The Effect of Various Parameters on the Crankshaft Torsional Deformation of a Turbocharged Diesel Engine Operating under Transient Load Conditions

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ABSTRACT: The modeling of transient diesel engine operation appeared in the early seventies and continues to be in the focal point of research, due to the importance of transient response in the everyday operating conditions of engines. The majority of the simulations have focused so far on thermodynamics, as this directly affects performance and pollutants emissions. As far as dynamic modeling is concerned, simplified approaches are usually adopted in order to save execution time of the simulation code. In this paper, one of the basic simplifications of transient dynamic modeling, i.e. the assumption of the crankshaft being sufficiently rigid is being investigated. In real engine operation, different, instantaneous torsional angles for the engine and load are experienced that lead to angular deformation of the crankshaft. This deformation can assume significant values, particularly during transients, and as such it is of great importance for safe engine operation. The crankshaft angular momentum equilibrium is analyzed in detail taking into account gas, inertia, friction, load as well as stiffness and damping torque contributions. An experimentally validated simulation code is used to study the effect of various parameters on the crankshaft torsional deformation during transients. The parameters considered are the magnitude of the applied load, the rigidity and damping of the crankshaft, the engine and load mass moments of inertia, the load change schedule, the type of the connected resistance and the ratio of crank radius to connecting rod length. Analytical diagrams are provided, which demonstrate the effect of each parameter on the transient crankshaft deformation.

Keywords: Diesel engine, Transient operation, Crankshaft, Torsional Deformation

NOMENCLATURE

A	Cross section area [m^2]
C	Coefficient
D	Cylinder bore [m]
F	Force [N]
G	Mass moment of inertia [kg m^2]
L_{crod}	Connecting rod length [m]
m	Mass [kg]
p	Pressure [bar]
r	Crank radius [m]
s	Exponent in load torque equation (3)
t	Time [sec]

Greek symbols

λ	Ratio of crank radius to connecting rod length
σ	Stress [N/m^2]

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τ Torque [Nm]
φ Crank angle [deg]
ω Angular velocity [rad/s]

Subscripts
D Damping
e Engine
fl Flywheel
fr Friction
g Gas
in Inertia
L Load
S Stiffness
T Torsional

Abbreviations
^{o}CA Degrees crank angle
rpm Revolutions per minute
TDC Top Dead Center

1. INTRODUCTION

The turbocharged diesel engine is nowadays the most preferred prime mover in medium and medium-large units applications. Moreover, it continuously increases its share in the highly competitive automotive market, owing to its reliability, which is combined with excellent fuel efficiency. Nonetheless, its transient operation, which comprises a significant portion of the everyday operating conditions of engines, is often linked with off-design (e.g. turbocharger lag) and consequently non-optimum performance and increased exhaust emissions. The latter point out the necessity for correct modeling of all individual engine processes.

During the last decades, diesel engine modeling has greatly supported the study of transient operation [1-4]. Ideally, a complete diesel engine transient simulation code should include analytical models for all thermodynamic and dynamic processes. However, when the main target of the simulation is engine performance and pollutants emissions, which is actually the case in most research works, it seems reasonable to focus primarily on thermodynamics. In these cases, simplified approaches are usually adopted for engine dynamic behavior in order to save execution time of the simulation code [3].

On the other hand, if the dynamic operation of the engine is the main object of research [5-9], advanced, non-linear models for the crankshaft and slider-crank mechanism need to be applied. Past research has shown that instantaneous engine torque fluctuates significantly during an engine cycle, even under steady state conditions due to the cyclic nature of gas pressures and inertia reciprocating forces [10]. On the other hand, resistance (load) torque remains practically constant during a cycle, owing to the low non-uniformity of rotation. As a result, a significant fluctuation occurs in the instantaneous net (engine minus load) torque that eventually leads to cyclic speed irregularities, twists between individual cranks of a multi-cylinder engine and, finally, torsional (angular) deformation of the whole elastic crankshaft. The crankshaft deformation is further enhanced during transient operation, owing to the dynamic instability induced by the considerable deficit of torque after the new, increased, load has been applied.

The object of this paper is to evaluate the effect of various parameters on the crankshaft torsional deformation and stress during transient operation. To this aim, an experimentally validated, non-linear, transient diesel engine simulation code that follows the filling and emptying modeling technique is used [3,4,11]. The crankshaft angular momentum equilibrium is analyzed in detail, taking into account the *instantaneous* values for gas, inertia, friction, load, stiffness and damping torque contributions. The parameters considered are the magnitude of the applied load, the rigidity and damping of the crankshaft, the engine and load mass moments of inertia, the load change duration, the type of the applied load and the ratio of crank radius to connecting rod length.

The results of the analysis are given in a series of analytical diagrams, which depict and quantify the effect of each parameter considered on the crankshaft torsional deformation. Due to the narrow speed range of the engine in hand, only load increases under constant governor setting are investigated, which, nonetheless, play a significant role in the European Transient Cycles of heavy duty vehicles.

2. SIMULATION ANALYSIS

2.1 General Process Description

The present analysis does not, at the moment, include prediction of exhaust gas emissions and on the other hand deals with transient operation calculations on a ^{o}CA (degree crank angle) basis. Therefore, a single-zone model following the filling and emptying approach is used for the thermodynamic processes evaluation. This approach is believed to be the best compromise between accuracy and limited PC program execution time [3].

For heat release rate predictions, the fundamental model proposed by Whitehouse and Way [12] is used. Especially during transients, the constant K in the (dominant) preparation rate equation of the model is correlated with the Sauter mean diameter (SMD) of the fuel droplets through a formula of the type $K \propto (1/SMD)^{2.5}$ [3,4].

The improved model of Annand [13] is used to simulate heat loss to the cylinder walls. During transient operation, the thermal inertia of the cylinder wall is taken into account, using a detailed heat transfer scheme that models the temperature distribution from the gas to the cylinder wall up to the coolant.

Various sophisticated sub-models, which analyze important engine features during transient operation, have been incorporated in the main simulation code [3,4,11]:

Multi-cylinder engine modeling. All the governing differential and algebraic equations are solved individually for every one cylinder of the six-cylinder engine under study. By following this approach, improved accuracy is accomplished since both the fueling and the air-mass flow rates differ for each cylinder during the *same* cycle of a transient event as a result of the continuous movement of the fuel pump rack and compressor operating point.

Fuel pump operation. Instead of using steady-state fuel pump data, an analytical fuel injection model, experimentally validated at steady-state conditions, is applied [14]; this simulates the fuel pump-injector lift mechanism using the method of characteristics, taking into account the delivery valve and injector needle motion.

Friction. For the computation of friction torque, at each ^{o}CA, the model proposed by Taraza *et al.* [15] is adopted, which describes the non-steady profile of friction torque during each cycle. The total amount of friction is divided into four parts (piston rings assembly, valvetrain, loaded bearings and auxiliaries) and it varies significantly during an engine cycle (especially around 'hot' TDC), unlike the usual 'mean fmep' approaches where friction torque remains constant throughout the cycle.

Connecting rod dynamics. The connecting rod is usually modeled as equivalent to two lumped masses concentrated at its ends, one reciprocating with the piston and the other rotating with the crank pin. In this work, a detailed model, based on rigid body dynamics, is used to simulate the complex movement of the rod's center of gravity that is produced by its simultaneous reciprocating and rotating motion [7]. Thus, a more accurate calculation of inertia torque is achieved.

2.2 Crankshaft Torque Equilibrium

The (crank)shaft is considered as a flexible, elastic body that may deform during engine operation. A condensed crankshaft model was chosen, i.e. rigid enough between the cylinders and elastic between flywheel and load. The angular momentum equilibrium for the whole system is described by the following two non-linear differential equations [9,3,7], with reference also to Fig. 1,

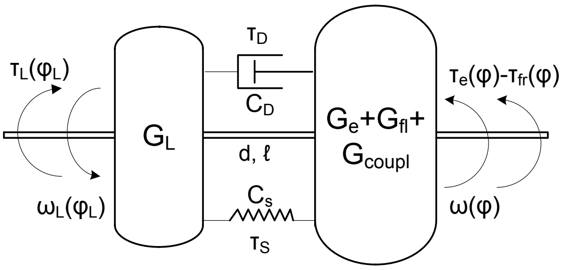

Figure 1: Schematic arrangement of engine-load dynamic system for crankshaft angular momentum equilibrium analysis

$$\tau_e(\varphi) - \tau_{fr}(\varphi) - \tau_S - \tau_D = (G_e + G_{fl} + G_{coupl})\frac{d\omega}{dt} \quad (1a)$$

$$\tau_S + \tau_D - \tau_L(\varphi_L) = G_L \frac{d\omega_L}{dt} \quad (1b)$$

Here, G_e, G_{fl}, G_{coupl} and G_L is the engine, flywheel, elastic coupling and load mass moments of inertia respectively, $\omega = d\varphi/dt$ is the engine angular velocity, $\omega_L = d\varphi_L/dt$ is the load angular velocity, and the crankshaft torsional deformation is defined as $(\varphi\text{-}\varphi_L)$. Further, $\tau_e(\varphi,\omega)$ denotes the instantaneous engine indicated torque (comprising of gas, and inertia forces contributions) that is given explicitly by

$$\tau_e(\varphi) = \tau_g(\varphi) + \tau_{in}(\varphi) = \left[\left(p_g(\varphi)\cdot A_{pist}\cdot\frac{u_{pist}(\varphi)}{r\omega}\right) + F_{Tin}(\varphi)\right] r \quad (2)$$

In the above relation, $p_g(\varphi)$ is the instantaneous cylinder pressure, F_{Tin} is the total torsional inertia force computed from the detailed connecting rod sub-model [7] and u_{pist} is the instantaneous piston velocity [3,7,10]. Also, in Eqs (1), $\tau_{fr}(\varphi)$ stands for the friction torque, $\tau_S = C_S(\varphi\text{-}\varphi_L)$ is the stiffness torque and $\tau_D = C_D(\omega\text{-}\omega_L)$ is the damping torque. Finally, τ_L is the load torque, which is approximated by the following relation,

$$\tau_L(\varphi_L) = C_1 + C_2\cdot[\omega_L(\varphi_L)]^s \quad (3)$$

For a linear load-type (i.e. electric brake, generator) s=1, for a quadratic load-type (i.e. hydraulic brake, vehicle aerodynamic resistance) s=2, with C_1 the speed-independent load term (e.g. road slope).

Simultaneous solution of Eqs (1) at each ^{o}CA, using the instantaneous values for each torque term, provides the engine-side and load-side rotational speeds as well as the respective angles φ and φ_L and, hence, torsional deformation $(\varphi\text{-}\varphi_L)$. Eqs (1) can be further expanded if we consider the torsional deformations between each cylinder, flywheel and load [5].

3. RESULTS AND DISCUSSION

Prior to the parametric study, a detailed experimental investigation was carried out on a six-cylinder, turbocharged diesel engine operating under steady-state and transient conditions. The basic data for the engine are given in Table 1. Since the particular engine has a narrow speed range, mainly load changes (increases) under constant governor setting were examined. Details and typical results of the experimental procedure can be found in [4]. After gaining confidence in the model's predictive capabilities, we proceeded to the parametric investigation of the effect of various engine parameters on the crankshaft torsional deformation during transients.

Table 1: Engine Data

Engine Type	6-cylinder, 4-stroke, turbocharged and aftercooled, heavy-duty diesel engine
Speed Range	1000÷1500 rpm
Bore / Stroke	140 mm / 180 mm
Maximum Power	236kW @ 1500 rpm
Moment of Inertia (engine and load)	15.60 kg m^2
Shaft rigidity	240,000 Nm/rad

Figure 2 illustrates a typical load increase transient of the order of 10-75%

commencing from an engine speed of 1180 rpm; this is the baseline for the analysis that follows. The instantaneous crankshaft torsional deformation during some intermediate cycles is given in Fig. 2a, while in Fig. 2b the development of engine speed and maximum and mean, over an engine cycle, deformation values and stress is depicted.

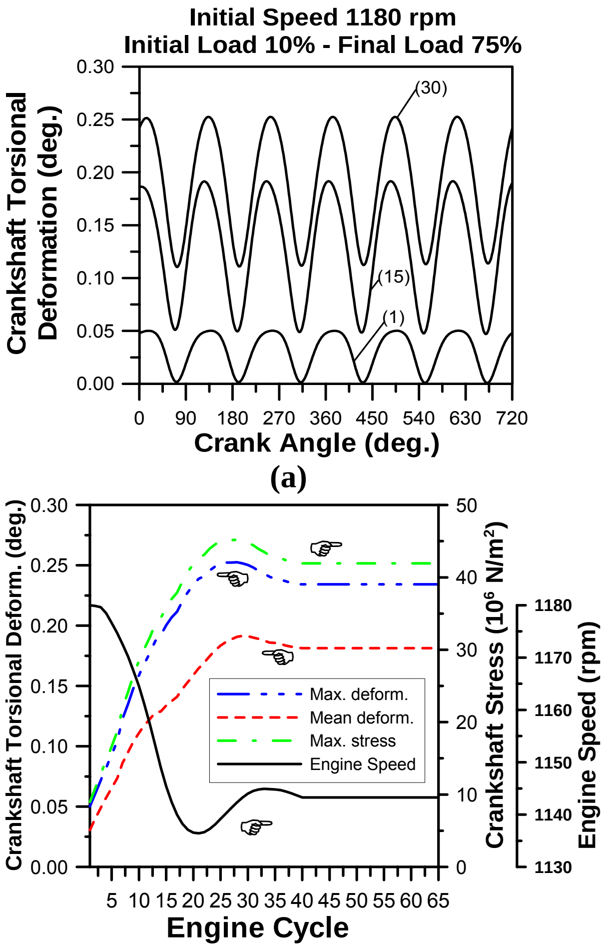

Figure 2: Nominal transient event – (a) In-cycle crankshaft torsional deformation (numbers in parentheses denote engine cycle), (b) Development of engine speed and maximum and mean, over an engine cycle, values of crankshaft torsional deformation, and stress

During the early cycles, the deformation is negligible owing to the low engine load. As the load increases, deformation increases too, due to higher gas pressures/torques, resulting from higher fuelings, following the governor response to the speed drop (Fig. 2b). Two important features should be underlined, with reference to Fig. 2:

- the increase in engine load leads in greater peak values and wider in-cycle fluctuations of the crankshaft deformations (Fig. 2a); the effect of the number of engine cylinders being dominant in these curves,
- the instantaneous maximum, over an engine cycle, deformation is considerably higher, (up to 85% for the cases examined in this work) than the respective mean value for the same transient cycle (Fig. 2b). This justifies the analysis on a degree crank angle basis, over the way classic torsional vibration handbooks deal with the subject, i.e. at *steady-state conditions*, on a *cycle* basis, and for the special, but very important, case where the engine operates at resonance with some harmonic order of the exciting engine forces [16]. The importance of the respective torsional (shear) stress is also indicated in Fig. 2b. This stress should be maintained low for safe engine operation. It is given by

$$\sigma = \frac{\Theta}{2}\frac{d}{\ell}\Delta\varphi_d = \frac{\Theta}{2}\frac{d}{\ell}(\varphi - \varphi_L) \qquad (4)$$

where Θ is the shear modulus and d, ℓ are the shaft diameter and length, respectively, between engine flywheel and load (Fig. 1).

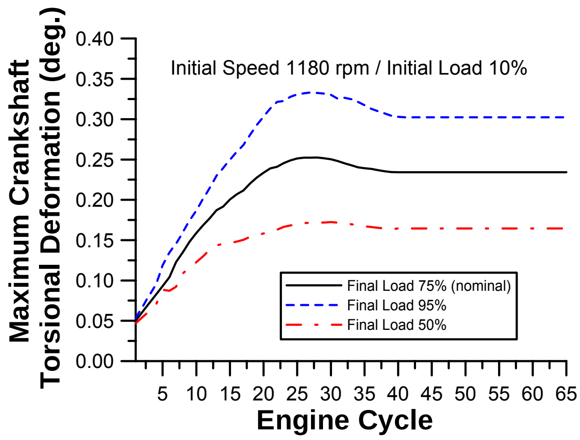

Figure 3: Effect of load change magnitude on the crankshaft torsional deformation during transient operation

Figure 3 demonstrates the effect of the magnitude of the final load on the crankshaft deformation. A higher applied load leads in greater speed drop, hence, larger governor displacement and fuel pump rack position, greater fueling rate and, finally, higher cylinder pressures/gas torques. Consequently, after the start of combustion and during expansion, the surplus of net engine torque, i.e. term $\tau_e(\varphi)$ - $\tau_{fr}(\varphi)$ in Eq. (1a) is higher, resulting in greater peaks of crankshaft deformation (cf. comments for Fig. 2).

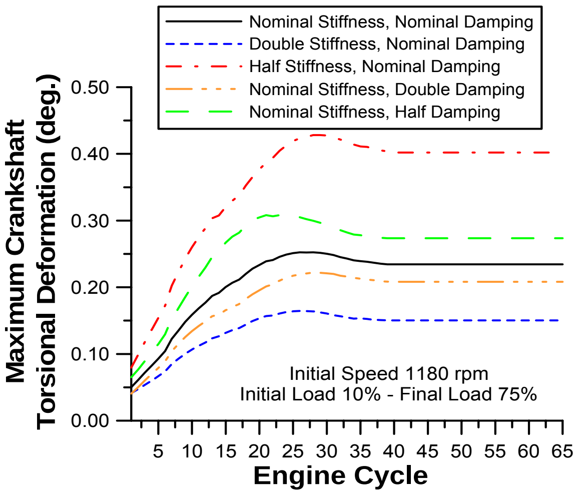

Figure 4: Effect of crankshaft stiffness and damping on the crankshaft torsional deformation during transient operation after a load increase

The effect of the main crankshaft properties, i.e. stiffness and damping, on the crankshaft deformation is presented in Fig. 4. As was expected, a more rigid shaft construction (greater values of C_S) results in lower deformations. The same holds for higher damping (greater values of C_D, realized via the use of elastic couplings), which is able to absorb the surplus of instantaneous net torque. Comparing the results between the cases of double and half stiffness coefficient, a difference of the order of 160% is observed regarding the peak value. On the other hand, the effect of damping appears to be less prominent for the current engine-load configuration.

Figure 5 illustrates the effect of engine (flywheel) and load mass moments of inertia on the crankshaft torsional deformation. The interesting finding here is the conflicting behavior of the engine mass moment of inertia compared to the load one. A smaller *engine* moment of inertia leads in greater deformation peaks, due to the higher acceleration rates dω/dt induced. This is the exact same phenomenon that applied to the case of low stiffness in Fig. 4. On the other hand, a smaller *load* mass moment of inertia results in lower values of deformation; the mechanism being the greater ease with which the load (resistance) 'follows' the rotary motion of the crankshaft.

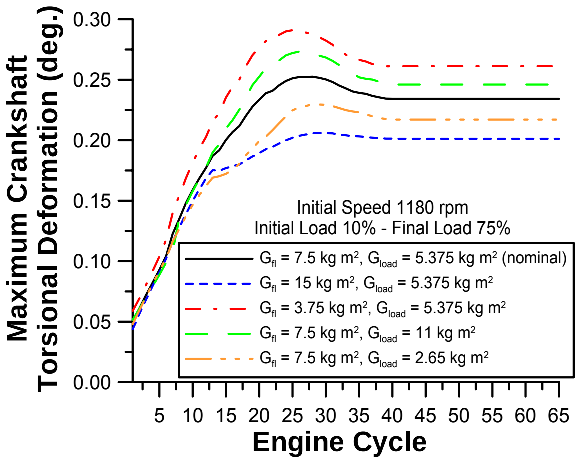

Figure 5: Effect of engine (flywheel) and load mass moments of inertia on the crankshaft torsional deformation during transient operation after a load increase

Figure 6 illustrates the effect of the load change duration on the crankshaft torsional deformation. The deformation profile follows closely the fueling one (upper sub-diagram of Fig. 6), and appears to be significantly affected by the load-time schedule. However, as was intuitively expected, the final equilibrium conditions are practically the same for all the cases examined. The most demanding case is the one with instant load application (Δt_{load}=0). Here, the load torque increases instantly reaching its final value during the first transient cycle, while its engine counterpart responds with a delay, following the governor response to the speed drop. Consequently, there is a considerable torque deficit during the first cycles of the transient event that leads in earlier, and considerably higher deformations compared to the nominal case (Δt_{load}=1.3 sec). A pulsating form of engine recovery is also observed in this case, which is usually unacceptable. On the other hand, a long load change duration (Δt_{load}=3.0 sec) causes smoother development of the crankshaft deformation, as the whole transient event develops at a much slower and, hence, 'safer' rate.

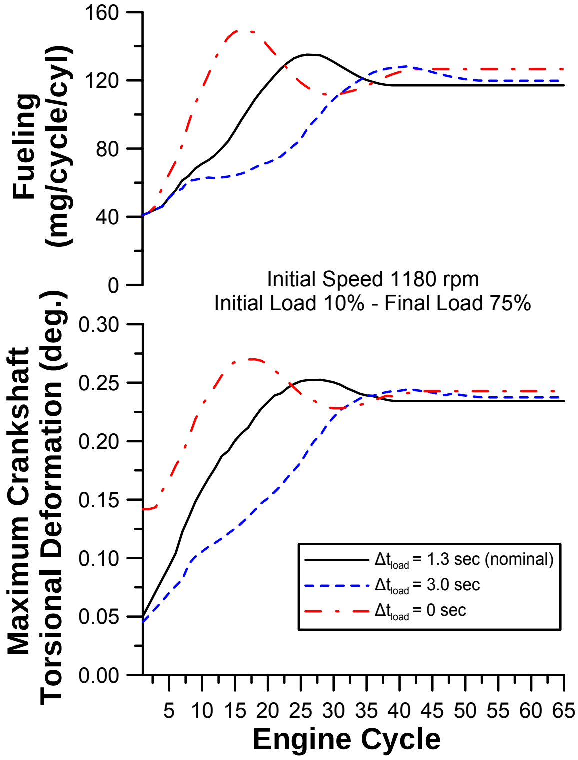

Figure 6: Development of crankshaft torsional deformation and fueling during transient operation after a load increase, for various load-time schedules

Another parameter that significantly affects the transient engine response is the load (resistance) type. Its effect on the crankshaft torsional deformation is depicted in Fig. 7. In the same figure, the engine speed response is also illustrated for comparison purposes. The stronger the dependence of the load torque on speed (Eq. (3)), the smaller the speed drop (as now the load torque $\tau_L(\varphi_L)$ 'follows' closely the engine speed response) and the faster the final equilibrium is achieved. On the other hand, larger speed drops (originating from smaller values of exponent 's' in Eq. (3)) lead in more abrupt governor displacements and, consequently, higher fueling rates and cylinder pressures/gas torques; thus, greater crankshaft deformation values are observed, together with longer duration of the transient event.

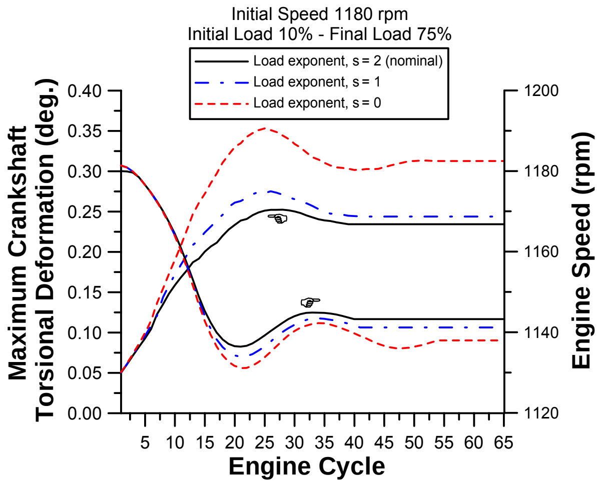

Figure 7: Effect of load type on the crankshaft torsional deformation, and the engine speed response during transient operation after a load increase

Finally, Fig. 8 presents the effect of the ratio of crank radius to connecting rod length, λ=r/L_{crod}, on the crankshaft torsional deformation. A lower value of λ (e.g. through longer connecting rod) results in higher values of the crankshaft deformation. A complicated mechanism lies behind this behavior. On the one hand, λ influences the slider-crank kinematics, i.e. displacement, velocity and acceleration of the piston [7,10]; on the other hand, the connecting rod length affects the inertia forces [7]. The combined effect of the above terms produces the results illustrated in Fig. 8. All in all, it is observed that ratio λ has only moderate effects on the crankshaft deformation.

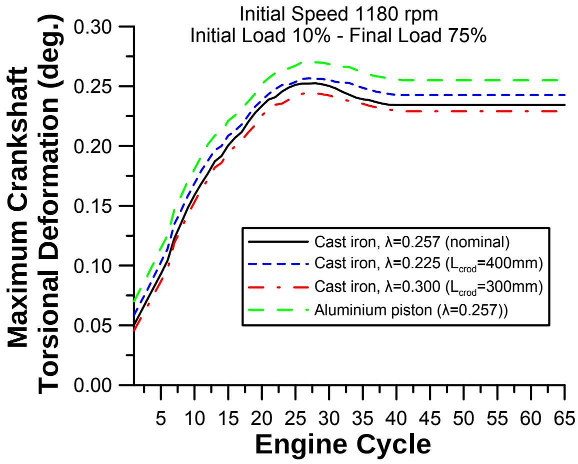

Figure 8: Effect of ratio λ and mass of piston on the crankshaft torsional deformation during transient operation after a load increase

Another interesting case is examined in Fig. 8, i.e. use of aluminium piston. Owing to the lower mass of aluminium compared to cast iron, the inertia forces developed are now lower. Consequently, the engine indicated torque is higher (Eq. (2)), resulting in slightly greater values of crankshaft deformation. The effect of inertia forces would be more pronounced for small spark ignition (car) engines operating at much higher rotational speeds than the current engine configuration (recall that inertia forces vary as the square of the engine speed [7,10]).

4. CONCLUSIONS

An experimentally validated simulation code developed has been used to study the effect of various dynamic parameters on the crankshaft torsional deformation of a turbocharged diesel engine operating under transient load conditions. To this aim detailed formulation of the crankshaft angular momentum equilibrium was accomplished, applying instantaneous values for engine, friction, load, stiffness, and damping torque contributions. From the analysis, the following were revealed for the present engine-load configuration:

- The magnitude of the applied load, the crankshaft stiffness and the total engine mass moment of inertia are the major parameters affecting the crankshaft deformation. The effect of another crankshaft property, i.e. damping, was found to be less prominent.
- The engine and the load mass moments of inertia exhibited conflicting behavior owing to the fact that they affect the engine angular momentum equilibrium in a different way.
- The load-time schedule influences strongly the profile and the peak values of maximum crankshaft deformation, with the final equilibrium conditions being more or less unaffected.
- The load type plays a significant role mainly when rigid loading is under study. Speed dependent loads are favorable for smaller crankshaft deformations owing to the lower fueling rates induced.
- The ratio of crank radius to connecting rod length was found to play a secondary role.
- Lighter piston construction increases crankshaft torsional deformation owing to the decrease in engine inertia torque.

All in all, the crankshaft deformation can assume significant values during transients, depending on the current engine-load configuration. This ultimately affects the stress that is experienced by the crankshaft being, thus, of great importance to the engine designer for safe engine operation.

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