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HEAT TRANSFER INVESTIGATION IN THE INTAKE PORT OF FOUR STROKE DIRECT INJECTION COMPRESSION IGNITION ENGINE

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ABSTRACT: Heat transfer is one important aspect of energy transformation in compression ignition engines. Fast transient heat flux between the combustion chamber and the cylinder wall must be investigated to understand the effects of the non-steady thermal environment. The objective of this paper is to present the development and application of heat transfer model to the intake manifold of four stroke direct injection diesel engine. One-dimensional (1D) gas dynamics was used to describe the flow and heat transfer in the components of the engine model. The engine model has been simulated with variable engine speed from 500 to 4500 rpm with increment of 500 rpm.
KEYWORDS: Diesel engine, computational simulation, heat transfer, 1D CFD, intake port

INTRODUCTION

In the last decades, the legislation on internal combustion engines (ICEs) has severely reduced the limits for pollutant and noise emissions. These requirements have established the research activity at design phase as a key stage in the engine production process. Therefore, an intensive investigation on ICEs has been carried out, focusing on the optimization of performances and fuel consumption. In particular, an important effort has been done seeking the improvement of the combustion and gas exchange processes, using tools such as Computational Fluid Dynamics (CFD).

Heat transfer is an important process in the intake manifold of engines. It increases the charge temperature which reduces the volumetric efficiency and also causes higher chemical reaction rates leading to increased NOx emissions. It also affects engine performance and emissions through enhancing the fuel evaporation and charge mixing process in cylinders. As a result, many experimental and theoretical studies have investigated heat transfer in the intake manifold of engines.

The heat transfer involved in the intake system occurs when air or an air-fuel mixture comes into the manifold. The intake manifold is hotter than the air-fuel mixture because of its proximity to the engine components or the design of the manifold. The intake manifold can be designed to heat the air-fuel mixture, so that the mixture can start to vaporize once it has entered the combustion chamber. One way of heating the manifold is to put it in close proximity with other hot components.

The manifold will heat through convective heat transfer. Electricity and hot coolant flow are other ways in which the manifold can also be heated. After the manifold is heated then the air-fuel mixture is heated through convective heat transfer. Equation (1) shows the heat transfer problem associated with the air-fuel mixture and the manifold walls.

$$\dot{Q} = hA(T_{wall} - T_{gas}) \qquad (1)$$

where, $\dot{Q}$ amount of heat transfer, h convective heat transfer coefficient, A inside surface area of intake manifold, T_{wall} wall temperature, T_{gas} gas temperature.

$$h = \frac{d}{k_f} C\,Re^m \qquad (2)$$

where, d cylinder diameter, k_f fluid thermal conductivity.

The classical, steady correlation are widely used for estimating the convective heat transfer coefficient in the intake manifolds of engines [1, 2, 3, 4], because of a correlations type easy to use as well as unsteady heat transfer model is not available. An equation (3) and (4) shows the classical steady correlations.

$$N_u = C\,Re^m \qquad (3)$$

$$N_u = C\,Re^m P_r^n \qquad (4)$$

where, C, m and n constants are adjusted to match the experimental data to account for unsteady heat transfer enhancements, surface deposits and surface roughness. The data of the steady state correlations constants was presented in Table (1).

Table 1. Steady state correlations constants

Reference of previous study	C	m	n
Dittus and Boelter (1930)	0.023	0.80	0.4
Bauer et al. (1998) for straight manifold	0.062	0.73	0.0
Bauer et al. (1998) for curved manifold	0.140	0.66	0.0
Depcik and Assanis (2002)	0.069	0.75	0.0
Shayler et al. (1996)	0.135	0.71	0.0

The frequencies based on valve events and pipe lengths, drastically alter the flow patterns and change the heat transfer relationship [3, 5]. The correlations provide good agreement with experimental data in fully-developed steady pipe flows and acceptable agreement with time-resolved experimental data in unsteady flows and slow velocity variation under the quasi-state assumption. It is important to indicate that these correlations can produce large errors in both phase and magnitude [6] for highly unsteady flows with rapid velocity variations. Different researchers suggest that the unsteady flow effect in the engine intake manifold enhances heat transfer by 50 to 100% over the prediction of the steady pipe flow correlations presented by Dittus and Boelter [1]. At different engine speed and load, the unsteadiness of the flow condition is different. Therefore, the constants C and m are usually optimized only for one operation condition of a given engine and hence compromised for other conditions.

FLUID DYNAMICS GOVERNING EQUATIONS

The flow model involves the solution of the Navier-Stokes equations, namely the conservation of continuity, momentum and energy equation. These equations are solved in one dimension, which means that all quantities are averages across the flow direction. There are two choices of time integration methods, which affect the solution variables and limits on time steps. The time integration methods include an explicit and an implicit integrator. The primary solution variables in the explicit method are mass flow, pressure and total enthalpy.

In broad terms, a model is created using two types of discritization. Firstly, the complete powertrain system is grouped into general components. These components consist of air cleaners, valves, piping, valves, fuel injectors, mufflers, resonators, catalytic converters, combustion chambers and resonators. The second aspect is separating each component into multiple control volumes. Each control volume is bounded by another control volume or wall. By discritizing the system into sufficiently small volumes, the properties of the fluid in that volume can be assumed to be constant. The scalar variables (pressure, temperature, density, internal energy, enthalpy, species concentration, etc.) are assumed to be uniform over each volume. The vector variables (mass flux, velocity, mass fraction fluxes, etc.) are calculated for each boundary. These type of discretization is referred to as a “started grid".

The conservation equation (5), energy equation (6), enthalpy equation (7) and momentum equation (8) are shown below.

$$\frac{dm}{dt} = \sum_{boundaries} \dot{m} \tag{5}$$

$$\frac{d(me)}{dt} = -p\frac{dV}{dt} + \sum(\dot{m}H) - hA_s(T_f - T_w) \tag{6}$$

$$\frac{d(\rho HV)}{dt} = \sum(\dot{m}H) + V\frac{dp}{dt} - hA_s(T_f - T_w) \tag{7}$$

$$\frac{d\dot{m}}{dt} = \frac{dpA + \sum_{boundaries}(\dot{m}u) - 4C_f\frac{\rho u|u|}{2}\frac{dxA}{D} - C_p\frac{1}{2}\rho u|u|\ A}{dx} \tag{8}$$

where, $\dot{m}$ boundary mass flux into volume, m mass of the volume, V volume, p pressure, ρ density, A flow area (cross-sectional), A_s heat transfer surface area e total internal energy (internal energy plus kinetic energy) per unit mass, H total enthalpy, h heat transfer coefficient, T_f fluid temperature, T_w wall temperature, u velocity at the boundary, C_f skin friction coefficient, C_p pressure loss coefficient, D equivalent diameter, dx length of mass element in the flow direction (discretization length), dp pressure differential acting across dx.

MAIN ENGINE DATA

The development of the four cylinder modeling and simulation for four-stroke direct-injection (DI) diesel engine was described in [7]. The specific engine characteristics are used to make the model are shown in Table (2).

Table 2. Specification of the engine

Engine Parameters	Value
Bore (mm)	100
Stroke (mm)	100
Displacement (cc)	3142
Number of cylinder	4
Compression ratio	18
Connecting rod length (mm)	152
Piston pin offset (mm)	0
Intake valve open (0CA)	340
Intake valve close (0CA)	-137
Exhaust valve open (0CA)	127
Exhaust valve close (0CA)	376
Brake power (KW)	120.9
Brake torque (Nm)	384.8
Model : Four Cylinder, Four-Stroke, Turbocharged, Vertical, Air Cooling.	

It should be noted that the intake and exhaust ports of the engine are modeled geometrically with pipes. The intake port characteristics of engine are shown in Table (3). The intake and and exhaust ports on the engine are modeled geometrically with pipes and the air enters trough a bell-mouth orifice to the pipe. The discharge coefficient of the bell-mouth orifice was set to 1 to smooth transition.

Table 3. Intake port characteristics

Intake port parameter (unit)	Value
Diameter at inlet end (mm)	47
Diameter at outlet end (mm)	47
Lenght (mm)	95
Surface roughness (mm)	0
Wall temperature. (K)	340

RESULT AND DISCUSSION

The heat transfer steady state and transient simulation of the intake port of diesel engine model was running on any different engine speed in rpm, there are 500, 1000, 1500, 2000, 2500, 3000, 3500, 4000 and 4500.

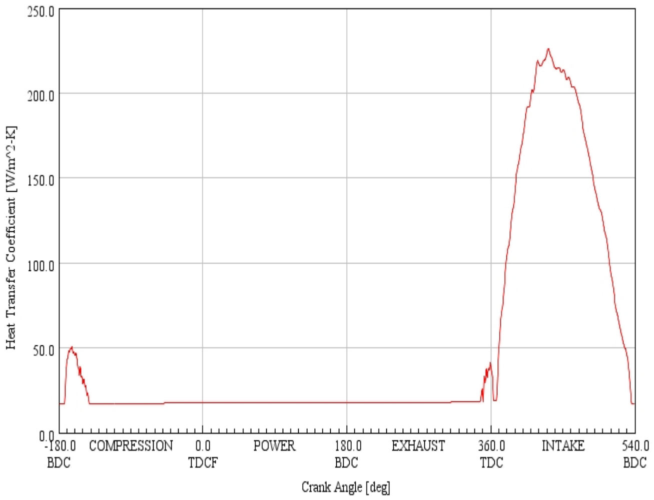

Figure 1. Heat transfer coefficient in intake port at 500 rpm

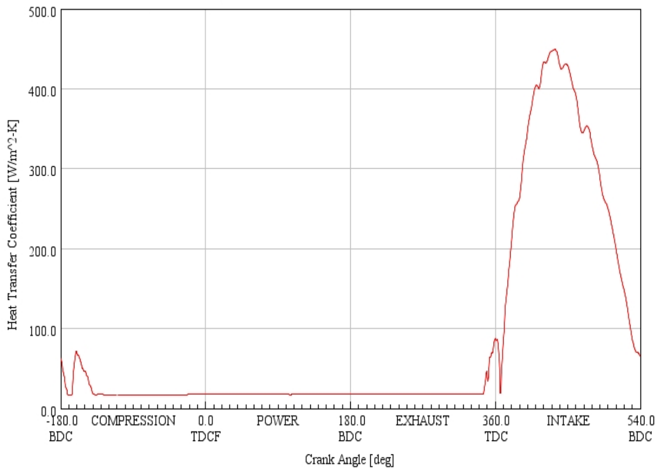

Figure 2. Heat transfer coefficient in intake port at 1000 rpm

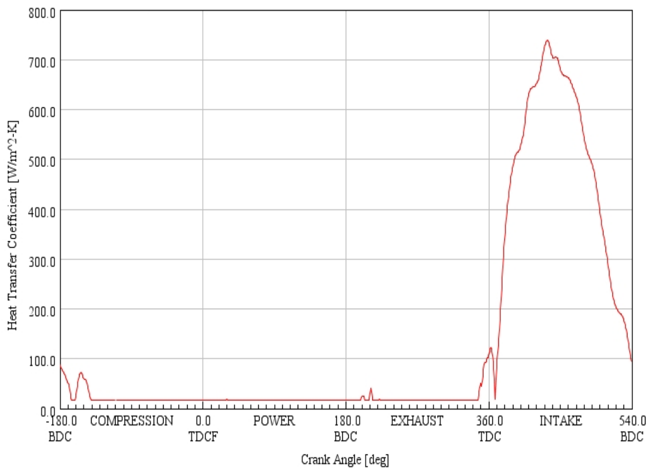

Figure 3. Heat transfer coefficient in intake port at 1500 rpm

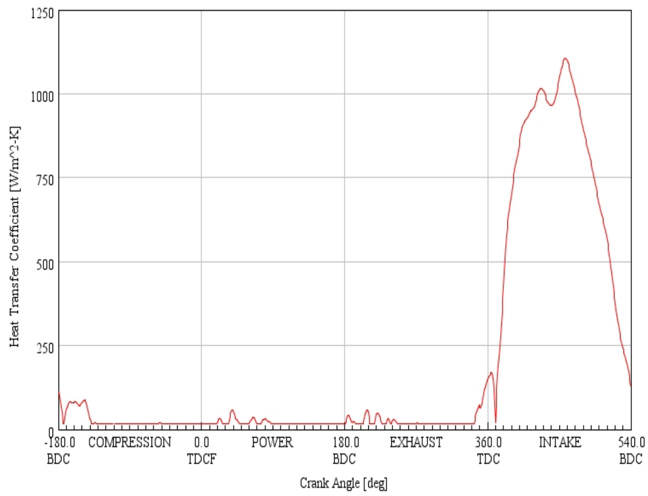

Figure 4. Heat transfer coefficient in intake port at 2000 rpm

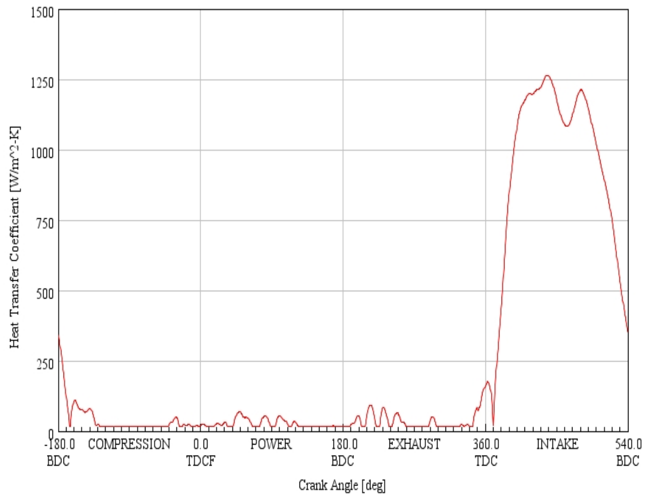

Figure 5. Heat transfer coefficient in intake port at 2500 rpm

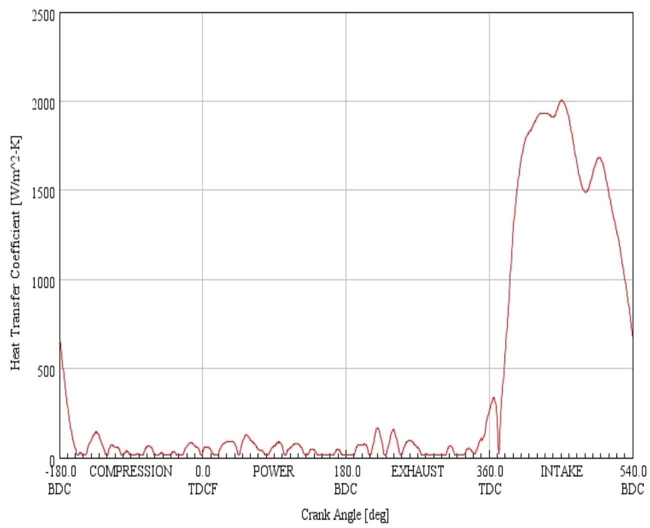

Figure 6. Heat transfer coefficient in intake port at 3000 rpm

The heat transfer coefficient in intake port simulation results are shown in Figure 1-9.
The fluid to wall heat transfer rate in intake port simulation results are shown in Figure 10-18.

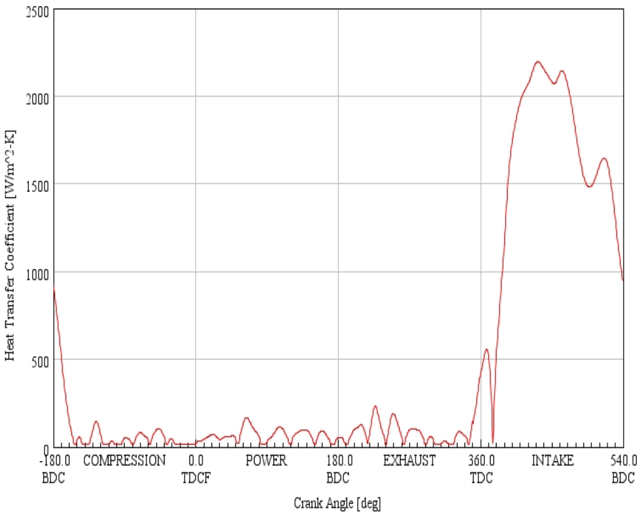

Figure 7. Heat transfer coefficient in intake port at 3500 rpm

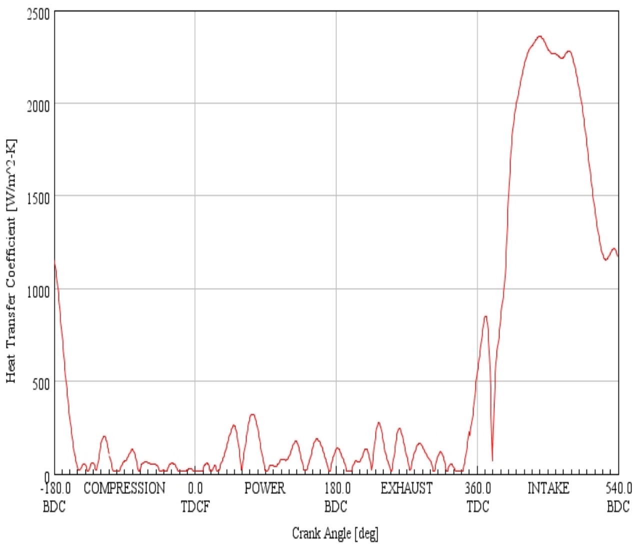

FIGURE 8. Heat transfer coefficient in intake port at 4000 rpm

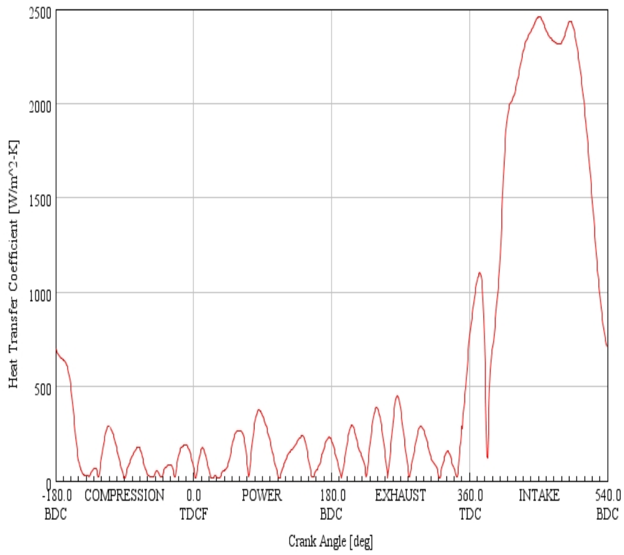

Figure 9. Heat transfer coefficient in intake port at 4500 rpm

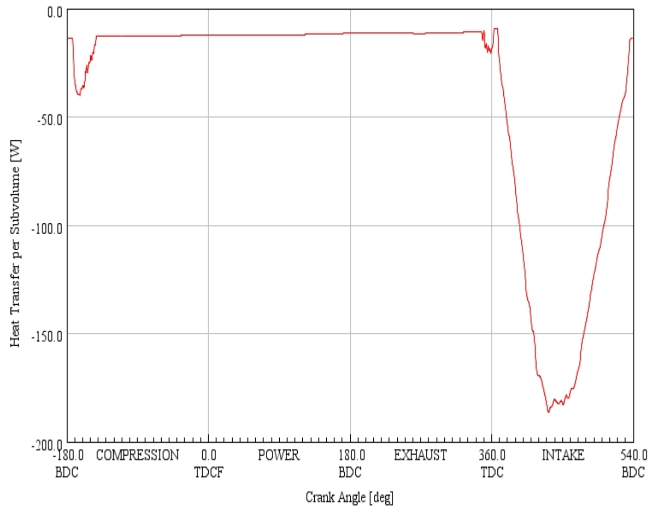

Figure 10. Fluid to wall heat transfer rate in intake port at 500 rpm

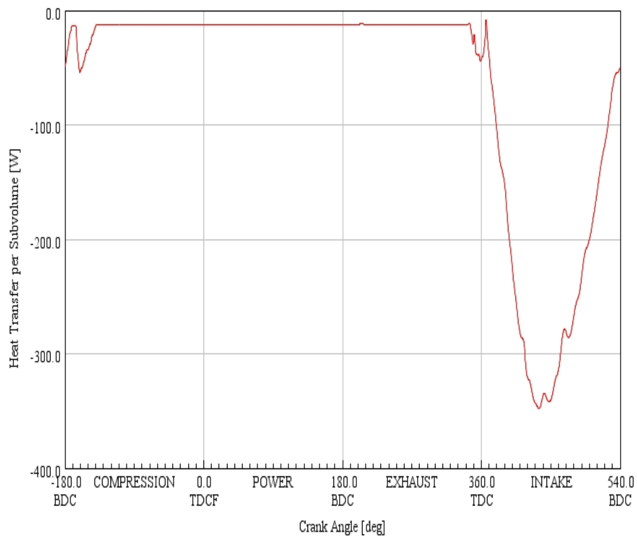

Figure 11. Fluid to wall heat transfer rate in intake port at 1000 rpm

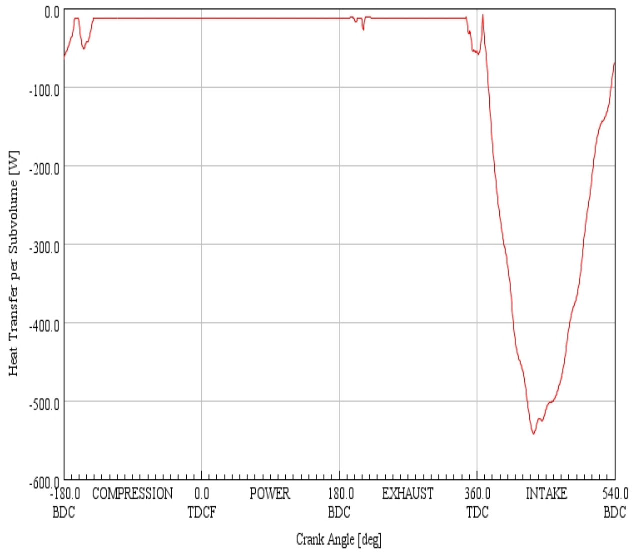

Figure 12. Fluid to wall heat transfer rate in intake port at 1500 rpm

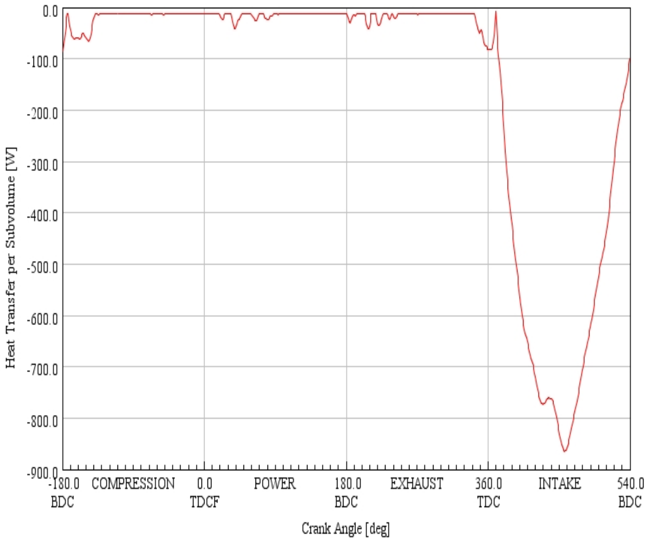

Figure 13. Fluid to wall heat transfer rate in intake port at 2000 rpm

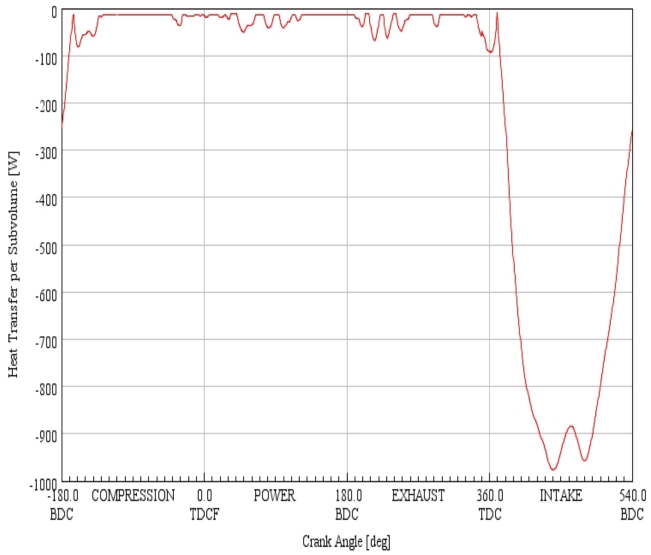

Figure 14. Fluid to wall heat transfer rate in intake port at 2500 rpm

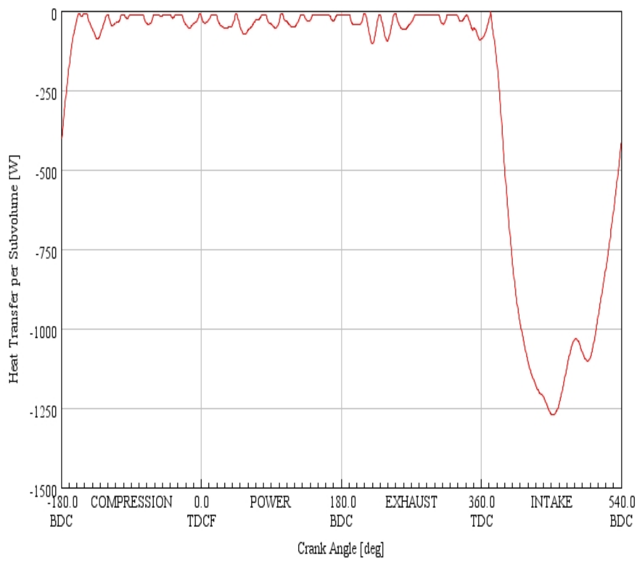

Figure 15. Fluid to wall heat transfer rate in intake port at 3000 rpm

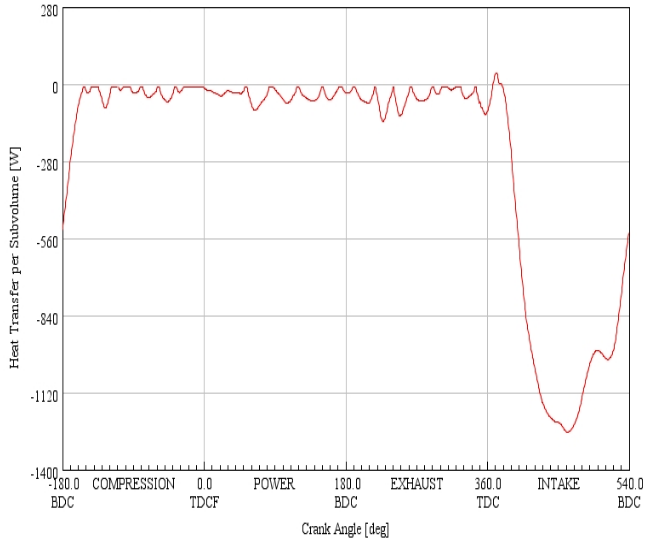

Figure 16. Fluid to wall heat transfer rate in intake port at 3500 rpm

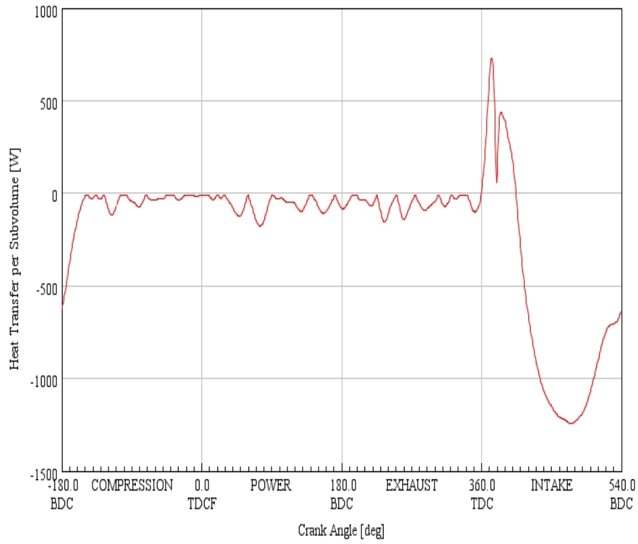

Figure 17. Fluid to wall heat transfer rate in intake port at 4000 rpm

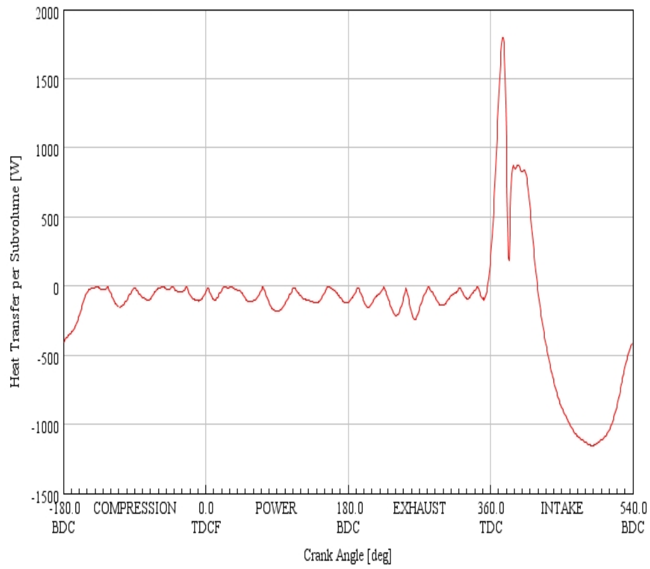

Figure 18. Fluid to wall heat transfer rate in intake port at 4500 rpm

The heat transfer coefficient obtained at different crank angle in intake port of engine in variation of engine speed shows that in turbocharged direct injection diesel engine, the values of heat transfer in compression stroke, power stroke and exhaust stroke is near to zero point at engine speed from 500 to 1500 rpm, because in this stroke the intake valve is closed. So, the heat from combustion process in engine cylinder is not interacted with intake port wall and heat gas from fluid combustion is not flow to intake port wall. The situation is quite different if the intake valve is opened, the heat from gas flow to exhaust valve is still in engine cylinder and quickly flows out to intake valve and intake port if the intake valve is opened. At engine speed over 1500 rpm, the values of heat transfer in the compression stroke, power stroke and exhaust stroke go to unstability (Fig. 13 - 18) to the down line and at engine speed from 2000 to 4500 rpm the values of heat transfer are only unstable during compression, power e and exhaust strokes. The probable reason for this effect of instability is increasing of the velocity and turbulence intensity of gas flow in intake port at high speeds. Heat transfer coefficient in intake port correlates with the engine speed - it is lowest at low engine speeds and highest at high engine speeds.

The fluid to wall heat transfer rate in intake port of engine versus crank angle degree at different engine speeds are shown in Fig. 10-18. It is evident that in turbocharged direct injection diesel engine, the nominal value of fluid to wall heat transfer rate in the compression, power and exhaust strokes is near to zero point at engine speed from 500 to 1500 rpm so the heat from combustion process in engine cylinder is not interacted with intake port wall and heat gas from fluid combustion is not flow to intake port wall. It is very different if the intake valve is opened. The heat from gas flow to intake valve is still in engine cylinder and quickly flows out to intake valve and intake port when the intake valve is opened. The trend result of the nominal fluid to wall heat transfer rate is shown that increasing of engine speed lead to increasing of fluid to wall heat transfer rate. The lowest fluid to wall heat transfer rate was obtained at 500 rpm and the highest - at 4500 rpm.

CONCLUSIONS

Heat transfer coefficient and fluid to wall heat transfer rate versus crank angle degree during the compression , power and exhaust strokes are stable at engine speed region 500 -1500 rpm. Its show that in these strokes the heat transfer is near to zero point and at engine speed region 2000 - 4500 rpm the values of heat transfer are only unstable. The probable reason for this effect of instability is increasing of the velocity and turbulence intensity of gas flow in intake port at high speeds. The heat transfer is maximal during intake stroke, because of interaction between gas in engine cylinder and intake port wall.

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