Modelling the effect of intake valve closing timing on exhaust thermal management of a turbocharged and intercooled diesel engine

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Abstract

Emissions from diesel engines have become recently a significant problem due to their positive effect on global-warming. There are strict restrictions on emissions and low-emission diesel engines nowadays are required to be developed. One of the solution for reduced-emission diesel engines is to utilize exhaust thermal management systems. However; these systems work efficiently mostly at temperatures above 250 ^oC and for diesel engines, especially at low speed and low load conditions, exhaust gas temperatures are not generally higher than 250 ^oC. That not only leads to inefficient aftertreatment systems, but also insufficient emission reduction. Variable valve timing (VVT) can be used to achieve those high exhaust gas temperatures by changing the opening and closing timings of the intake&exhaust valves at any speed and any load. Therefore, the aim of this study is to try to increase the turbine exit temperature of a diesel engine above 250 ^oC at 2.50 bar brake mean effective pressure (bmep) and 1200 rpm engine speed condition by changing intake valve closing (IVC) timings. Diesel engine system is simulated by using Lotus Engine Simulation (LES) program. The model is then validated with experimental results. It is seen that exhaust gas temperatures can be raised higher than 250 ^oC for the studied particular engine loading case when IVC is advanced or retarded. The method results in fuel consumption saving in comparison to nominal valve timing by decreasing the required fuel injection rate for the constant engine loading. However, earlier and later closing of intake valve also causes exhaust flow rate to drop off.

Keywords: Diesel engines, exhaust thermal management, variable valve timing, turbine exit temperature.

1. Introduction

The internal combustion engines have a leading role in the world as a power plant for about a century. Thanks to advances in materials, manufacturing, computer analysis and design tools in the current 30 years; the internal combustion engines can attain higher standards of performance. Especially, strict legislations on emissions, fuel scarceness and social and economic factors are bringing new targets for current engine systems for the following years. Therefore, improved diesel engine models via computer simulations will be needed to supply those demands in the future.

Variable valve timing (VVT) is one of the solutions for achieving higher torque, brake power, volumetric efficiency (η_{vol}) and reduced brake specific fuel consumption (bsfc) on diesel engine systems. VVT concept has long been searched for gasoline and diesel engines and significant improvements are gained in these studies (Gray, 1988), (Dresner&Barkan, 1989), (Ahmad&Theobald, 1989), (Leonard et al., 1991), (Stone et al., 1995), (Ozsoysal et al., 1995), (Lancefield et al., 2000), (Parvate-Patil et al., 2004), (He et al., 2008), (Deng&Stobart, 2009) and (Tomoda et al., 2010). However, in this study, particularly effect of VVT on exhaust thermal management is examined. Some of the previous studies related with this subject are briefly explained in the following paragraphs.

Honardar et al. (2011) investigate the effects of different ways (exhaust valve timing phase, main injection variation, post injection variation and throttle valve variation) to improve the exhaust temperature management on emissions, bsfc rise and external EGR requirements on diesel engines. It is asserted in the study that when exhaust valve phasing is advanced, exhaust temperature can be increased up to approximately 40 ^oC, but there is also 11 % bsfc growth in the system.

Wickström (2012) studies the use of VVA for improving the thermal management of exhaust gases especially at low load conditions in diesel engines. Different VVT strategies are tested in different engine loads on a single-cylinder research engine. Loading is kept constant while VVT is implemented. Early and late IVC result in higher exhaust temperatures with lower NO_x emissions, but greater fuel consumption (up to 11 %). Exhaust and intake valve phase shifts are also rising the exhaust temperature and cause a NO_x fall of up to 8 g/kWh, however, bsfc growth can go up to 25 g/kWh.

Gehrke et al. (2013) tested a single cylinder MAN D20 research engine in order to investigate the potential benefits of VVA on exhaust thermal management. When early IVC is applied, there is up to 60 ^oC exhaust temperature rise. But Particulate Matter (PM) and CO increase in the system and also a slight bsfc growth is seen. Same exhaust temperature rise and emission increase are observed for late IVC too. For the negative valve overlap (crank angle (CA) between IVO and EVC) case; although exhaust gas temperature gain can become up to 70 ^oC, bsfc and PM can go up rapidly for the high negative overlap values. Finally, for the earlier EVO, greater bsfc, PM and CO emissions are the penalties so as to climb the exhaust gas temperature more than 60 ^oC.

Roberts et al. (2014) search the effect of early EVO on exhaust temperature and fuel consumption growth for a constant torque operation in a turbocharged, charge cooled, exhaust gas recirculated (EGR) six cylinder Cummins diesel engine. The analysis claims that the method results in lower (max. 5 % decrease) brake thermal efficiency, BTE, (therefore, higher fuel consumption) at low speeds and high loads in order to increase the turbine out temperature. However, for high speeds and low loads, a lower (about 2 % decrease) BTE is required to raise the turbine exit temperature. In the study, 30 ^oC to 100 ^oC exhaust temperature increase is obtained by advancing EVO 90 CA from the nominal position and that proves early EVO as a useful method for exhaust thermal management.

Magee (2014) uses cylinder deactivation (CDA) and late IVC (LIVC) in order to raise the exhaust gas temperature above 250 ^oC for efficient thermal management in a diesel engine for different engine loading cases. It is shown in the study that when CDA and LIVC are applied together, exhaust temperature values at low loads can become higher than 250 ^oC for various operating speeds of the diesel engine. Lower NO_x levels are yielded in the study, however, brake thermal efficiency cannot be increased significantly in the system.

Garg et al (2015) investigate the effects of early and late intake valve closing timings (EIVC&LIVC) on exhaust thermal management of a six-cylinder turbocharged&intercooled diesel engine. The engine loading is kept constant in the study and it is shown that turbine exit temperature (TET) can be increased to higher than 250 ^oC (more than 60 ^oC TET rise is obtained) for a low loading and low engine speed condition both with EIVC&LIVC timings. It is seen that TET is inversely proportional with the η_{vol} for either advanced or retarded IVC timings. It is also demonstrated in the study that EIVC&LIVC result in fuel-saving condition in comparison to nominal IVC timing due to the increase in open-cycle efficiency. However, this method causes to reduction in exhaust flow rate and it decreases the heat transfer from the exhaust flow to the catalyst substrate.

In this study, VVT will be used to increase the exhaust gas temperatures for more efficient exhaust thermal management. Particularly, at low load and low speed cases in diesel engines, the exhaust temperatures become lower than 250 ^oC which is insufficient for an efficient thermal management. Therefore, the aim of the study is to rise the exhaust gas temperatures above that limit for constant engine loading cases by advancing and retarding IVC timings. Firstly, specifications of the diesel engine studied are explained. Mathematical formulations and validation of the simulation will lead to that. Finally, application of VVT will be shown with various figures of IVC timings.

2. Diesel engine specifications and the simulation model

Diesel engine specifications and experimental data are explained first. Simulation model will be shown next.

2.1. Diesel engine specifications

The engine used for the study is Cummins type 6 cylinder turbocharged and intercooled diesel engine. The detailed specifications are given on Table 1 below. Performance curves of the diesel engine (η_{vol} and TET values) are obtained from Garg (2013). These graphs are used for the validation of the simulation model of the diesel engine with experimental results for 2.5 bar engine loading (taken as bmep) at 1200 rpm engine speed.

The experimental data for η_{vol} and TET for the studied diesel engine is shown in Table 2 below. As it is seen, results are given for earlier and later IVC timings. 0 CA denotes the nominal IVC timing.

The aim of the simulation model will be to approach those experimentally obtained numbers as close as possible so as to obtain a reliable model. Then, effect of IVC timings on diesel engine performance and exhaust gas flow rate can be examined.

Table 1. Specifications of the diesel engine.

No of cylinders	6
Bore (mm)	107
Stroke (mm)	124
Connecting-Rod Length (mm)	192
Displacement (L)	6.7
Compression Ratio	17.3
Firing Order	1-5-3-6-2-4
Fuel System/Type	Direct Injection / Diesel
Calorific Value of Fuel (kJ/kg)	42700
Intake Method	Turbocharged & Air-Air Intercooled

Table 2. Experimental data of the diesel engine.

IVC Timing (CA)	η_{vol} (%)	TET (°C)
-65	63,00	254,5
-50	72,30	232,0
-40	78,00	220,5
-20	88,50	206,5
0	93,60	198,0
20	94,50	195,0
50	89,60	203,0
70	82,30	217,0
90	74,20	235,0
100	69,30	249,0

2.2. The simulation model

In this study, Lotus Engine Simulation (LES) program is used for the simulation of the studied turbocharged&intercooled diesel engine (LES, 2013). The model is shown on Figure 1 below. Cylinders, valves, ports, intercooler, turbocharger, plenums, sensors and also pipes connecting these elements can all be seen in this figure.

The simulation model is set to operate at 1200 rpm engine speed and 2.5 bar bmep. The specifications given on Table 1 above are used in the model. However, other required data for the model construction is defined appropriately on the simulation in order to obtain the experimental performance values (η_{vol} & TET) at 2.5 bar engine loading and 1200 rpm engine speed condition on Table 2. This is explained explicitly on the validation of the model part.

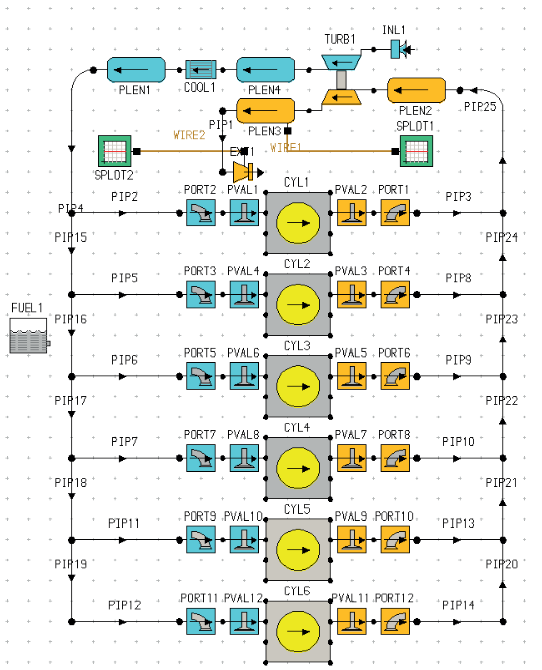


Figure 1. LES model of the diesel engine.

3. Mathematical formulations

3.1. Governing equations of gas flow

In the diesel engine simulation, one-dimensional model of pipe gas dynamics are applied for the gas flow in pipes. Conservation equations for mass, momentum and energy are solved for calculating the conditions within the pipe elements.

The governing equations (continuity, momentum and energy) for the 1-dimensional flow of a compressible fluid in a pipe with area variation, wall friction and heat transfer are (Winterbone, 2000):

$$\frac{\partial(\rho F)}{\partial t}+\frac{\partial(\rho u F)}{\partial x}=0 \tag{1}$$

$$\frac{\partial(\rho u F)}{\partial t}+\frac{\partial\left(\rho u^{2}+p\right)F}{\partial x}-p\frac{dF}{dx}+\frac{1}{2}\rho u^{2} f\pi D=0 \tag{2}$$

$$\frac{\partial(\rho e_{0} F)}{\partial t}+\frac{\partial(\rho u h_{0} F)}{\partial x}-q\rho F=0 \tag{3}$$

The equations above can be shown in vector form as:

$$\frac{\partial(W)}{\partial t}+\frac{\partial F(W)}{\partial x}+C=0 \tag{4}$$

where,

$$W = \begin{bmatrix} \rho F \\ \rho u F \\ \rho e_0 F \end{bmatrix} \quad (5)$$

$$F(W) = \begin{bmatrix} \rho u F \\ (\rho u^2 + p)F \\ \rho u h_0 F \end{bmatrix} \quad (6)$$

$$C = \begin{bmatrix} 0 \\ -p\dfrac{dF}{dx} \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ \rho G f \\ -q\rho F \end{bmatrix} \quad (7)$$

The two-step Lax-Wendroff (Richtmyer) is used to solve the governing equations above. The formulas for solution can be expressed with the following equations below:

$$W_i^{n+1} = W_i^n - \frac{\Delta t}{\Delta x}\left(F_{i+1/2}^{n+1/2} - F_{i-1/2}^{n+1/2}\right) - \frac{\Delta t}{2}\left(C_{i+1/2}^{n+1/2} + C_{i-1/2}^{n+1/2}\right) \quad (8)$$

$$W_{i-1/2}^{n+1/2} = \frac{1}{2}\left(W_i^n + W_{i-1}^n\right) - \frac{\Delta t}{2\Delta x}\left(F_i^n - F_{i-1}^n\right) - \frac{\Delta t}{4}\left(C_i^n + C_{i-1}^n\right) \quad (9)$$

$$W_{i+1/2}^{n+1/2} = \frac{1}{2}\left(W_{i+1}^n + W_i^n\right) - \frac{\Delta t}{2\Delta x}\left(F_{i+1}^n - F_i^n\right) - \frac{\Delta t}{4}\left(C_{i+1}^n + C_i^n\right) \quad (10)$$

3.2. Calculation of performance parameters

Diesel engine performance parameters in LES are calculated with the following equations below (Heywood, 1988)&(Pearson et al., 2002).

The brake mean effective pressure (bmep) of the diesel engine is found with:

$$bmep = imep - fmep \quad (11)$$

Sandoval&Heywood engine friction model is used to obtain the friction mean effective pressure (fmep) shown in formula (11) above (Sandoval&Heywood, 2003).

Indicated mean effective pressure (imep) is calculated with:

$$imep = W_c / V_d \quad (12)$$

In the equation above, W_c (kJ) represents the net indicated work per cycle and V_d shows the cylinder displacement volume. W_c can be determined with the formula below:

$$W_c = \int pdV \tag{13}$$

Displaced volume, V_d, is obtained with the following equation:

$$V_d = S(\pi B^2 / 4) \tag{14}$$

where S and B are engine stroke and cylinder bore.

The brake power, P_e (kW), and torque, τ_e (Nm), are calculated with the equations given below:

$$P_e = \left(bmepV_d NZ \middle/ n_r 60 \right) \tag{15}$$

$$\tau_e = \left(10^3 P_e \middle/ \omega \right) \tag{16}$$

where N is engine speed (rpm) and Z is the cylinder numbers. Also, n_r is the revolution per cycle and is taken as 2 for four-stroke engines and angular speed of the engine, ω (rad/s), is defined as:

$$\omega = 2\pi N / 60 \tag{17}$$

The brake specific fuel consumption, bsfc (g/kWh), is calculated with the following formula below:

$$bsfc = \dot{m}_f / P_e \tag{18}$$

where m_f shows the fuel mass flow rate (g/h). The brake thermal efficiency and volumetric efficiency of the system are found with:

$$\eta_{th} = \left[3600 P_e \middle/ Q_{LHV} \dot{m}_f \right] \tag{19}$$

$$\eta_{vol} = \left[2 \dot{m}_{ia} 10^3 \middle/ 30 \rho_{ia} V_d N \right] \tag{20}$$

where Q_{LHV} is the lower heating (calorific) value of fuel, m_{ia} is inlet air mass flow rate (g/h) and ρ_{ia} is inlet air density (kg/m^3) which is calculated by using the ideal gas law:

$$\rho_{ia} = \left[10^3 p_{ia} \middle/ RT_{ia} \right] \tag{21}$$

3.3. In-cylinder calculations

The conditions within cylinder are calculated at each crank angle by solving the energy equation (Benson, 1982):

$$\frac{dQ}{\partial t}+\frac{dB}{\partial t}-\frac{dW}{\partial t}=\frac{dE}{\partial t}+\sum \delta H \tag{22}$$

In the equation above, Q is the net rate of heat energy transfer into the system, B is the heat release due to combustion, δH is the enthalpy change due to gas flows. W and E represent the displacement work and internal energy.

3.4. Combustion system

A single zone heat release model is applied to the system. The combustion rate is determined by two-part Wiebe function (Watson&Pilley, 1980). The Wiebe function defines the mass fraction burned as:

$$m_{frac}=1.0-exp^{-A\left(\frac{\theta}{\theta_b}\right)^{M+1}} \tag{23}$$

where A and M are coefficients in Wiebe equation. θ is actual burn angle (after start of combustion), θ_b is the total burn angle (0-100 % burn duration).

In two-part Wiebe equation, total combustion period includes two periods; premixed combustion period and diffusion combustion period. The mass fraction burned in the premixed combustion is given as:

$$m_{frac,premixed}=1.0-\left[1.0-\left(\frac{\theta}{\theta_b}\right)^{C_1}\right]^{C_2} \tag{24}$$

and for diffusion combustion period, it is:

$$m_{frac,diffusion}=1.0-exp^{-A\left(\frac{\theta-\Delta}{\theta_b-\Delta}\right)^{M+1}} \tag{25}$$

Finally, mass fraction value is calculated with the following formula for two-part Wiebe:

$$m_{frac}=\beta\left(m_{frac,premixed}\right)+(1-\beta)\left(m_{frac,diffusion}\right) \tag{26}$$

In the equations above, C_1 and C_2 are coefficients in Watson&Pilley equation. β and Δ are the fraction of premixed combustion to total combustion and delay angle between premixed and diffusion combustion values. Typical values for a turbocharged DI diesel engine are given as A=10.0, M=0.4, C_1=2.0, C_2=5500, β=0.05 and Δ=0.0.

3.5. Heat transfer

In LES, heat transfer can be modeled with Annand, Woschni and Eichelberg formulations. Annand heat transfer model is chosen in the simulation for the cylinders.

The connective heat transfer model defined by Annand can be stated as (Annand, 1963):

$$\frac{hD_{cyl}}{k} = ARe^{B} \qquad (27)$$

In (27); h is heat transfer coefficient (W/m^{2}K), k is thermal conductivity of gas in the cylinder (W/mK), D_{cyl} is cylinder bore, Re is Reynolds number and A and B are Annand open or closed cycle coefficients which are taken for open cycle as 1.1 and 0.7 and for closed cycle as 0.2 and 0.8. The heat transfer per unit cylinder area can be calculated with:

$$\frac{d\tilde{Q}}{A} = h\left(T_{gas} - T_{wall}\right) + C\left(T_{gas}^{4} - T_{wall}^{4}\right) \qquad (28)$$

where A is area, T is temperature and C is Annand closed cycle coefficient, taken as 4.29x10^{-9}.

4. Validation of the model

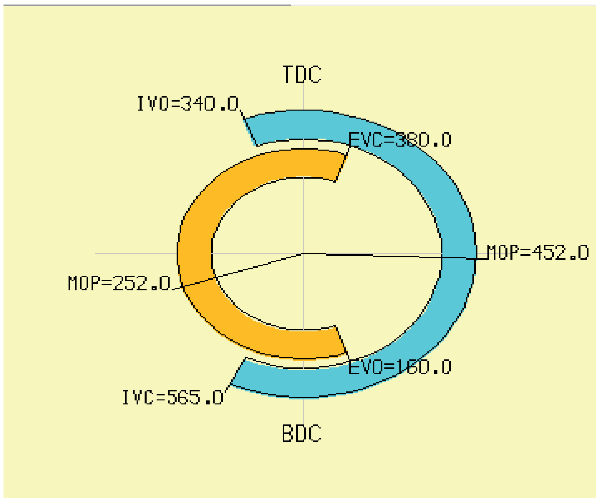
Technical data stated in Table 1 are inserted into the simulation specifications. However, some other input variables are required in the program and these values are taken as appropriately in order to achieve the experimental data in Table 2.

The inlet and exhaust port diameters are taken as 24,00 mm and 22,00 mm, fuel type is diesel and calorific value is 42700 kJ/kg. The sensors seen in Figure 1 are put on the model for TET (^{o}C) and exhaust flow rate (kg/min). One is located under the turbocharger; linked with a wire to the plenum after the turbine and obtains the TET values. Other one is connected to the exit and measures the exhaust flow going directly to the exhaust thermal management.

Maximum valve lifts for inlet and outlet valves are 8.5 mm and 9.90 mm. Also nominal valve timings for the simulation are given below on Figure 2.

When these nominal valve timing, valve maximum lift and other required parameters above are taken, the variation of the in-cylinder pressure (bar) through a cycle (0 CA to 720 CA) is obtained as shown on Figure 3 below. This pressure variation is valid for IVC565 and it is sufficient to produce 2.50 bar engine loading at 1200 rpm.

The experimental results on Table 2 are yielded via changing IVC timings as shown on Figure 4 below. Nominal IVC timing is closed later or earlier when all other valve timings and maximum valve lifts are kept constant. Also, at 1200 rpm engine speed, engine loading is kept fixed too.



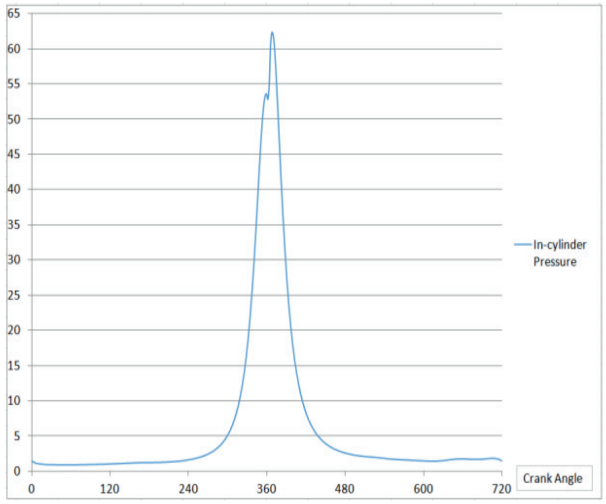


Figure 2. Nominal valve timings of the simulation. **Figure 3**. In-cylinder pressure (bar) variation.

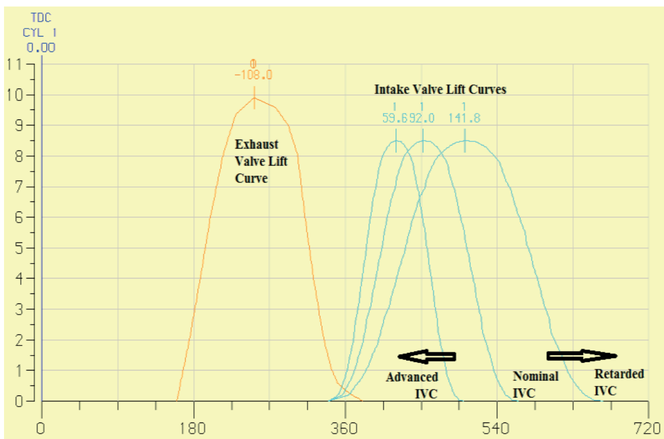
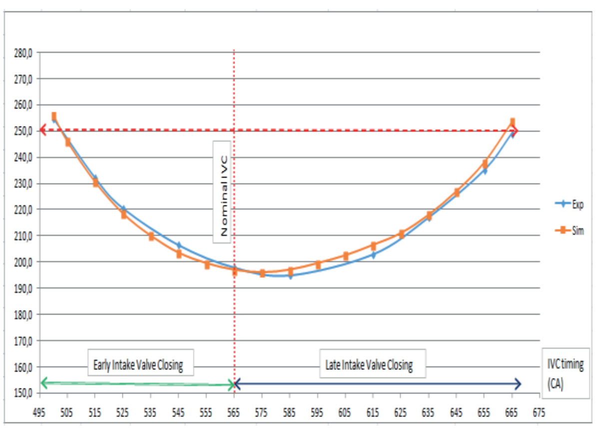


Figure 4. Advanced and retarded IVC timings.

By using the assumptions explained above and mathematical expressions stated on previous part, the performance graphs of the simulation for TET and volumetric efficiency are yielded as seen in Figure 5 and Figure 6 below.



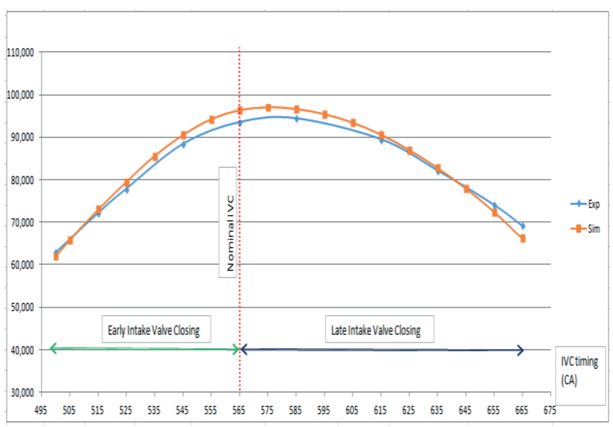


Figure 5. TET (^{o}C) comparison between sim & exp. **Figure 6**. η_{vol} (%) comparison between sim & exp.

As it is seen from Figures 5&6, simulated TET and volumetric efficiency values are generally compatible with the experimental data for 2.50 bar bmep and 1200 rpm. That proves that the simulation is reliable for examining the effect of IVC timing on exhaust thermal management. It is evident from the verification above that changing IVC timing is very useful for rising TET from nominal 195 ^{o}C to more than 250 ^{o}C. Early and late IVC are both beneficial. Advancing IVC timing 65 CA from the nominal value can result in 60 ^{o}C TET increase. However, for the same TET rise, IVC timing must be retarded 100 CA from the nominal closing timing. It can be derived that TET is directly related with the η_{vol}. Also, the results are consistent with the recent Miller process (earlier and later closing of IVC) studies (Dembinski&Lewis, 2009), (De Ojeda, 2010), (Modiyani, 2010).

5. Effect of IVC on diesel engine performance

As explained on the previous section that IVC is definitely practical for reaching higher than 250 ^{o}C TET and hence more efficient exhaust thermal management. However, as seen on Figure 6 volumetric efficiency is decreasing sharply for both EIVC and LIVC cases. It goes down lower than 70 % with LIVC and even lower than 65 % with EIVC. It can be derived that those extra air close to nominal IVC (at timings where volumetric efficiency is high) causes a decrease on TET. Also, when Figure 5 and Figure 6 are compared, it is evident that TET is inversely proportional with the volumetric efficiency. The lower the volumetric efficiency is, the higher the TET is for constant engine loading case.

Now, effect of IVC timing on diesel engine efficiency and exhaust flow rate will be examined. These performance values must also be considered while reaching greater than 250 ^{o}C TETs.

5.1. Diesel engine efficiency

In the study, as explained earlier, engine loading is kept constant (bmep equal to 2.50 bar) at 1200 rpm for different IVC timings. While IVC sweeps through earlier and later timings, fuel injection rate (mm^3/inj) at nominal case must be increased or decreased in order to manage the bmep fixed. The change in fuel injection rate per cylinder is shown below on Figure 7.

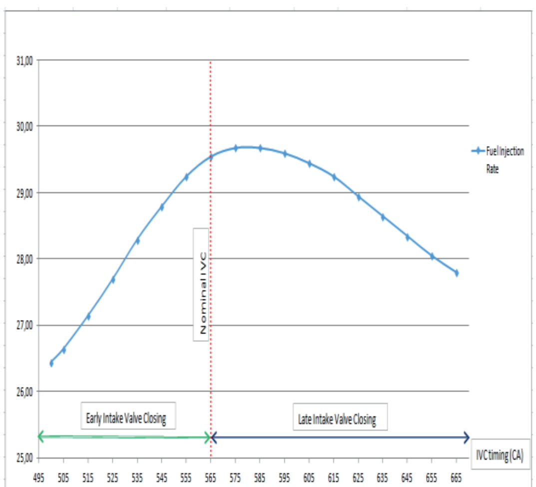
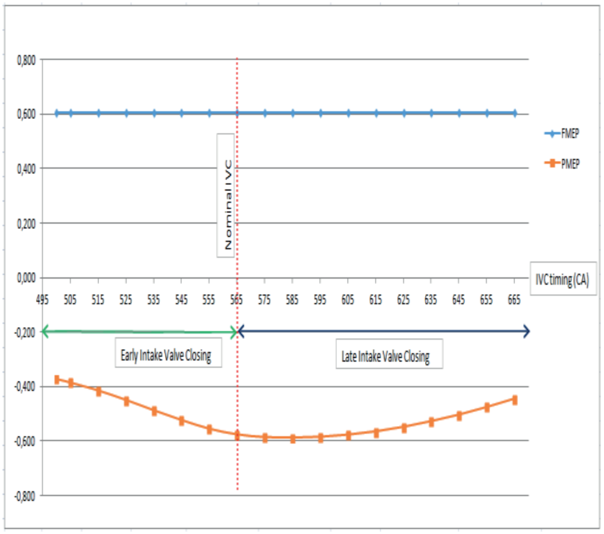


Figure 7. Fuel injection rate (mm^3/inj) change with IVC.

It is seen from figure above that fuel injection rate is going down for both advanced and retarded IVC timings. Less fuel is required to obtain 2.50 bmep when IVC is swept earlier and later than nominal. EIVC becomes more fuel-saving than LIVC. In order to understand the reason behind this fuel consumption reduction, pumping mean effective pressure (PMEP) , friction mean effective pressure (FMEP) and indicated mean effective pressure (IMEP) change for the same IVC timing alterations must be analyzed below on Figures 8&9.



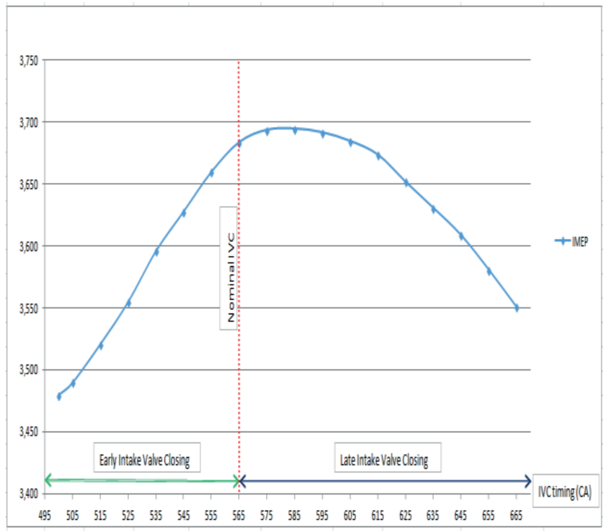


Figure 8. FMEP (bar) & PMEP (bar) change with IVC. **Figure 9**. IMEP (bar) change with IVC.

As it is seen from the figures above, FMEP is not affected significantly but PMEP is decreasing for earlier and later IVC timings. That means there is lower pumping loss during the cycle. This stems from the reduction of air flow through the engine which is evident from the decrease of volumetric efficiency demonstrated in Figure 6. Therefore, lower IMEP is needed to reach 2.50 bmep in comparison to nominal valve timing condition and lower fuel consumption is sufficient to maintain constant engine loading. The method is fuel-saving for both advanced and retarded IVC timings.

5.2. Exhaust flow rate

Sweeping IVC timing forward and backward from the nominal timing is definitely beneficial for reaching higher than 250 ^{o}C TETs. Moreover, it increases the efficiency by diminishing the demanded fuel injection rate for the same engine loading. However, it also has a negative effect. It leads to reduction on exhaust flow rate. Exhaust flow (kg/min) change is given on Figure 10 below. Similar to volumetric efficiency, it drops sharply too.

Decreased exhaust flow rate causes to reduction of heat transfer from the exhaust gases leaving the turbine to the catalyst substrate. It also slows down the process and affects efficiency negatively. However, 2.50 bar bmep is a low engine loading condition. Exhaust flow probably will not decrease that much for higher engine loading cases when the same method is applied to rise the TET above 250 ^{o}C. At lower engine loadings, this is a serious problem.

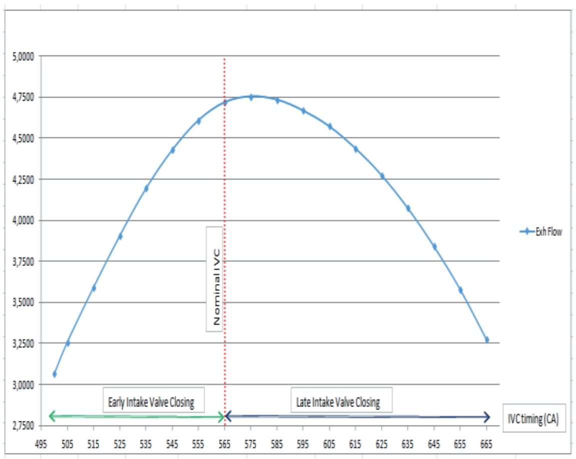


Figure 10. Exhaust flow (kg/min) change with IVC timings.

6. Conclusion

In this study, effect of earlier and later closing of intake valve on exhaust thermal management of a 6-cylinder turbocharged&intercooled diesel engine system is examined. The diesel engine model is constructed via using LES program and it is then validated with the experimental results of the same diesel engine for different intake valve closing timings. It is seen that TET can become higher than 250 ^{o}C when IVC is advanced 65 CA from nominal timing or is retarded 100 CA from stock valve timing. Up to 60 ^{o}C TET rise is achieved with this method for both earliest and latest IVC timings. Effect of IVC timing on diesel engine performance is also studied. Sweeping IVC timing from nominal results in fuel-efficiency for both advanced and retarded cases. Decreased volumetric efficiency lowers the pumping losses and decreased PMEP leads to fuel-saving. But, reducing volumetric efficiency also causes exhaust flow rate to drop sharply which decreases the heat transfer from the exhaust gases to the catalyst substrate. Examining VVT effect on exhaust thermal management efficiency should continue in order to raise TET above 250 ^{o}C without causing a significant drop in exhaust flow rate.

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Zorlanmış yalpa hareketi yapan iki boyutlu cisimlerin hidrodinamik katsayılarının interpolasyonlu parçacık metodu ile hesaplanması

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Özet

Su yüzeyinde salınım yapan cisimlerde meydana gelen hidrodinamik kuvvetlerin ve momentlerin tahmini için birçok analitik, deneysel ve sayısal çalışma yapılmaktadır. Bu çalışmada, sayısal yöntemlerden biri olan İnterpolasyonlu Parçacık Hidrodinamiği (Smoothed Particle Hydrodynamics) kullanılarak zorlanmış yalpa hareketi yapan 2-Boyutlu cisimlere ait ek-su kütlesi ve sönüm katsayıları için bir hesaplama altyapısı geliştirilmiş ve elde edilen sonuçların literatürde yer alan diğer sonuçlar ile mukayesesi yapılmıştır. İnterpolasyonlu Parçacık Hidrodinamiği (İPH) yönteminin uygulamasında değişik algoritmalar kullanılmakta olup, bu çalışma çerçevesinde geliştirilmiş bilgisayar programı temelde, Yapay Viskozite Terimi'ni (Artificial Viscosity Term) içeren Euler Hareket Denklemi ve Süreklilik Denklemini, Zayıf Olarak Sıkıştırılabilir İPH Yaklaşımı (WCSPH) yardımı ile çözmektedir. Geliştirilmiş olan bilgisayar kodunda; WCSPH kullanımında basınç değerlerinin değişimini düzenleyen Yoğunluk Düzeltmesi (Shephard Filtering) ile Özbulut (2013a) tarafından geliştirmiş olan Birleşik Serbest Su Yüzeyi ve Suni Parçacık Ötelemesi çözüm algoritması kullanılmıştır. Ayrıca problem sınırlarının eğik olduğu durumlarda ise çözüm algoritması içinde parçacıkların çözüm bölgesine daha homojen dağıtılmasına yarayan Colagrossi ve diğerleri (2012) tarafından geliştirilen Parçacık Paketleme Algoritması (Particle Packing Algorithm) kullanılmıştır. Herhangi bir ağ sistemine ihtiyaç duymayan ve Lagrange temelli doğası ile parçacıkların her birinin çözüm süresi boyunca yoğunluk, basınç, hız, vb. kinematik ya da dinamik büyüklüklerinin takibine dayanan İPH yönteminin kullanılması ile elde edilen hidrodinamik katsayıların, literatürde yer alan diğer sayısal yöntemlere oranla deney sonuçlarına daha yakın değer verdiği gözlenmiştir.

Anahtar kelimeler: Yalpa hareketi, hidrodinamik katsayılar, interpolasyonlu parçacık hidrodinamiği, sönüm katsayısı, ek-su kütlesi.

1. Giriş

Dünya ticaretinin %90'ının deniz yolu ile gerçekleştirildiği, savunma sanayi harcamalarının %30'una yakının deniz platformlarına yapıldığı düşünüldüğünde deniz taşıtlarının ticari, askeri ve mali olarak ne kadar önemli olduğu ortaya çıkmaktadır. Ülkelerin savunmasında, ticari faaliyetlerinde ve mali