Effect of heat transfer correlation on wet cylinder liner temperature distribution when converting an old engine into a turbocharged engine

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Abstract For conventional diesel engines, two of the most widely used global correlations are due to Woschni and Hohenberg. Besides, the modern diesel engines used a new heat transfer coefficient correlation was proposed by Finol and Robinson. In Vietnam, improving engine power density is a trend of improving non-turbocharged base engines by using a supercharg-ing system with exhaust gas energy recovery. Increasing engine power by the turbocharger is limited for two reasons: mechanical stress and thermal stress of the components surrounding the combustion chamber. In general, the heat transfer coefficient has a major effect on heat transfer rate, es-pecially during the combustion process. So, the purpose of this study is to compare the cylinder distribution results from the simulation using the equations of Woschni and Hohenberg and compare to the experiment re-sults when converting an old heavy-duty engine into a turbocharged engine. Results show that the cylinder distribution using Hohenberg’s correlation has a good agreement with the experiment results, especially in the case of a turbocharged engine.

Keywords: Heat transfer correlation; Turbocharged engine; Cylinder liner distribution; Supercharging

Nomenclature

A – surface area, m^2
a, *b* – parameters
B – cylinder bore, m
C_1, C_2 – constants
htm – heat transfer multiplier
C_u – peripheral gas velocity, m/s
k_g – thermal conductivity of the gas, W/m·K
L – characteristic length, m
N_{an} – swirl anemometer speed, rad/min
Nu – Nusselt number
n – engine rotational speed, rpm
PiK – pressure ratio, the ratio between start and end pressure of the compression
p – instantaneous cylinder gas pressure, bar
$\dot{Q}$ – heat transfer rate, W
Re – Reynolds number
T – temperature, K
V – characteristic velocity, m/s
V_p – mean piston speed, m/s
V_i – instantaneous cylinder volume, m^3
d_{sp} – sphere diameter, m
V_s – swept volume, m^3

Greek symbols

α – convective heat transfer coefficient, W/m^2K
ν – kinematic viscosity, kg/m·s

Subscripts

g – gas
m – motored condition
r – reference state
wall – wall surface
an – anemometer
p – piston
s – swept
i – instantaneous
sp – sphere

Acronyms

TDC – top dead center
BDC – bottom dead center
CMP – compression process
EXH – exhaust process
TDCF – top dead center firing
ATDCF – after top dead center firing
BTDCF – before top dead center firing

1 Introduction

The cylinder heat transfer is one of the most important processes of inter-nal combustion engines and a universal convective heat transfer correlation is not available. The engine heat transfer process is complex due to some items such as the unsteady, compressible and turbulent flow field, spatial and temporal variations of boundary conditions, varying fluid properties, surface details, and the interactions with the equally complex combustion process. The global correlation for the cylinder heat transfer are often sin-gle expressions that are applied for the complete cylinder and are used on the instantaneous basis. The actual engine cylinder heat transfer is a com-plex process, and has extreme spatial and temporal variations. To expect a single, global correlation to capture all of these details is unrealistic. On the other hand, engineers often need practical, approximate correlations that can help complete evaluations that could not be done otherwise. For these reasons, the pursuit of appropriate, approximate correlations contin-ues [1, 2].

There are many overviews of engine cylinder heat transfer published over the years, some of the earliest works provided global correlations for the convective heat transfer correlation. Nusselt in 1923 described one of the first heat transfer correlations for a diesel engine [3]. The data used by Nusselt for this correlation were obtained from the constant volume com-bustion facilities. Although this expression is not often used anymore, the basic approach of Nusselt has still continued throughout the history of de-veloping correlations for engine cylinder heat transfer, such as, Eichelberg in 1939, Annand in 1963, and Woschni in 1967 in which, the global cor-relation of Woschni was widely used. His correlation was based on diesel engine data [4–6]. Woschni suggested several different characteristic veloci-ties depending on the portion of the cycle. During intake, compression, and exhaust, he assumed that the characteristic velocity would be proportional to the mean piston speed. During combustion and expansion, he assumed the velocity would scale with some measure of the combustion intensity. Therefore, he proposed that the velocity for this period would be propor-tional to the increase in-cylinder pressure above the motoring pressure.

Hohenberg in 1979 re-examined much of the previous cylinder heat trans-fer information, including that from Woschni, and proposed a new correla-tion from his investigation on four different direct injection diesel engines and completed extensive experiments that included measurements of heat flux, heat balance, and component temperatures [7].

Finol and Robinson have been examined several empirical models to predict the in-cylinder heat transfer in internal combustion engines [8]. The investigation showed that the main difficulty remains the right selection of the significant parameters, namely, flow velocity and to a lesser extent characteristic length and fluid properties. The choice of these quantities still depends on empirical approaches because the entire flow and thermal fields in the cylinder, particularly in the boundary layer, cannot be fully interpreted by analytical methods.

The application of existing methods to modern diesel engines is questionable because key technologies found in current engines did not exist or were not widely used when those methods were developed. Examples of such technologies include high-pressure common rail and variable fuel injection strategies including retarded injection for nitrogen oxides emission control; etc. Thus, Finol and Robinson proposed a new heat transfer coefficient cor-relation to predict the gas-side heat transfer coefficient in modern diesel engines [9]. Their correlation is a simple relationship between the Nusselt and Reynolds numbers calibrated to predict the instantaneous spatially av-eraged heat transfer coefficient at several operating conditions using air as gas in the model. The results showed that the new correlation adequately predicted the instantaneous coefficient throughout the operating cycle of a high-speed diesel engine. They also showed that the complexity of the transient turbulent process during intake and exhaust, which cannot be properly described by the Woschni model. On the other hand when using the Hohenberg correlation, the results showed good agreement between predicted values of heat flux and measurements at various speed and load conditions. The investigation confirmed that in the case of high-speed diesel engines, Woschni's correlation underestimates the heat flux during the compression and expansion phases and overestimates the maximum value of the heat flux induced by combustion. Results presented by Hohenberg's corre-lation revealed improvements in those respects as shown in [10].

Several papers have been published on gas to wall heat transfer process suggesting some correlations to calculate the heat transfer coefficient in internal combustion engines [3–8]. In this study, two equations of Woschni and Hohenberg were used to predict the spatially-averaged gas-side instantaneous heat transfer coefficient to evaluate the temperature distribution of the cylinder liner when converting a naturally aspirated old engine into a turbocharged engine by using a supercharging system with exhaust gas energy recovery [11]. Adding the turbocharger to the engine does not increase its capacity but the power.

2 Methodology

2.1 Heat transfer correlations

In general, the engine cylinder heat transfer is a set of complex processes that involves turbulent flow, time and spatial temperature differences, che-mically reacting flow, and other unknown properties. The general form of the cylinder overall heat transfer rate is [1]

$$\dot{Q} = \text{htm}\,\alpha\,A(T - T_{\text{wall}}),$$

where α is the convective heat transfer coefficient, A is the surface area, and $T - T_{\text{wall}}$ is the temperature difference, and htm is a 'heat transfer multiplier'.

The heat transfer coefficient is generally given as a function of a Reynolds number [1]

$$\text{Nu} = \frac{h_c L}{k_g} = a\text{Re}^b = a\left(\frac{VL}{v}\right)^b,$$

where Nu is the Nusselt number, k_g is the thermal conductivity of the gas, Re is the Reynolds number, a and b are parameters selected to provide agreement with the experimental values, L and V are the characteristic length and velocity, respectively, and v is the kinematic viscosity. The se-lected correlations are Woschni's correlation and Hohenberg's correlation.

Woschni's correlation is as follows [6]:

$$\alpha = 129.9B^{-0.2}p^{0.8}T_g^{-0.53}\left(C_1 V_p + C_2 \frac{T_g V_s}{p_r V_r}(p - p_m)\right)^{0.8},$$

where B is the cylinder bore, p and T_g are the pressure and temperature of gas, and correspond to instantaneous in-cylinder conditions, p_r is known working value corresponding to the volume V_r, of a reference state such as inlet valve closure or beginning of combustion, V_p is the mean piston speed, V_s is the swept volume, and p_m is the motoring pressure. The constants C_1 and C_2 are necessary to consider changes in gas velocity over the engine cycle, are as follows:

$$C_1 = 6.18 + 0.417\frac{C_u}{V_p} \quad \text{for the gas exchange process,}$$

$$C_1 = 2.28 + 0.308\frac{C_u}{V_p} \quad \text{for the rest of the cycle, with}$$

$$C_u = \frac{\pi B N_{an}}{60},$$

where C_u is the peripheral gas velocity and N_{an} is the swirl anemometer speed, which has been obtained at 0.7B below the cylinder head in a steady flow test.

Hohenberg's correlation based on experimental observations, was ob-tained after a detailed examination of Woschni's original formula. The mod-ified correlation is [7]

$$\alpha = C_1 V_i^{-0.06} p^{0.8} T_g^{-0.4} (V_p + C_2)^{0.8},$$

where p is the indicated gas pressure (in bar), T_g is the mean gas tempera-ture, $V_i = \frac{\pi}{6} d^3_{sp}$ is the instantaneous cylinder volume, and d_{sp} is the sphere diameter. Mean values of the constants C_1 and C_2 were given as:

$$C_1 = 130, \qquad C_2 = 1.4.$$

2.2 The simulation models

In this study, the old V-12 engine is equipped with a pair of turbocharg-ers to increase the capacity of the engine. The engine simulation models performed using the industry standard GT-Power Engine Simulation soft-ware [12] were assessed for reliability [13]. Based on this model, the author investigates the temperature and the convective heat transfer coefficient according to the two thermal transfer models of Woschni and Hohenberg, these are the two important parameters that serve as the basis for calcu-lating cylinder liner thermal distribution.

The original engine used in the investigation was a V-12 engine that consists of 12 cylinders where two banks of six cylinders are arranged in a V configuration around a common crankshaft. The V-12 engine was equipped on the tanks of Russia and Vietnam. The main specifications of the V-12 diesel engine are summarized in Table 1.

To calculate the cylinder temperature distribution of the V-12 engine, the author built a calculation program in the form of the script, using the commercial Ansys Parametric Design Language (APDL) [14] based on finite element method (FEM) written as jobname.txt, then run by Ansys soft-ware. To match the evaluation with the experimental results, the assump-tion that set that the heat-transfer process is quasi-steady. The boundary conditions of the heat exchange coefficient and temperature as input pa-rameters used to calculate the temperature field by finite element method can be referenced in [15].

Table 1: Specifications of a V-12 diesel engine under study [13].

Parameters	Symbol	Value	Unit
Number of cylinders	–	12	–
Engine type	V-12	Diesel, the V arrangement, the twelve cylinders are arranged in two banks of six, with a 60° angle between their axis	
The firing order of the cylinders	$1^L–6^R–5^L–2^R–3^L–4^R–6^L–1^R–2^L–5^R–4^L–3^{R*}$		
Compression ratio	ε	15 ± 0.5	–
Maximum power	–	387.4/2000	kW/rpm
Maximum torque	–	2256.3/1200	Nm/rpm
Intake valve open (crank angle ATDCF[a])	IVO	340	°
Intake valve close (crank angle BTDCF[b])	IVC	–132	°
Exhaust valve open (crank angle ATDCF)	EVO	132	°
Exhaust valve close (crank angle ATDCF)	EVC	380	°
Specific fuel consumption	$G_{e.\,min}$	0.265 ± 0.005	kg/kWh

[*]L – left, R – right;
[a]ATDCF – after top dead center firing;
[b]BTDCF – before top dead center firing.

The heat transfer coefficient and the temperature of the in-cylinder gases are determined when running a simulation model built by GT-Power software [13]. Based on the calculation results and by the method of graph integration, we can determine the values of heat transfer coefficient, α_i and the temperature, T_i, (i = 1, . . . , 12) corresponding to each region of the cylinder liner due to exposure to the in-cylinder gases, conduction, convec-tion as shown in Fig. 1.

To determine the temperature distribution of the cylinder liner by experiment, direct measurement methods have been used by applying mea-suring techniques based on arrays of thermocouples at the given locations in the cylinder liner. The temperature measurement positions of the eight characteristic points on the cylinder sleeve have been presented in Fig. 2. The connection diagram and wiring diagram in the temperature measure-ment system at the survey locations in the engine cylinder liner was given in [15].

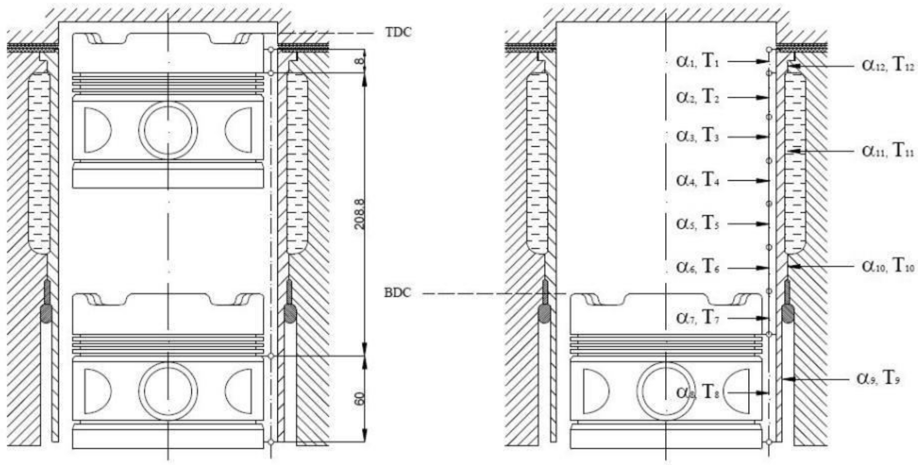

Figure 1: Surfaces of heat exchange of the wet cylinder liner [15]: TDC – top dead center, BDC – bottom dead center.

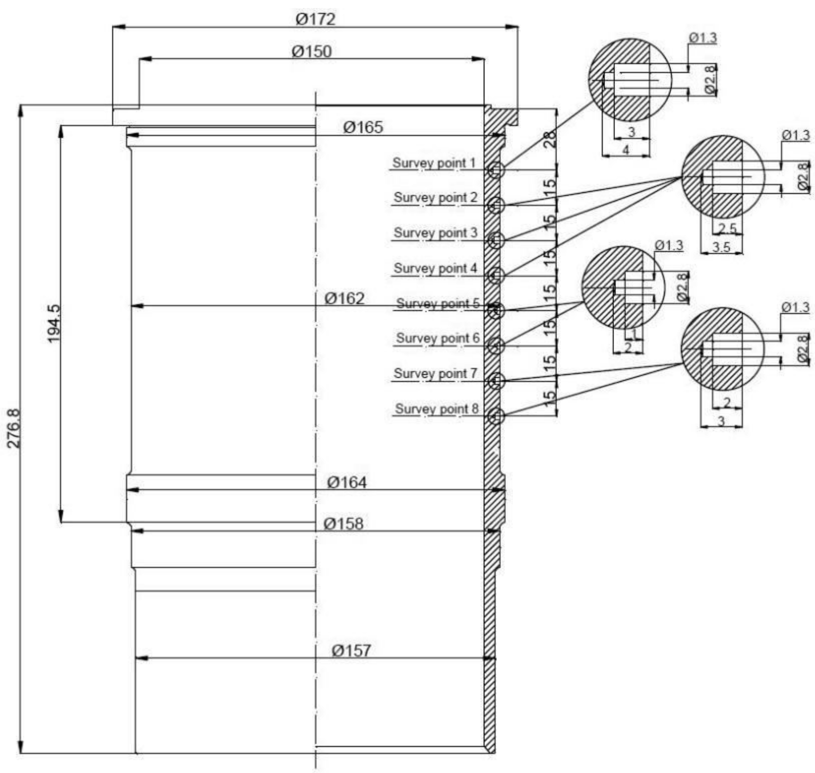

Figure 2: Schematic location of thermocouple sets in the wet cylinder liner [15].

3 Results and discussion

In the content of this paper, the cylinder liner distribution according to instantaneous spatially-averaged heat transfer coefficients given by Hohenberg and Woschni is presented and compared to the experiment. As shown in Fig. 3, in steady-state operating conditions the heat transfer correlations seem to have no substantial influence on the in-cylinder gas temperature.

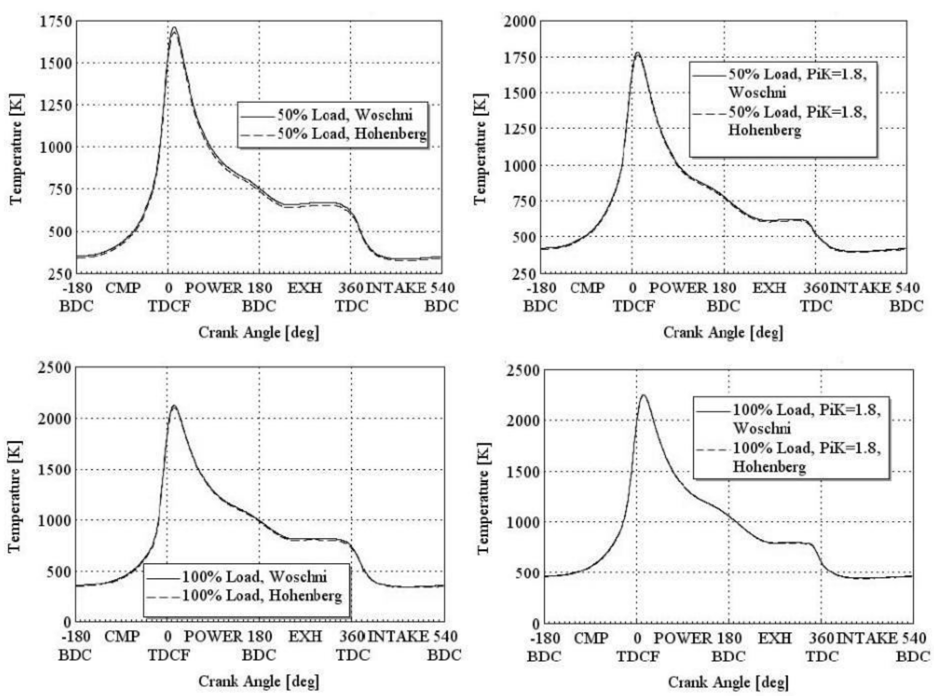

Figure 3: The in-cylinder gas temperature of an original engine and a turbocharged engine under the different loads and heat transfer correlations: BDC – bottom dead center, TDC – top dead center, TDCF – top dead center firing, CMP – compression process, EXH – exhaust process.

The heat transfer correlations to predict the gas-side heat transfer coefficient were evaluated at an engine speed of 2000 rpm and load conditions of 50% and 100% of the limiting torque according to the pressure ratio (PiK = 1.8). Fig. 4 shows the instantaneous heat transfer coefficient from Woschni and Hohenberg as a function of crank angle for the base case con-ditions. The coefficient increases rapidly during combustion to attain its highest value slightly after TDCF, and then it decreases during the expansion process. The high values during combustion are at least partly due to the high cylinder pressures and the high estimated cylinder gas veloc-ity. The Woschni's correlation predicted a peak heat transfer coefficient larger than the one given by the Hohenberg equation. Besides, the cor-relation by Woschni seems to underestimate the heat transfer coefficient during compression and expansion and overestimate the coefficient dur-ing the gas exchange process in the test engine concerning the Hohenberg correlation. In contrast, during compression and expansion, the estimates given by Woschni's equation were lower. However, this correlation seems to over-predict the heat transfer coefficient during combustion at surveyed conditions. These results are in agreement with [6, 7].

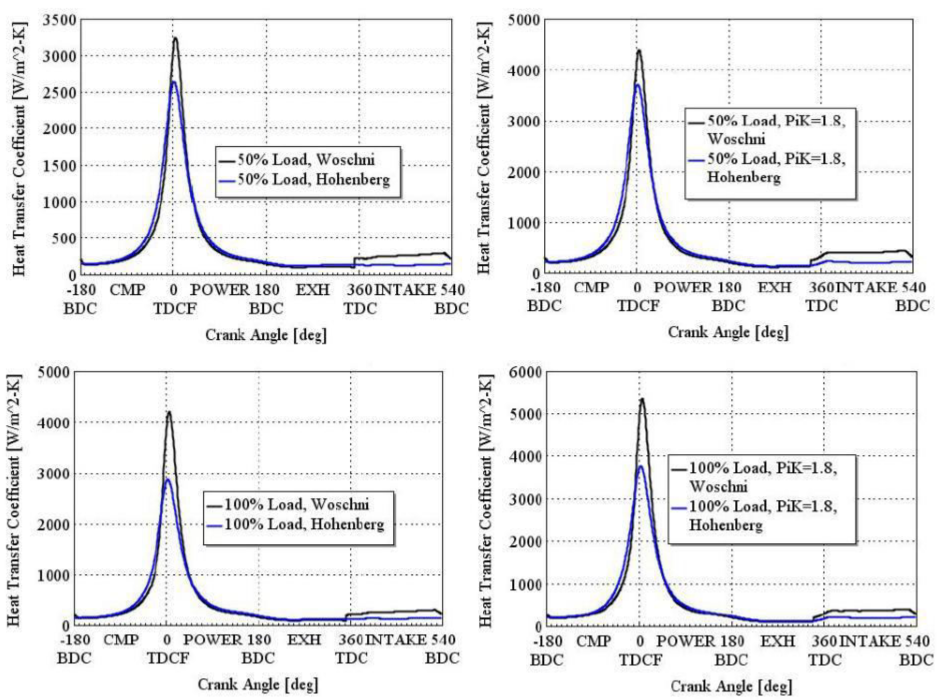

Figure 4: The instantaneous heat transfer coefficient in an original engine and a tur-bocharged engine under the different loads and heat transfer correlations.

The calculated results show how the temperature varies along the length of a V-12 diesel engine liner as shown in Figs. 5 and 6. From these results, the highest temperature area is the top inside edge of the cylinder liner. The temperature distribution decreases gradually from the mirror surface to the back of the liner, and the temperature significantly decreases with distance from the cylinder head. These results agree well with the results found in existing literature because this surface that forms the volume of the combustion chamber is exposed to high-temperature in-cylinder burned gases [16]. The lower regions of the liner are only exposed to combustion products for a part of the cycle after the substantial gas expansion has occurred.

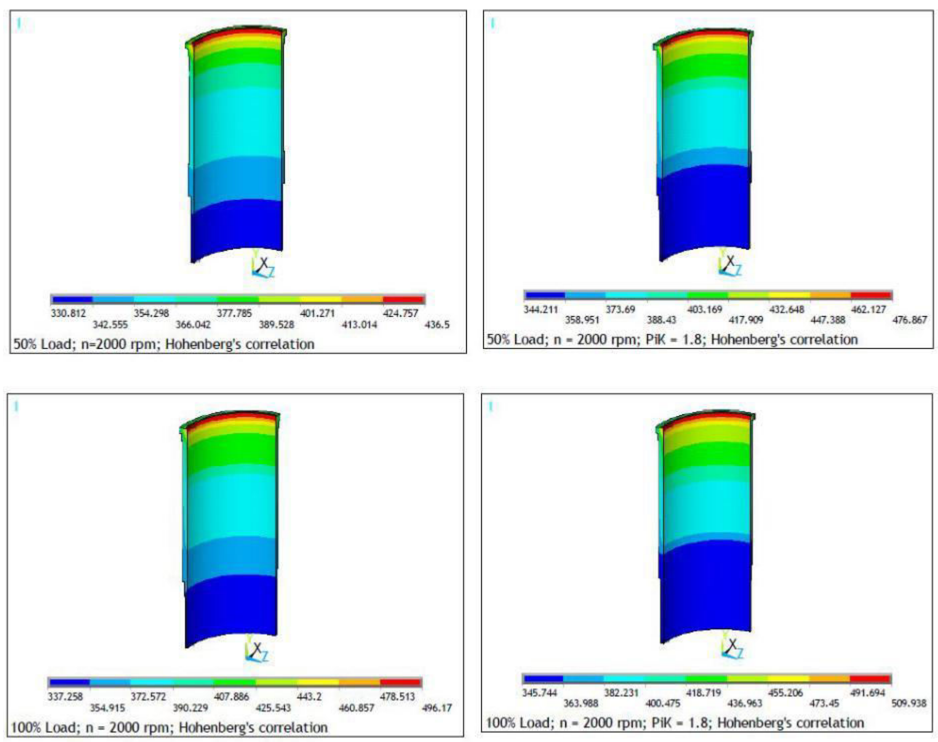

Figure 5: The cylinder temperature distribution in Kelvins, using Hohenberg's correlation in an original engine and a turbocharged engine under different loads.

Measured and calculated temperature distributions of the cylinder liner in an original engine and a turbocharged engine under different loads and comparison with experimental results at eight survey points of the V-12 engine cylinder liner are presented in Figs. 7 and 8, respectively. In terms of the difference between each of the calculations and the experimental values, the correlation by Hohenberg was the most consistent over the investigated conditions, especially in the turbocharged engine.

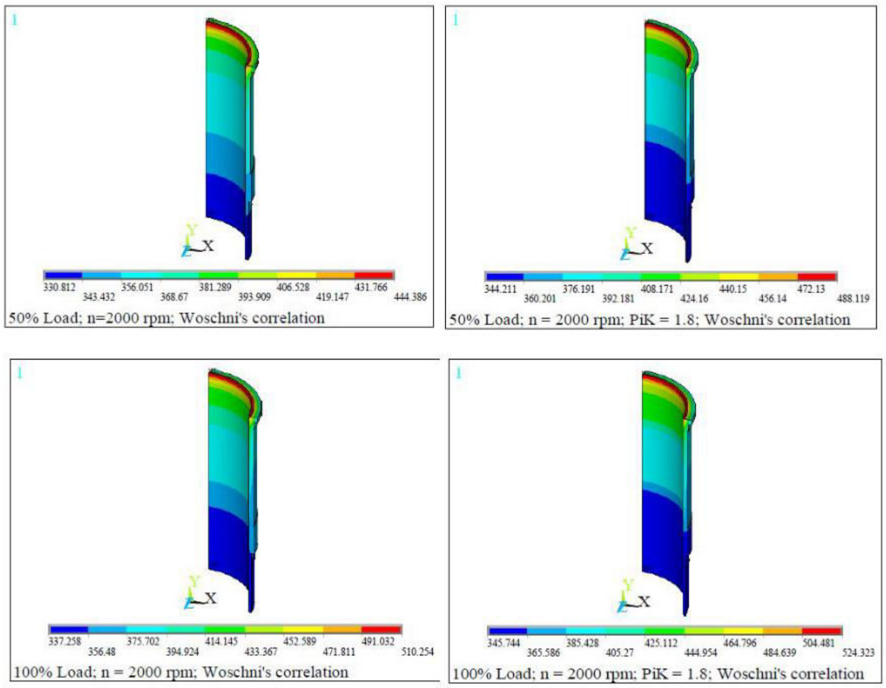

Figure 6: The cylinder temperature distribution in Kelvins, using Woschni's correlation in an old engine and a turbocharged engine under different loads.

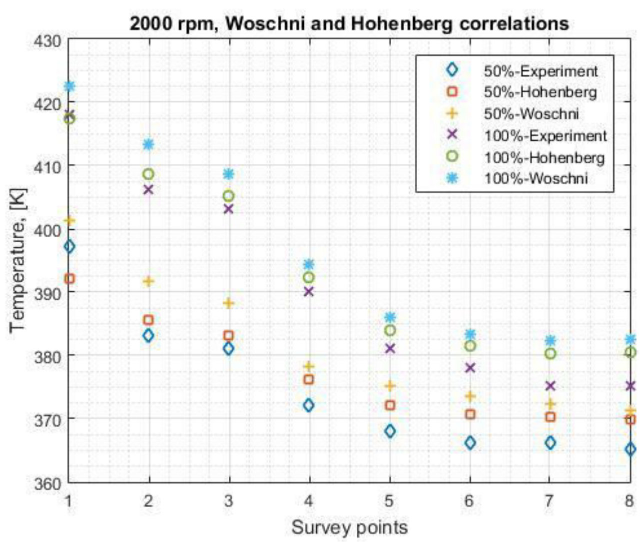

Figure 7: Measured and calculated temperature distributions of the cylinder liner in an original engine under different loads.

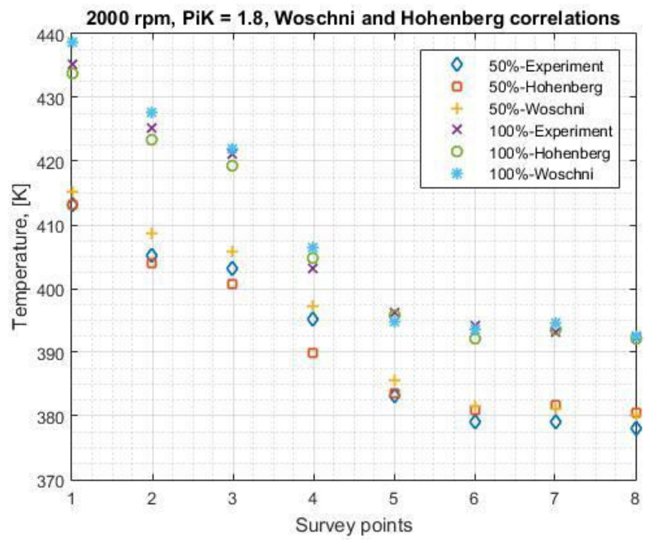

Figure 8: Measured and calculated temperature distributions of the cylinder liner in a turbocharged engine under different loads.

4 Conclusions

The study completed a comparison of the use to two typical existing global heat transfer correlations to calculate the cylinder liner distribution of an old heavy-duty diesel engine when converting into a turbocharged diesel engine. The results show that the Hohenberg heat transfer correlation was the most consistent over the investigated conditions, especially in a turbocharged engine. Therefore, this is the basis for accurately calculating the thermo-mechanical stress of the cylinder when supercharging the engine.

Received 25 May 2021

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