Modeling of Quasi-Steady State Heat Transfer at Intake Port of Real IC Engine and its Application to 1-D Engine Simulation

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Received on July 6, 2018

ABSTRACT: Overall efficiency of internal combustion engines are heavily depended on intake air temperature which is directly related to the heat transfer inside an intake system. Previously, authors developed an equation by using port model setup to calculate Nusselt number with introduction of Graetz and Strouhal numbers. This study modified the port model equation to improve its accuracy in a real engine experimental setup. Predicted intake air temperature was compared to the measured data with a maximum error of 5.6%. Additionally, 100 K of temperature difference was found between the boost pressure values of 944hPa and 678hPa from 1-D engine simulation results.

KEY WORDS: heat engine, spark ignition engine, intake and exhaust / heat transfer, thermal boundary layer [A1]

1. Introduction

Thermal efficiency measurements are dependent on intake air's mass and temperature for internal combustion (IC) engines. A major influence is made on the mass of air flowing (from now on referred as intake air) inside an intake system, as the air density varies with changing intake air temperature. This temperature change on intake air also affects another important aspect called charging efficiency, which basically determines the amount of fresh mixture (intake air and fuel) that is directed to the displaced volume for the combustion process. Additionally, heat transfer phenomenon at intake system also affects the combustion efficiency, which acts as a crucial aspect for engine performance as well as hazardous gas emissions, such as NO_x, particulate matter (PM), CO, and so forth. Thus, it is clear that the heat transfer phenomenon at an intake system acts as one of the first indicators on both thermal efficiency and performance measurements for IC engines and needs to be studied with great care.

Heat transfer phenomena at intake systems have been analyzed in previous studies ^{(1) – (4)}, where Colburn analogy has been used broadly. This conventional analogy describes the heat transfer as steady-state flow with forced convection. However, this approach was found to be limited when characterizing the actual physical phenomenon that is taking place in a real intake system. Hence, it cannot be applied directly to the heat transfer at the intake system of an IC engine ^{(5) – (7)}. Intake air inside the manifold of an IC engine flows as an oscillating intermittent flow due to opening and closing of the intake valves and rapid pressure variations. In the previous studies ^{(8) – (10)} authors suggested new empirical equations by assuming the heat transfer phenomenon as quasi-steady state in order to model the physical aspect with considering the region outside of the developed thermal boundary layer and the intermittent flow effects based on

Colburn analogy. Results showed a good correlation with previous study, the port model approach ⁽¹⁰⁾, however, frequency levels for the experimental model was found to be inadequate when compared to the actual working frequency (several tens Hz) of an intake valve.

In the present study, experiments were conducted on a real engine setup, where higher working frequency levels for intake valves were attained. Temperature measurements were also conducted in a fired engine for the formulation of the quasi-steady heat transfer. An empirical formula for deriving the Nusselt number, Nu , of the intake port arranged as an expression having Reynolds number, Re , Graetz number, Gz , and Strouhal number, St , as dimensionless variables. The purpose of this paper was to calculate the outlet air temperature of the intake port, as accurately as possible, before entering into the engine cylinder. Furthermore, commercially available, 1-D engine simulation software was used in order to calculate the outlet intake air temperatures at the intake port by implementing the present equation, the port model equation and the Colburn's equation to the software and compare the obtained results.

2. Experimental Apparatus and Conditions

2.1. Instrumentation for engine experiments

Figure 1 shows schematic view of the experimental apparatus used during this study. Figure 2 illustrates a detailed view of the engine intake system, which consisted of an air filter, a surge tank, a throttle valve, an intake manifold, intake port, an intake valve and a fuel injector. Intake port was made from aluminum, 122 mm in length, 32 mm in inner diameter. Figure 3 illustrates a detailed representation of the intake port. Air flow rate inside the manifold was measured with a laminar air flow meter (Tsukasa Sokken Co. Ltd, LFE-75B). Gas temperature (T_{gup} , T_{gmid} , T_{glow})

inside the intake port, and port's wall surface temperature (T_{sup} , T_{smid} , T_{slow}) were measured at specified locations as shown in Figure 3. In order to measure the gas temperature inside the intake port, K-type thermocouples (Okazaki Manufacturing Company, AEROPAK), with a wire diameter of 0.15 mm, were attached at the centre of the cross section of each indicated position. 0.15 mm diameter was chosen for eliminating excessive air drag which would cause additional turbulence on air flow inside the intake manifold. Surface temperature of the intake port wall was measured with coaxial surface thermocouples (E-type, Mueller Instruments, MCT-36), which were attached to the same three locations on the circumference of the intake port.

Fig. 1 Schematic view of experimental apparatus

Fig. 2 Detailed view of intake system

Fig. 3 Location of thermocouples installed on intake port

2.2. Engine Specifications

The experiments were conducted in-line single cylinder OHC 2 valve direct-injection gasoline engine, with a variable valve timing mechanism. Details about the engine are given in Table 1.

Table 1 Engine specification for experiments

Type	4 stroke , Gasoline
Valve Type	OHC , 2 valves
Injection Type	Direct injection
Bore × Stroke [mm]	85.0 × 85.0
Cylinder Number	1
Displacement [cc]	482
Compression Ratio [-]	8

2.3. Experimental conditions

Under the experimental conditions shown in Table 2, gas temperature at each position was measured. Engine speed was controlled and measured by ECU. Injected fuel was adjusted depending on the engine speed by a digital burette. Engine load was adjusted by the help of dynamometer, with five different loads. Additionally, a thermal regulator was used to control cooling water temperature, thus thermal boundary layer effects could be studied thoroughly. Intake port surface wall temperature and intake air temperature were changed depending on the variations on the intake manifold pressure, engine speed, and cooling water temperature.

Table 2 Experimental conditions

Engine speed [rpm]	1200, 1600, 2000, 2400
Boost pressure [hPa]	411, 544, 678, 811, 944
Cooling water temperature [K]	343, 353, 363
Air-to fuel ratio	14.7
Ambient air temperature [K]	293
Injection timing	MBT

2.4. Numerical Simulation Setup

Commercially available 1-D engine simulation software (referred as 1-D simulation onwards), GT-Power, was used to validate the usage of suggested empirical equations. Details about simulated engine specifications and simulation conditions are given in Tables 3 and 4, respectively.

Table 3 Engine specification for numerical simulations

Type	4 stroke , Gasoline
Valve Type	OHC , 16 valves
Injection Type	Direct injection
Bore × Stroke [mm]	86.0 × 86.0
Cylinder Number	4
Displacement [cc]	1998
Compression Ratio [-]	9.8

Table 4 Numerical simulation conditions

Engine speed [rpm]	1500, 2000, 2250
Boost pressure [hPa]	678, 811, 944
Cooling water temperature [K]	343, 353, 363

This software can be edited by its user to define necessary correlations that are being used to calculate heat transfer phenomena and so forth. This was done by changing the heat transfer coefficient calculations in pre-defined subroutines inside the 1-D simulation. Authors used FORTRAN language to define and edit these heat transfer correlations. Empirical equations (16) and (17) were implemented whose detail was shown in section 4.3, and simulations were conducted accordingly. Empirical equations' simulation results were compared to the conventional heat transfer correlation, Colburn analogy.

A naturally aspirated direct fuel injected gasoline engine with compression ratio of 9.8 was created on 1-D engine simulation. Ambient temperature and pressure were set to be 1atm and 300 K, respectively. Three different coolant temperatures similar to the real engine experiment were used to change wall temperature and analyzed their effect on heat transfer phenomena at the intake system. Similar to the experimental setup, intake ports were also made from aluminum, 140 mm in length, 26 mm in diameter and 5 mm in wall thickness.

3. Modeling Approach

3.1. Modeling of Intake Air Temperature

Intake air temperature which changes due to the friction caused as the fresh air passes through air filter, surge tank, and throttle valve, is denoted by ΔT_i , and shown in Figure 1. Temperature difference inside the intake port was denoted as ΔT_p , and temperature change due to latent heat of vaporization of fuel in the case of port injection was denoted as ΔT_f . Thus, temperature difference between inlet T_i , and outlet, T_o , of the intake port could be expressed by the following equation:

$$T_o = T_i + \Delta T_i + \Delta T_p + \Delta T_f \quad (1)$$

In the previous study [8] temperature difference due to friction, ΔT_i , was found to be 0.22%, thus, omitted. In addition, since engine used during the experiments was a direct injection fuel type, latent heat of vaporization of fuel, ΔT_f , was also left out from the equation. However, ΔT_p could not be neglected as the change in the intake air temperature was considerably large when compared to other heat transfer phenomenon inside the engine's intake system. From the previous equation, intake port's outlet temperature was determined by the inlet air temperature, T_i , and temperature change in the intake port, ΔT_p , thus equation (1) was revised as the following:

$$T_o = T_i + \Delta T_p \quad (2)$$

3.2. Modeling of Heat Transfer at Intake Port

In the present study, after treating the heat transfer phenomenon inside the intake port as a quasi-steady state, influence of the intermittent flow caused by the intake valve's opening and closing frequency was examined separately, which was expressed by the following equation based on the first law of thermodynamics:

$$\dot{Q} = \dot{m} c_p \Delta T_p = \frac{\rho u \pi d^2}{4} c_p (T_o - T_i) \quad (3)$$

where $\dot{m}$: mass flow rate [kg/s], c_p : specific heat capacity at constant pressure [J/(kg·K)], ρ : density of air [kg/m³], u : average gas velocity inside the port [m/s], d : port diameter [m], T_o : outlet temperature [K], T_i : inlet temperature [K]. Additionally, based on Newton's cooling law, heat transfer rate from the intake air to port's wall surface was expressed by the following equation:

$$\dot{Q} = \alpha A (T_g - T_s) = \alpha \pi d l (T_g - T_s) \quad (4)$$

$$T_g = \frac{T_{sup} + T_{gmid} + T_{slow}}{3} \quad (5)$$

$$T_s = \frac{T_{sup} + T_{smid} + T_{slow}}{3} \quad (6)$$

where α : heat transfer coefficient [W/(m²·K)], A : heat transfer area [m²], l : port length [m], T_g : average gas temperature [K], T_s : average port wall surface temperature [K]. From equations (5) and (6) average wall surface temperature and the average gas temperature were calculated based on the thermocouples at each position shown in Figure 2. From the above equations (5) and (6), it is possible to evaluate the Nusselt number by using following equation:

$$Nu = \frac{\alpha l}{\lambda} = \frac{\dot{m} c_p (T_o - T_i)}{\lambda \pi l (T_g - T_s)} \quad (7)$$

where λ : thermal conductivity of air [W/(m·K)]. These Nusselt number values were obtained from the experiments and then compared to the suggested empirical equations results. Heat transfer from the port to the intake air can also be expressed by the following empirical equation called Colburn analogy:

$$Nu = 0.023 Re^{0.8} Pr^{\frac{1}{3}} = 0.02 Re^{0.8} \quad (8)$$

$$Re = \frac{\rho u d}{\mu} \quad (9)$$

where Re : Reynolds number [-], Pr : Prandtl number [-], μ : air viscosity [Pa·s]. For fresh air, Prandtl number values varies between 0.7-0.76, thus the first term was simplified to 0.02, as shown in equation (8). In this paper, effects of the undeveloped thermal boundary layer and intermittent flows caused by the intake valve, on heat transfer were based on conventional Colburn analogy, equation (8), and modeled as the following equation:

$$Nu = 0.02 Re^{0.8} \left(1 + \left(\frac{1}{Gz} \right)^{n_1} \right) m (St)^{n_2} \quad (10)$$

$$Gz = \frac{Re \cdot Pr \cdot d}{l} \quad (11)$$

$$St = \frac{f \cdot l}{u} \quad (12)$$

where Gz : Graetz number [-], St : Strouhal number, f : opening and closing frequency of the intake valve [Hz], m : undetermined coefficient [-], n_1, n_2 : exponential index [-]. Since reciprocal of Graetz number represents the development process of the thermal boundary layer was added to the Colburn equation. Additionally, intermittent flow caused by the opening and closing of the intake valve was added to the equation by using Strouhal number.

In the previous study, the authors derived the empirical equation (13) from the experiments conducted on an intake port model (simulated by a throttle valve, using a blower) [9].

$$Nu = 0.02Re^{0.8} \left(1 + \left(\frac{1}{Gz} \right)^{-0.34} \right) \times 1.01St^{-0.02} \quad (13)$$

Strouhal number in equation (13) was in the range of $1.04 < 1.01St^{-0.02} < 1.1$, which approximately corresponded to engine speeds of 240-400 rpm. This engine speed range was considered to be noticeably inadequate to real driving conditions. Hence, the present study was conducted on a real engine setup and results were discussed in the next session.

4. Experimental Results and Discussion

4.1. Temperature Change inside the Intake Port

For the investigation of intake air temperature variation inside an intake port, real engine experiments were conducted. Figure 4 illustrates air and wall surface temperature variation from the inlet towards the outlet of the intake port under different engine speeds.

Fig. 4 Temperature change of (a) wall surface and (b) average gas when changing engine speed under condition of boost pressure of 544 hPa and coolant temperature of 353 K

Figure 4 shows the temperature variation with respect to changing cooling water temperatures. 15 mm corresponds to the upstream, and 121 mm corresponds to the downstream of the intake port. From Figure 4, it was observed that the port's wall surface temperatures did not show significant changes with varying engine speed, whereas, intake air temperature increased considerably. Higher engine speeds led to higher in-cylinder gas temperature, due to elevated combustion energy, which transferred its intensity towards the intake port due to valve-overlap and displacement backflows.

Additionally, from Figure 4, larger temperature gradients at engine speeds of 1200-1600 rpm were observed when compared to higher engine speeds of 2000-2400 rpm. This was a result of the increased time interval of the opening and closing frequency of the intake valve at lower engine speeds, which resulted in higher amount of backflow gases penetrating into the intake system.

Figure 5 shows wall surface temperature and intake air temperature variations along the intake port with different cooling water temperatures.

Fig. 5 Temperature change of (a) wall surface and (b) average gas when changing coolant temperature under engine speed of 1600 rpm and boost pressure of 544 hPa

Higher cooling water temperature resulted in higher surface and gas temperature values, since more heat was dissipated to the intake air. However, temperature gradients at upstream and downstream of the intake port did not show considerable variations under different cooling water temperatures. Reason behind this result was due to the same boost pressure and engine speed values were used in the experiments, which played more important role on intake air temperature due to the effect of the backflow gases.

4.2. Relation between Nusselt Number and Dimensionless Numbers

Figure 6 illustrates relation between Nusselt number and Reynolds number along with the results obtained by using the Colburn equation (8), and the port model equation (13), under all experimental conditions. Results were obtained at the downstream of the intake port. From Figure 6, it was clear that as the Reynolds number got larger and Nusselt number also increased. These results indicated that the heat transfer rate increased as the gas flow velocity in the intake port increased. When the experimental results were compared to the Colburn analogy, approximately 40 times difference was observed. As a result of this, expression of Colburn's analogy was understood to be insufficient when it was used to characterize the heat transfer phenomenon at an intake system of an IC engine, thoroughly. Colburn equation was used to examine the heat transfer in circular pipes where the thermal boundary layer was sufficiently developed. However, in a real engine, thermal boundary layer develops gradually. In other words, the development process of the thermal boundary layer seems to have a great influence on the heat transfer phenomenon inside the intake port, and portrays an important role, however, neglected in the previous studies.

Fig. 6 Relation between Nusselt number and Reynolds number under all experimental conditions obtained from experimental data and estimated from Colburn equation (8) and port model equation (13)

Figure 7 depicts summarized results of the Nusselt number with respect to the reciprocal of the Graetz number. Once again, results at the outlet of the intake port were shown.

From Figure 7, it could be seen that Nusselt number decreased as the reciprocal number of the Graetz number increased. This result implied that the heat transfer rate decreased due to the development of the thermal boundary layer. In addition, it was understood that the Nusselt number became almost constant when the reciprocal of the Graetz number exceeded 0.0008. This was an indication of fully developed thermal boundary layer, where heat transfer was decreased due to the boundary conditions.

Fig. 7 Relation between Nusselt number and reciprocal of Graetz number under all experimental conditions

Figure 8 illustrates the results of the Nusselt number relative to the Strouhal number. As an example of the results, experimental data at the downstream of the intake port are shown. As depicted in Figure 8, Nusselt number decreased as Strouhal number increased. This result implied that the heat transfer coefficient decreased as the flow unsteadiness increased which was caused by the intake valves movement.

Fig. 8 Relation between Nusselt number and Strouhal number under all experimental conditions

Results from Figures 6, 7, and 8 were used to calculate the undetermined coefficient m , and indices n_1 and n_2 in equation (10) which were derived by the least mean square method. Introduced empirical equation characterized thermal boundary layer development and unsteadiness of the intake flow, which was used to determined the heat transfer rate at an intake system of an IC engine:

$$Nu = 0.02Re^{0.8} \left(1 + \left(\frac{1}{Gz} \right)^{-0.29} \right) \times 3.4St^{-0.035} \quad (14)$$

In this study, experiments were performed within the range of $3000 < Re < 30000$, $500 < Gz < 5000$, $0.02 < St < 0.2$. Strouhal number values for equation (14) resulted in the range of $3.6 < 3.4St^{-0.035} < 3.8$, which corresponded to engine speeds of 1200-2400 rpm. These results were approximately 3.5 times larger than the port model equation (13), which suggested that a real engine was being affected by the intermittent flow.

4.3. Outlet Air Temperature Estimation

Intake port's outlet air temperature were calculated by using three approaches. Previously, outlet temperature dependence on inlet air and heat transfer phenomenon inside the port were explained in equation (2). From Colburn's equation, estimation of the outlet temperature (15) was derived by using the equations (7) and (8):

$$T_o = T_i + \frac{\lambda \pi l (T_g - T_s)}{\dot{m} c_p} \times 0.02 Re^{0.8} \quad (15)$$

Additionally, estimation equation of the intake air temperature at the outlet of the port was obtained by using the intake port model [9] which was expressed by the following expression from the equations (7) and (13):

$$T_o = T_i + \frac{\lambda \pi l (T_g - T_s)}{\dot{m} c_p} \times 0.02 Re^{0.8} \left(1 + \left(\frac{1}{Gz} \right)^{-0.34} \right) \times 1.01 St^{-0.02} \quad (16)$$

Estimation of the intake air temperature derived from the real engine formula was expressed by the following equation from equations (7) and (14):

$$T_o = T_i + \frac{\lambda \pi l (T_g - T_s)}{\dot{m} c_p} \times 0.02 Re^{0.8} \left(1 + \left(\frac{1}{Gz} \right)^{-0.29} \right) \times 3.4 St^{-0.035} \quad (17)$$

Figure 9 shows the results of the measured and estimated intake air temperature with varying boost pressure values, obtained from the equations (15), (16), and (17) at engine speed of 1600 rpm. Horizontal axis indicates measured data and vertical axis indicates estimated values. Pressure inside the intake port was used as the altering variable for comparing different conditions.

Fig. 9 Comparison of outlet air temperatures estimated by using equations (15), (16), and (17) when changing boost pressure, at coolant temperature of 353 K.

From Figure 9, it can be seen that by using the Colburn equation (15) and the empirical equation (16) of the intake port model, estimated gas temperature values had a bias error with respect to the measured values. On the other hand, using the empirical formula (17), it has been found that the estimated results of the intake air temperature agreed well with the measured data. Additionally, usefulness of the equation (17) was

checked by varying engine speed and coolant temperature. Estimated results were once again compared to the measured data.

Figure 10 shows the result of changing the cooling water temperature with a fixed value of engine speed, at 1600 rpm. From Figure 10, it can be understood that the intake air temperature increases as the cooling water temperature increases. Estimated values from equation (17) coincided with good precision to the measured values. Maximum error was found to be 5.6%, where minimum error was only 1.6%. On the other hand, the accuracy of the estimated results obtained from equation (16) was 13.9% in maximum, and 9.9% in average error. These differences between equations (16) and (17) were caused by the working frequency differences (several Hz - Port model experiment [9] vs. several tens Hz - Real engine experiments). For equation (15), maximum and average errors were found to be 27.7% and 18.1%, respectively.

Fig. 10 Comparison of outlet air temperatures estimated using equations (15), (16), and (17) and measured using thermocouples when changing coolant temperature

4.4. Numerical Simulation Results and Discussions

Similar to the results shown in Figure 4, there was a big difference on Nusselt number results calculated from the experimental data and Colburn analogy.

Table 5 Reynolds number, Nusselt number and heat transfer coefficient obtained from 1-D simulation using equations (8), (13) and (14)

	Equation (8)	Equation (13)	Equation (14)
Reynolds	5.1×10^4	4.9×10^4	4.7×10^4
Nusselt	1.1×10^2	2.5×10^3	5.8×10^3
HTC [W/m ² ·K]	1.2×10^2	2.7×10^3	6.4×10^3

As expected, comparable results were also obtained in 1-D engine simulation. Table 5 depicts Reynolds number, Nusselt number and heat transfer coefficient results which were obtained from equations (8), (13) and (14) in 1-D simulation.

Figure 11 illustrates instantaneous gas temperature variation during the intake stroke of the simulated engine in 1-D simulation with different boost pressure values. Engine speed was 1500 rpm and coolant temperature was set to 353 K. As

shown in Figure 11, rapid rise in temperature was observed during valve-overlap phase for all cases. As the boost pressure got higher, gas temperature values were observed to decrease along with the duration of the backflow caused during the valve-overlap phase, which roughly took place in between 0° - 29° CA. Higher boost pressure values lowered the pressure difference between the engine cylinder and the intake port, resulted in less amount of backflow gas penetrating into the intake port. Approximately 100 K difference on intake air temperature was observed between highest and lowest boost pressure values at valve-overlap phase.

Fig. 11 Air temperature change during intake stroke, at 1500 rpm, coolant temperature of 353 K - IVO at 0° CA, IVC at 262° CA

Figure 12 shows intake air temperature changes along the intake port of the simulated engine, calculated by equations (15), (16) and (17). Inlet gas temperature differences at 0 mm were caused by the heat transfer that took place at the upper stream of the engine's intake system.

Fig. 12 Comparison of intake port gas temperature values obtained from equations (15), (16) and (17). Engine speed at 1500 rpm, coolant temperature of 353 K, with boost pressure 678 hPa.

As depicted in Figure 12, outlet gas temperature variation between equations (16) and (17) only showed 5 K difference, whereas predicted results in the real engine experiments showed 30 K difference, as shown in Figures 8 and 9. This was a result of the mentioned backflow occurrences, which were not characterized in the empirical equations (15), (16) and (17). However, in 1-D simulation, backflow gas effect was included in

the calculations, thus outlet gas temperature differences between the real engine equation (17) and the port model equation (16) were smaller than the results that were obtained during the engine experiments. 1-D engine simulation conclusion displayed the importance of the backflow effect on intake air temperature and further research is needed to define its impact on the outlet gas temperature of the intake port.

5. Conclusions

In order to formulate the heat transfer at an intake system, intake air temperature measurements were done on a real engine, and the following knowledge was obtained accordingly:

- 1) Temperature differences of the intake port's surface wall temperature and intake gas temperature were measured with respect to changes in intake manifold pressure, engine speed and cooling water temperature. It was observed that the influence of the thermal boundary layer and intermittent flow should be taken into consideration in order to formulate an accurate intake port heat transfer.
- 2) Empirical equation for determining the Nusselt number inside the intake port was arranged by using the Reynolds number with the addition of Graetz number, and Strouhal number as dimensionless variables. Maximum and average errors of the empirical equation to the experimental data were found to be 5.6% and 1.6%, respectively. Applicable range of empirical equation was $3000 < Re < 30000$, $500 < Gz < 5000$, $0.02 < Sr < 0.2$.
- 3) 1-D engine simulation results revealed the effect of the backflow on outlet air temperature of the intake port, which was related to the pressure difference between intake manifold and engine cylinder. Approximately 100 K of temperature difference was encountered between the boost pressure values of 944 hPa and 678 hPa. These results from the 1-D simulation illustrated the importance of the backflow gas effect, which will be studied in detail as the next step for determining the outlet air temperature of the intake port.

* This paper is written based on the proceedings presented at 2018 JSAE Annual Congress (Spring).

Acknowledgements

This research was subsidized by the Japan Society for the Promotion of Science Grant-in-Aid for Scientific Research and Basic Research (C) (No.16 K06129).

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