

An Integrated Model of Energy Transport in a Reciprocating, Lean Burn, Spark Ignition Engine

Peter A. Dennis, Michael J. Brear, Harry C. Watson, Pedro J. Orbaiz, and Payman Abbasi Atibeh
University of Melbourne

ABSTRACT

This paper presents a combined experimental and numerical method for analysing energy flows within a spark ignition engine. Engine dynamometer data is combined with physical models of in-cylinder convection and the engine's thermal impedances, allowing closure of the First Law of Thermodynamics over the entire engine system. In contrast to almost all previous works, the coolant and metal temperatures are not assumed constant, but rather are outputs from this approach. This method is therefore expected to be most useful for lean burn engines, whose temperatures should depart most from normal experience.

As an example of this method, the effects of normalised air-fuel ratio (λ), compression ratio and combustion chamber geometry are examined using a hydrogen-fueled engine operating from $\lambda = 1.5$ to $\lambda = 6$. This shows large variations in the in-cylinder wall temperatures and heat transfer with respect to λ . In keeping with our other works, thermal efficiency also appears to be limited by in-cylinder heat transfer on the rich side of optimum λ , and diminishing combustion quality on the lean side.

By comparing different compression ratios, this method confirms the widely-reported significance of in-cylinder heat transfer in determining the optimal compression ratio of hydrogen-fueled engines. As part of this, while increased in-cylinder pressures contribute to increased heat transfer at higher compression ratio, the accompanying increased piston surface area was also found to be significant. As a final contribution, potential improvements in piston design are therefore investigated numerically, and suggestions for high efficiency, hydrogen fuelled engine designs are made.

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INTRODUCTION

The transfer of energy through a reciprocating, internal combustion engine is complex. Numerous physical processes occur simultaneously. These processes occur in gaseous, liquid, two-phase and solid states of matter. Computationally practical yet rigorous, integrated modelling of energy transfer from the fuel into work and waste energy therefore remains a substantial challenge for the engine community.

A significant part of this challenge relates to the proper selection of boundary conditions between engine sub-systems. This is a particularly so for arguably the key sub-system - the combustion chamber. In this case, measurement of surface temperatures *throughout the chamber* (preferably including spatial and temporal variations) is excessively difficult, if not impossible. Nonetheless, combustion chamber surface temperatures play an important role in determining the performance of the entire engine system in several respects, and are governed by heat transfer both from the combustion chamber and to the rest of the engine.

When estimating in-cylinder heat transfer in reciprocating engines, use of Woschni's correlation [1], like that of Annand [2], is widespread. However, while Woschni intended that calibrated

constants for the correlation should remain valid as long as engine geometry remains similar, the literature suggests that this is not true for hydrogen-fueled engines, where calibration is often required even for changes in air-fuel ratio [3, 4]. These short-comings also appear to be shared with more recent correlations (e.g. [5, 6, 7]), particularly if one intends to estimate peak heat flux [8, 9]. Furthermore, any sensitivity to air-fuel ratio is exacerbated by hydrogen's wide λ operating range. Finally, as experimental studies concerning in-cylinder surface temperatures for spark ignition engines are usually performed at or near stoichiometry (e.g. [10, 11, 12]), temperatures reported are often not directly transferable to very lean operation.

However, while these heat transfer correlations may offer only limited predictive power, their application remains useful given careful calibration. A number of recent studies provide estimates of wall heat transfer in hydrogen spark ignition engines. Some (e.g. [13, 14]) develop such estimates using a calibrated convection model based on in-cylinder pressure measurements. Others (e.g. [9, 15]) focus on direct measurement of temperatures and temporal variations in heat flux at particular points in the chamber. However,

generalisation of local measurements to an entire chamber is challenging, given the spatial variations that should occur over the head, valve, bore and piston surfaces [9].

This work therefore proposes a systematic, physically justifiable approach for calibrating an in-cylinder convection model and for estimating thermal conditions throughout an engine. This requires the inclusion of a modified version of Woschni's correlation for in-cylinder heat transfer [16] in a thermal impedance model of the overall engine. The results from this process are then used to examine energy transport and combustion behaviour under hydrogen-fueled operation, in particular wall heat transfer and combustion stability, and discuss how these relate to engine performance.

EXPERIMENTAL METHOD AND RESULTS

Testing was conducted at the Advanced Centre for Automotive Research and Testing (ACART) facility at the University of Melbourne using a Horiba *Titan T460* dynamometer (Table 1). A 4.0L I6 engine provided by the Ford Motor Company of Australia was used for the experiments (Table 2). Variants of this engine have powered some of Australia's most popular vehicles, and its relatively large displacement makes ultra-lean burn strategies providing sufficient torque more easily attainable.

Table 1. Engine dynamometer facility

Horiba <i>Titan T460</i> Engine Dynamometer Features	
Max Torque	1484 Nm
Power Absorption	460 kW
Speed Range	0–8000 rev/min
Control System	Horiba SPARC/STARS

Engine Conditioning Components	
Oil/Water Conditioning	Horiba STARS
Air Conditioning	External Ventilation System

Data Acquisition Systems	
General Analogue	ATI 16AI EDAQ Module
Temperature	ATI 16T EDAQ Module
Emissions Analysis	ADS9000 (O ₂ , NO, UHC)
Emissions Calibration	Horiba Vehicle Certification Bench
Combustion Analysis	ACART LabVIEW Application
Engine Instrumentation	Ford ECU Intervention

Miscellaneous Instrumentation	
Exhaust Gas Oxygen (λ)	MOTEC PLM with Bosch LSU4.9
Thermocouples	1.5 mm and 3 mm K-type
Pressure Transducers	Kistler 603B11
Charge Amplifier	Kistler SCP2854A with 3x 5064B
Comb. Analysis DAQ	National Instruments PCI-6123

Control of the engine was achieved through ECU intervention via ATI's *Vision* software suite. All data used in this paper was obtained at 1500^{rev}/_{min} and MBT spark timing with the cams fixed at their midpoint positions. The coolant inlet temperature was set to 90°C using the Horiba management system. The blowby rates measured were typical of spark ignition engines, at approximately 1% of the air mass flow rate.

In-cylinder pressure data underpin much of the analysis, including estimation of indicated work, burn rates, cylinder wall heat transfer and unburned fuel fractions. Analysis was performed using a representative pressure trace; a single cycle chosen from 300 recorded at each operating point to minimise the cost function

$$\text{cost} = 1.2 \left| \frac{\overline{\text{IMEP}}_{\text{cycle}} - \overline{\text{IMEP}}}{\overline{\text{IMEP}}} \right| + \left| \frac{\overline{\text{PP}}_{\text{cycle}} - \overline{\text{PP}}}{\overline{\text{PP}}} \right|, \quad (1)$$

where PP indicates peak pressure and the overbar represents the mean of all recorded traces.

Table 2. Ford *Barra* 4.0L I6 specifications

Cylinders	
Configuration	Inline 6
Firing Order	1-5-3-6-2-4
Capacity	3984cc
Bore	92.25 mm
Stroke	99.31 mm
Compression Ratio	10.3:1

Valve-Train	
Configuration	Independent DOHC VCT
Valve Lift	11.0 mm
Valve Diameter (I)	35.0 mm
Valve Diameter (E)	32.0 mm
Intake Opening	27.5° BTDC to 32.5° ATDC
Intake Closing	48.5° ABDC to 108.5° ABDC
Exhaust Opening	78.5° BBDC to 18.5° BBDC
Exhaust Closing	2.5° BTDC to 57.5° ATDC
Intake Centreline	99.5° ATDC to 159.5° ATDC
Exhaust Centreline	131.5° BTDC to 71.5° BTDC

Other Specifications	
ECU	Ford BF Falcon
Calibration	BF NA/Turbo with Modifications
H ₂ Gas Control	Prins VSI (2 Modules)
H ₂ Gas Injectors	Keihin LPG/CNG (2 per port)
Spark Plugs	Ford/Denso OEM Plugs
Exhaust System	No catalyst, high-flow mufflers

Figure 2 shows representative pressure traces for the two data sets considered in this paper. The pressure measurements for the reference case (Figure 2a) were obtained at wide-open throttle (WOT) and show no significant variation in pumping or compression, but combustion rates and indicated work decrease with output torque as λ increases. Figure 2b shows the in-cylinder pressure measurements for two different compression ratios, achieved by fitting higher-crowned pistons (Figure 1). Data was captured within 3% of WOT airflow to achieve the best performance at each compression ratio [17]. The pumping work, the polytropic index during compression and burn rates appear to be similar for the two compression ratios, while greater peak pressures are observable for the 13.3:1 cases, as expected.

A number of time-averaged measurements taken on the engine dynamometer also contribute to this analysis (Figure 3). The decreasing values for these quantities with increasing λ are expected. However, despite near identical fuel flows for the two compression ratios (Figure 3d), torque deviates significantly for these engine variants around $\lambda = 4$ (Figure 3b), while in Figure 3f the exhaust temperatures differ around $\lambda = 2$, then converge at the leanest condition for reasons discussed later.

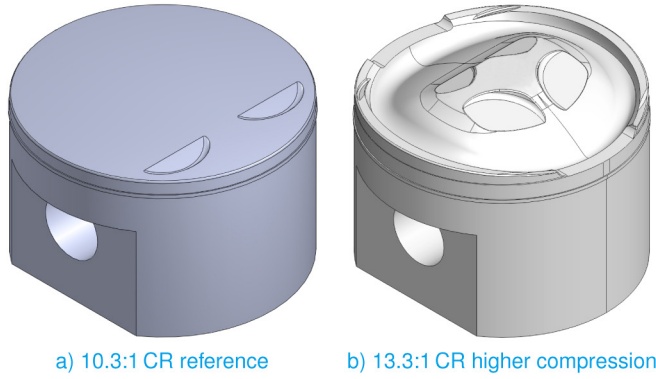


Figure 1. Pistons used for experimental testing of compression ratios

METHOD FOR DETERMINING ENERGY FLOWS

Definition of Control Volumes

Three control volumes are defined: the engine, the combustion chamber, and the aspiration volumes (Figure 4).

Engine Control Volume

Considering the control volume around the entire engine (Figure 4), conservation of energy may be expressed as

$$\overline{H}_i = \overline{Q} + \overline{W}_b + \overline{H}_e, \quad (2)$$

where $\overline{W}_b$ is the brake work, $\overline{Q}$ represents all undetermined heat losses and $\overline{H}$ is the enthalpy of the fluid flows. Overlined values indicate averaged quantities, and subscripts i and e designate intake and exhaust flows respectively. By setting intake conditions as the reference state, and separating $\overline{H}_e$ into enthalpies of formation $\overline{H}_{f,e}$ and sensible enthalpies $\overline{H}_{s,e}$, our energy balance becomes

$$0 = \overline{Q} + \overline{W}_b + \overline{H}_{s,e} + \overline{H}_{f,e}, \quad (3)$$

which in terms of the lower heating value of the fuel is

$$\overline{n}_{iH_2} \text{LHV}_{H_2} = \overline{Q} + \overline{W}_b + \overline{H}_{s,e} + \overline{H}_{f,e} - \overline{n}_{iH_2} h_{fH_2O}, \quad (4)$$

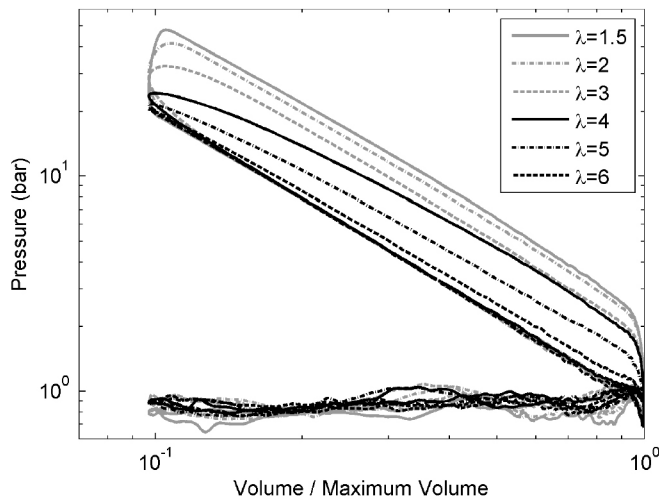
where $\overline{n}_{iH_2}$ is the molar flow rate of hydrogen at the intake, h_{fH_2O} is the molar-specific heat of formation of water and LHV_{H_2} is the molar-specific lower heating value of hydrogen. It may then be shown that Equation 4 is equivalent to

$$\overline{n}_{iH_2} \text{LHV}_{H_2} = \overline{Q} + \overline{W}_b + \overline{H}_{s,e} + \overline{n}_{eH_2} \text{LHV}_{H_2}. \quad (5)$$

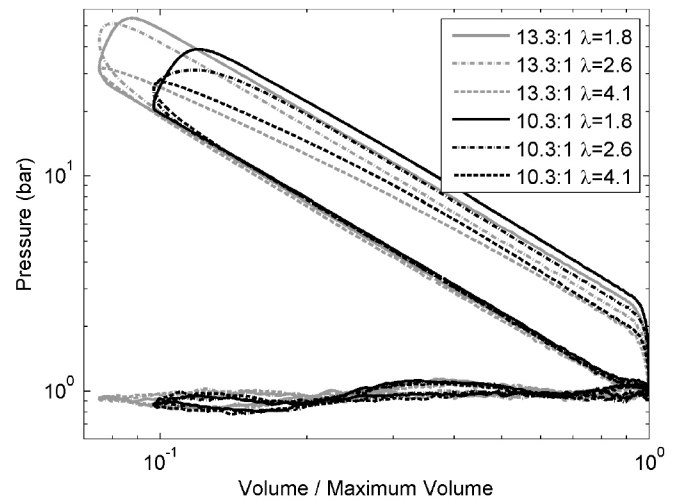
Finally, to allow direct use of mass flow measurements, we express the energy balance in mass-specific form:

$$\overline{m}_{iH_2} \text{LHV}_{H_2} = \overline{W}_b + \overline{Q} + \overline{H}_{s,e} + \overline{m}_{eH_2} \text{LHV}_{H_2}, \quad (6)$$

where $\overline{m}$ represents the mass flow rate and the terms H_s and LHV_{H_2} are now on a mass basis.



a) Indicator diagram for naturally aspirated WOT operation at a compression ratio of 10.3:1



b) Indicator diagram for near-WOT comparison of 10.3:1 and 13.3:1 compression ratio pistons (turbocharged)

Figure 2. Measured log p -log V indicator diagrams

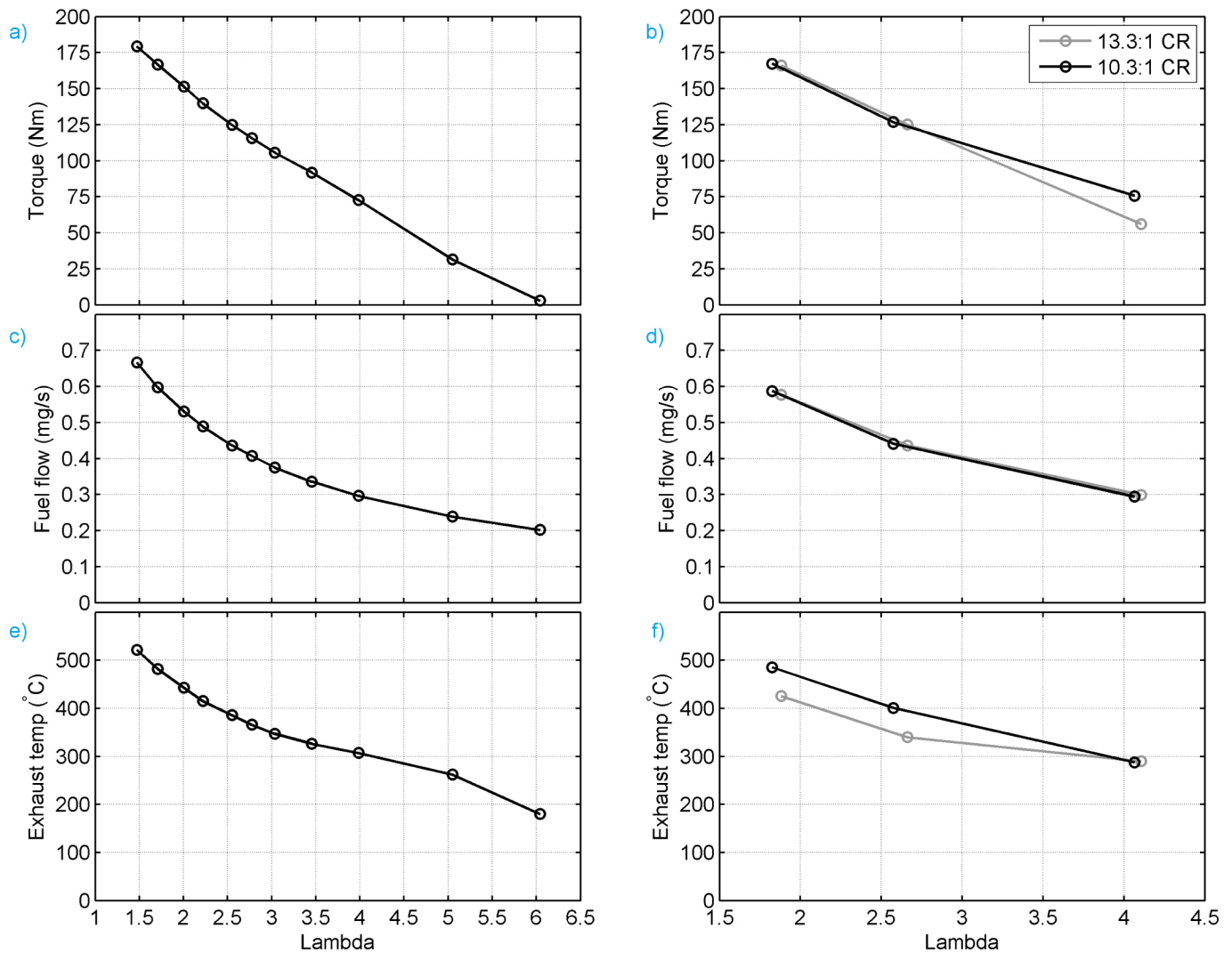


Figure 3. Dynamometer testing results for WOT, naturally aspirated operation at 10.3:1 compression ratio (left) and for comparison of 10.3:1 and 13.3:1 compression ratio pistons in a turbocharged engine (right). Plots a) and b) show brake torque, c) and d) show fuel flow rates, and e) and f) show exhaust temperatures sampled at the cylinder head exhaust port.

In order to determine these quantities, measurements are taken of fuel mass flow rate, air mass flow rate, shaft speed, shaft torque and exhaust temperature. The term $\bar{m}_{eH_2}$, and thus $\bar{m}_{eH_2} \text{LHV}_{H_2}$, cannot be determined directly without exhaust hydrogen measurements.

Combustion Chamber Control Volume

All six cylinders are analysed as one control volume. Thus, the energy balance over the combustion chamber may be expressed in a similar form to Equation 6,

$$\bar{m}_{iH_2} \text{LHV}_{H_2} = \bar{W}_i + \bar{Q}_{wall} + \bar{m}_{eH_2} \text{LHV}_{H_2} + \bar{H}_{s,e}^{cyl} - \bar{H}_{s,i}^{cyl}, \quad (7)$$

where $\bar{W}_i$ is the indicated work and $\bar{Q}_{wall}$ is the heat transferred from the gases to the cylinder walls. Due to heat transfer and ejection of combustion chamber gases into the intake manifold while the inlet valves are open, the last term is not necessarily equal to zero.

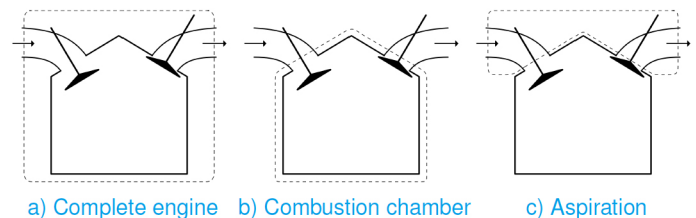


Figure 4. Boundaries for the control volumes used

Of course, $\bar{W}_i$ is measured using in-cylinder pressure data, while $\bar{Q}_{wall}$ and $\bar{m}_{eH_2} \text{LHV}_{H_2}$ can be solved for by simulating wall temperatures, heat transfer coefficients, burn rates and unburned fuel fractions. $\bar{H}_{s,e}^{cyl} - \bar{H}_{s,i}^{cyl}$ is then the remaining energy after all other terms are evaluated.

Aspiration Control Volume

No work is performed in this control volume, so the energy balance may be expressed as



$$\bar{H}_i = \bar{Q}_{asp} + \bar{H}_e, \quad (8)$$

where $\bar{Q}_{asp}$ is heat transfer to walls of the intake and exhaust ports and manifolds. It is also assumed that no chemical reactions occur. The aspiration control volume has four mass transfer interfaces, two shared with the engine control volume, and two with the combustion chamber control volume. Consequently, Equation 8 is equivalent to

$$\bar{H}_{s,e}^{cyl} - \bar{H}_{s,i}^{cyl} = \bar{Q}_{asp} + \bar{H}_{s,e}, \quad (9)$$

using the terms developed earlier.

Turbocharger Control Volume

The control volume boundaries for the engine and aspiration volumes are downstream of the compressor and upstream of the turbine when a turbocharger is present. A potential fourth control volume containing the turbocharger would allow further analysis if of interest. This is not done in this study.

Combining the Control Volumes

Combining Equations 6, 7 and 9 leads to the final expression of the energy balance over the whole engine

$$\bar{m}_{iH_2} \text{LHV}_{H_2} = \bar{W}_b + \bar{Q}_{fric} + \bar{Q}_{wall} + \bar{Q}_{asp} + \bar{H}_{s,e} + \bar{m}_{eH_2} \text{LHV}_{H_2}, \quad (10)$$

where $\bar{Q}_{fric} = \bar{W}_i - \bar{W}_b$, and $\bar{Q} = \bar{Q}_{fric} + \bar{Q}_{wall} + \bar{Q}_{asp}$ in equation 6.

Modelling the Thermal Impedance Network

Calculation of the in-cylinder heat transfer term $\bar{Q}_{wall}$ requires modelling of a network of heat fluxes and thermal impedances between the coolant and in-cylinder gases. The proposed model is shown in Figure 5. In this model, heat entering the head and cylinder bore from the in-cylinder gases is directly conducted to the coolant. Heat entering the piston must either pass through the cylinder bore or via some other path to the coolant. Heat transfer directly to the environment is considered small and is ignored.

$\bar{Q}_{wall}$ is assumed to consist of three components, $\bar{Q}_h$, $\bar{Q}_p$ and $\bar{Q}_b$, representing heat transfer from the in-cylinder fluid to the surfaces of the head, piston and cylinder bore respectively. As these terms will be estimated using a modified version of the Woschni heat transfer correlation, it is not necessary to calculate the effective gas temperature T_g , or the related gas-to-wall impedances Z_{gh} , Z_{gp} and Z_{gb} explicitly. However, the Woschni model requires the input of wall temperatures, which themselves depend on $\bar{Q}_h$, $\bar{Q}_p$ and $\bar{Q}_b$, as well as on the thermal impedances between the combustion chamber surfaces and the coolant. The estimation of these thermal impedances is detailed later in the section "Modelling metal temperatures".

In order to simplify the analysis, three further assumptions were made regarding the surface temperatures of the piston T_p , the cylinder head T_h and the cylinder bore T_b :

1. Cylinder wall temperatures do not vary during the cycle.
2. T_p and T_h may be represented as a single effective temperature T_{ph} .
3. $T_b = T_c + 0.5 (T_{ph} - T_c)$, where T_c is the measured coolant temperature.

The first assumption is consistent with studies of chamber wall temperatures (e.g. [11]). It also means that the temperatures only depend on the mean, and not the time-resolved, heat fluxes. The second assumption is appropriate when the in-cylinder convection model only considers a single, averaged in-cylinder fluid state, and because both the piston surface and head surface are in contact with the fluid throughout the cycle. T_{ph} should be evaluated such that the heat flux through both the piston and head Q_{ph} is equal to $Q_p + Q_h$ at each crank angle, i.e.,

$$h (T_g - T_{ph}) (A_p + A_h) = h (T_g - T_p) A_p + h (T_g - T_h) A_h, \quad (11)$$

where A is the area of the designated surface. Then, as the convection coefficient h is assumed to be independent of location in the cylinder, this may be simplified to

$$T_{ph} = \frac{T_p A_p + T_h A_h}{A_p + A_h}. \quad (12)$$

A separate cylinder bore temperature is modelled according to the third assumption, as T_b is generally significantly lower than T_{ph} ([10, 11, 18]). Furthermore, unlike the piston and head surfaces, the bore surface is only exposed to the charge as the piston moves away from top dead centre. Additionally, modelling of the dissipation of heat from the piston through the bore requires a temperature gradient.

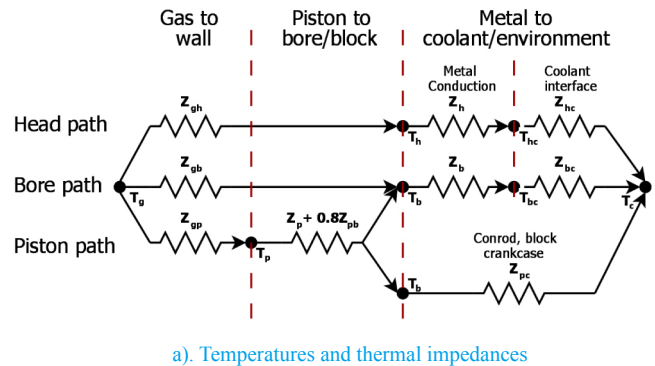
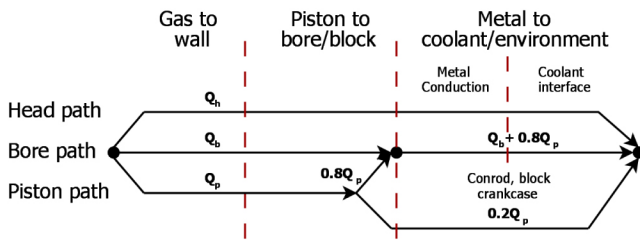


Figure 5. The engine thermal impedance model



b). Magnitude of heat fluxes. Q_p , Q_h and Q_b denote the heat fluxes through the in-cylinder piston, head and cylinder bore surfaces respectively.

Figure 5 (cont). The engine thermal impedance model

We may now create two models relating the effective wall temperature T_{ph} to average heat transfer rates $\bar{Q}_h$, $\bar{Q}_p$ and $\bar{Q}_b$,

$$T_{ph} = f_1(\bar{Q}_h, \bar{Q}_p, \bar{Q}_b), \quad (13)$$

$$[\bar{Q}_h, \bar{Q}_p, \bar{Q}_b] = f_2(T_{ph}). \quad (14)$$

Equation 13 is created from the model shown in Figure 5 using the thermal impedance values calculated in the later section “Modelling metal temperatures”. Equation 14 is produced for each engine operating point via calibration of the Woschni heat transfer correlation over a range of plausible wall temperatures. This process is described in detail in the following sections.

The wall temperature and heat transfer rates where these two models are in agreement is found via iteration. Beginning with an initial estimate of T_{ph} , named $\hat{T}_{ph0}$, Equation 14 is used to obtain the corresponding heat transfer rates,

$$[\hat{\bar{Q}}_{h_k}, \hat{\bar{Q}}_{p_k}, \hat{\bar{Q}}_{b_k}] = f_2(\hat{T}_{ph_k}). \quad (15)$$

Equation 13 is then used to update the estimate of T_{ph} ,

$$\hat{T}_{ph_{k+1}} = f_1(\hat{\bar{Q}}_{h_k}, \hat{\bar{Q}}_{p_k}, \hat{\bar{Q}}_{b_k}). \quad (16)$$

The sequence defined by Equations 15 and 16 is then

$$\hat{T}_{ph_{k+1}} = f_1\left(f_2\left(\hat{T}_{ph_k}\right)\right), \quad (17)$$

$$= f_3\left(\hat{T}_{ph_k}\right). \quad (18)$$

Assuming a solution to Equations 13 and 14 exists, and that the initial estimate $\hat{T}_{ph0}$ is within the domain of attraction for this solution, the sequence described in Equation 18 will converge to the solution for T_{ph} . These assumptions appear to be valid, as convergence was observed for all cases considered.

Trapped Mass In-Cylinder Heat Transfer

Neglecting blowby, the trapped mass is constant between inlet valve closure (IVC) and exhaust valve opening (EVO), and thus the energy balance equations do not need to consider mass transfer over the system boundaries. *GT-Power* (part of *GT-Suite* [19]) can analyse measured pressure traces during this period, and must be supplied with the crank angles of the start of combustion (SOC) and the end of combustion (EOC), the molar concentrations of all species at these points, the engine geometry and the trapped mass. A heat transfer model must also be specified. The “reverse run” model supplied by *GT-Power* may then be used to simulate piston movement and cylinder wall heat transfer between IVC and EVO, inferring the burn rate at each timestep to match the measured pressure trace where possible. Blowby was measured during engine testing, and found to be similar to that expected in hydrocarbon-fueled engines, and neglected due to the limited impact such rates would have on the energy flow analyses.

The engine geometry is known and the fuel and air flows are measured, while a *GT-Power* 1D flow model of the engine (validated against WOT airflows) was used to estimate the residual fraction. The SOC was assumed equal to spark timing, and the EOC was estimated by examining plots of $d \log p(\theta) / d \log V(\theta)$ to determine where the polytropic index returns to a constant value. In the absence of equipment to determine hydrogen concentrations in the exhaust, establishing the unburned fuel fraction is more difficult, and relies on an iterative analysis of heat transfer rates and in-cylinder energy balance. This process is covered in the section “Determination of unburned fuel fraction” below, while this section and the following section assume that the unburned fraction is known.

GT-Suite Implementation of the Woschni Correlation

Relative to Woschni's original formulation [1], the heat transfer model selected contains additional modeling of heat transfer while the inlet or exhaust valves are open, as described in the *GT-Suite* User's Manual [16]. Elsewhere in the cycle, the heat transfer coefficient h is modeled as per Woschni [1]. The in-cylinder heat flux may then be estimated from the correlation using

$$q = C_0 h (T_g - T_w), \quad (19)$$

where T_w is the temperature of a wall segment, and C_0 is a dimensionless calibration constant. Henceforth, C_0 is termed the h -multiplier, and $C_0 h$ is a corrected heat transfer coefficient.

This implementation of Woschni's model thus has two main inputs - the wall temperatures and the h -multiplier - for which suitable values are initially unknown. As discussed, very lean combustion means typical engine metal temperature measurements are unlikely to be applicable and their significance is not known. Also, appropriate values for the h -multiplier are not easily derived from published work, as hydrogen-fueled operation often presents significant challenges for in-cylinder heat transfer correlations, including that of Woschni [1, 3, 4].