Thermodynamic Simulation Model of the Compression Ignition Engine

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Abstract: In this work we carried out a comparative study of indicated mean effective pressure, mean effective pressure, power, torque and brake specific fuel consumption obtained by the analytical model for thermodynamic cycle simulation of a turbocharged diesel engine with the computer program developed in the language FORTRAN and those with the GT-Power software.The language FORTRAN program developedis currently usedin the course ofmodeling and simulation ofengine performance.

Key words: One zone model; Ignition compression engine; Heat transfer; Friction; Turbocharged Diesel engine; GT-Power.

Introduction: The objective of the present work is to predict and analyze the performance of a turbocharged compression ignition engine using a developed computational thermodynamic model. This simulation model predicts in-cylinder temperatures and pressures as function of the crank angle, with the application of modified Vibe function for combustion model and Woschni correlation modified by Hohenberg for heat transfer at cylinder walls. It also takes into consideration the effects of heat losses and temperature dependent specific heats. In this work, we will compare the effects on performance characteristics mentioned above of changing values for example; engine speed, crank angle, injection timing and compression ratio, respectively predicted by the developed simulation program and the selected known commercial GT-Power software.

Method: The developed process simulation program, written in FORTRAN, taking into account the specifications of diesel engines. To compare and validate the theoretical results obtained, we chose the commercial software GT-Power. The assumptions that have been made in developing the in-cylinder model for the direct injection diesel engine are:

- The pressure and temperature of cylinder charge are assumed to be uniform throughout the cylinder and vary with crank angle.
- The unburned mixture at any instant is composed of air and residual gases without chemical reaction.
- No gas leakage through the valves and piston rings so that the mass remains constant.
- The heat transfer region is limited by the cylinder head, the bottom surface of the piston and the instantaneous cylinder wall.
- The temperature of the surfaces mentioned above is constant during the cycle.
- The rate of heat transfer of gases to the wall is calculated from the temperature of the combustion gases and the wall.
- The heat transfer of gas-wall is changing rapidly due to the motion of the gas during the piston motion and to the geometry of the combustion chamber. The correlation of Hohenberg is used to calculate the rate of heat transfer cylinder. We consider uniform crank speed (steady state engine).

In this simulation, we chose the single zone combustion model proposed by Watson and al. This model reproduces in two combustion phases; the first is the faster combustion process, said the premixed combustion and the second is the diffusion combustion which is slower and represents the main combustion phase.

Figures 1 present the schematic flowchart of the developed computer simulation program.

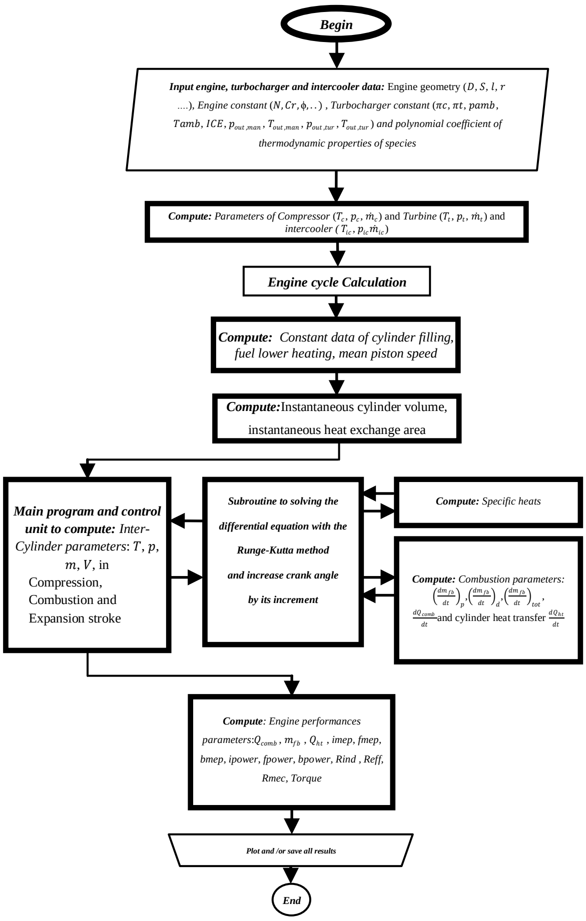

Fig.1:*schematic flowchart of the developed computer simulation program.*

Results and discussions

Figures 2 and 3 show the evolution of the fuel burning rate and the normalized burned fuel mass for premixed, diffusion and total combustion at a speed of 1400 rpm and full load.

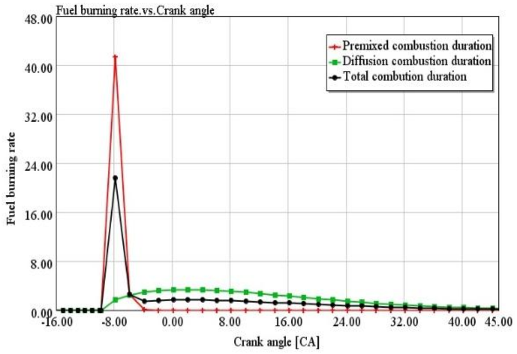

Fig.2:*Fuel burning rate for premixed, diffusion and total combustion.*

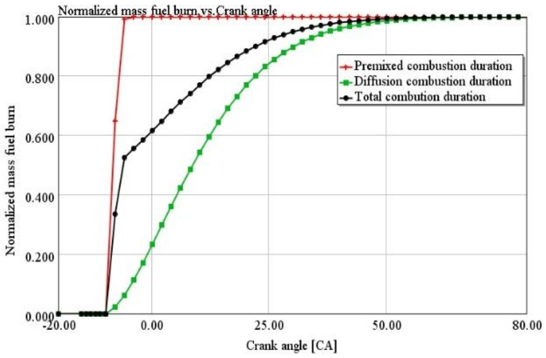

Fig.3:*Normalized burned fuel mass in premixed, diffusion and total combustion.*

Figures4and 5presents the cylinder pressure and Temperature traces for half and full load at N = 1400 rpmobtained using the developed simulation model and GT-Power. The irregularities shown in the tow diagrams may be due to the residual gas portion in the combustion chamber and also the vaporization of diesel fuel in the cylinder walls causing a cooling effect which affects the pressure and temperature. The appearance of both curves is almost identical.

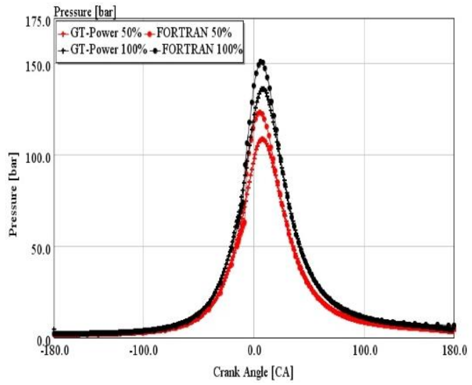

Fig.4:*Comparison of cylinder pressure at N=1400 rpm and different loads.*

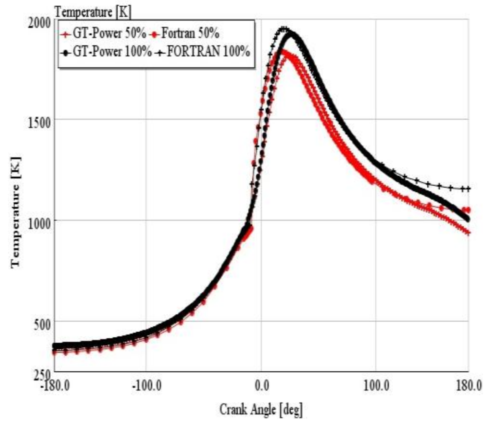

Fig.5:*Temperature cylinder for N=1400rpm.*

The evolution of the friction power as a function of engine speed for different loads is shown in Figure 6. The friction power increases parabolically with an engine speed due to friction losses of moving parts and pumping losses, which correspond to the polynomial function found by Hendricks and Sorenson. The friction Power also increases with the engine load due to increaseof lateral forces on the piston. We can observe that the engine speed and load has the major influence over the engine friction. The friction powers according to the GT-Power model are smaller than those of the developed simulation and this is due to the choice of adopted friction model, Chen-Flynn friction model. For a speed of 2100 rpm, the middle gap between the two results, developed simulation model and GT-Power, is lower than 8%.

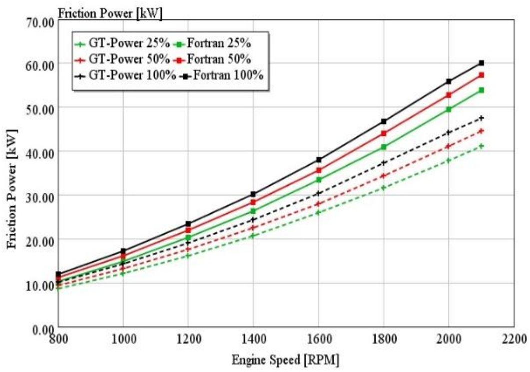

Fig.6:*Friction Power at full load and partial loads.*

Figure 7 shows the change of brake power engine upon engine speed for different loads. The effective powers increase with the engine speed and loads. The gap between the two results, the developed simulation model and with GT-Power grows with the engine speed and loads, but the middle gap is lower than 8%.

The variation of the brake efficiency and themechanical efficiency with the engine speed and for different engine loads are shown by Figures 8.*a* and 8.*b*.

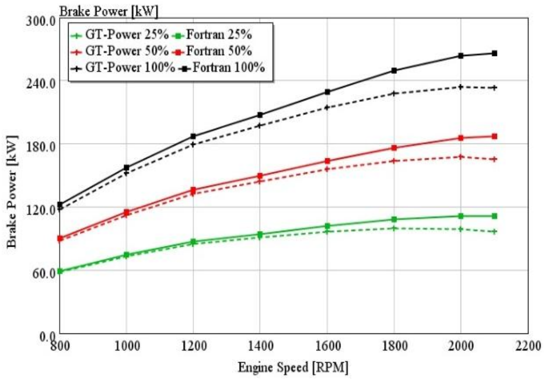

Fig.7:*Brake power at full load and partial loads.*

There is some difference in results between the developed simulation model and the GT-Power model due to the pressure losses with the engine speed, and essentially tothe chosen friction losses model.

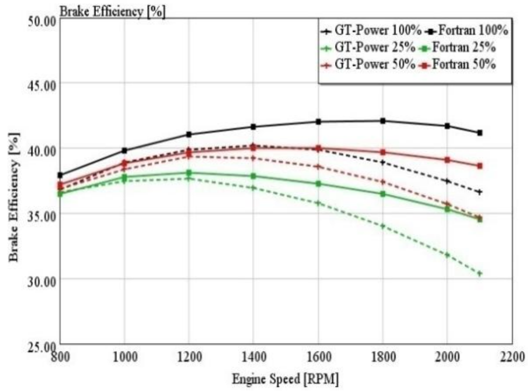

Fig.8-*a*:*Brake efficiency at full and partial load.*

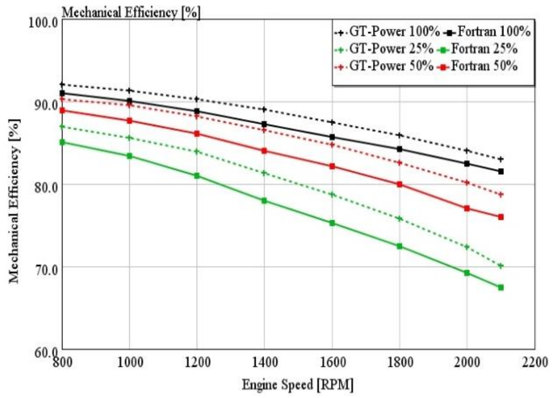

Fig.8-*b*:*Mechanical efficiency at full and partial load.*

In Figures 9.*a* and*9.b*, we can see that the friction pressure increases linearly and the mean effective pressure drops with the engine speed,regardless of the engine load.

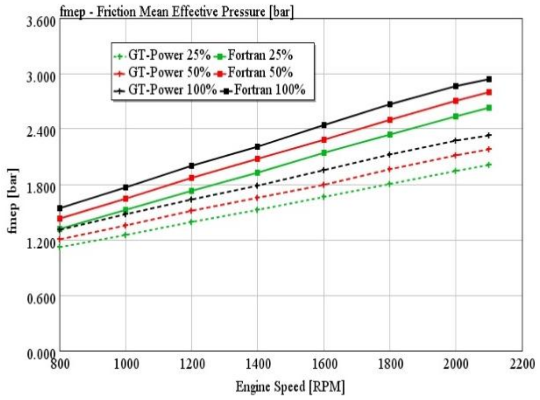

Fig.9-*a*:*Friction pressure at full and partial load.*

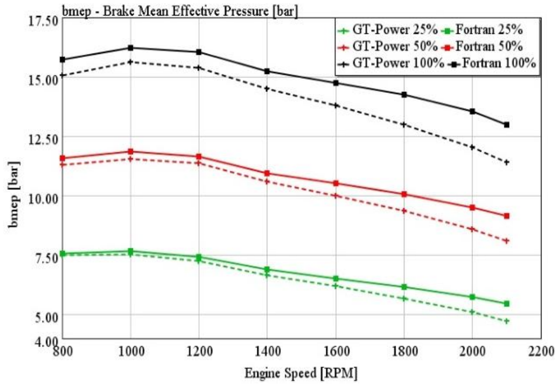

Fig.9-*b*:*Mean effective pressure at full and partial load*

Conclusions

The comparative study showed that the results obtained with the developed simulation program are sufficiently similar to those with the used commercial software GT-Power. The results from the present numerical simulation model could be used to improve the design and control strategy of the engine in terms of performance.

In future work, we will try to develop a semi-empirical model describing more accurately the real operating conditions of a turbocharged diesel engine, incorporating for example real values for timing, injection rate, combustion rates, real engine geometry, heat transfer in combustion chamber, input and the exhaust manifold, and engine-turbocharger interaction. The present study leaves open many possibilities for future researches based upon the engine modeling.

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