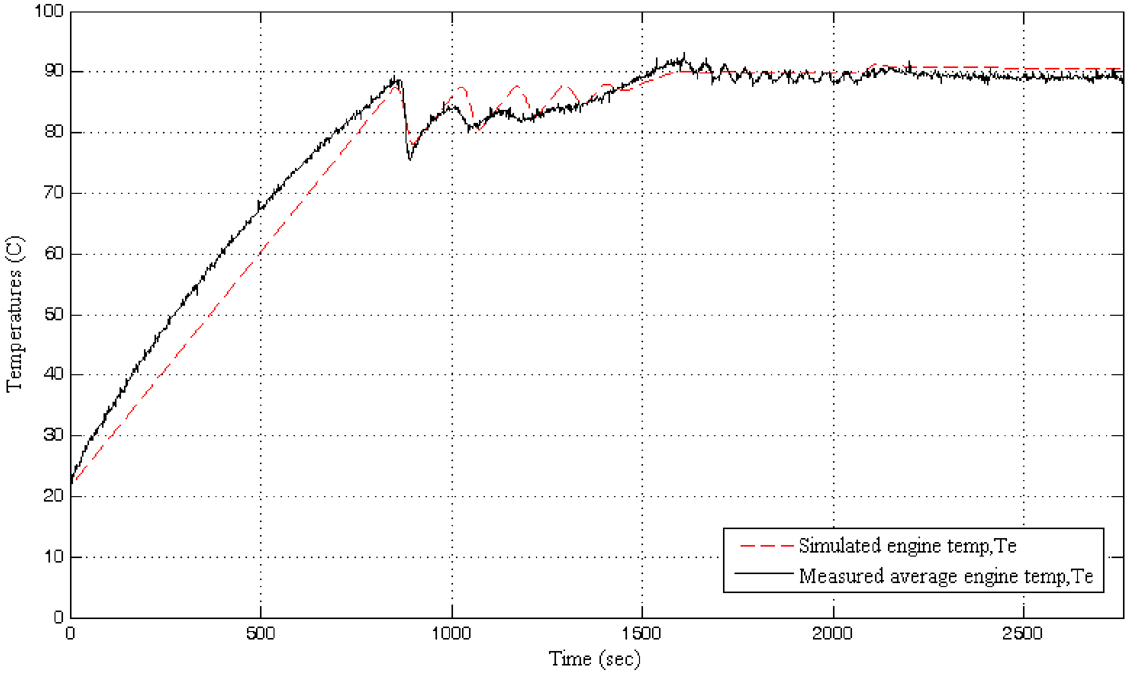


Fig. D.11: Simulated engine and coolant temperature in comparison with experimental average engine temperature for engine test no. 3

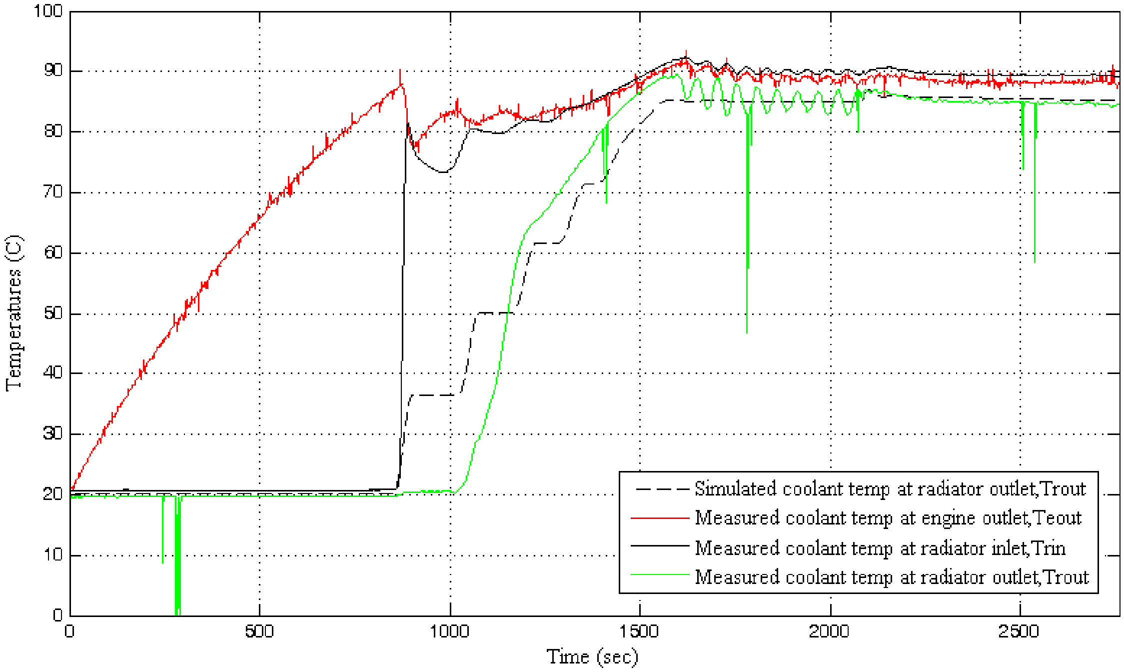


Fig. D.12: Simulated coolant temperatures at various locations in comparison with the experimental coolant temperatures at engine outlet, radiator inlet and radiator outlet for engine test no. 3

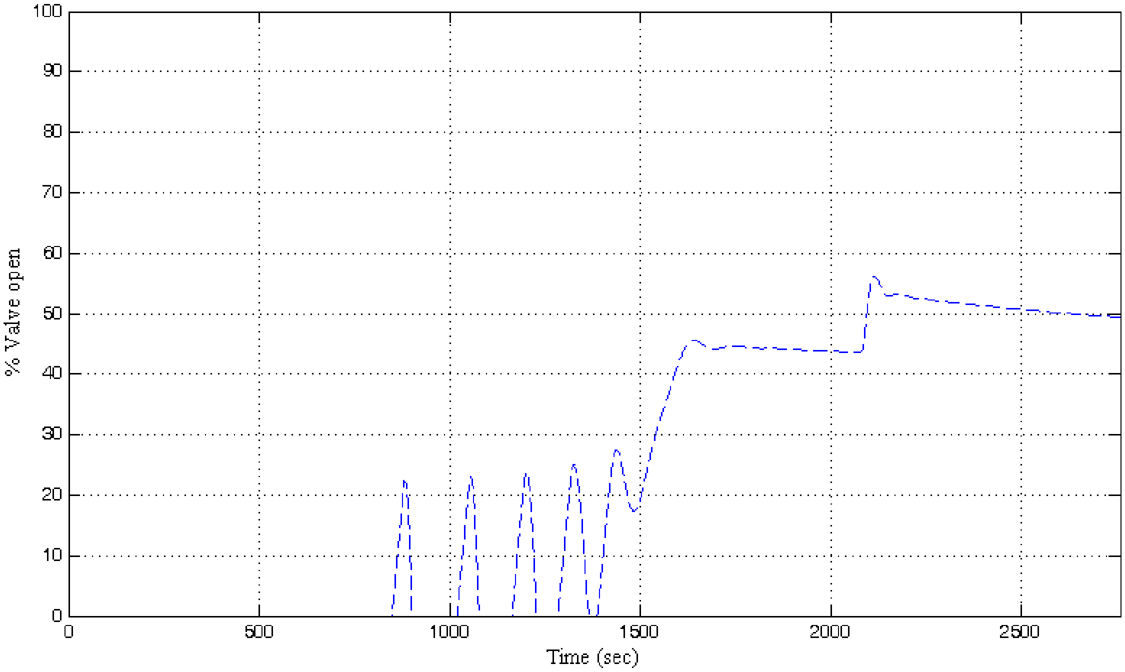


Fig. D.13: Simulated valve opening for engine test no. 3

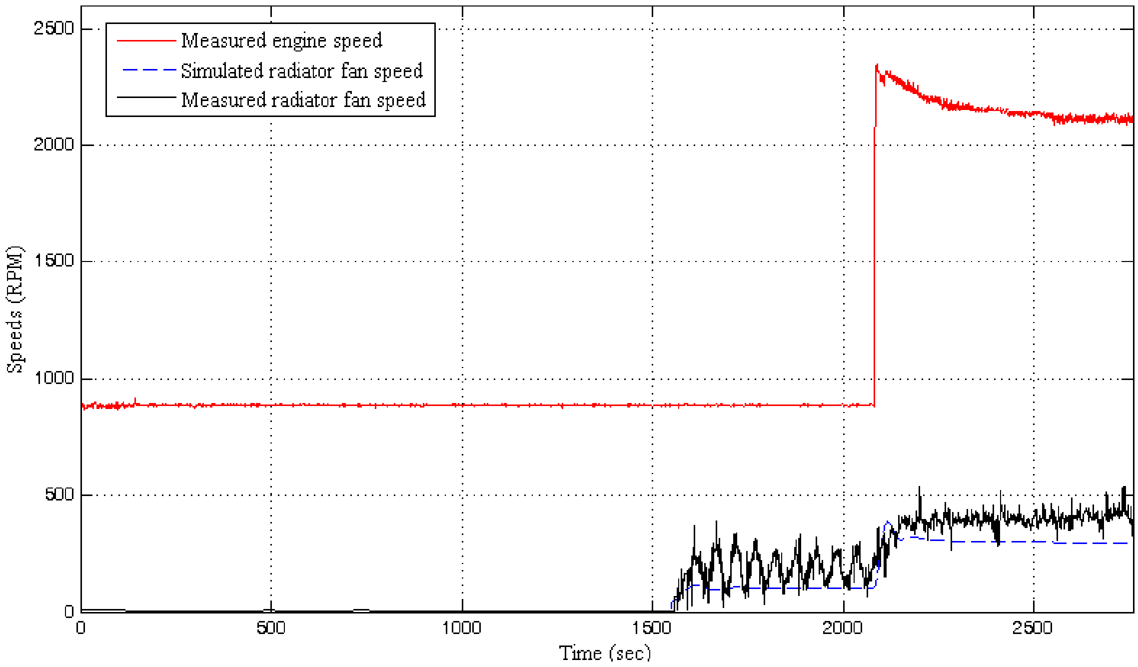


Fig. D.14: Simulated radiator fan speed in comparison with the experimental radiator fan speed and engine speed for engine test no. 3

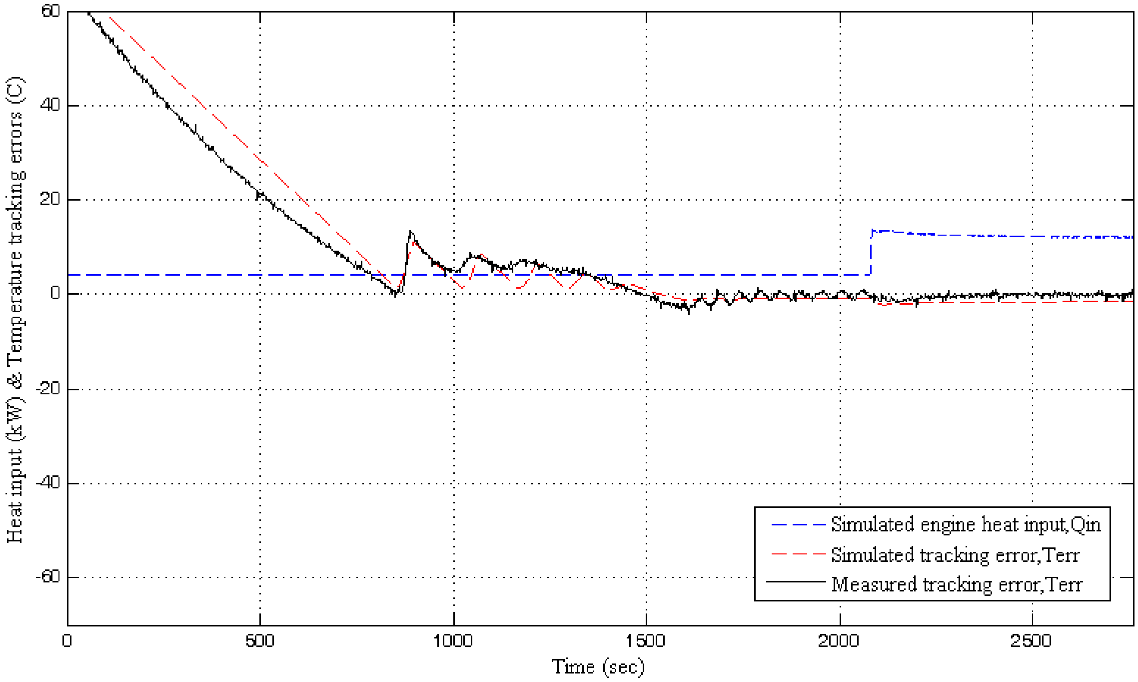


Fig. D.15: Estimated engine heat input and temperature tracking errors for simulated and experimental engine test no. 3

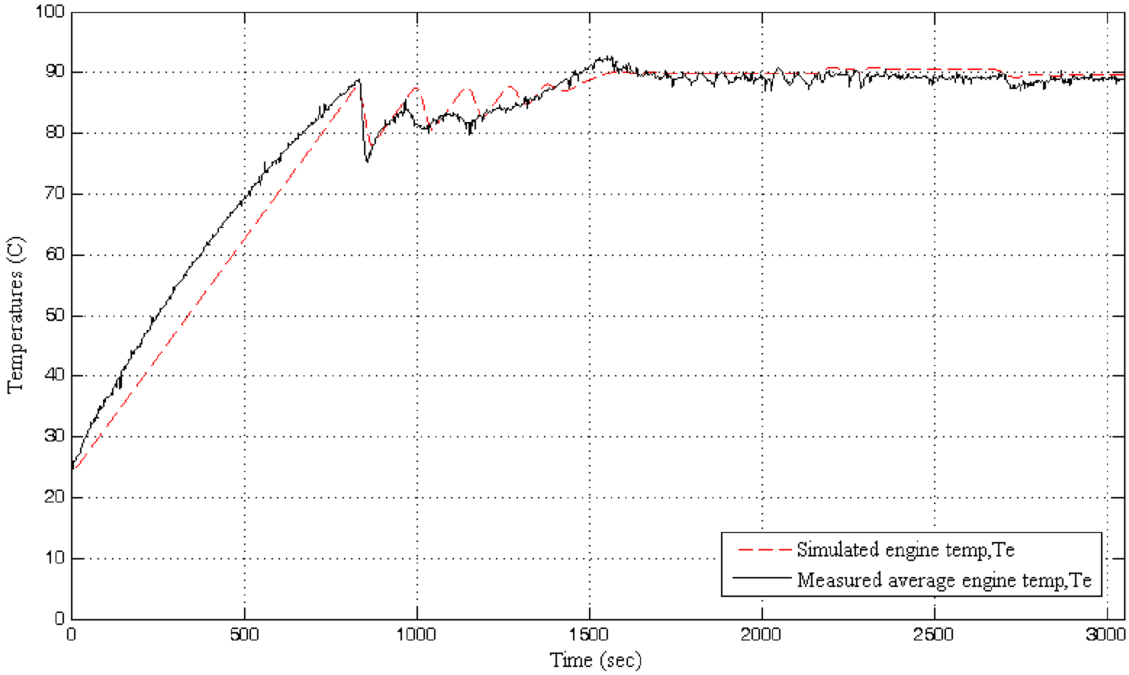


Fig. D.16: Simulated engine and coolant temperature in comparison with experimental average engine temperature for engine test no. 4

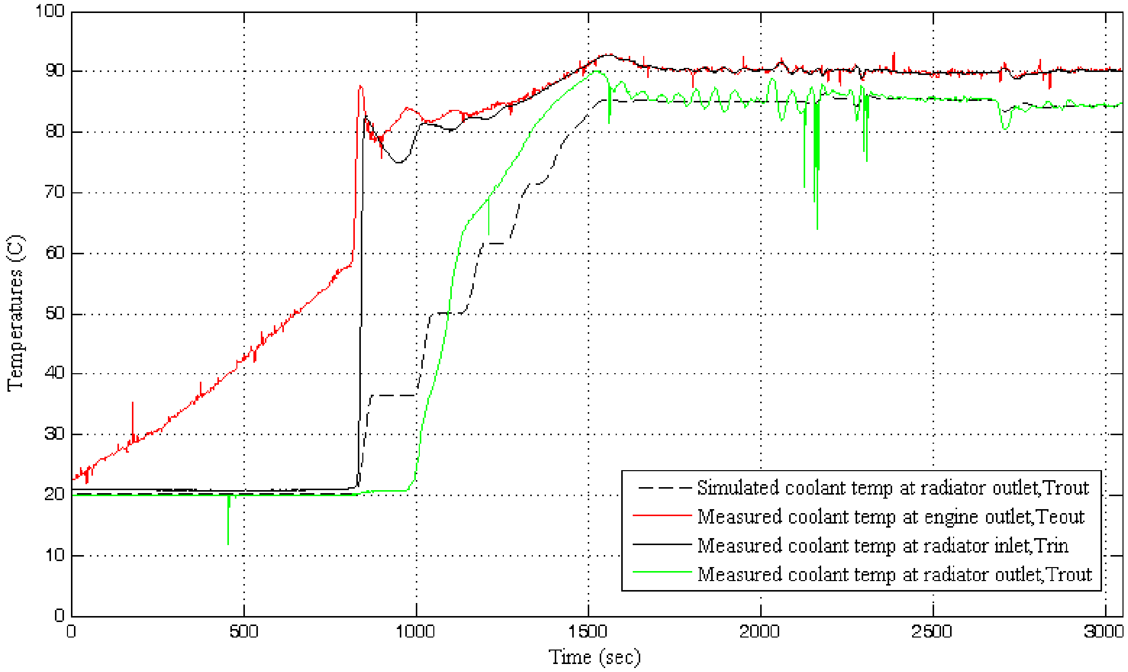


Fig. D.17: Simulated coolant temperatures at various locations in comparison with the experimental coolant temperatures at engine outlet, radiator inlet and radiator outlet for engine test no. 4

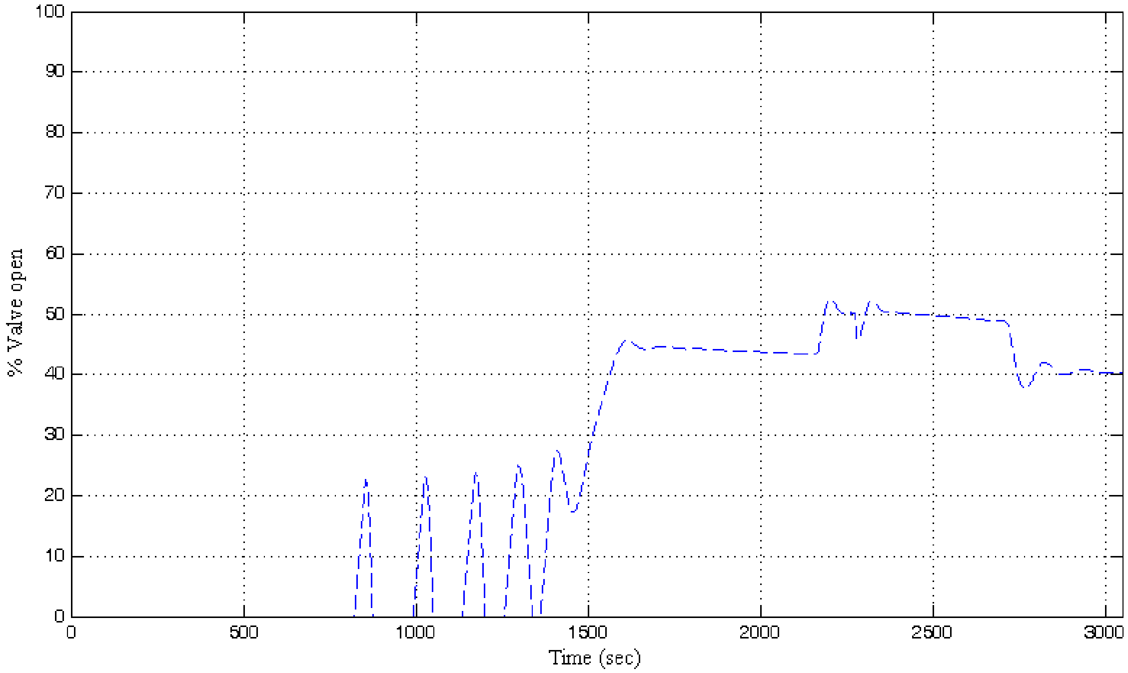


Fig. D.18: Simulated valve opening for engine test no. 4

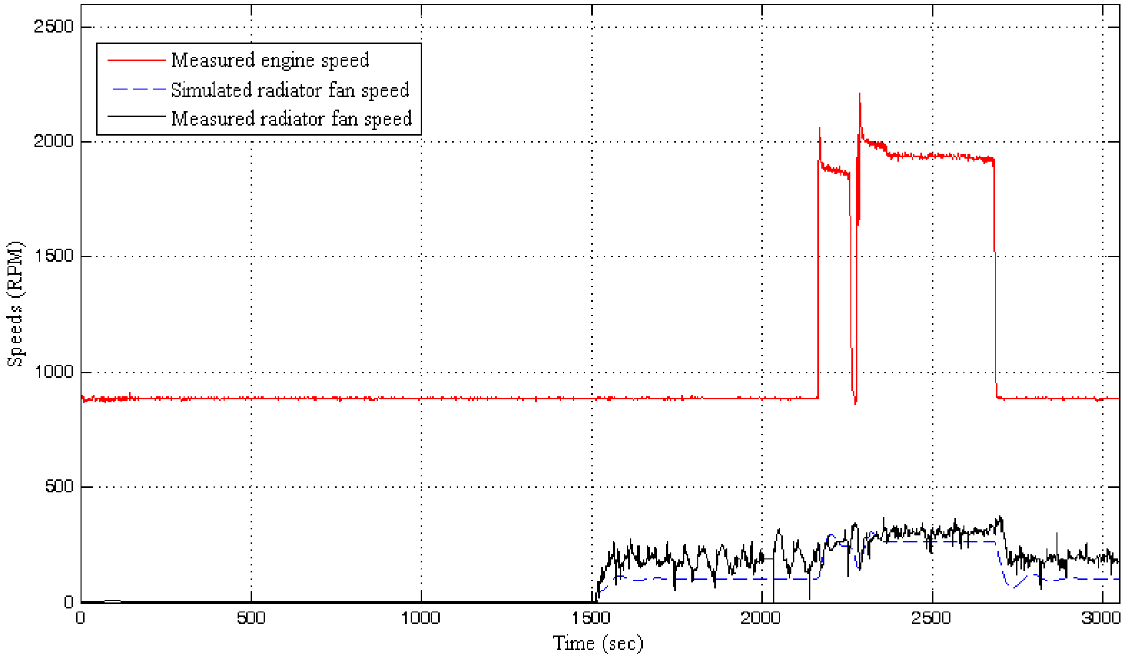


Fig. D.19: Simulated radiator fan speed in comparison with the experimental radiator fan speed and engine speed for engine test no. 4

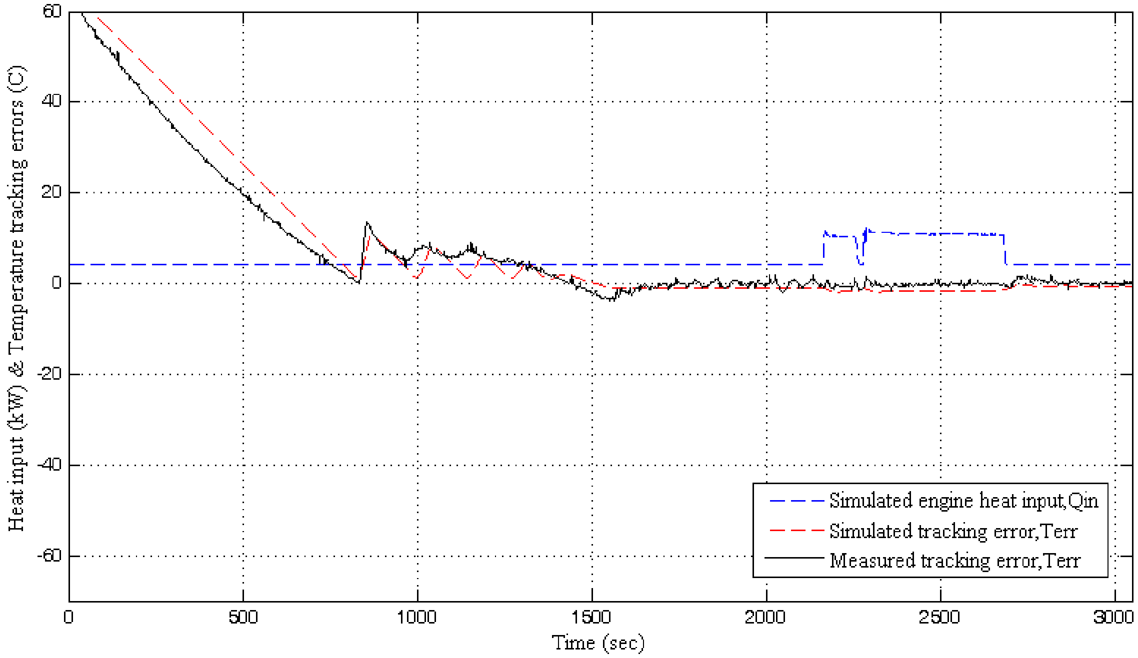


Fig. D.20: Estimated engine heat input and temperature tracking errors for simulated and experimental engine test no. 4

APPENDIX E: SIMULATION TEST PARAMETERS

Appendix E provides the values of various parameters used for simulated testing and for running simulation model of hydraulic based engine cooling system in the MATLAB/SimulinkTM.

```
clear all
clc
%% Defining parameters for engine thermal system management model.
% Hydraulic fluid properties
rho=0.03251;                % Hydraulic fluid density (lb/in^3)
                            % Beta=2.2*10^5;
                            % Hydraulic fluid Bulk modulus (lbs/in^2)
                            % mu2=2*10^-6; % Absolute
                            % viscosity (lb*s/in^2)

% Solenoid Model Parameters
L=0.02;                     % Coil inductance (H)
R=4.5;                      % Coil Resistance (Ohms)
N=1600;                     % Number of coil turns
mu0=4*pi*10^-7;             % Solenoid armature permeability (Henries/inch)
a=0.5394;                   % Solenoid contact length (in)
lg=0.0393;                  % Reluctance gap (in)

% Spool Valve Model Parameters
Cd=0.63;                    % Flow coefficient
w=1.963;                    % Approximate area gradient of orifice(in^2/in)
ms=10;                      % Mass of the spool (lbs)
bv= 40;                     % Spool damping (lb*s/in)
kv= 300;                    % Spool spring constant (lb/in)
phi=69*pi/180;              % Hydraulic fluid flow angle (rad)
Ld=0.5;                     % Damping length (in)

% Hydraulic Radiator Fan Motor Parameters
Dmf= 0.711/(2*pi);          % Motor Displacement (in^3/rad)
Cd = 0.63;                  % Discharge Coefficient
w = 3.35*0.425;             % Area Gradient (in^2/in)
Cimf =1/9506.97;            % Internal motor Leakage coefficient (in^5/(lb*s));
Betaf= 100000;              % Bulk Modulus (psi)
Vtf = 2250;                 % Total Compressed Volume (in^3)
Jf = 0.001;                 % Fan Inertia (lb*in^2)
```

```
Bmf = 0.47;                    % Motor Damping (lb*s/in)
eta_hm= 0.98;                  % Hydro-Mechanical efficiency
Ps_f=1000;                     % Hydraulic fluid supply pressure (psi)
Pt_f=0;                        % Hydraulic fluid tank return pressure (psi)

% Engine & Radiator Thermal Model Parameters
Cpc=2.36;                      % Specific heat of coolant (kJ/kgK)
Ce=54;                         % Engine thermal block capacity (kJ/K)
Cpa=1.01;                      % Air specific heat (kJ/kgK)
E=0.1;                         % Effectiveness of radiator fan (%)
Cr=44;                         % Radiator thermal capacity (kJ/K)
rho_a= 1.18;                   % Density of air (kg/m^3)
rho_c= 9.97e+02;               % Density of coolant (kg/m^3)
A_rad=4.1;                     % Radiator front area (ft^2)
T_inf= 293.35;                 % Ambient temperature (23 degree Celcius)
Engine_Load=0;                  % 147.5122 lb-ft i.e. 200 Nm

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% For determining the empirical relationship between control voltage %% and Hydraulic
% Line Pressures Pa, Pb, Ps, Pt

Fan_valve_Vf=[ 0 ; 0.27; 0.6; 0.96; 1.33; 1.76; 2.19; 2.65; 3.16; 3.7;
         4.3; 4.9; 5.28; 5.7; 6.1; 6.51; 6.93; 7.34 ];
Fan_Pa_f=[340; 380; 390; 394; 392; 389; 375; 371; 370; 370; 370; 370;
      375; 377; 377; 380; 383; 387];
Fan_Pb_f=[316; 350; 354; 346; 335; 323; 298; 288; 279; 267; 255; 242;
      237; 230; 220; 213; 206; 200];
Curve_Pa_coeff=polyfit(Fan_valve_Vf,Fan_Pa_f,6);
Curve_Pb_coeff=polyfit(Fan_valve_Vf,Fan_Pb_f,6);

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Engine Test 1 Data
% Opening the experimental engine testing data and defining the variable names for data
data=open('C:\Rajwardhan\Dr. Wagner\Engine Thermal Management
Project\Data\trial_data_16_may_fan_pid_control.mat');
% Time duration of experimental data set
time=data.trial_data_16_may_fan_pid_contr.X.Data;
% Engine speed recorded in experimental testing
engine_rpm=[time' data.trial_data_16_may_fan_pid_contr.Y(1,3).Data'];
% Radiator fan speed recorded in experimental testing
fan_rpm= [time' data.trial_data_16_may_fan_pid_contr.Y(1,2).Data'];
% Storing in a format which can be used in Simulink
fan_rpm1=data.trial_data_16_may_fan_pid_contr.Y(1,2).Data;
% Average engine temperature recorded in experimental engine testing
```

```
aver_engine_temp= data.trial_data_16_may_fan_pid_contr.Y(1,1).Data;
% Coolant temperature at engine outlet recorded
eng_out= data.trial_data_16_may_fan_pid_contr.Y(1,8).Data;
% Coolant temperature at radiator inlet recorded
radia_in= data.trial_data_16_may_fan_pid_contr.Y(1,10).Data;
% Coolant temperature at radiator outlet recorded
radia_out= data.trial_data_16_may_fan_pid_contr.Y(1,11).Data;
% Average temperature of radiator
radia=(radia_in+radia_out)/2;
size_of_matrix=length(time);
ref_temp=ones(1,size_of_matrix);
ref_temp=88*ref_temp;
% Tracking error between desired and actual engine temperature recorded
Terr=ref_temp-aver_engine_temp;
Simulation_time= 2888; % Unit: sec
% Reference temperature for tracking
Te_Ref=88;
Te_inf=273.15+aver_engine_temp(1); % Coverting temperature from celcius to kelvin
Tc_inf=Te_inf;
Ce=60;
% Program to remove recorded entries of infinity during experimental testing
for i=1:1:size_of_matrix
   if fan_rpm1(i)==inf
      fan_rpm(i,2)=0;
   end
end

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% The above mentioned comments will be same for rest of the experimental data set
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

%% Engine Test 2 Data
% data=open('C:\Rajwardhan\Dr. Wagner\Engine Thermal Management
% Project\Data\trial_data_20_may_new_eng_out_location.mat');
% time=data.trial_data_20_may_new_eng_out_l.X.Data;
% engine_rpm=[time'.data.trial_data_20_may_new_eng_out_l.Y(1,3).Data']
% fan_rpm= [time' data.trial_data_20_may_new_eng_out_l.Y(1,2).Data'];
% fan_rpm1=data.trial_data_20_may_new_eng_out_l.Y(1,2).Data;
% aver_engine_temp= data.trial_data_20_may_new_eng_out_l.Y(1,1).Data;
% coolant_temp= data.trial_data_20_may_new_eng_out_l.Y(1,8).Data;
% radia_in= data.trial_data_20_may_new_eng_out_l.Y(1,10).Data;
% radia_out= data.trial_data_20_may_new_eng_out_l.Y(1,11).Data;
% radia=(radia_in+radia_out)/2;
% eng_out=coolant_temp;
```

```
% size_of_matrix=length(time);
% ref_temp=ones(1,size_of_matrix);
% ref_temp=89*ref_temp;
% Terr=ref_temp-aver_engine_temp;
% Simulation_time= 3210; % Unit: sec
% Te_Ref=89;
% Te_inf=273.15+aver_engine_temp(1);
% Tc_inf=Te_inf;
% for i=1:1:size_of_matrix
%    if fan_rpm1(i)==inf
%        fan_rpm(i,2)=0;
%    end
% end

%% Engine Test 3 Data
% data=open('C:\Rajwardhan\Dr. Wagner\Engine Thermal Management
% Project\Data\trial_data_22_may_new_eng_out_location1.mat');
% time=data.trial_data_22_may_new_eng_out_l.X.Data;
% aver_engine_temp= data.trial_data_22_may_new_eng_out_l.Y(1,1).Data;
% fan_rpm= [ time'.data.trial_data_22_may_new_eng_out_l.Y(1,2).Data'];
% fan_rpm1=data.trial_data_22_may_new_eng_out_l.Y(1,2).Data;
% engine_rpm= [ time' data.trial_data_22_may_new_eng_out_l.Y(1,3).Data'];
% radia_in= data.trial_data_22_may_new_eng_out_l.Y(1,10).Data;
% radia_out= data.trial_data_22_may_new_eng_out_l.Y(1,11).Data;
% coolant_temp=data.trial_data_22_may_new_eng_out_l.Y(1,8).Data;
% radia=(radia_in+radia_out)/2;
% eng_out=coolant_temp;
% size_of_matrix=length(time);
% ref_temp=ones(1,size_of_matrix);
% ref_temp=89*ref_temp;
% Terr=ref_temp-aver_engine_temp;
% Simulation_time= 2760; % Unit: sec
% Te_Ref=89;
% Te_inf=273.15+aver_engine_temp(1);
% Tc_inf=Te_inf;
% for i=1:1:size_of_matrix
%    if fan_rpm1(i)==inf
%        fan_rpm(i,2)=0;
%    end
% end

%% Engine Test 4 Data
% data=open('C:\Rajwardhan\Dr. Wagner\Engine Thermal Management
% Project\Data\trial_data_may_19_pid_flow_per_sec.mat');
```

```
% time=data.trial_data_may_19_pid_flow_per_.X.Data;
% aver_engine_temp= data.trial_data_may_19_pid_flow_per_.Y(1,1).Data;
% fan_rpm1= data.trial_data_may_19_pid_flow_per_.Y(1,2).Data;
% fan_rpm=[time' fan_rpm1'];
% engine_rpm1= data.trial_data_may_19_pid_flow_per_.Y(1,3).Data;
% engine_rpm=[time' engine_rpm1'];
% eng_out= data.trial_data_may_19_pid_flow_per_.Y(1,8).Data;
% coolant_flow=data.trial_data_may_19_pid_flow_per_.Y(1,9).Data;
% radia_in= data.trial_data_may_19_pid_flow_per_.Y(1,10).Data;
% radia_out= data.trial_data_may_19_pid_flow_per_.Y(1,11).Data;
% radia=(radia_in+radia_out)/2;
% Coolant_ent=(eng_out+radia_out)/2;
% size_of_matrix=length(time);
% ref_temp=ones(1,size_of_matrix);
% ref_temp=89*ref_temp;
% Terr=ref_temp-aver_engine_temp;
% Simulation_time= 3049; % Unit: sec
% Te_Ref=89;
% Te_inf=273.15+aver_engine_temp(1);
% Tc_inf=Te_inf;
% for i=1:1:size_of_matrix
%     if fan_rpm1(i)==inf
%         fan_rpm(i,2)=0;
%     end
% end

%% Running the simulation in Simulink
open_system('ETSM_Simulation_for_thesis1.mdl');
sim('ETSM_Simulation_for_thesis1.mdl');

%% Plotting the simulation results along with the experimental data
figure(1) % For plotting simulated and experimental engine temperature
plot(simout.time,simout.signals.values(:,3),'--r');
grid on
hold on
plot(time,aver_engine_temp,'-k');
legend1=legend('Simulated engine temp,Te','Measured average engine temp,Te');
set(legend1,'Location','Best','FontSize',11,'FontName','Times New Roman');
axis([0 Simulation_time 0 100]);
xlabel('Time (sec)','FontSize',11,'FontName','Times New Roman');
ylabel('Temperatures (C)','FontSize',11,'FontName','Times New Roman');

figure(2) % To plot simulated coolant and experimental temperature at various locations
plot(simout.time,simout.signals.values(:,6),'--k');
```

```
hold on
plot(time,eng_out,'-r');
hold on
plot(time,radia_in,'-k');
hold on
plot(time,radia_out,'-g');
grid on
legend1=legend('Simulated coolant temp at radiator outlet,Trout','Measured coolant temp
at engine outlet,Teout','Measured coolant temp at radiator inlet,Trin','Measured coolant
temp at radiator outlet,Trout');
set(legend1,'Location','Best','FontSize',11,'FontName','Times New Roman');
axis([0 Simulation_time 0 100]);
xlabel('Time (sec)','FontSize',11,'FontName','Times New Roman');
ylabel('Temperatures (C)','FontSize',11,'FontName','Times New Roman');

figure(3) % To plot simulated valve opening percentage
plot(simout.time,simout.signals.values(:,4)*100,'--b');
grid on
axis([0 Simulation_time 0 100]);
xlabel('Time (sec)','FontSize',11,'FontName','Times New Roman');
ylabel('% Valve open','FontSize',11,'FontName','Times New Roman');

figure(4) % For plotting measured engine speed, simulated radiator fan speed along with
          % measured radiator fan speed
plot(simout.time,simout.signals.values(:,5),'-r');
grid on
hold on
plot(simout.time,simout.signals.values(:,7),'--b');
hold on
plot(time,fan_rpm1,'-k');
legend1=legend('Measured engine speed','Simulated radiator fan speed','Measured
radiator fan speed');
set(legend1,'Location','Best','FontSize',11,'FontName','Times New Roman');
axis([0 Simulation_time 0 2600]);
xlabel('Time (sec)','FontSize',11,'FontName','Times New Roman');
ylabel('Speeds (RPM)','FontSize',11,'FontName','Times New Roman');

figure(5) % To plot engine temperature tracking errors of simulation and experimental
          % testing of engine, and estimated heat input to system
plot(simout.time,simout.signals.values(:,1),'--b');
grid on
hold on
plot(simout.time,simout.signals.values(:,2),'--r');
hold on
```

```
plot(time,Terr,'-k');
legend1=legend('Simulated engine heat input,Qin','Simulated tracking
error,Terr','Measured tracking error,Terr');
set(legend1,'Location','Best','FontSize',11,'FontName','Times New Roman');
axis([0 Simulation_time -70 60]);
xlabel('Time (sec)','FontSize',11,'FontName','Times New Roman');
ylabel('Heat input (kW) & Temperature tracking errors
(C)','FontSize',11,'FontName','Times New Roman');
```

APPENDIX F: SIMULATION ALGORITHM FOR ENGINE THERMAL SYSTEM MANAGEMENT

Appendix F presents the simulation algorithm in the Matlab/Simulink software package for the hydraulic based engine cooling system (Fig. F.1 – Fig. F.16).

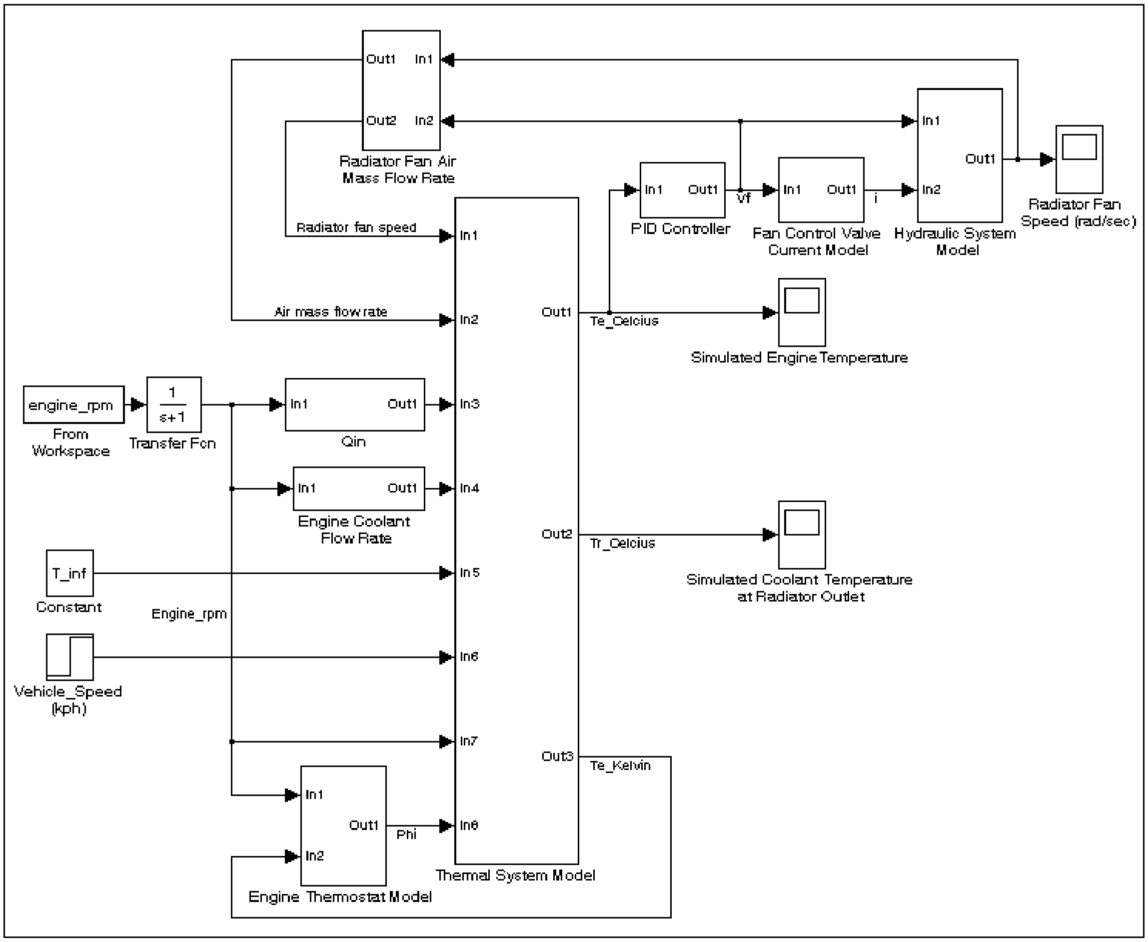


Fig. F.1: Simulink algorithm for simulated model of hydraulic based engine thermal system management, 1^{st} level

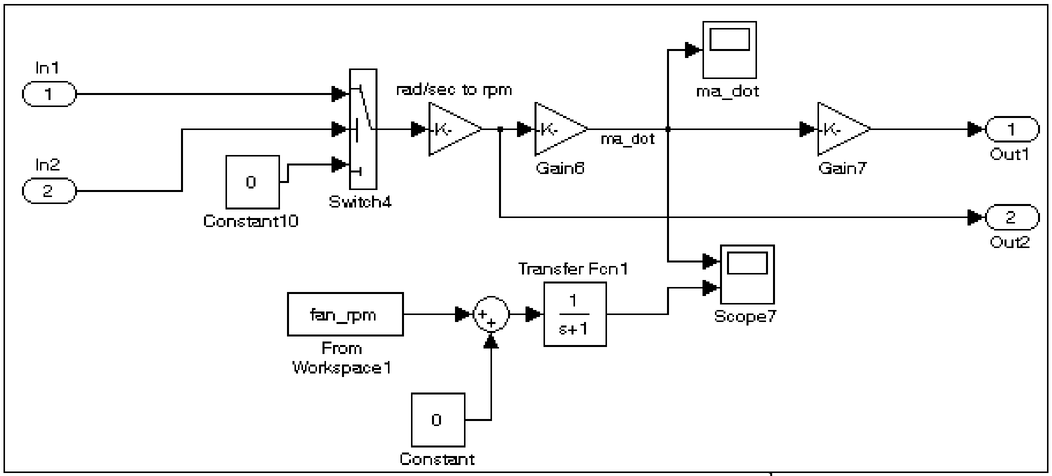


Fig. F.2: Radiator fan air mass flow rate, 2nd level

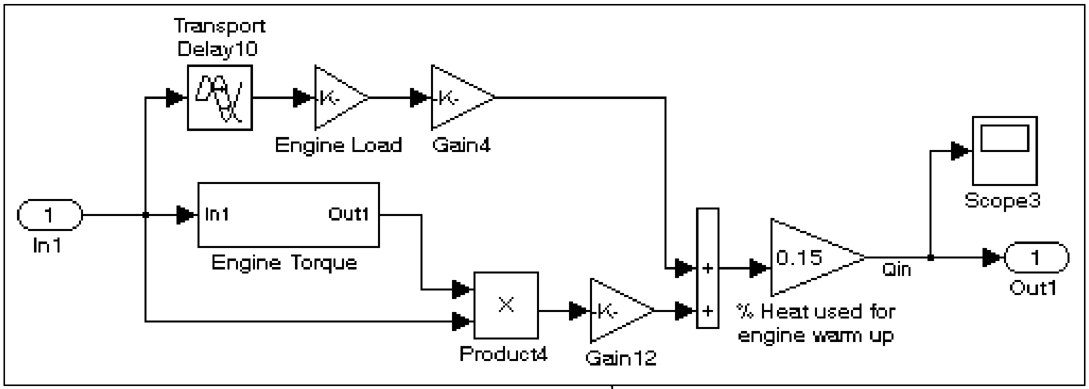


Fig. F.3: Q_{in}, 2nd level

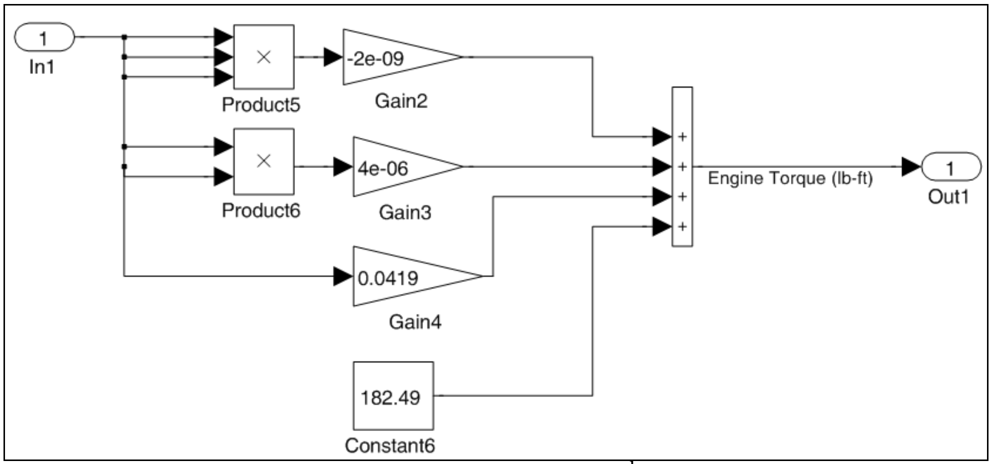


Fig. F.4: Engine torque, 3rd level

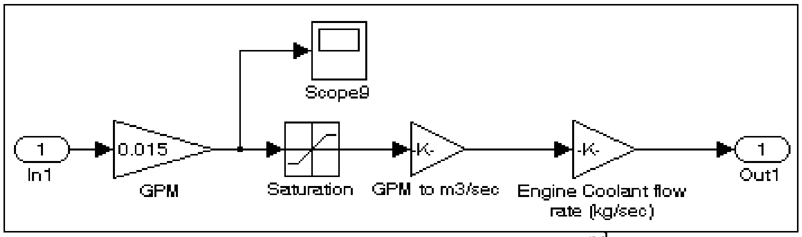


Fig. F.5: Engine coolant flow rate, 2nd level

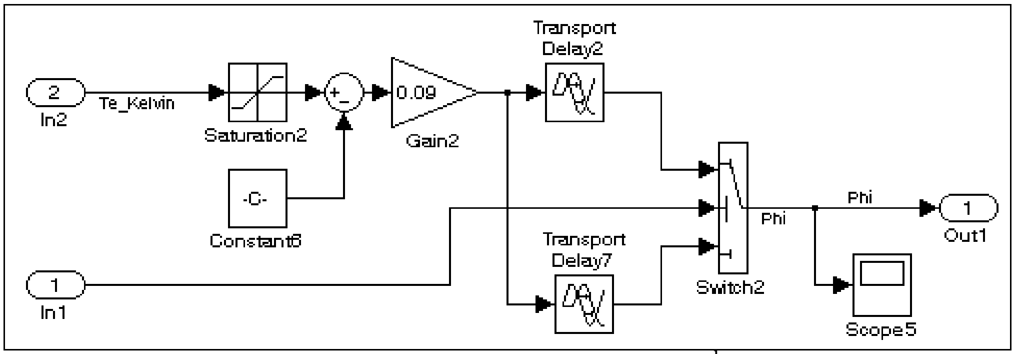


Fig. F.6: Engine thermostat model, 2nd level

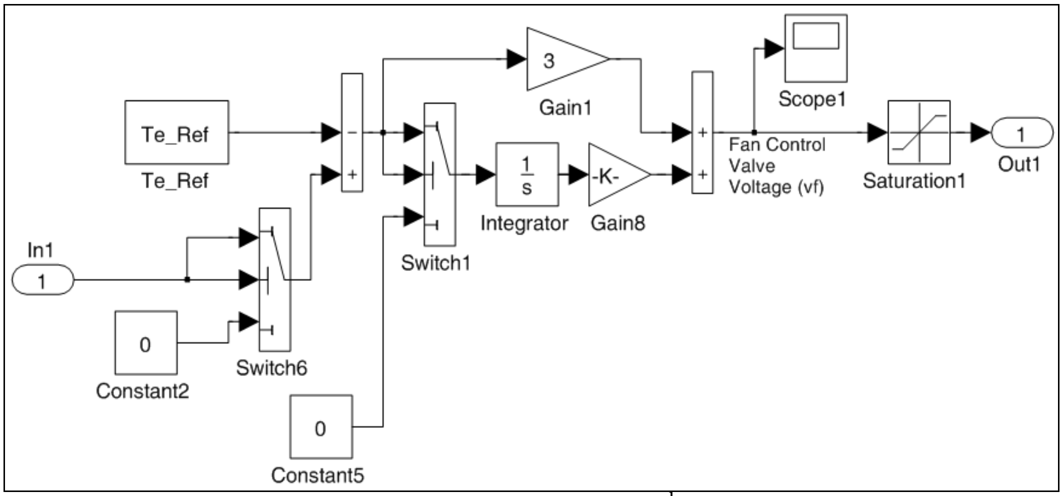


Fig. F.7: PID controller, 2nd level

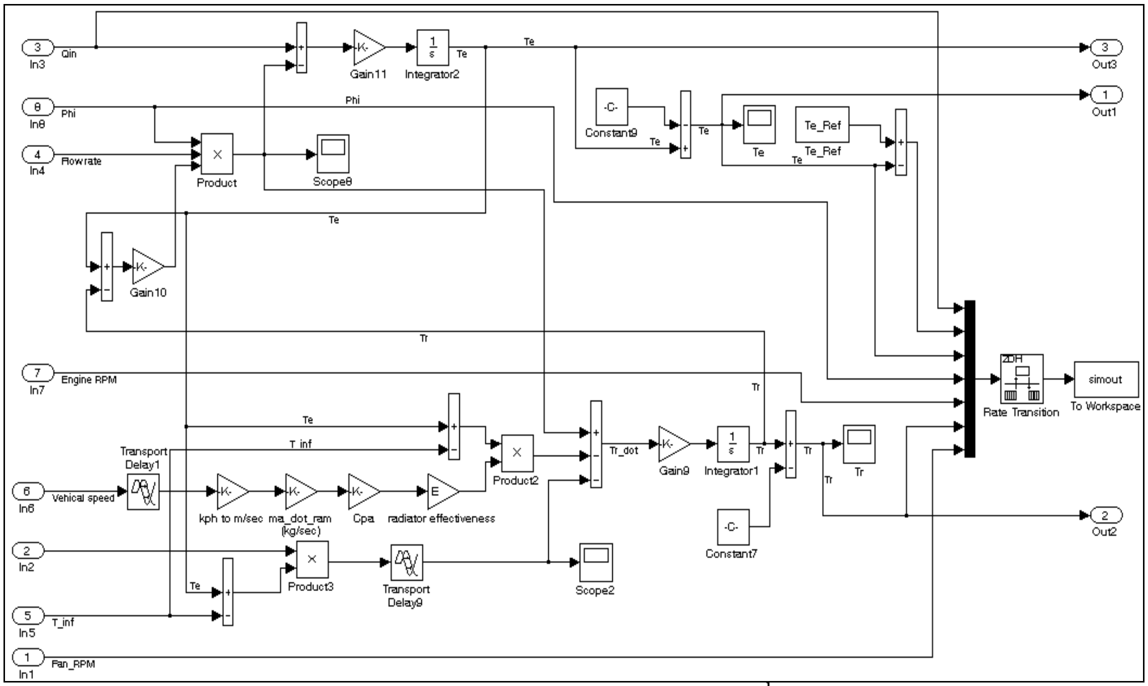


Fig. F.8: Thermal system model, 2^{nd} level

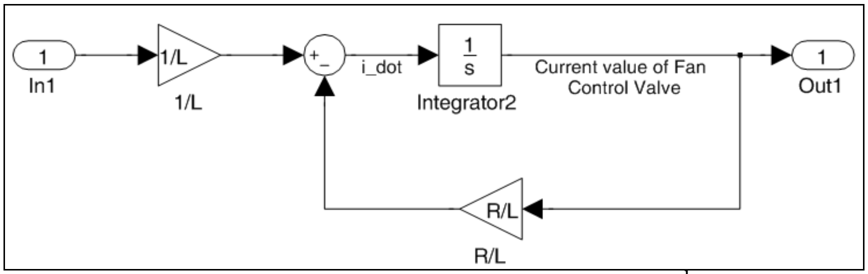


Fig. F.9: Fan control valve current model, 2^{nd} level

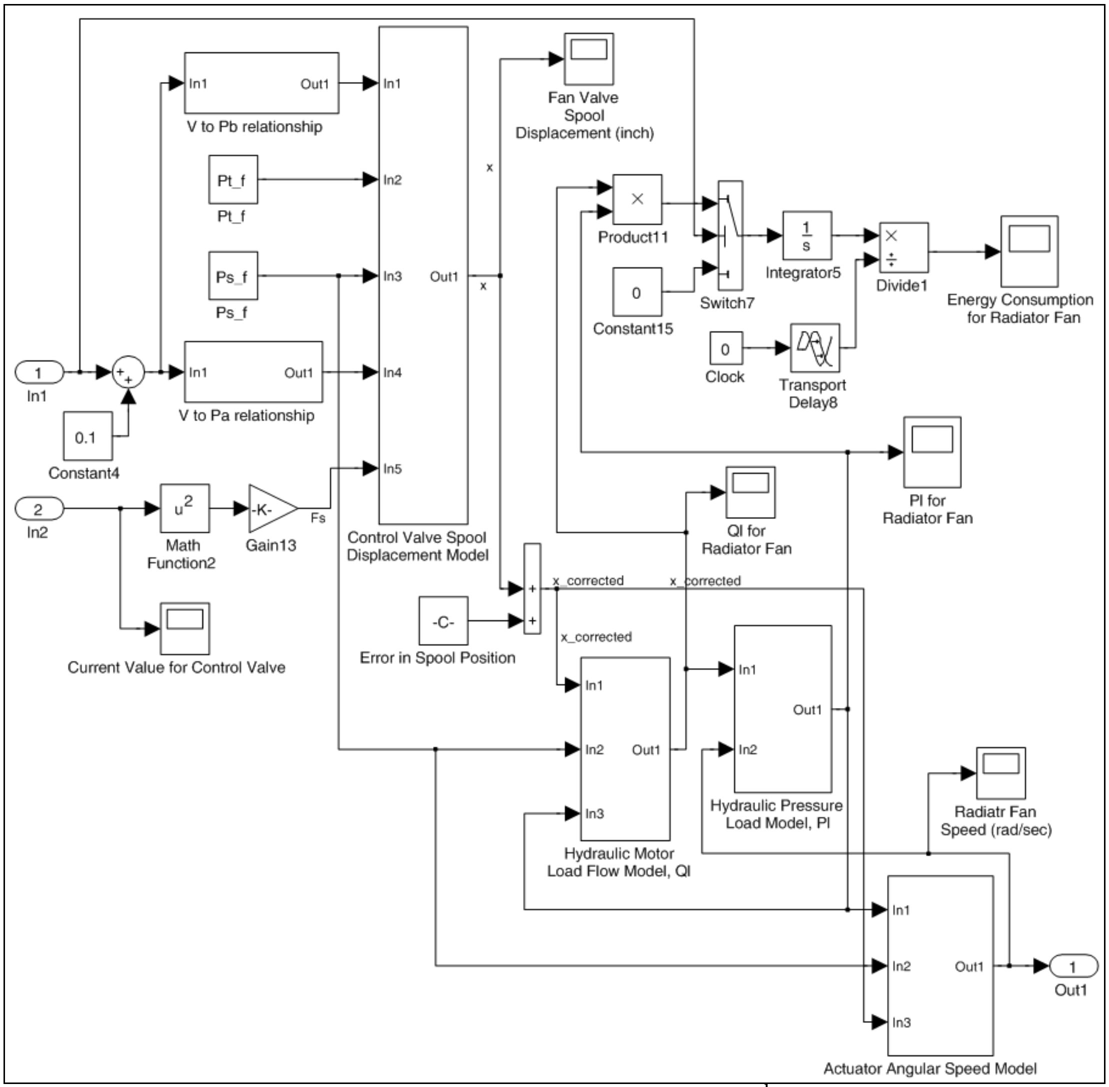


Fig. F.10: Hydraulic system model, 2nd level

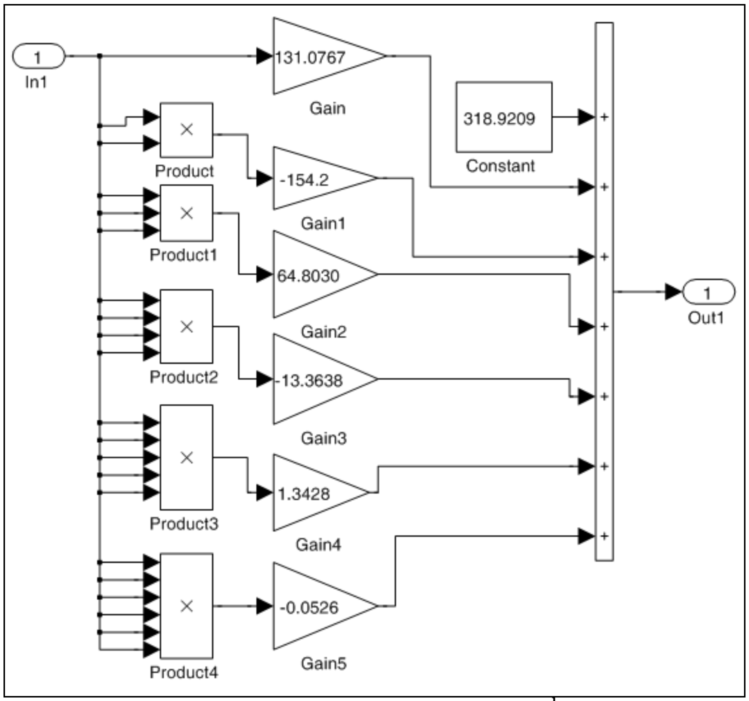


Fig. F.11: V to P_b relationship, 3rd level

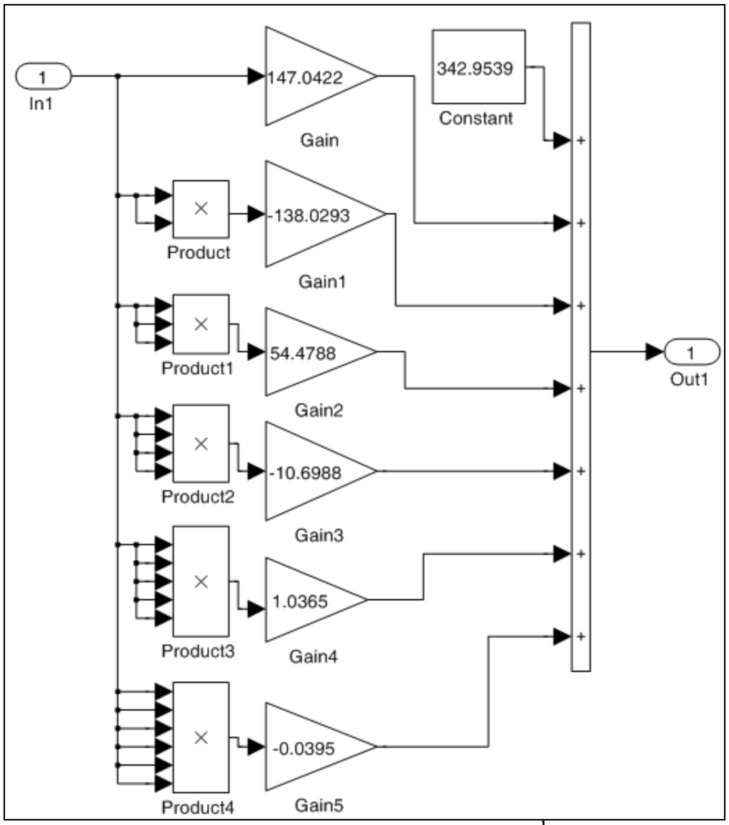


Fig. F.12: V to P_a relationship, 3rd level

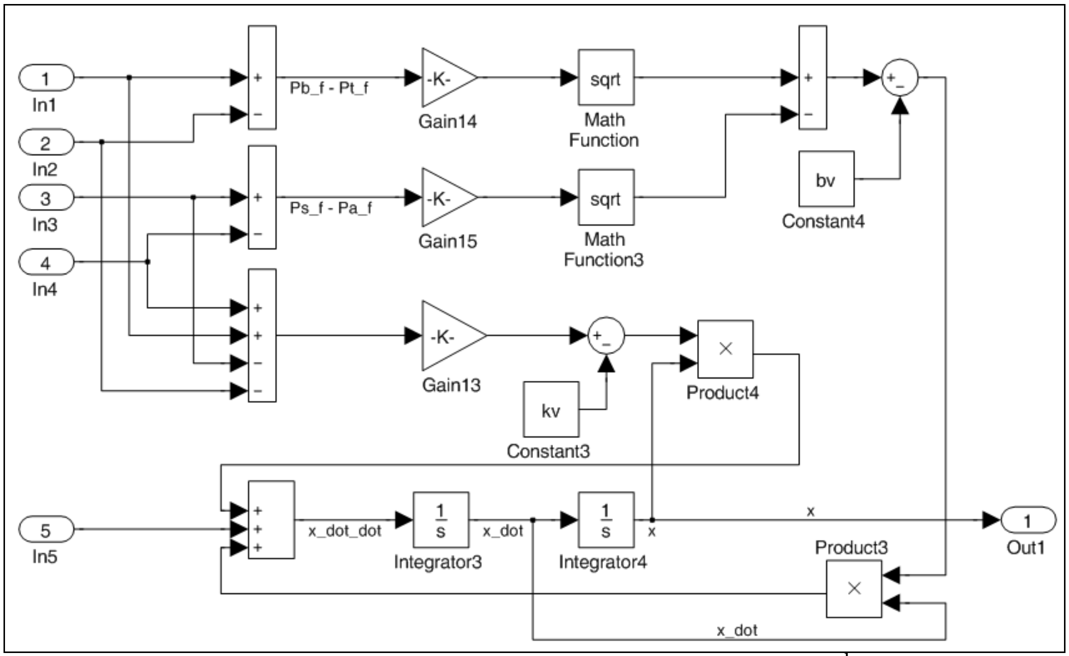


Fig. F.13: Control valve spool displacement model, 3^{rd} level

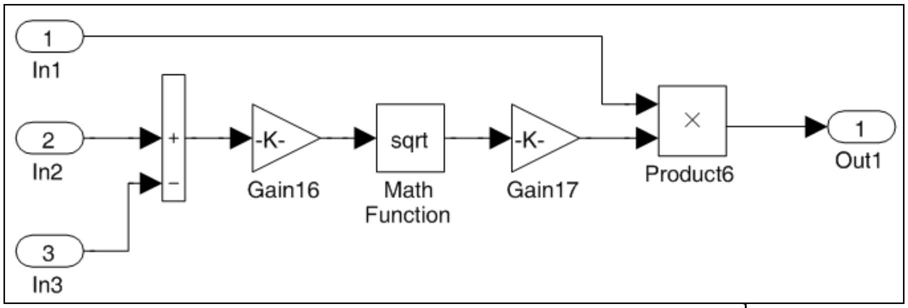


Fig. F.14: Hydraulic motor load flow model, Q_l, 3^{rd} level

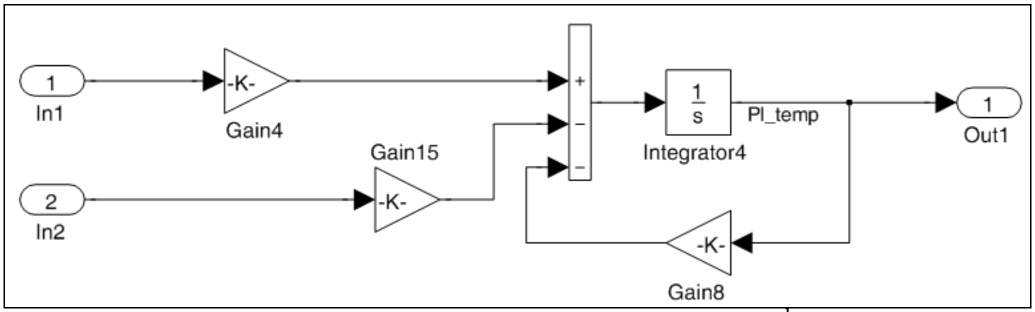


Fig. F.15: Hydraulic pressure load model, P_l, 3^{rd} level

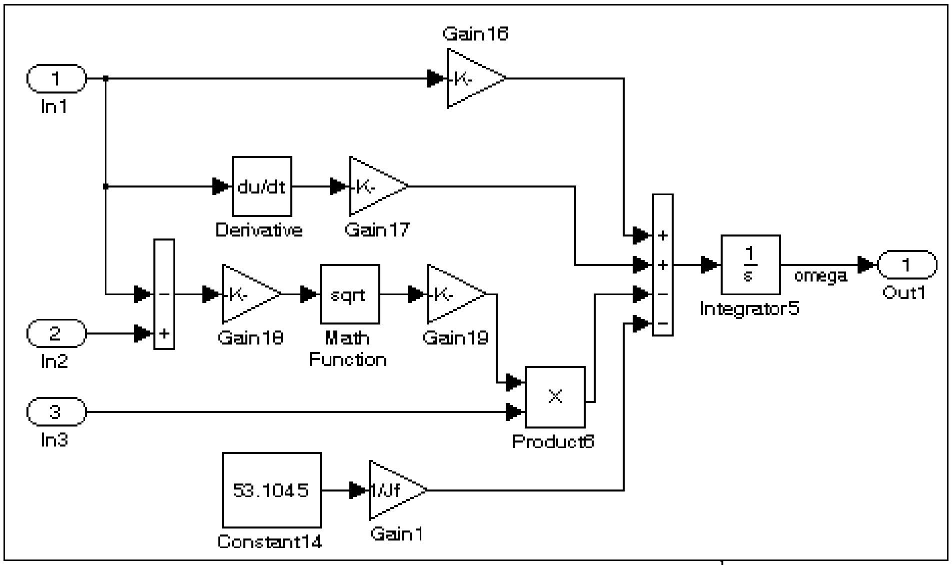


Fig. F.16: Actuator angular speed model, 3rd level

APPENDIX G: ALTERNATIVE APPROACH TO MODEL ENGINE THERMAL MANAGEMENT SYSTEM

Appendix G provides the alternative approach to refine the mathematical models presented in the Section 2.3. The mathematical models used in the analysis and control of the proposed engine thermal management system can be modeled using a lumped parameter approach to realize the governing differential equations.

Engine and Radiator Thermal Dynamics

The thermal behavior of the engine has been modeled using a lumped parameter modeling strategy. The removal of heat from the engine block due to fuel combustion is primarily through convective heat transfer. Four assumptions have been imposed to derive the governing equations:

A.1 No heat losses occur in radiator hoses due to insulations.

A.2 Lumped temperatures for engine block, engine coolant, and radiator block are adequate to describe general thermal behavior.

A.3 Radiator fluid has been separated into two temperature nodes to reflect fluid inlet and exit.

A.4 Primary heat transfer mode is convection with secondary heat transfer modes as radiation and conduction.

The heat balance equations can be well understood by considering the temperature nodes and thermal system network as shown in Fig. G.1 and Fig. G.2. The thermal resistance and capacitance equations used to define the thermal system are as follows (Paradis, 2001, Frick *et al.*, 2006).

$$C_e\dot{T}_e = Q_{in} - \frac{1}{R_1}(T_e - T_C) - \frac{1}{R_4}(T_e - T_\infty)$$

$$C_C\dot{T}_C = \frac{-1}{R_1}(T_C - T_e) - \frac{1}{R_2}(T_C - T_{rin}) - \frac{1}{R_3}(T_C - T_{rout})$$

$$C_{rin}\dot{T}_{rin} = \frac{-1}{R_2}(T_{rin} - T_C) - \frac{1}{R_6}(T_{rin} - T_r) - \frac{1}{R_5}(T_{rin} - T_{rout})$$

$$C_{rout}\dot{T}_{rout} = \frac{-1}{R_5}(T_{rout} - T_{rin}) - \frac{1}{R_3}(T_{rout} - T_C) - \frac{1}{R_7}(T_{rout} - T_r)$$

$$C_r\dot{T}_r = -Q_o - \frac{1}{R_6}(T_r - T_{rin}) - \frac{1}{R_7}(T_r - T_{rout}) - \frac{1}{R_8}(T_r - T_\infty)$$

where the thermal components used in above equations are presented as shown in the Table G.1.

Table G.1: Formulas for thermal heat transfer model

Thermal Resistance and Capacitance	Formula
Convection	$R = \frac{1}{h_{fluid}A_{fluid}}$ or $R = \frac{1}{C_{p,fluid}\dot{m}_{fluid}}$
Conduction	$R = \frac{L_{cd}}{k_{cd}A_{cd}}$
Radiation	$R = \frac{T_i - T_j}{A_{rd}\sigma\varepsilon_{rd}(T_i^4 - T_j^4)}$
Capacitance	$C = m_{node}C_{p,node}$

The values for the thermal resistances, *R*, are given as

$$R_1 = \frac{1}{C_{pc}\dot{m}_c(1-\phi)} + \frac{1}{h_{ein}A_{ein}}, \quad R_2 = \frac{1}{C_{pc}\dot{m}_c\phi} + \frac{L_{hose,inlet}}{k_C A_{in,hose}},$$

$$R_3 = \frac{1}{C_{pc}\dot{m}_c\phi} + \frac{L_{hose,outlet}}{k_C A_{in,hose}}, \quad R_4 = \left[\left(\frac{1}{h_\infty A_e} \right)^{-1} + \left(\frac{(T_e - T_\infty)}{A_e \sigma \varepsilon_e \left(T_e^4 - T_\infty^4 \right)} \right)^{-1} \right]^{-1},$$

$$R_5 = \frac{1}{C_{pc}\dot{m}_c\phi} + \frac{L_{tube}}{k_C A_{in,tube}}, \quad R_6 = \frac{1}{C_{pc}\dot{m}_c\phi} + \frac{t_{tube}}{k_{al} A_{inlet,tube}},$$

$$R_7 = \frac{1}{C_{pc}\dot{m}_c\phi} + \frac{t_{tube}}{k_{al} A_{outlet,tube}}, \quad R_8 = \frac{1}{\varepsilon C_{pa}\dot{m}_{a_{fan}}} + \frac{1}{h_\infty A_{r,fin}}.$$

The system variables used in the above equations, $Q_{in}(t), Q_o(t), \dot{m}_c = f(\omega_e)$, and $\dot{m}_{a_{fan}} = f(\omega_{fan}, V_{speed})$ represent the rate of heat used to warm up the engine, rate of heat lost at the radiator due to ram air flow (considering vehicle speed), empirically derived coolant and air mass flow rate, respectively. The rate of heat used to warm up the engine, $Q_{in}(t)$ has been estimated to be proportional to the rated engine power output for a given engine speed and load. The rate of heat lost by the radiator due to ram air can be given as (Incropera and DeWitt, 2002)

$$Q_o(t) = \frac{1}{R_9}(T_r - T_\infty)$$

where $R_9 = \frac{1}{\varepsilon C_{pa}\dot{m}_{a_{ram}}} + \frac{1}{h_\infty A_{r,fin}}$, and $\dot{m}_{a_{ram}} = V_{speed} A_{rad} \rho_a$.

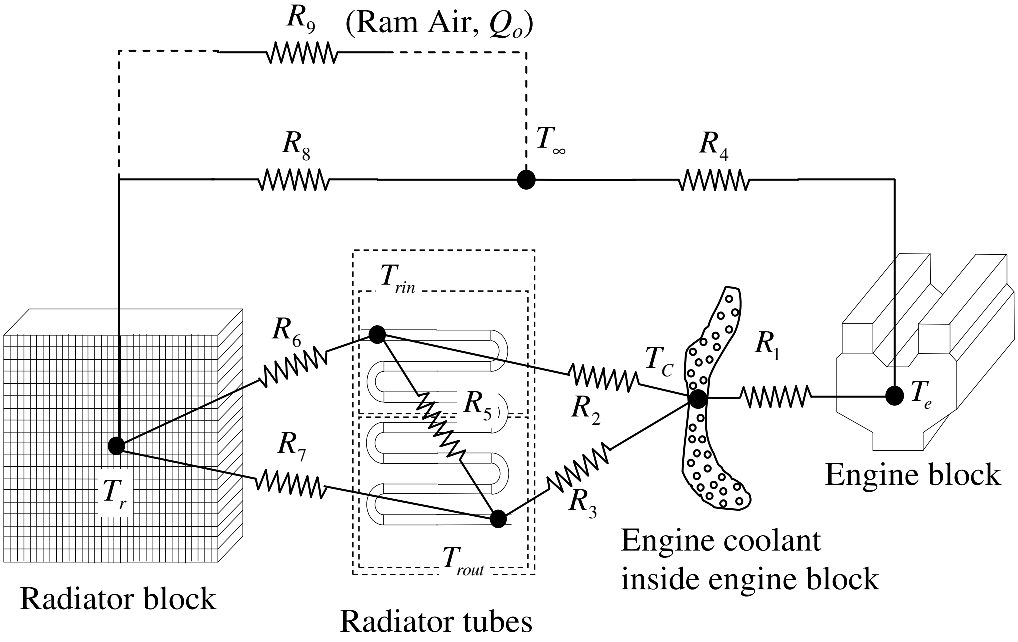


Fig. G.1: Thermal system network showing the locations of the thermal nodes and thermal resistances used in the model

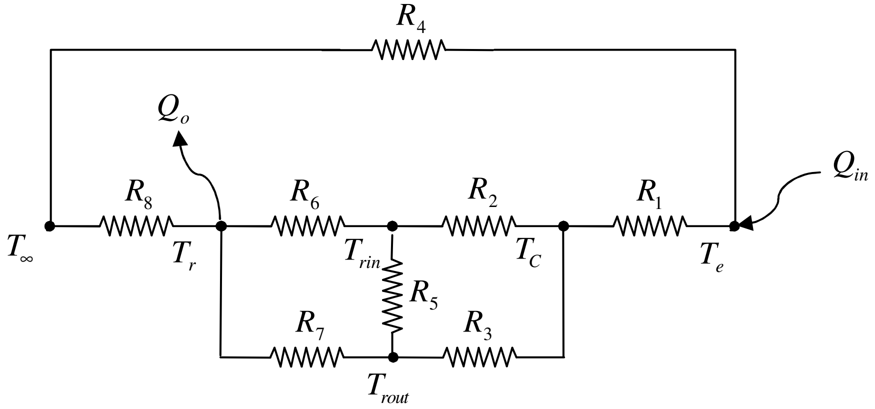
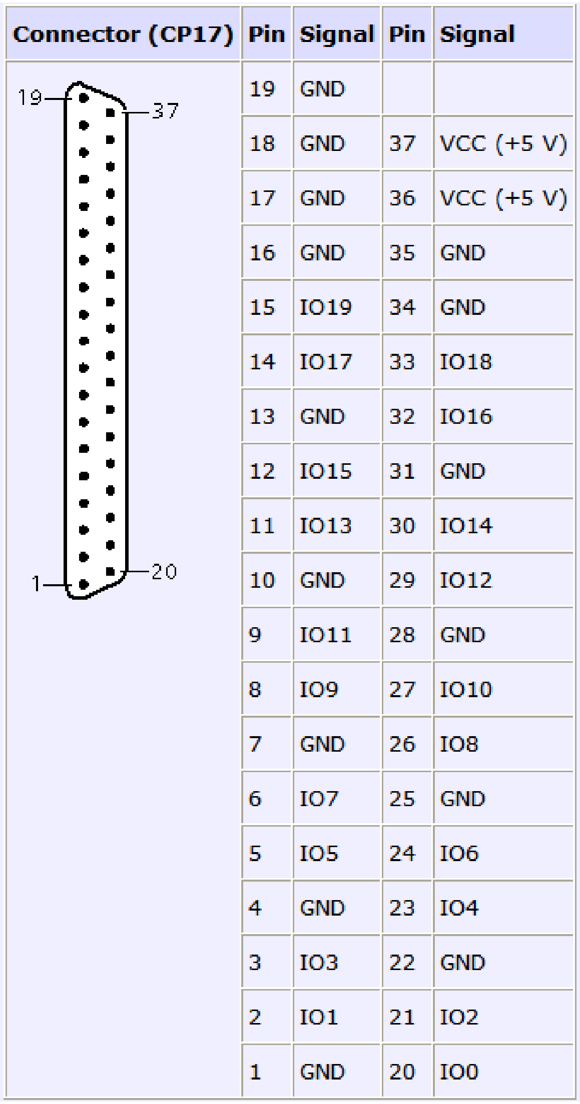


Fig. G.2: Thermal system network for the engine block, coolant, radiator, and ambient temperature nodes with resistance elements

APPENDIX H: DATA ACQUISITION SYSTEM DETAILS

Appendix H provides the information about the dSpace - DS1104RTLib data acquisition system and multiplexer for collecting the real time data from experimental tests (Fig. H.1 – Fig. H.4).



Connector (CP17)	Pin	Signal	Pin	Signal
	19	GND		
	18	GND	37	VCC (+5 V)
	17	GND	36	VCC (+5 V)
	16	GND	35	GND
	15	IO19	34	GND
	14	IO17	33	IO18
	13	GND	32	IO16
	12	IO15	31	GND
	11	IO13	30	IO14
	10	GND	29	IO12
	9	IO11	28	GND
	8	IO9	27	IO10
	7	GND	26	IO8
	6	IO7	25	GND
	5	IO5	24	IO6
	4	GND	23	IO4
	3	IO3	22	GND
	2	IO1	21	IO2
	1	GND	20	IO0

Fig. H.1: DS1104RTLib connector CP17 pin diagram

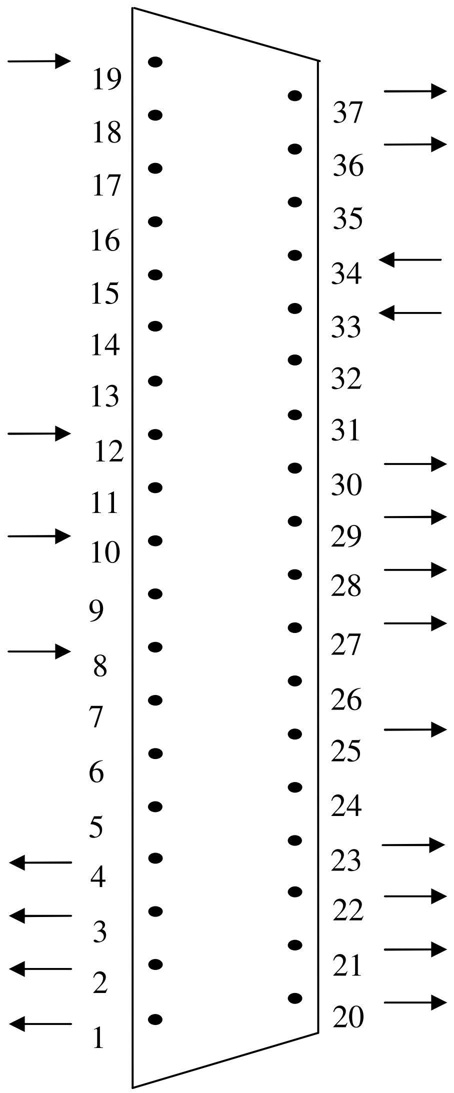


Fig. H.2: Directions of signal coming in and out of Pin CP17

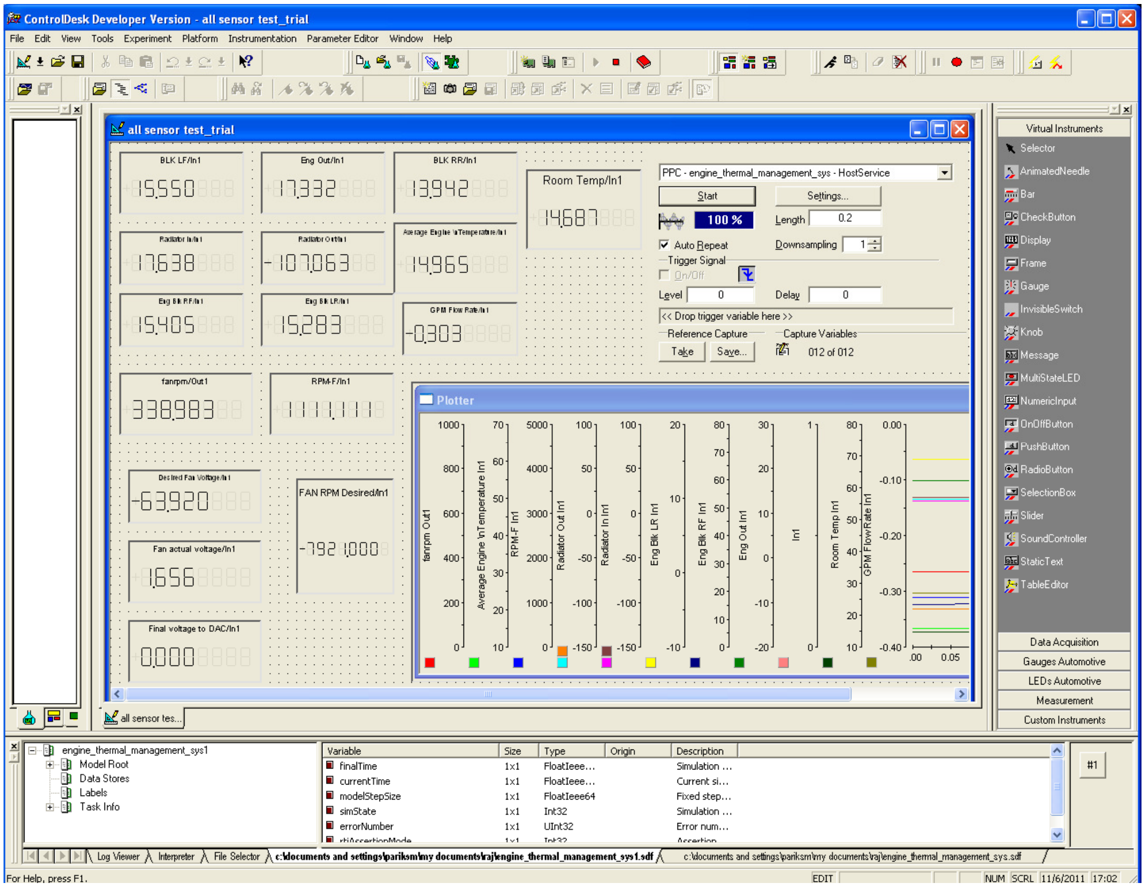
Table H.1: Description of signals coming in and going out of Pin CP17

Signal Input / Output	**Pin No.**	**Signal Input / Output**	**Pin No.**
Ground (GND)	1	Signal out to rotate hydraulic motor for coolant pump in reverse direction	23
Signal out to rotate hydraulic motor for radiator fan in reverse direction	2	Ground (GND)	25
Signal out to rotate hydraulic motor for coolant	3	Multiplexer control input A	27

pump in forward direction			
Ground (GND)	4	Ground (GND)	28
Signal in from coolant pump rpm sensor	8	Multiplexer control input C	29
Ground (GND)	10	Multiplexer control input INHIBIT	30
Signal in from radiator fan rpm sensor	12	Signal in from engine rpm sensor	33
Multiplexer control input B	20	Ground (GND)	34
Signal out to rotate hydraulic motor for radiator fan in forward direction	21	+5 V Power Supply	36
Ground (GND)	22	+5 V Power Supply	37

Table H.2: Control inputs to multiplexer chip to receive particular signal at the output

Control Inputs to Multiplexer chip			Multiplexer chip input number	Temperature signals coming in
A	B	C		
0	0	0	X0	Engine coolant at left rear location
1	1	0	X3	Engine block right front location
1	1	1	X7	Engine coolant out from thermostat
1	0	0	X1	Engine block right rear location
0	0	1	X4	Engine block left front location
1	0	1	X5	Engine room temperature



BLK LF/In1: 15.550
Eng Out/In1: 17.332
BLK RR/In1: 13.942
Room Temp/In1: 14.687
Radiator In/In1: 17.638
Radiator Out/In1: -107.063
Average Engine \nTemperature/In1: 14.965
Eng Blk RF/In1: 15.405
Eng Blk LR/In1: 15.283
GPM Flow Rate/In1: -0.303
fanrpm/Out1: 338.983
RPM-F/In1: 111.111
Desired Fan Voltage/In1: -63.920
FAN RPM Desired/In1: -7921.000
Fan actual voltage/In1: 1.656
Final voltage to DAC/In1: 0.000

PPC - engine_thermal_management_sys - HostService
Length: 0.2
☑ Auto Repeat
Downsampling: 1
Level: 0
Delay: 0
Capture Variables: 012 of 012

Variable	Size	Type	Origin	Description
finalTime	1x1	FloatIeee...		Simulation ...
currentTime	1x1	FloatIeee...		Current si...
modelStepSize	1x1	FloatIeee64		Fixed step...
simState	1x1	Int32		Simulation ...
errorNumber	1x1	UInt32		Error num...

c:\documents and settings\pariksm\my documents\raj\engine_thermal_management_sys1.sdf

Fig. H.3: Control Desk software data collection representation

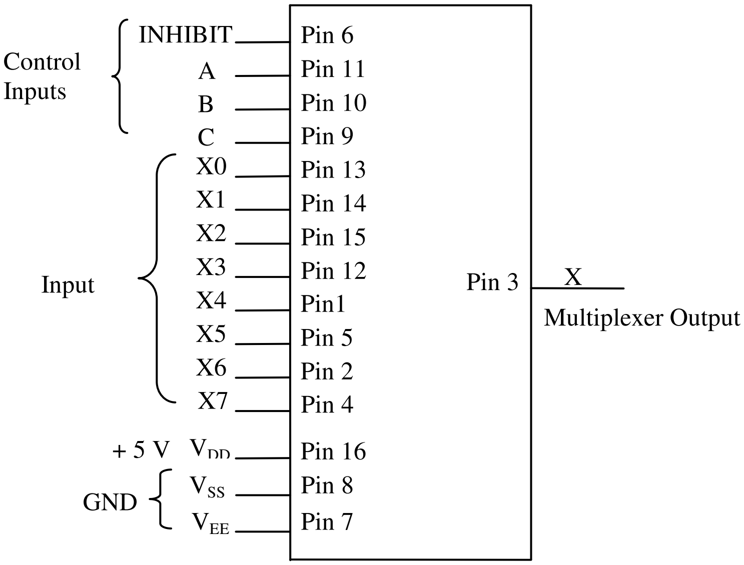


Fig. H.4: Multiplexer pin diagram

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