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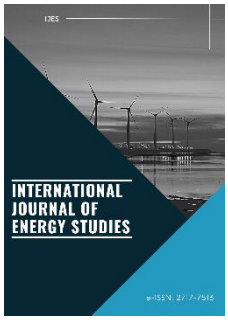

Research Article

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Investigation of conversion of a diesel engine to homogeneous charge compression ignition engine using n-heptane: A zero-dimensional modeling

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Highlights

- Model verification of the engine with known specifications.
- Completing the model by determining combustion profiles using experimental pressure data.
- Comparing the AVL Boost model output to experimental data.
- Completing numerical simulation of HCCI combustion with a zero-dimensional model.
- Investigating the compression ratio and inlet temperature effect for HCCI combustion

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ABSTRACT

Numerical methods are commonly used for analyzing combustion characteristics. Besides, they provide an opportunity to predict emissions of an engine. In this paper, the conversion of a single-cylinder diesel engine to HCCI combustion model is investigated with AVL Boost v2021R2. The model was used to simulate the power, emission, bsfc value. In-cylinder pressure results also are obtained from the model. Verification of the model is conducted with experimental data from the literature. The differentiation between the numerical and experimental results remained below 8.9% for the power, below 6.62% for bsfc. In addition to this, the model gave maximum pressure values with an accuracy of ± 1%, and maximum HRR values with an accuracy of ± 2%. Maximum HRR values and in-cylinder pressure curves for 1200-2400 rpm were obtained with acceptable accuracy. Besides, the operating range of an HCCI engine fueled with n-heptane was investigated using a zero-dimensional single-zone model with reduced fuel chemistry. The compression ratio and inlet air temperature effect on HCCI combustion were analyzed. The increasing the air inlet temperature to 40°C from 20°C, increases the lowest air-fuel ratio about 6.6% which the engine can operate without entering the knock zone, at 17.5 CR at 1200rpm.

Keywords: Diesel engine, AVL Boost, Numerical simulation, HCCI

1. INTRODUCTION

Energy is one of the most important requirements of the world. Besides that, the energy requirement for transportation rises enormously because of the significant increase in the number of vehicles. Because of this, experimental and numerical studies are conducted to improve performance and decrease the emission of internal combustion engines (ICE). Experimental methods are generally more expensive and slower than numerical methods [1]. As a result, numerical methods are widely used for internal combustion engine studies. The use of 0/1 dimensional thermodynamic algorithms have greatly aided the development of internal combustion engines. These programs use optimized mathematical models to reproduce the thermodynamic processes occurring within the engine, allowing engineers to evaluate and improve engine efficiency. This method captures key elements of complex combustion and energy transfer phenomena by considering the engine as a collection of connected 1D volumes. These programs were used, for example, to study cylinder pressure changes, heat transfer, and combustion efficiency in engines. They provided insightful information on adjustments to valve timing, ignition timing, and fuel injection technology that resulted in increased engine efficiency and reduced emissions [2] . In addition to these advantages, numerical methods can provide further information about in-cylinder processes. In the literature, several studies are carried out on internal combustion engines through various software programs. Alqahtani et. al conducted a study to compare simulation software programs and their advantages. One -dimensional engine and gas dynamics simulation software is relatively cheaper tools used for studies of the engine. It is indicated that AVL Boost software can use for complicated and advanced engine design and cycle simulations, even though it is a more difficult program than other one-dimensional software in the study [3]. Aktas et al. conducted a numerical study to research the effect of ethanol, methanol into diesel fuels in a diesel engine with AVL Boost software. According to their study, ethanol, and methanol addition to diesel causes a decrease in power compared to pure diesel. The NO_X reduction rate is between %25 and %30 [4]. Iclodean and Burnete conducted a study to optimize the combustion parameters of AVL MCC combustion model by computer modeling. Their study indicated that determining the combustion parameters significantly influences the results of simulations [5]. Jukneleviius et al. conducted a study in which they used a hydrogen-diesel mixture in a CI engine, compared the performance and emission characteristics of diesel. The main limitations of hydrogen-diesel mixture usage were analyzed with the validated model of AVL Boost software. AVL Boost simulation revealed the influence of the H_2 flow rate on the

autoignition delay and combustion duration. The simulation verified that autoignition delay reduced and combustion duration increased with a rise in H_2 flow rate [6].

The primary purpose of the studies carried out on combustion is to reduce emissions and increase efficiency and performance. There are many studies on both spark-ignition combustion and diesel combustion. Homogeneous charged compression ignition combines the characteristics of both combustion and has the potential to rise efficiency and drop emissions. Homogenous charge compression ignition (HCCI) exists when air and fuel fill in the cylinder as a homogenous mixture and self-ignite due to rising temperatures during compression. HCCI combustion has the characteristics of both the use of a homogeneous mixture for combustion and self-ignition [7]. HCCI engines are predicted to be more efficient than SI engines cause of the higher compression ratio and the elimination of throttle throttling losses. The presence of many ignition sites in the HCCI engines combustion chamber prevents the production of high-temperature zones, the formation of soot particles, and the emission of high NOx and particulate matter [8]. When literature studies are examined, it is seen that HCCI has remarkable potential in terms of fuel economy, efficiency, and emissions [9].

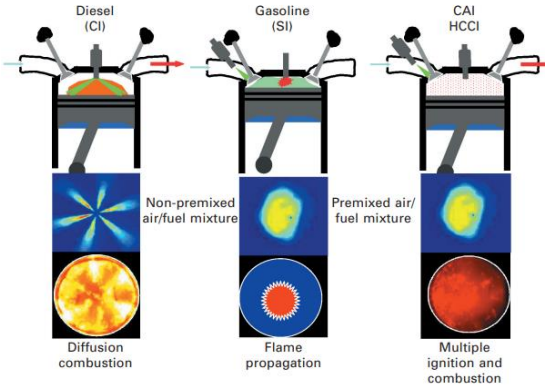


Figure 1. Three combustion modes [7]

The region of operation is quite narrow for stable combustion than conventional combustion for HCCI engines. Above the high load limit, the charge ignites through the whole combustion chamber. For this reason, it increases the pressure rise rate, noise, and vibration. The limits of knocking zones at high loads must be determined to specify the appropriate operating range [10, 11]. The low exhaust temperature and high AFR at the low load limit can cause a misfire and

increase the ignition delay [12]. Ignition delay and HRR can't be controlled directly for stable HCCI combustion as a result, the operating range of these engines is considerably narrow.

Dimensionless and single-zone combustion modeling is the simplest model used and creates an ideal HCCI combustion. The Wiebe function, which is used in engine research because it is a simple combustion model and has high numerical competence, is also used as a combustion function in HCCI combustion modeling [13] In dimensionless and single-domain modeling, all features are assumed to be uniform inside the cylinder and some relations are used to determine the heat transfer [14, 15]. The equation to calculate of heat transfer constants by Woschni is one of the equations widely used in combustion modeling [16]. Dimensionless thermo-kinetic models form the basis of detailed mechanism studies of HCCI combustion.

Dimensionless modeling studies have been used to examine the HCCI combustion properties of different fuels. These fuels are generally gasoline, diesel, n-heptane, n-heptane/isooctane /toluene mixtures, dimethyl ether, ethanol, natural gas, and many other fuels [17, 18].

Some of the automotive companies are working on HCCI engines. The Skyactiv-X SPCCI engine currently available can be specified as a commercial engine that is most similar to the HCCI engine. HCCI engines are one of the important research topics of automotive companies due to their potential [19, 20].

In this paper, the conversion of a single-cylinder diesel engine to an HCCI combustion model is investigated with AVL Boost v2021R2. The simulation results are compared with the announced experimental data for diesel combustion and model were validated. Then, the operation region of an engine for HCCI combustion with n-heptane was investigated. A 0-D single-zone numerical model was used for investigation. Converting diesel engines to HCCI with a high compression ratio may result in knocking. It is crucial to identify the optimal compression ratio and AFR range for ensuring a successful conversion process. This study aims to establish the operational range for a converted engine and assess the impact of intake air inlet temperature on HCCI combustion. The findings will serve as a foundation for future experimental studies.

2. AVL BOOST ENGINE SIMULATION MODEL

AVL Boost program offers approximate one-dimensional solutions to problems that are modeled by using gas dynamics equations. AVL Boost program makes use of basic conservation equations, thermodynamic and heat transfer models while solving the defined conditions. Woschni's 1978 heat transfer model and AVL MCC combustion model are employed for diesel combustion. Single-Zone HCCI Auto-Ignition model was utilized for HCCI combustion. The combustion and heat transfer models are briefly summarized below using the AVL Boost theory and user manuals [21, 22].

2.1. Heat Transfer Model

Heat transfer to the walls of combustion chamber is determined with following equation. The combustion chamber walls are piston, cylinder liner and head.

$$Q_{wi} = A_i . \alpha_w . (T_c - T_{wi}) \quad (1)$$

α_w: heat transfer coefficient.
T_c: gas temperature in-cylinder
Q_{wi}: wall heat flow
T_{wi}: wall temperature
A_i: surface area,

The axial temperature alteration between TDC and BDC position is considered when calculating the liner wall temperature:

$$T_L = T_{L,TDC} . \frac{1-e^{-c.x}}{x.c} \quad (2)$$

$$c = ln\left(\frac{T_{L,TDC}}{T_{L,BDC}}\right) \quad (3)$$

$T_{L,BDC}$: liner temperature at BDC
T_L: liner temperature
x: relative stroke (actual piston position related to full stroke)
$T_{L,TDC}$: liner temperature at TDC

The heat transfer coefficients were determined with the Woschni model (1978) [16]. This model was released for the high-pressure cycle.

$$\alpha_w = 130 . D^{-0.2} . p_c^{0.8} T_c^{-0.53} . \left[C_1 . c_m + C_2 . \frac{V_D . T_{C,1}}{p_{c,1} . V_{c,1}} \left(p_c - p_{c,0}\right)\right]^{0.8} \quad (4)$$

D: bore
$V_{D:}$ cylinder displacement
c_m: mean piston speed
$c_{u:}$ circumferential velocity
$T_{c,1}$: in-cylinder temperature at IVC
$p_{c,0}$: cylinder pressure of the motored engine (bar)
$p_{c,1:}$ in-cylinder pressure at IVC (bar)

$$C_1 = 2.28 + 0.308\frac{c_u}{c_m} \quad (5)$$

$$C_2 = 0.00324 \text{ (DI engine)} \quad (6)$$

2.2. Combustion Models

AVL Boost program combustion models are defined as predefined heat release, calculated heat release, predefined pressure curve (analysis), ideal heat release, and user-defined heat release. These combustion models have their sub-models. Mixing Controlled Combustion (MCC) and Single-Zone HCCI Auto-Ignition are two of the sub-models of calculated heat release models. MCC model is used for diesel combustion.

2.2.1. Combustion model of diesel combustion

Estimation of combustion properties for direct injection CI engines can be performed using Mixed Controlled Combustion (MCC) model. The influence of both premixed (PMC) and diffusion (MCC) controlled combustion processes is considered in the model.

$$\frac{dQ_{total}}{d_\alpha} = \frac{dQ_{MCC}}{d_\alpha} + \frac{dQ_{PMC}}{d_\alpha} \quad (7)$$

The fuel quantity available and turbulent kinetic energy density is respectively defined as (f1) and (f2). The heat release is determined with the function of (f1), (f2). f_1 and f_2 functions can be expressed as follows.

$$\frac{dQ_{MCC}}{d\alpha} = C_{Comb}.f_1(m_F, Q_{MCC}).f_2(k, V) \quad (8)$$

$$f_1(m_F, Q) = \left(m_F - \frac{Q_{MCC}}{LCV}\right).\left(w_{oxygen,avaliable}\right)^{C_{EGR}} \quad (9)$$

$$f_2(k, V) = C_{Rate}.\frac{\sqrt{k}}{\sqrt[3]{V}} \quad (10)$$

Q_{MCC}: cumulative heat release (kJ)
m_F: vaporized fuel mass (actual) (kg),
C_{Comb}: combustion constant (kJ/kg/deg CA)
C_{Rate}: mixing rate constant (s)
LCV: lower heating value (kJ/kg),
k: local density of turbulent kinetic energy (m^2/s^2)
V: volume (m^3)
$w_{Oxygen,available}$: mass fraction of available Oxygen at SOI (-)
α: crank angle (deg CA)
C_{EGR} : EGR influence constant (-)

The kinetic energy from the distribution of squish and swirl is significantly less than the kinetic energy input from the fuel injection and it is neglected. Therefore, the quantity of kinetic energy

delivered to the cylinder charge is calculated by the injection rate ($0.5.C_{turb}\dot{m}_F.v_F^2$). The dissipation is accepted in proportion to the kinetic energy ($C_{Diss}.E_{kin}^{1.5}$).

$$\frac{dE_{kin}}{dt} = 0.5.C_{turb}\dot{m}_F.v_F^2 - C_{Diss}.E_{kin}^{1.5} \quad (11)$$

$$k = \frac{E_{kin}}{m_{F,I}(1+\lambda_{Diff}m_{stoich})} \quad (12)$$

n: speed (rpm)
E_{kin}: kinetic jet energy (J)
v: injection velocity=$\frac{\dot{m}_F}{\rho_F.\mu A}$ (m/s)
C_{Turb}: turbulent energy production constant (-)
μA: effective nozzle hole area (m^2)
C_{Diss}: dissipation constant; [$J^{-0.5}/s$]
ρ_F: fuel density (kg/m^3)
m_{stoich}: stoichiometric mass of fresh charge (kg/kg)
$\dot{m}_{F,I}$: injected fuel mass (actual) (kg)
λ_{Diff}: air excess ratio for diffusion burning (-)
t: time (s)

The ignition delay is calculated with following differential equation in Andree and Pachernegg [23]model. When the ignition delay integral I_{id} arrives the value of 1.0 (=at α_{id}) at the ignition delay τ_{iD} is determined with the equation below.

$$\frac{dI_{id}}{d\alpha} = \frac{T_{UB}-T_{ref}}{f_{id}.Q_{ref}} \quad (13)$$

$$\tau_{id}=\alpha_{id}-\alpha_{SOI.} \quad (14)$$

Iid: ignition delay integral (-)
Tid ignition delay (s)
αid: ignition delay timing (degCA),
fid: ignition delay calibration factor (-).
Tref: reference temperature = 505.0 (K)
T_{UB}: unburned zone temperature (K)
Qref: reference activation energy (K)
αSOI: start of injection timing (degCA)

2.2.2. Single-zone HCCI auto-ignition

In this model,$\frac{dQ_f}{d\alpha}$ (fuel heat input) is expressed as follows.

$$\frac{dQ_f}{d\alpha} = \sum_{i=1}^{nSpcGas} u_i.MW_i.\dot{\omega}_i \quad (15)$$

The species mass fractions are determined as:

$$\rho\frac{d\omega_i}{d\alpha} = MW_i.\dot{\omega}_i \quad (16)$$

$nSpcGas$: number of species in the gas phase (-)
ρ: mixture density (kg/m^3)
MW: species molecular weight (kg/kmol)
ω : species mass fraction (-)
u : species inner energy (J/kgK)
$\dot{\omega}_i$:species reaction rate ($kmol/m^3s$).

The auto-ignition process define by chemical reactions and these chemical reactions are used to calculate the reaction rate of species $\dot{\omega}_i$. In this combustion model, a reduced chemical mechanism for C_7H_{16}(n-heptane) is used [24].

2.3. AVL Boost Engine Simulation Model Set Up

In this work, a 1-D model of a one-cylinder naturally aspirated diesel engine was made using the commercial software AVL Boost. Figure 2 shows all the modeled engine's components and connections. Connection points (1-4) that used for the relation between elements, plenums (PL1) exhaust section, measurement points(MP(1-4)) to get the required data (pressure, temperature, etc.), air filter (CL1) for cleaning the incoming air; cylinder (C1), system boundary condition definitions (SB1-SB2). The physical specification of the cylinder, the pressure at EVO and temperature, etc. combustion model selection, emission parameters, heat transfer model, heat transfer surface area information, intake and exhaust valve information are defined in submenus of cylinder. Combustion model is also specified in that section. In this study, AVL MCC combustion model needs the information about the injector and injection system (the number of holes and radius, rail pressure).

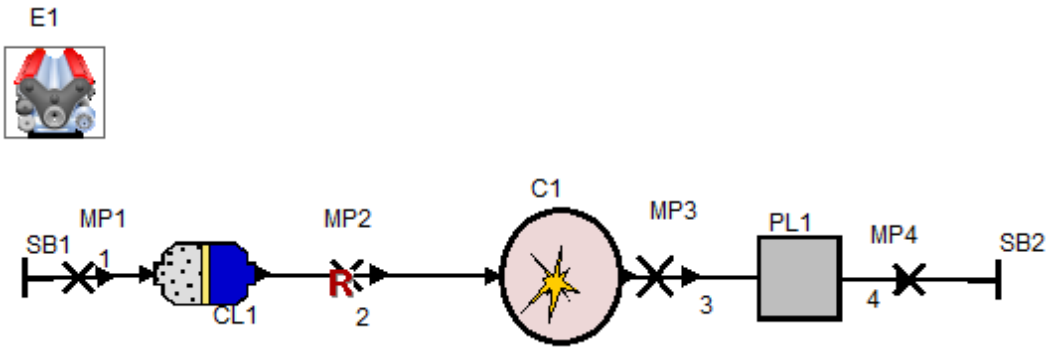


Figure 2. AVL Boost image of modeled one-cylinder diesel engine.

Table 1. Engine specifications [25]

Engine	3 LD 510 (ANTOR)
Type	4-stroke, Direct Injection
Cylinder	1
Volume	510 cm3
Bore	85 mm
Stroke	90 mm
Compression Ratio	17,5:1
Maximum speed	3000 rpm
Torque	32.8Nm @1800 rpm
Power	9 kW @3000 rpm
Con-rod length	145 mm
Number of injector holes	4
Injection pressure	190bar

3. VALIDATION OF MODEL FOR DIESEL COMBUSTION

The model was verified by using the experimental data taken from literature [26, 27] . The in-cylinder pressure at 1200-2400 rpm under full load, power, fuel consumption, and emission values at different speeds were simulated with AVL Boost model.

When Figure 3 is checked, it is noticed that the obtained power values are more than 91% compatible with the experimental results. While the maximum power value is determined as 7.56 kW at 2400rpm, the maximum difference is seen at 1800rpm with 8.9%. In addition, although the torque decreases with the effect of increasing mechanical and thermal losses after the maximum torque is obtained, it is seen that the power follows a constant path depending on the increase in speed. There is a harmony for the specific fuel consumption values of experimental and numerical. It is observed that the maximum difference is less than 6.5% at 1800 rpm.

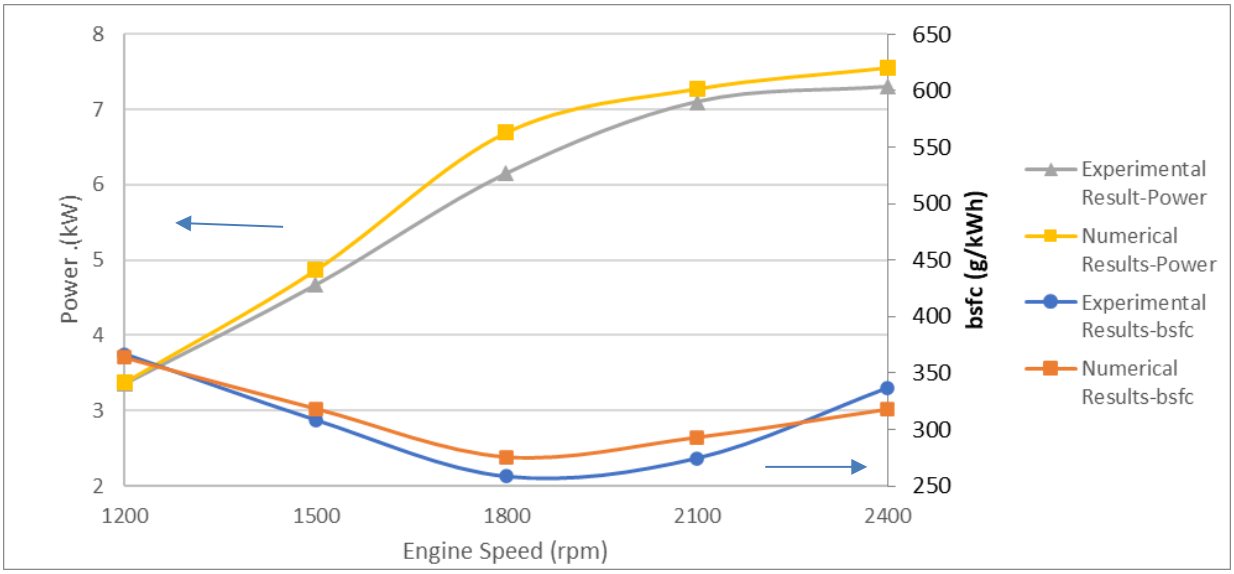
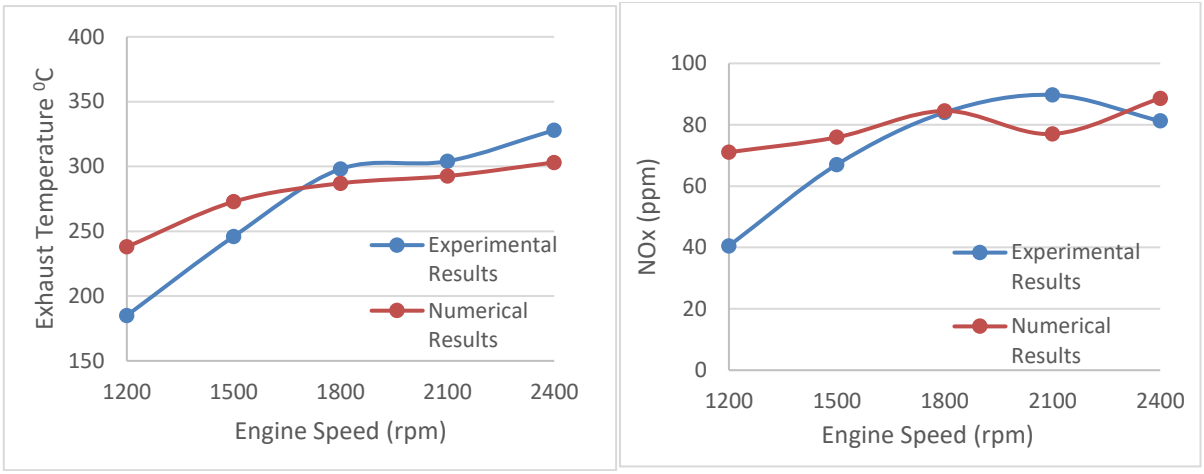


Figure 3. Power and specific fuel consumption validation graphs

Exhaust temperature results are shown in Figure 4 (a). The graph shows that there is a more difference than expected for the exhaust temperatures and NOx emissions at 1200 rpm. Although it is stated that the exhaust temperature was measured from the closest and thinnest walled place of the exhaust manifold to the combustion chamber in the reference study, the fact that it was measured with an infrared thermometer. This situation indicates that there may be errors in the experimental measurements. The absence of any uncertainty analysis in the relevant experimental study also shows that this difference between numerical study and experimental results at low speeds is understandable. The model results for NOx emission agree with the experimental results.

However, since it is a one-dimensional model study, it should be considered that the emission values are obtained by using the calibration factor. As it should be, a single calibration factor was used for all engine speeds. It should be taken in to account that one-dimensional numerical software program usually just predict trends. On the other hand, when the model results were examined, it can be understood that the maximum error rate is around 15% for 1500-2400 rpm.



(a) **(b)**

Figure 4. Exhaust a) gas temperature and b) NOx emissions

Maximum torque is occurred at 1800 rpm. Therefore, Figure 5 shows in-cylinder pressure and HRR results of 1800 rpm at first. There is a harmony between the experimental and numerical results in Figure. Maximum in-cylinder pressure occurred at the same CA in experimental and numerical results curves.

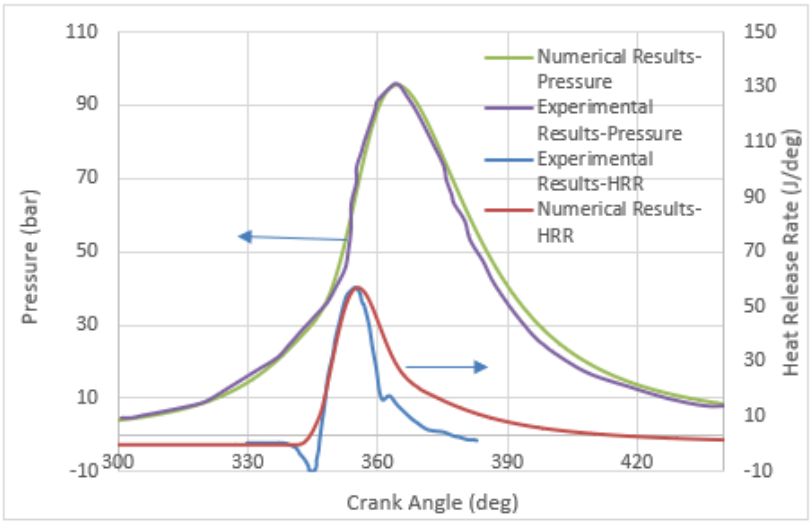
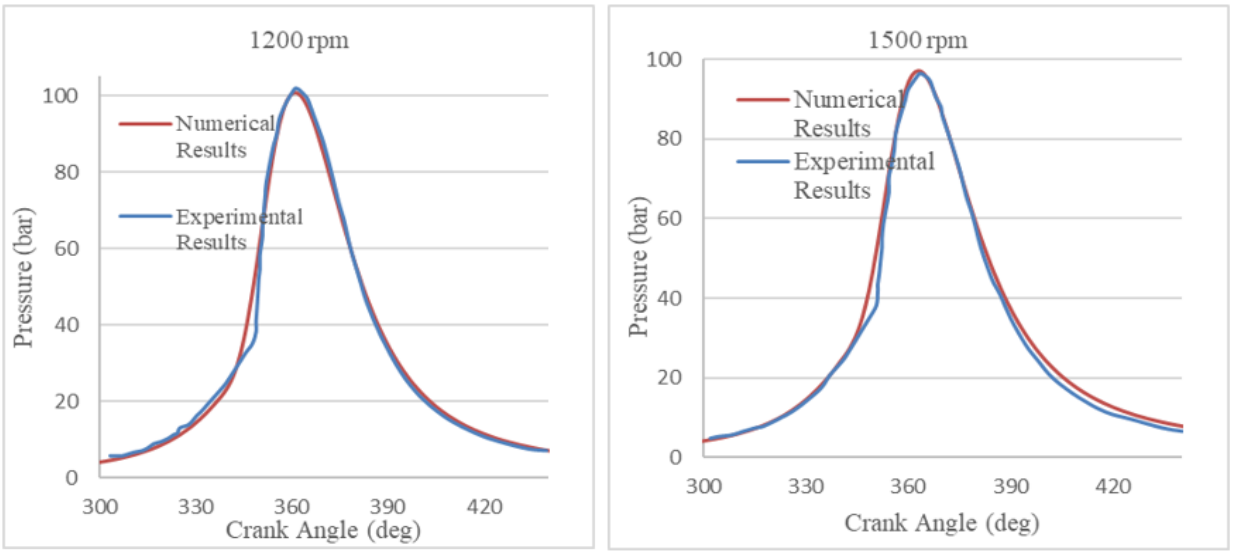


Figure 5. In-cylinder pressure and HRR at 1800 rpm

AVL MCC model parameters are used to optimize the pressure and HRR was modified to align with the experimental results. These parameters are ignition delay calibration factor, combustion, turbulence, dissipation, EGR influence and premixed combustion parameters. The first calibration factor affects ignition delay. The combustion parameter influences combustion duration and has the greatest effect on the ROHR shape. The turbulence parameter directs the effect of kinetic energy density. The dissipation parameter affects kinetic energy dissipation. The dissipation parameter controls the turbulence dissipation. The premixed combustion parameter determines the fraction of fuel that is injected during ignition delay and burns during premixed combustion [21, 22]. Ignition delay calibration factor, combustion, turbulence, dissipation, and premixed combustion parameters was defined respectively 0.57,0.87,0.87,1 and 0.7 for 1800 rpm. Heat transfer, surface area factors etc. are hold steady for other engine speeds to validate the model. AVL MCC combustion model parameters are used to optimize and change the pressure and HRR. The combustion parameters were modified to align pressure with the experimental results. Figure 6 and 7 shows the in-cylinder pressure at 1200-2400 rpm.



(a) **(b)**

Figure 6. In-cylinder pressure at a) 1200 and b) 1500rpm

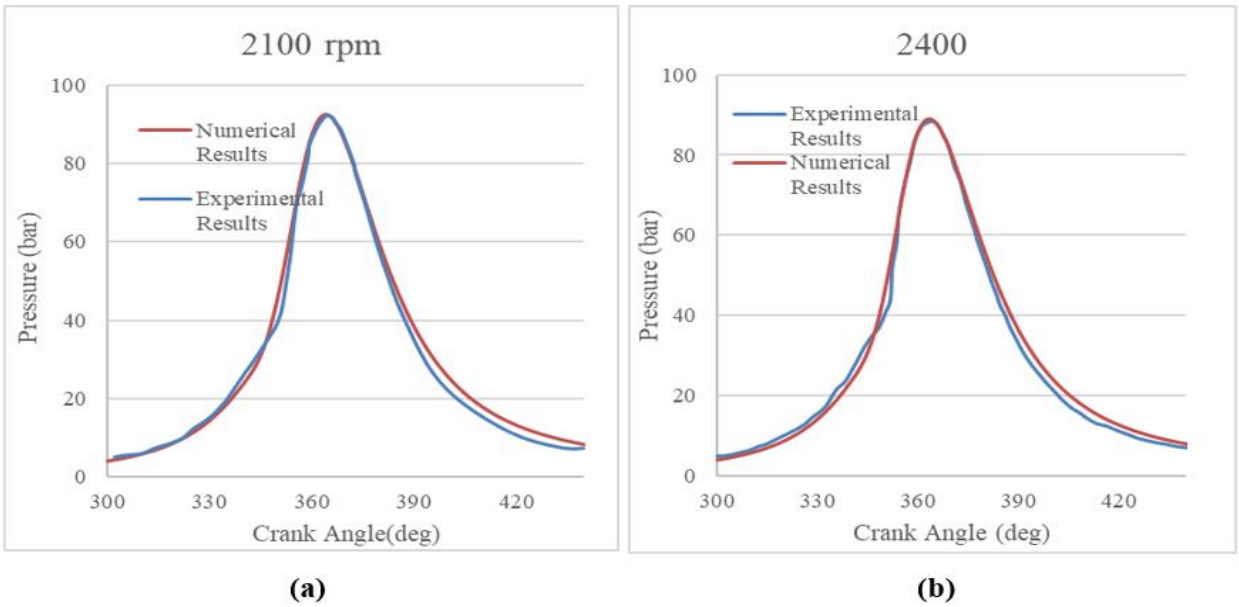


Figure 7. In-cylinder pressure at a) 2100 and b) 2400rpm

The model gave maximum pressure value with an accuracy of ± 2% HRR value with an accuracy of ± 1%. In Table 2, P_{max}(bar), HRR_{max} ($J/^0C$) and HRR_{max} (CAD) (bar) values are shown.

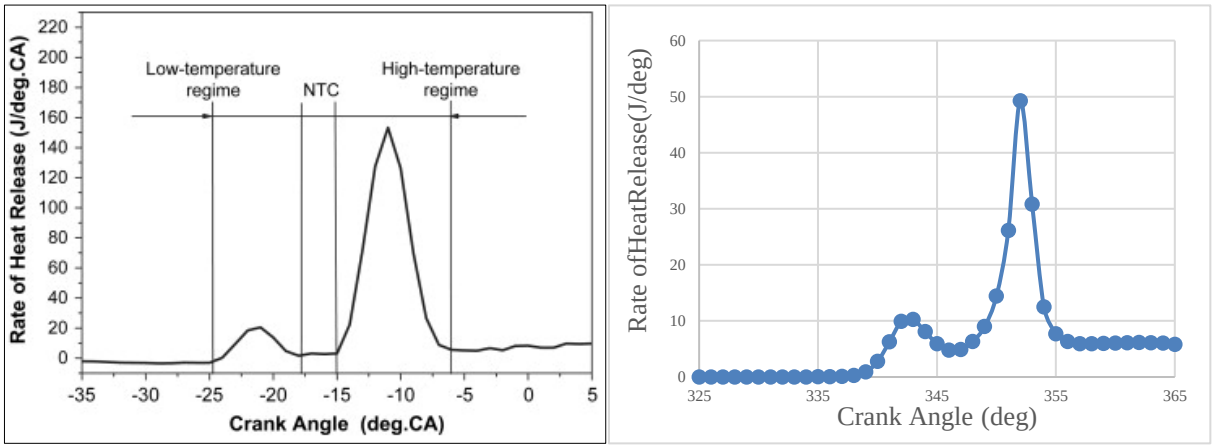
Table 2. Pressure and Heat Release Rate Data

		P_{max} (bar)	HRR_{max} ($J/^0C$)	HRR_{max} (CAD)
1200	Experimental	101.91	57.15	10 bTDC
	Numerical	100.9	57.8	10 bTDC
1500	Experimental	96.49	54.88	7 bTDC
	Numerical	97.15	54.72	7 bTDC
1800	Experimental	95.85	56.8	5 bTDC
	Numerical	95.41	56.68	5 bTDC
2100	Experimental	92.09	46.99	5 bTDC
	Numerical	92.56	47.11	5 bTDC
2400	Experimental	88.26	46.33	6 bTDC
	Numerical	88.86	46.49	6 bTDC

4. SINGLE-ZONE HCCI AUTO-IGNITION MODEL RESULTS

HCCI combustion exhibits distinctive two-phase heat release for diesel-like fuels. This distinctive heat release is shown in Figure 8 (a). The initial phase is related to low temperature kinetic reactions. There is a time delay between the initial and main heat release. This situation is related to the "negative temperature coefficient (NTC) regime" that lies between the two phases. Even though the in-cylinder temperature rises, the total reaction rate reduces in this regime. In this situation, the reactivity of regime gets lower [28]. Figure 8 (b) shows the heat release curve of the

combustion model for 1200 rpm, Pin=0.95 bar Tin=20ºC, AFR=45. In Figure 8 (b), the model achieves a two-stage heat release of HCCI combustion.



a) **b)**

Figure 8. Heat Release Curve a) Typical curve for HCCI combustion with n-heptane [28] b) obtained from the model for 1200 rpm, Pin=0.95 bar Tin=20ºC, AFR=45

Table 3 shows the characteristics of n-heptane fuel. Cetane index of n-heptane is quite similar to diesel, cetane index and its high reactivity make it a suitable choice for HCCI combustion.

Table 3. n-heptane properties [29]

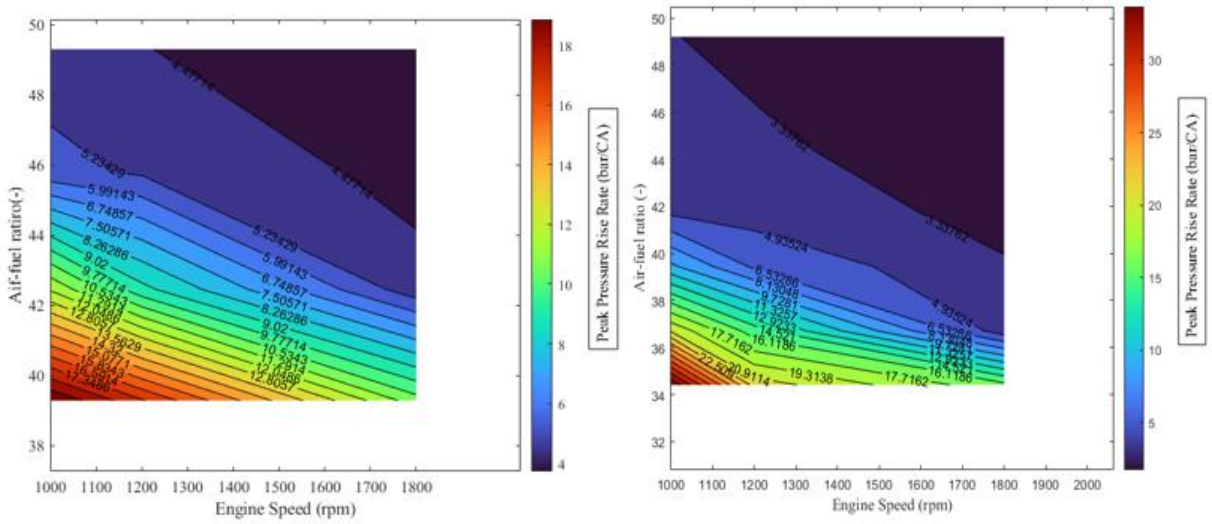
Chemical formula	C7H16
Density [kg/m3]	684
Boiling point [°C]	98
Auto-ignition temperature [°C]	223
Stoichiometric air/fuel ratio	15.1
Research octane number	0
Molar mass [g/mol]	100.2
Cetane number	56
Lower heating value [MJ/kg]	44.56
Heat Capacity (cp) [J/mol-K] (at 298.15 K)	224

4.1. Compression Ratio Effect on HCCI Operating Range

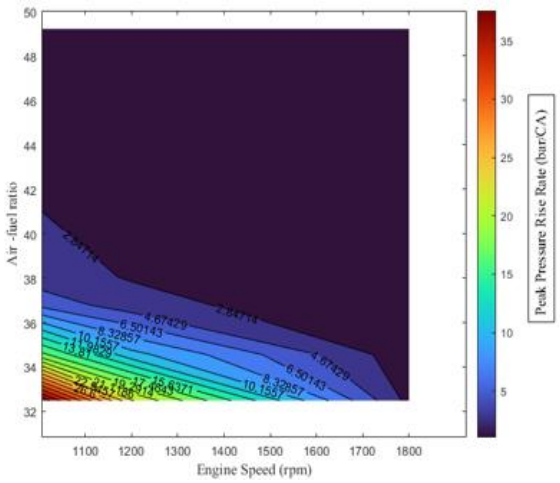
The modeling was conducted at engine speed of 1000, 1200 and 1800 rpm, at 20º C intake air temperature and 0.95 bar inlet pressure. In the study, the fuel was defined directly, and the results were obtained in the range of AFR value of 50-30 and compression ratio of 17.5-13. Knocking problem occurs in relatively rich mixture conditions, low air-fuel ratio. Therefore, peak pressure rise rate was examined and the knock limit value is accepted as the value where peak pressure rise

rate is 10bar/ºCA. According to the numerical results obtained, the reduction of the compression ratio provided that working at a lower air-fuel ratio without knocking. It was seen that it was possible to work at a compression ratio of 17.5,1800 rpm, 40 AFR without knocking and knock area decreased when the engine speed increased. Compression ratio effect on HCCI operating range and peak pressure rise rate for n-heptane fuel is shown in Figure 9. Reducing the compression ratio (CR) in HCCI engines reduces the gas temperature on compression stroke. The ignition delay time is extended and thus prevents unstable self-ignition. Therefore, reduction of CR reduces the knocking of the engine.

A reduction in the AFR means a higher fuel concentration in the combustion chamber. Therefore, increasing fuel concentration causes the ignition to occur much earlier than TDC, resulting in unstable combustion. As a result, lowering the air-fuel ratio causes it to approach the knock zone. In addition, the decrease in compression ratio causes the maximum IMEP to rise from 2.5 to 3.8 bar for 1000 rpm.

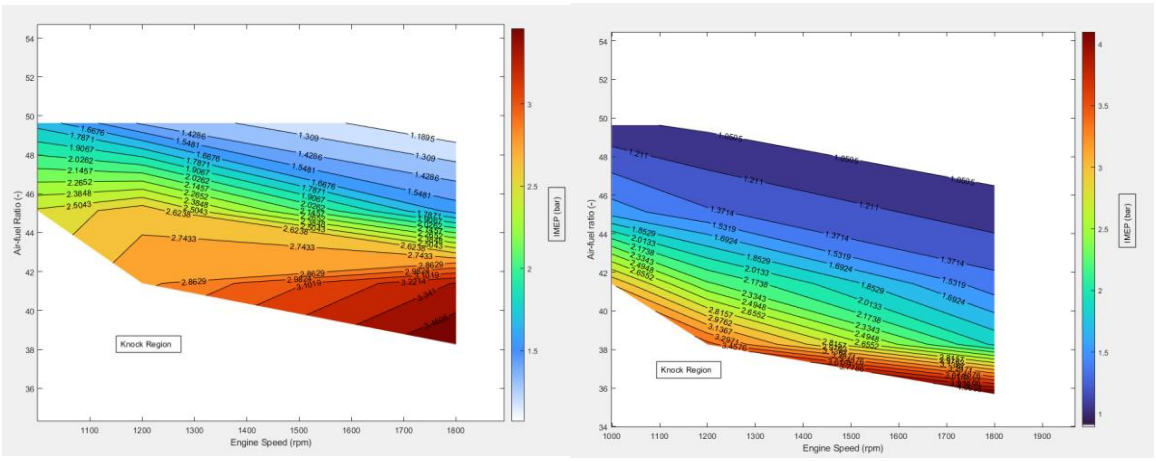


a) **b)**

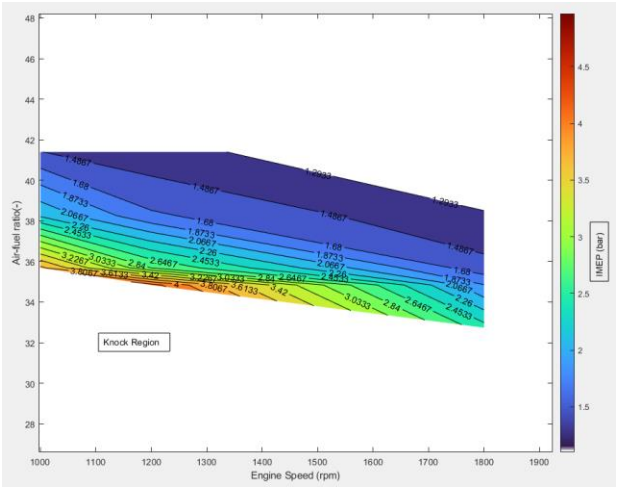


c)

Figure 9. Compression ratio effect on HCCI operating range and peak pressure rise rate (Tin=20ºC Pin=0.95 bar) a) 17.5 b) 15 c) 13



a) **b)**



c)

Figure 10. Compression ratio effect on HCCI operating range and IMEP (Tin=20ºC Pin=0.95 bar) a) 17.5 b) 15 c) 13

4.2. Inlet Air Temperature Effect on HCCI Operating Range

The experimental studies indicate that in-cylinder gas temperature rises due to the high intake air temperatures, and it causes acceleration in reactions [30]. Therefore, it was aimed to determine the AFR and compression ratio at which the engine can operate as HCCI without entering the knock zone for higher inlet temperatures.

When the obtained results are analyzed, it is understood that increasing the air inlet temperature to 40ºC increases the lowest air-fuel ratio value, which the engine can operate without entering the knock zone, from 42 to 45 at 17.5 CR at 1200rpm. At 80ºC inlet temperature, knocking occurs at all 50-30 AFR values for 17.5 compression ratio. At a CR of 15, there appears to be no knocking at only 50 AFR for 80ºC inlet temperature.

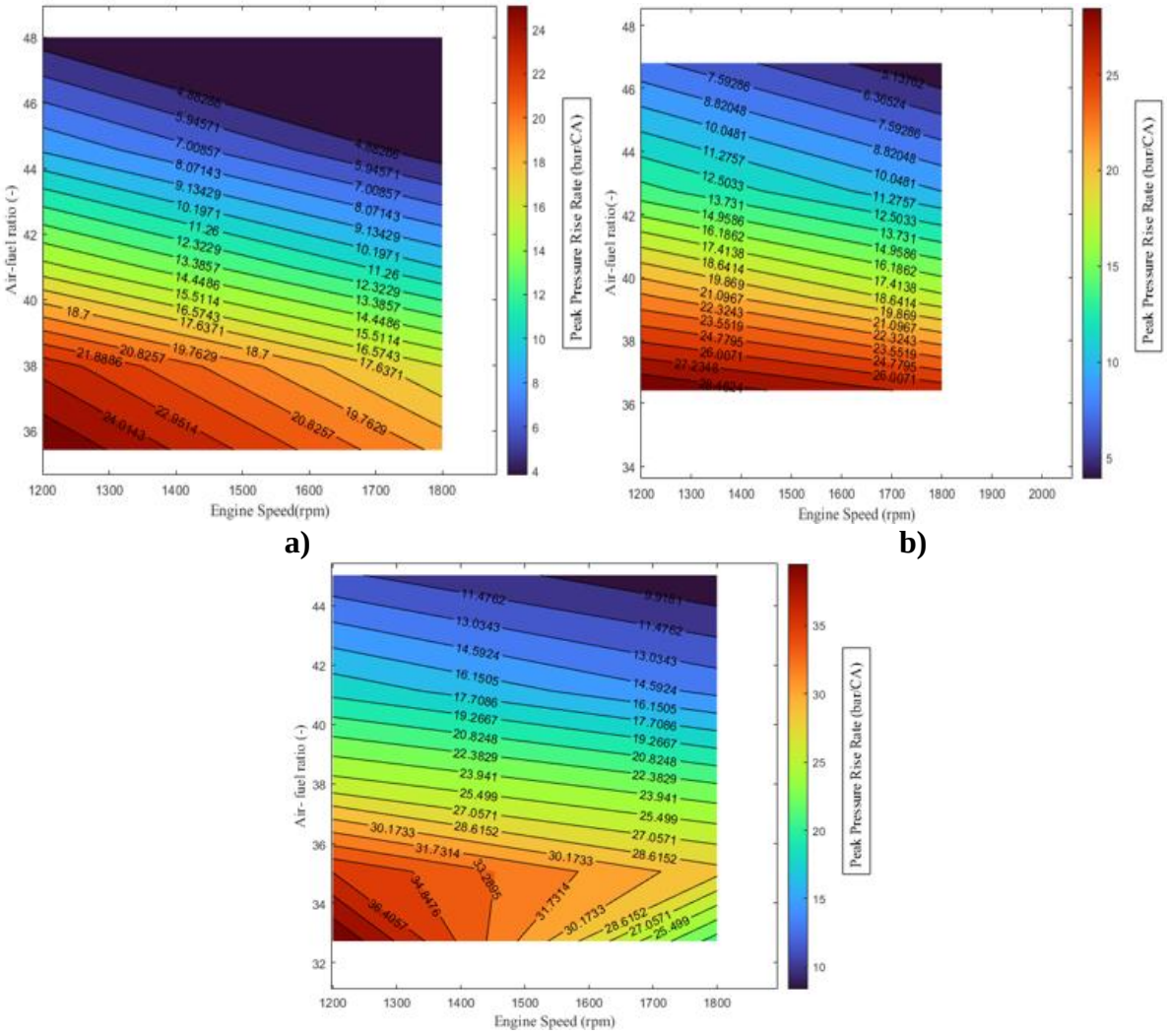


Figure 11. Inlet air temperature effect on HCCI operating range and peak pressure rise rate (Pin=0.95 bar CR=17.5) a) Tin=40ºC b) Tin=60ºC c) 80 ºC

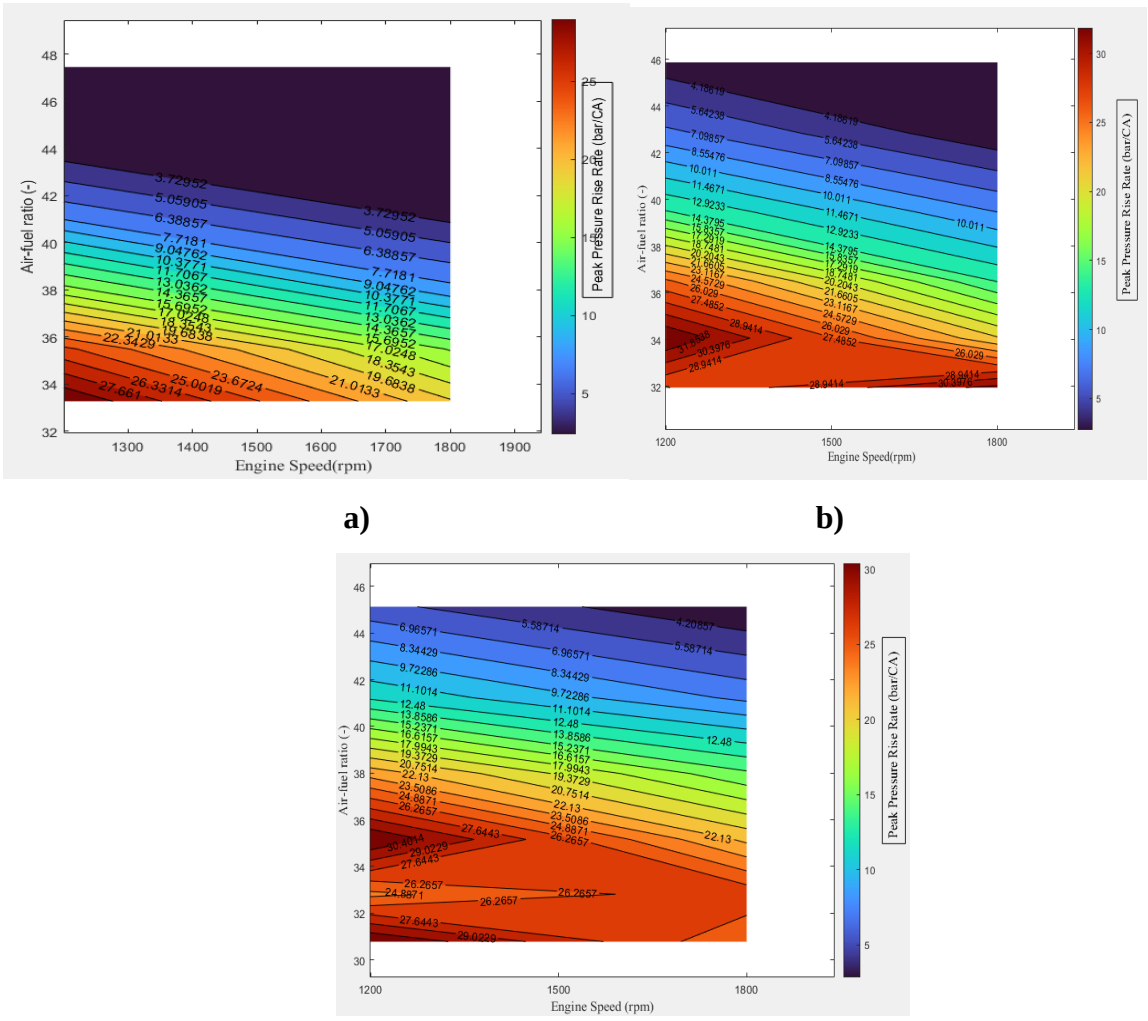


Figure 12. Inlet air temperature effect on HCCI operating range and peak pressure rise rate (Pin=0.95 bar CR=15) a) Tin=40^{o}C b) Tin=60^{o}C c) 80 ^{o}C

By increasing the inlet air temperature, the IMEP values for the relatively low AFR values have increased. From the data shown in Figure 12, it can be observed that raising the inlet temperature from 20^{o}C to 40^{o}C caused an increase in the IMEP value from 2.85 bar to 3 bar for 42 AFR at 1000 rpm.

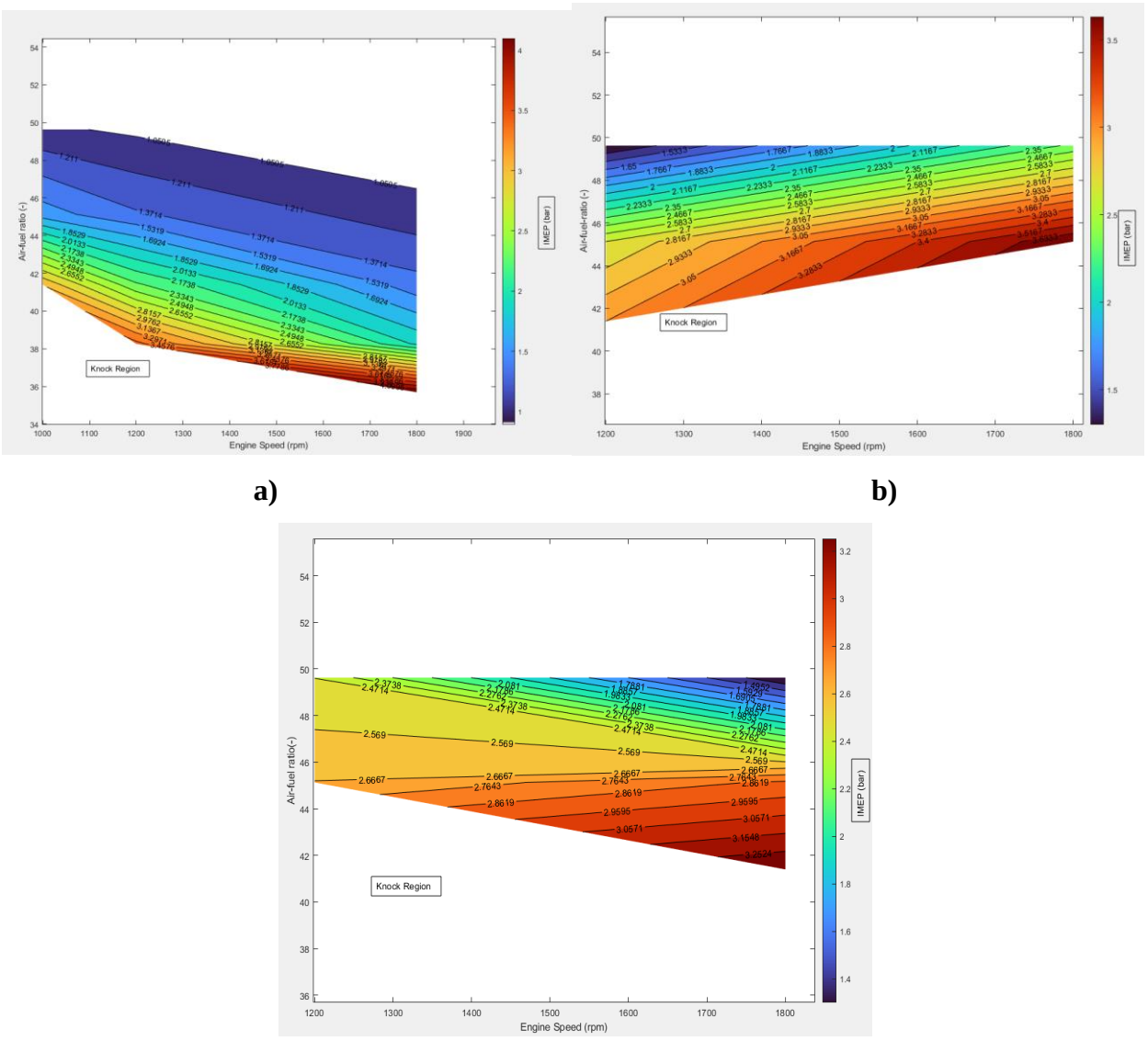


Figure 13. Inlet air temperature effect on HCCI operating range and IMEP (Pin=0.95 bar CR=15) a) Tin=20^{o}C b) Tin=40^{o}C c) 60 ^{o}C

5. CONCLUSIONS

In this study, a 0/1-dimensional numerical simulation were performed for a naturally aspirated compression ignition engine. A zero-dimensional single-zone numerical simulation with reduced fuel chemistry was used to investigate the operating range for an HCCI engine fueled with n-heptane. The optimization of the combustion model parameters takes a time because of the combustion parameters significantly influence the results of the model. On the other hand, the result of this study shows that AVL MCC combustion model can provide an opportunity to simulate the compression ignition combustion engine performance with high accuracy.

A zero-dimensional single-zone numerical simulation shows that, HCCI operating range for n-heptane is quite narrow. When the n-heptane auto-ignition temperature and other fuels are compared, it can be said that n-heptane has the lowest auto-ignition temperature. The usage of n-heptane with high octane fuels can provide the operating range to be extended.

1. Numerical and experimental values differ by less than 6% for power and less than 10% for bsfc.
2. The model results for NOx emission agree with the experimental results, the deviation is about 15%.
3. The in-cylinder pressure curves for 1200-2400 rpm and maximum HRR values were obtained with acceptable accuracy.
4. The model achieves a two-stage heat release of HCCI combustion.
5. By reducing the compression ratio, it is possible to operate at a lower AFR without experiencing knocking for n-heptane.
6. The increasing inlet air temperature cause wider knock region.

NOMENCLATURE

aBDC	After bottom dead center
AFR	Air-fuel ratio
aTDC	After the top dead center
bBDC	Before the bottom dead center
bsfc	Brake specific fuel consumption
bTDC	Before the top dead center
CA	Crank angle
CI	Compression ignition
CR	Compression ratio
DI	Direct injection
EVO	Exhaust valve opening
HCCI	Homogeneous charge compression ignition
HRR	Heat release rate
IMEP	Indicated mean effective pressure
IVC	Inlet valve closing

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DECLARATION OF ETHICAL STANDARDS

The authors of the paper submitted declare that nothing which is necessary for achieving the paper requires ethical committee and/or legal-special permissions.

CONTRIBUTION OF THE AUTHORS

Gonca Kethüdaoğlu: Construction of the model, consideration and interpretation of the results, manuscript preparation, check of the paper template, and proofreading**.**

Fatih Aktaş: Construction of the model, consideration and interpretation of the results, manuscript preparation, check of the paper template, and proofreading.

Salih Karaaslan: Consideration and re-interpretation of the results, check of the paper template, and proofreading.

Seyfi Polat: Consideration and re-interpretation of the results, final check of the paper template, and proofreading

Nureddin Dinler: Consideration and re-interpretation of the results, final check of the paper template, and proofreading

CONFLICT OF INTEREST

There is no conflict of interest in this study.

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