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# UNSTEADY HEAT TRANSFER DURING THE RAPID COMPRESSION AND EXPANSION OF AIR

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## ABSTRACT

The instantaneous unsteady heat transfer during rapid compression and expansion of air within a pneumatically driven piston in a cylinder arrangement which offers simple,well-controlled and known boundary conditions was examined. Values of the instantaneous apparent overall heat flux from the cylinder gas to the wall surfaces were calculated using a thermodynamic analysis of the experimentally measured pressure and volume temporal development. Corresponding heat flux values were also calculated through the application of a zero-dimensional model that incorporates the use of the $k-e$ turbulence model. Comparison of the results of the model with corresponding experimental data showed fair to good agreement for the wide range of compression ratio values used (8.4~24.3). Also, correlation of the derived data using an effective velocity which is based on the distribution of mean kinetic energy, turbulence energy and piston motion and a characteristic length that is a function of the instantaneous height between the piston top and the cylinder head and bore diameter as the parameters to use when calculating the Reynolds, Prandtl and Nusselt numbers resulted in the following workable relationship: $Nu=0.01Re^{0.205}Pr^{0.033}$.

**Keywords:** $k-e$ turbulence model, unsteady heat transfer, rapid compression-expansion

## NOTATION

| | |
|---|---|
| B,D | cylinder bore (m) |
| $C_\mu$ | constant |
| K | mean kinetic energy (kW) |
| m | mass (kg) |
| $\dot{m}_i$ | mass flow rate (kg/s) |
| $\dot{m}_e$ | mass flow rate at exit (kg/s) |
| V | characteristic velocity (m/s) |
| k | turbulent kinetic energy (kW) |
| P | rate of turbulent k.e. production (kW/s) |
| ε | rate of turbulent k.e. dissipation (kW/s) |
| ρ | density (kg/m$^3$) |
| V | volume (m$^3$) |
| Ve | mean gas velocity (m/s) |
| $v_p$ | piston velocity (m/s) |
| u′ | turbulence intensity (m/s) |
| A | area of surface exposed to heat transfer (m$^2$) |
| $h_g$ | convective heat transfer coefficient (kW/m$^2$K) |
| Nu | Nusselt number |

Re Reynolds number
Pr Prandtl number
Pre mean gas velocity (m/s)
$q^o$ instantaneous heat transfer rate (kW)
t time (s)
T temperature (K)
$T_g$ mass-averaged bulk temperature (K)

**Subscripts and Superscripts**

air air
g gas
m motored operation
a,b exponent constants
p piston
w wall

## INTRODUCTION

Engine models describe the thermodynamic, fluid flow, heat transfer, combustion, and pollutant-formation phenomena. These models may be classified as zero-dimensional, quasi-dimensional and multi-dimensional. They also can be categorized as mainly thermodynamic or fluid dynamic in nature, depending on whether the equations which give the model its predominant structure are based on energy conservation or on a full analysis of fluid motion.

Zero-dimensional and phenomenological models assume at any instant a uniform state of the in-cylinder gas. In the absence of fluid flow modeling, geometric features of the fluid motion cannot be predicted. Once a model is calibrated, however it can represent a particular engine and can be used in a database. This type of model may be used for predicting the trends of some design changes of an engine on a relative basis and providing guidance to designers and operators.

In Multi-Dimensional models, the full governing differential conservation equations with time and spacial dimensions are solved for the engine processes. Many sub-models, such as those of turbulence and combustion, are still needed to avoid the need for direct solution of the complex Navier-Stokes and chemical reactions equations.

The heat transfer in internal combustion engines has always drawn much attention from engine researchers because it affects engine design,thermal loading of structural components,performance and emissions. The engine specific power and thermal efficiency are affected by the magnitude of engine heat transfer due to the losses of available energy. The changes in gas temperature due to heat transfer affect emissions formation,(especially for NOx and unburnt hydrocarbons), both within the engine cylinder and in the exhaust system. Therefore, the processes of gas-to-wall heat transfer, heat conduction and wall-to-coolant heat transfer need to be modelled correctly so as to provide the necessary information for the optimization of the performance of internal combustion engines. Of these processes, the gas-to-wall heat transfer is the most difficult to predict. This is mainly due to the very complex processes proceeding inside the engine cylinder that involve rapid and large changes in the temperature of the charge, the pressure and flow fields throughout the cycle. Under normal engine operation, heat losses from the working gases in the cylinder to the walls are due to primarily forced-convection heat transfer which is the major component of heat transfer, especially in spark ignited engines.

In-cylinder heat transfer models have been proposed over the years on the basis of heat transfer measurements and used for the analysis of in-cylinder heat transfer [Borman and Nishiwak, 1987]. For practical engineering purposes, most of the global and thermodynamic gas-to-wall convective heat transfer models employ the simplifying assumption that the instantaneous gas-to-wall heat transfer process in the reciprocating engine can still be regarded as quasi-steady and may be described by the following expression:

$$q^{\cdot}(t) = h_g(t)A(t)[T_g(t)-T_w(t)] \qquad (1)$$

Where $q^{\cdot}(t)$ is the instantaneous heat transfer rate, $h_g$ is the convective heat transfer coefficient, $A$ is the area of surface exposed to heat transfer, $T_g$ is the mass-averaged or bulk mean gas temperature and $T_w$ is the wall surface temperature, often assumed for simplicity to be uniform and constant. The main difference between various global and zonal gas-to-wall convective heat transfer models stem from the definition and derivation of the convective heat transfer coefficient employed in equation (1). Most of the convective heat transfer coefficient correlations proposed so far, which are basically empirically based on the measurement of local wall surface temperatures as well as local surface heat flux at the various measuring points chosen. One of the earliest dimensional convective heat transfer models for engine applications is that due to Nusselt [1923], which is based on experiments carried out in a spherical combustion bomb. He proposed the following convective heat transfer coefficient for internal combustion engines, (kW/m2K):

$$h_g=5.41x10^{-3}(1+1.24V_p)(Pre^2T_g)^{1/3} \qquad (2)$$

Where $V_p$ is the mean piston speed (m/s) and (Pre) is the cylinder pressure (MPa). Others proposed other expressions later. One of these is due to Eichelberg [1939], who correlated measurements of the instantaneous heat transfer rate, (kW/m2K) in an operating engine into the following empirical expression:

$$h_g=7.67x10^{-3}V_p^{1/3}(Pre.T_g)^{1/2} \qquad (3)$$

Empirical models of this type, being simple and explicit were widely used, but lacked accuracy and physical credibility and presented scaling problems. Some convective heat transfer models adopted dimensional analysis to develop the functional relationship governing the gas-side heat transfer coefficient. The basis of these models is the assumption that the following relationship between the Nusselet, Reynolds and the Prandtl numbers, which was developed originally for turbulent flow in pipes or over flat plates, may be used similarly to predict the instantaneous unsteady convective heat transfer in internal combustion engines:

$$Nu = Constant \times Re^{a} Pr^{b} \qquad (4)$$

The Prandtl number, Pr, for the engine cylinder content varies little during a cycle. The Reynolds number, Re, thus becomes the major parameter affecting the instantaneous unsteady convective heat transfer. However, the evaluation of its value during the various stages of the cycle remains a constant source of uncertainty and error. The widely used heat transfer models of this kind are those due to Annand[1963] and Woschni[1967]. From a re-analysis of the limited experimental data available at the time on instantaneous heat fluxes to selected cylinder head locations, Annand proposed the following instantaneous convective heat transfer coefficient correlation:

$$h_g = ak/D\ Re^{0.7} \qquad (5)$$

Where k is the mean thermal conductivity, D is the cylinder bore diameter and the value of ‘a’ varies with engine design and increases with increasing intensity of charge motion in the engine cylinder (from 0.35 to 0.80). Another term is added to equation (5) to account for radiation heat transfer when considered to be relatively significant. Mainly for convenience, the Reynolds number was based on the mean piston speed and cylinder bore diameter, with the gas properties evaluated at the bulk mean charge temperature obtained from the ideal gas law. Only data from cylinder head thermocouple locations were used to develop this correlation, but they were often used to estimate the instantaneous spatially averaged heat fluxes for the entire cylinder chamber of engines.

Woschini[1967] proposed another widely used heat transfer model that was similar to correlations of turbulent heat transfer in pipes; “$Nu \sim Re^{0.8}$” . A heat balance was used to determine the total instantaneous heat transfer between the cylinder gas and walls for a complete engine cycle. After taking the cylinder bore diameter as the characteristic length and assuming the thermal conductivity to be $k \sim T^{0.75}$, the following heat transfer coefficient, (kW/m2K), correlation without an explicit term for radiation was proposed for the intake, compression and exhaust strokes:

$$h_g = 0.82 D^{-0.2} P^{0.8} Ve^{0.8} T_g^{-0.53} \qquad (6)$$

where (Ve) represents the mean gas velocity(m/s) affecting heat transfer which was assumed to be proportional to the mean piston speed during intake, compression and exhaust. During combustion and expansion, an additional gas velocity term proportional to the pressure rise due to combustion, “$P-P_m$” was added where $P_m$ is the motored cylinder maximum pressure.

Most models of this kind characterize the instantaneous unsteady convective heat transfer by dimensionless parameters and represent the actual in-cylinder flow velocities through the use of the mean piston speed. However, although they are simple and are widely used, they are very approximate and do not represent adequately the effects of fluid motion and the variations of the in-cylinder gas flow velocities. They also use simplistically the constant value of the cylinder bore diameter as the characteristic length. For example, comparisons between heat fluxes predicted by the various formulae for the same engine and operating conditions showed very large differences [Lefeuvre et al.,1969 and Jackson,et al. 1990]. The constants in these models must be adjusted empirically to give reasonable overall instantaneous heat transfer values from engine to engine and even from one set of operating conditions to another, which restricts the applicability and accuracy of these models. Moreover, virtually all these correlations have been made for fired internal combustion engines while accounting for the relatively simple case of heat transfer from motored engines or reciprocating compressors has been relatively ignored.

A successful instantaneous convective heat transfer model must take into account all the key in-cylinder fluid motions such as intake flow, swirl, squish, piston driven axial flow, turbulence and motion generated by fuel injection and combustion which require the use of more complex flow-based models[Morrel and Keribar, 1985]. Knight [1964] proposed the use of gas velocity calculated from the mean kinetic energy of the gas charge in the cylinder. In order to account for the effect of air swirl, LeFeuvre et al. 1969] and Dent and Suliaman [1977] considered the gas flow pattern due to swirl as a solid-body forced vortex having many similarities to the flow near rotating discs. They proposed the use of gas swirl velocity as a characteristic velocity and the radius of the thermocouple position from the bore axis where the temperature is measured, as the characteristic length in the Reynold number definition. The instantaneous local convective heat transfer coefficient obtained from their correlations varied with radius along the piston and head but is constant along the liner. They could not include the important effects of squish, turbulence, and deceleration and dissipation of the air swirl due to the action of viscous drag on the surrounding walls. Davis and

Borgnakke[1988] considered the effects of turbulence on convective heat transfer explicitly in a two-zone heat transfer model where the characteristic velocity was taken to be the square root of the turbulent kinetic energy and the characteristic length was the turbulence length scale. However, the effect of the mean flow on the heat transfer model of Poulos and Heywood[1983] considered convective heat transfer to be driven by both the mean gas motion and turbulence. Their model defined the characteristic length as the macroscale of turbulence and the characteristic velocity as an effective velocity due to contributions from the mean flow kinetic energy, the turbulence kinetic energy and the instantaneous piston speed. They did not include the effects of swirl and squish that are critically important. Ikegami et al.[1986] considered the effects of swirl and squish motion on local heat fluxes in an axisymmetric numerical model in a motored engine cylinder. The computed surface averaged heat flux was used to obtain the Nusselt-Reynolds numbers correlation. Morel and Keribar [1985] and Morel et al. [1982] proposed the use of kinetic-energy-based gas velocity model.

In principle,the flow-based models are expected to be superior to the dimensional and dimensionless models. However, the flow-based models have not yet been sufficiently evalutaed, and the key fluid motion responsible for instanteous convective heat transfer in engines are still inadequatly described in thses models. Accordingly, further work is needed to provide adequate theoretical analysis of the complex process of the instantaeous unsteady heat transfer in reciprocating devices. The present contribution examines the instantaeous unsteady heat transfer in a circular cylinder closed by a moving flat piston of a rapid compression-expansion machine that offers much simpler, well-controlled and better known boundary conditions than in a fired or motored engines.

## EXPERIMENTAL SET-UP

Rapid compression-expansion machines have been recognized as a useful research tool that can offer advantages over the direct use of fired or motored engines, since they provide a better control of the various experimental parameters. For example, not only can the compression ratio and piston speed be varied independently over a wide range, but also the pressure, temperature, velocity distribution and costituent composition at the start of compression can be controlled precisely. These can enable a reliable estimation of the heat loss from the charge to the cylinder walls to be made. Uncertain engine effects, such as induction swirl, blow-by and contamination of the charge by lubricants and residuals from previous cycles can be eliminated. Furthermore, the temperature and state of the cylinder surface may be kept invariant. The piston motion, if suitably controlled pneumatically, can also permit in principle the simulation of density-time relationships encountered in reciprocating engines.

All the experimental data necessary for the present work were obtained by Karim & Watson [1968] in the engine-like rapid compression-expansion apparatus, having a 5.08 cm bore and a variable stroke of up to 34.29 cm while using dry air only as the working medium.

The initial charge temperature control is obtained by using a jacketed working cylinder with heated oil circulating through the jacket. In the present study, the wall of the cylinder remained at the initial temperature of about 353 K throughout the compression and expansion of the gas. The initial charge was of closely controlled composition with no lubricating oil used. The initial pressure was normally atmospheric. The charge pressure-time records were obtained using a pre-calibrated flush-mounted piezoelectric, temperature and vibration-compensated pressure transducer mounted centrally and flush in the flat and smooth cylinder head. The pressure-time and piston displacement-time data were measured over a wide range of compression ratios (8.4~24.3).

The pressure and volume-time data were obtained experimentally. Through the application of thermodynamic analysis the instantaneous apparent spatially averaged heat flux and heat transfer coefficient were derived [Chen and Karim, 1998].

## ZERO-DIMENSIONAL $k-e$ TURBULENCE MODELING

A heat transfer model of the zero-dimensional that uses the $k-e$ turbulence model requires an estimate of the characteristic flow velocity and length scales. The model which incorporates the key physical mechanisms affecting charge motion in the cylinder, was used to estimate these scales. The model consists of a zero-dimensional turbulence energy cascade where the mean kinetic energy of the flow is supplied normally to the cylinder through the intake and converted into turbulent kinetic energy through turbulent flow motion and the turbulent kinetic energy is converted to internal energy through viscous dissipation. When mass flows out of the cylinder, it carries out with it both mean and turbulent kinetic energy. Details of this model are described by Mansouri and Bakhshan [2000,2001]. Fig. 1 shows schematically the energy cascade of this model. The swirl in the cylinder is amplified during compression. In this study the swirl at the start of compression was taken to be zero and the kinetic energy is of a very low value.

At any time during the cycle, the mean flow velocity and turbulent intensity are found from a knowledge of the respective mean and turbulent kinetic energies:

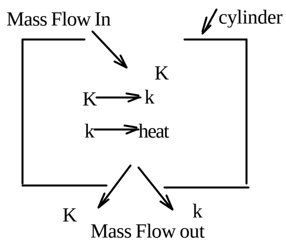


**Fig. 1 Turbulent energy cascade used, K :mean kinetic energy; k: turbulent kinetic energy**

$K=1/2\, mu^2$ and $k=3/2\, mu'^2$ (7)

$$\frac{dk}{dt} = P - m\varepsilon - k\frac{\dot{m_e}}{m} + RD \qquad (8)$$

$$\frac{dK}{dt} = \frac{1}{2}\dot{m_i}V_i^2 - P - K\frac{\dot{m_e}}{m} \qquad (9)$$

$$\varepsilon = \frac{u'^3}{l} = \frac{(2k/3m)^{3/2}}{l} \qquad (10)$$

Where the "RD" is rapid distortion and " ε " is dissipation rate. During compression the turbulent kinetic energy decays owing to viscous dissipation but is amplified owing to the rapid distortion that the cylinder charge undergoes with rising cylinder pressure. The rate of turbulence amplification due to rapid distortion can be calculated by assuming that conservation of mass and angular momentum can be applied to the large scale eddies during the rapid distortion period [Ramos, 1988].

$$\frac{du'}{dt} = \frac{u'}{3\rho}\frac{d\rho}{dt},\; RD = 3mu'\frac{du'}{dt} = \frac{2}{3}\frac{k}{\rho}\frac{d\rho}{dt} \qquad (11)$$

The macroscale of turbulence and geometric length scale are assumed to be given by:

$$L = l = V/\left(\pi B^2/4\right) \qquad (12)$$

where B is the bore and V is the instantaneous volume of cylinder. Heat transfer due to convection from the turbulent flow in the cylinder to the combustion chamber walls can be expressed as:

$$\dot{Q}_w = hA(T_g - T_w) \qquad (13)$$

The value of h is calculated by using the following relationship

$$Nu = a(Re)^d(Pr)^e \qquad (14)$$

In this model, the characteristic length scale is taken to be the macroscale of turbulence, as calculated equation(12 ) and V is postulated to be an effective velocity due to contributions from the mean and turbulent kinetic energies and piston motion:

$$V = \left[u^2 + u'^2 + \frac{1}{2}(v_p)^2\right]^{1/2} \qquad (15)$$

The gas temperature in equation (13) is assumed to be the mass-averaged mean temperature. The viscosity and thermal conductivity of gases in equation (14) is found from correlations in reference [Mansouri and Heywood, 1980].

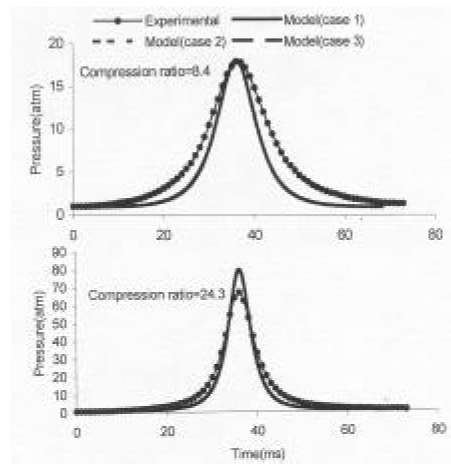


**Fig. 2 Comparison of calculated mean cylinder pressure and measured cylinder pressure for compression ratios of 8.4 and 24.3**

It is shown that there is a fair agreement between the calculated and measured values in general with small differences occurring around TDC. A better level of agreement is obtained with the lower compression ratio. The differences between the calculated values for the three cases were relatively negligible. A number of possible factors that may include the underestimation of wall heat transfer by the model and inaccuracies associated with the experimental data such as the calculated pressure does not necessarily represent the measured value at the center of the cylinder head., could have contributed. Also the validity of applying the basic relationship of equation(14) remains strictly uncertain.

Fig. 3 shows the difference between the computed and measured peak pressure values increases with increasing the compression ratio and a better level of agreement was obtained for compression ratios that are less than 17:1.

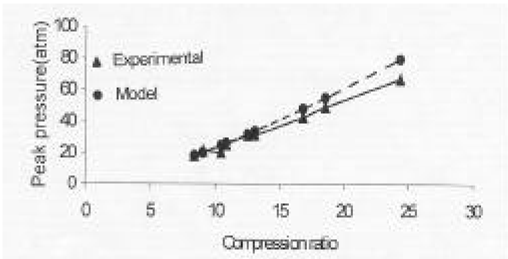


**Fig. 3 The variation of peak pressure with compression ratios.**

Through the application of a zero-dimensional $k-e$ turbulence model the associated instantaneous unsteady heat transfer from the cylinder gas to the wall surfaces could be obtained. Figure 4 shows the temporal variations of the calculated apparent overall heat flux for two cases involving a relatively low compression ratio of 8.4 and a much higher compression ratio of 24.3. the heat flux values expectedly are much higher with the high compression ratio. A better level of agreement between the experimental values and the corresponding calculated values were obtained when the characteristic length in the modeling was chosen to be the instantaneous height between the piston top and the cylinder head (case 1) or when it was chosen as a combination of this height and bore diameter (case 3). In case 2, when the characteristic length was chosen to be the bore, there were large differences occurring around the top dead center (TDC) region. For the higher compression ratio the experimental and model values are in good agreement only when the characteristic length was chosen to be a combination of the instantaneous height between the piston top and differences cylinder head and the bore diameter (case 3). The higher cylinder pressure predicted by the model with the high compression ratio could be partly due to the limitations to describe correctly turbulence when in-cylinder flow intensity increases to outside the range of current turbulent model available in the simulation.

These results show that the cylinder bore diameter cannot represent adequately the effects of the periodic change in the cylinder volume and surface area on heat transfer during compression and expansion at least for the configuration of the set up considered. On the other hand, the use of the instantaneous height tends to over predict. Thus, it appears that a combination of this height and the bore diameter is more suitable.

Values of the surface-mean heat transfer rates are usually expressed in terms of a function linking the Nusselt, Prandtl and Reynolds numbers. In this investigation, when using as a characteristic velocity, an effective velocity which is a contribution of the mean kinetic, turbulence intensity and piston motion and as a characteristic length made up of the combined instantaneous height between the piston top and the cylinder head and the bore diameter, these results were then concluded in terms of the following formula:

$$Nu=0.01Re^{0.205}Pr^{0.033} \tag{16}$$

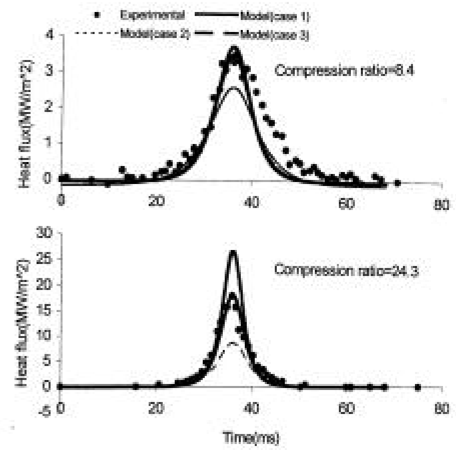


**Fig. 4 Comparison of calculated apparent overall heat flux with experimental for compression ratios of 8.4 and 24.3**

## CONCLUSIONS

The predictive model for the apparent heat flux in a rapid compression-expansion apparatus used showed fair to good agreement with experiment over a wide range of compression ratio values,(8.4~24.3).

Correlation of derived data, using an effective velocity, which is the distribution of mean kenitic, turbulence intensity and piston speed, as a characteristic velocity and a characteristic length based on a combination of the instantaneous height between the piston top and cylinder head and the bore diameter for calculation of the effective Reynolds, Prandtl and Nusselt numbers, resulted in: $Nu=0.01Re^{0.205}Pr^{0.033}$.

A best combination of the instantaneous height between the piston top and cylinder head and the cylinder diameter for use as a characteristic length was found to be:

$$L= 0.25*(\text{bore}) + 0.75*(\text{instant height}) \tag{17}$$


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