PROBLEMS WITH DETERMINATION OF INSTANTANEOUS VALUES OF TORQUE GENERATED BY A COMBUSTION ENGINE

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Abstract

By its nature an internal combustion engine is a machine that transmits a variable torque to the power receiver. Engine of less cylinders performs greater variations of torque within the work cycle. Both value and direction of the torque change themselves. Especially the change of direction is important because of clearances in the drive chain. Definition of the specific tangential force, the quantity directly affecting the torque generated by the engine belongs to the basic tasks during engine design process. The specific tangential force graph is being computed ignoring vibrations in the engine–power receiver set. It is a quite substantial simplification which leads to results that could not be confirmed in a course of experiments. The following paper presents diagrams of the tangential force variations obtained in a course of measurements, calculations and computations taking into account the elasticity of drive chain.

Results of efforts described in this paper brought about the development of a measuring–computational method which allows to determine the engine generated torque with sufficient accuracy. It is possible to determine this torque on the ground of measured engine to a power receiver coupling torque and the instantaneous value of angular velocity of both engine shaft and the shaft of power receiver.

Keyword: *transport, internal combustion engine, vibrations*

1. Introduction

The course of torque transmited by the shaft connecting the engine to the power receiver is one of the most important signals determining the basic parameters of engine operation. It should be noted that the torque measurement is not an easy task in this case. Not mentioning the historical development of measuring methods one can conclude that at present the most reliable results could be achieved using torquemeters installed on the shaft. Such a torquemeter consists of a rotating shaft sector and a steady housing equipped with measuring wires. By the way, the best torquemeters are those in which the signal is transmitted telemetrically from the shaft to housing. Alas, such torquemeters belongs to the most expensive ones but the signal is not disturbed by the operation of connector system. Fig. 1 presents a torquemeter installed on the shaft coupling the engine to the power receiver.

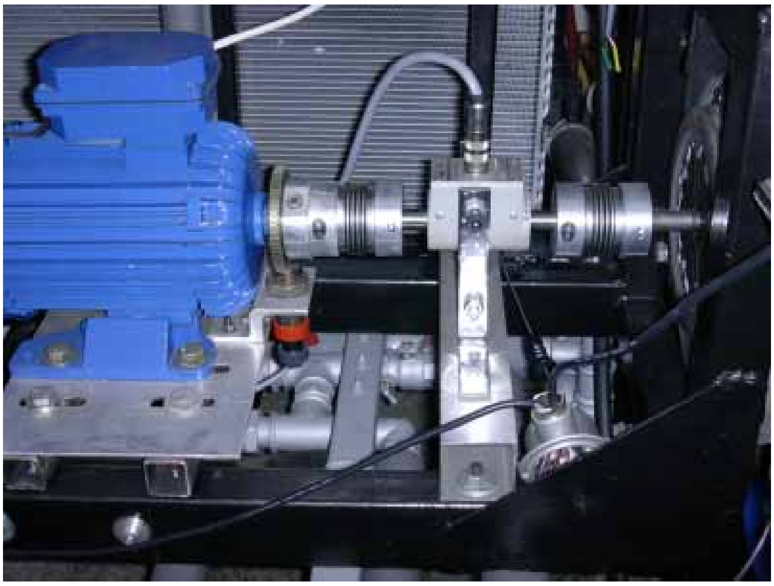

Fig. 1. Engine to power receiver coupling shaft equipped with torquemeter and two elastic joints

On the basis of specific tangential force course an instantaneous rotational speed can be easily determined digitally. The method of engine instantaneous rotational speed determination using the course of specific tangential force has been described in literature [1]. Just accomplished results should be mentioned here. In Fig. 2 the course of measured tangential force has been presented with the wider line while the narrower one presents the course of instantaneous rotational speed.

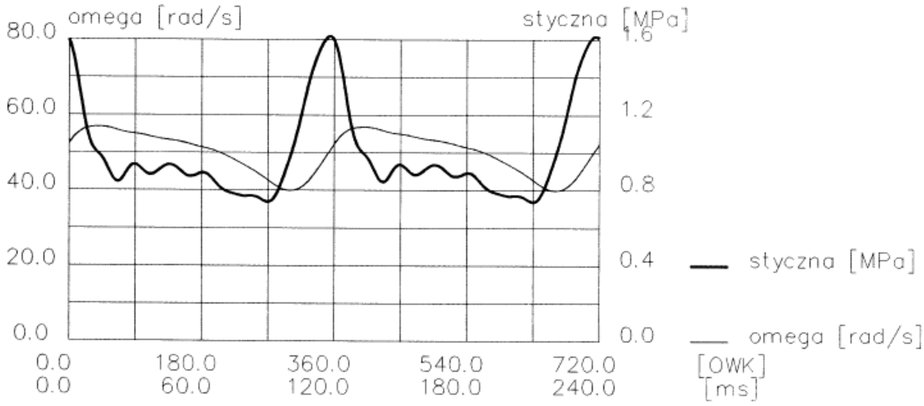

Fig. 2. Course of measured specific tangential force and resulting rotational speed within an individual work cycle of a 2-cylinder engine [1]

Though the example presented in Fig. 2 quite accurately projects the actual engine parameters, in a number of cases results are dissatisfying. Therefore a more accurate analysis of errors committed when determining the engine shaft instantaneous rotational speed on the basis of engine-to-power receiver torque becomes necessary.

As it turned out, a more precise analysis of error committed when attempting to define the instantaneous value of engine rotational speed on the ground of the coupling torque between the engine and power receiver becomes necessary. A pretty obvious phenomenon of engine – power receiver shaft vibrations makes that the course of engine torque is not the same as the course of coupling torque between engine and power receiver. The analysis of occurring phenomena requires a repetition of certain issues of elastic vibrations theory.

2. Results of engine – to – power receiver coupling torque computations

Computations of engine – to – power receiver coupling torque require formulation of suitable model. Unfortunately, those available in literature do not take into consideration the complexity of drive chain in the engine – power receiver assembly. The simplest case assumes the joining of engine shaft of Θ_1 inertia moment to a shaft of power receiver of Θ_2 inertia moment with a power shaft of e_1 yield. In this case the torque brought about by the natural vibrations of frequency:

$$\nu = \frac{1}{2\pi}\sqrt{\frac{1}{e_1 \frac{\Theta_1\Theta_2}{\Theta_1+\Theta_2}}}, \tag{1}$$

combines with the static torque.

As it can be noticed, even in the simplest case the freequency formula is quite complicated. This formula will be used for verification of the numerical model. In a real system vibrations are damped and omitting this phenomenon leads to the errors difficult to assess. In the simplest case the damping torque is proportional to the speed which leads to the equation of damped motion of the following form:

$$\Theta\ddot{\beta} + c\dot{\beta} + \frac{1}{e}\beta = 0, \tag{2}$$

where:

Θ – one mass system inertia moment,
β – angular displacement,
c – damping factor,
e – shaft yield.

Formulas (1) and (2) are the starting point to the theory of vibrations and they are usually used for solving various problems of this theory [2]. Alas, phenomena encountered in the engine – power receiver system are so complicated that the classical theory is not able to solve them. Analytical solutions presented in this paper will serve for the verification of proposed numerical method of engine – power receiver vibration parameters determination.

Equation (1) is one of the solutions of equation (2). Solution of any form of equation (2) in numerical notation reduces itself to the Procedure 1 written in the Pascal language.

Procedure 1

```
for i:=0 to iii do
      begin
           moms[i]:=(b[2,i]-b[1,i])/e[1];
           om[1,i+1]:=om[1,i]+
           (-tlum*(om[1,i]-om[2,i])+
           mom[i]+moms[i])
           /tet[1]*delt;
           om[2,i+1]:=om[2,i]-moms[i]/tet[2]*delt;
           b[1,i+1]:=b[1,i]+om[1,i+1]*delt;
           b[2,i+1]:=b[2,i]+om[2,i+1]*delt;
      {1}  if b[1,i]*b[1,i+1]<0 then
           begin
                ttt[kk]:=i;
                inc(kk)
      {2}  end;
           if (i=0) or (i=500) then writeln(b[1,i]:8:6,' ')
      end;
```

where:

i – steering variable,
moms – masses of Θ1 and Θ2 inertia coupling moment,
b – angle of mass revolution,
e – yield of shaft,
om – angular speed of masses,
tlum – damping factor denominated as c in Eq. (2),
mom – moment of external influence, eg. engine pistons,
delt – sampling time,
tet – Θ.

Time in which a mass relative torsion reaches 0, what allows to determine the vibration period T, is looked for in instructions from the range *{1}-{2}*.

Diagrams presented in Figs. 3 through 5 achieved for damping factors c equal to 0, 2 and 10 [Nms/rad] respectively, are drawn using Procedure 1.

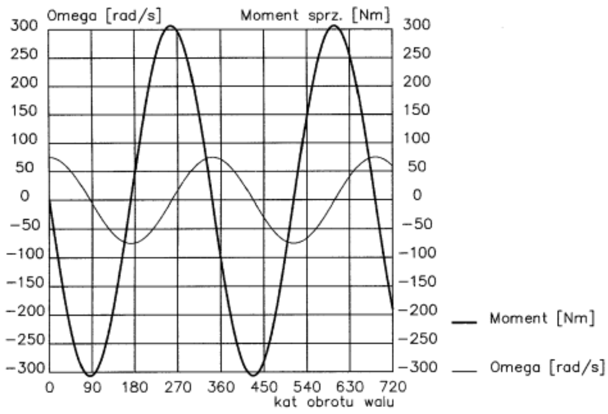

Fig. 3. The course of two vibrating masses relative angular speed– narrower line and the course of coupling torque between those masses – wider line; undamped vibrations

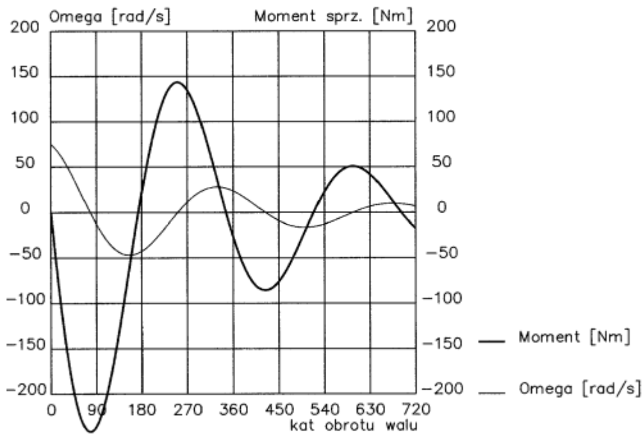

Fig. 4. The course of two vibrating masses relative angular speed– narrower line and the course of coupling torque between those masses – wider line; vibrations damped with intensity of c=2 [Nms/rad]

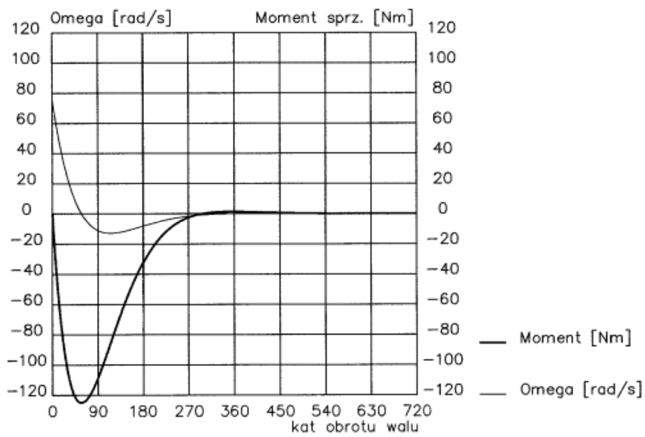

Fig. 5. The course of two vibrating masses relative angular speed– narrower line and the course of coupling torque between those masses – wider line; vibrations damped with intensity of c=10 [Nms/rad]

The period of natural vibrations calculated according to the example in Fig. 3 is 1.0260 [ms], while the value accomplished according to Eq. (1) is – 1.0252 [ms]. This mean, that the error of numerical method is basically insignificant. Obviously, as in any numerical method, the assumed time of sampling considerably influence the error. Diagrams in Figs. 3 to 5 have been accomplished for sampling time 3 [μs]. The inertia moments are 0.001 [kgm^2] and 0.002 [kgm^2], while the yield of coupling shaft is 0.00004 [rad/Nm].

The above presented simple example of vibrating system puts in order the basic knowledge and introduces damping to the vibrating systems of combustion engine, what is usually omitted when estimating natural frequency of vibrations [2]. A quite substantial error is committed in this case because as it has been calculated when preparing data for Fig. 4 the period of consecutive natural oscillations is being lengthen to 1.101 [ms], i.e. by about 8% relatively to undamped vibrations. For greater factors of damping – Fig. 5 – the idea of vibration period losses its sense.

3. The engine produced torque vs. power receiver coupling torque

The course of engine generated torque is a basic diagram plotted for the initial evaluation of designed engine. A question arises if measurements could prove correctness of engine torque courses presented in every handbook on internal combustion engines. Alas, the experimental practice gives the negative answer [3]. In order to clear the cause of discrepancies between theory and experimental practice a new method of determination the engine to power receiver coupling torque taking into consideration coupling shaft yield and possible damping factor will be presented.

Computations of engine to power receiver coupling carried out according to Procedure 1 gave results presented in Fig. 6..

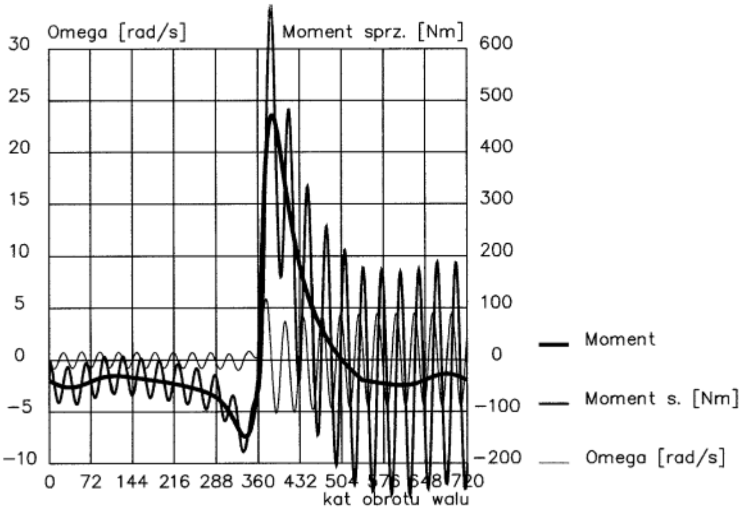

Fig. 6. Course of a single cylinder engine static torque – the widest line and the course of coupling torque between engine and power receiver – the line of medium width; the course of relative engine and power receiver speed has been additionally marked with the thinnest line in this figure

Despite the assumption that the engine has been coupled to the power receiver with a drive shaft of only 0.00001 [rad/Nm] yield, which corresponds to the shaft of 50 mm in diameter and 0.5 m long, the coupling torque substantially differs from the engine generated torque. Such strong disturbance of coupling torque with natural vibrations makes the course of this torque completely useless for the evaluation of engine generated instantaneous torque. In this context it is very advantageous to apply a torsional vibrations damper between those two shafts. As it comes out, application of damper of c constant equal to 10 [Nms/rad] causes the change in the course of coupling torque to the form presented in Fig. 7.

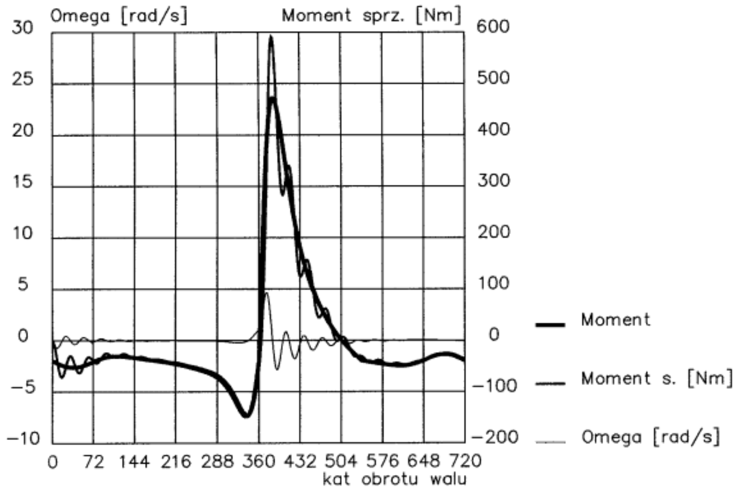

Fig. 7. Course of a single cylinder engine static torque – the widest line and the course of coupling torque between engine and power receiver – the line of medium width; the course of relative engine and power receiver speed when the damper of c=10 [Nms/rad] has been installed was additionally marked with the thinnest line in this figure

4. Summary and conclusions

Considerations presented in the paper highlight the difficulties one comes across when trying to use the couple torque between an engine and power receiver as the signal characterizing engine operation or diagnostic signal. The couple torque is commonly dependable to the high extent on random values of coupling shaft parameters. In the case of high yield shafts the couple torque losses its similarity to the courses of torque generated cyclicly by the reciprocating engine. However, certain conclusions can be withdrawn from the carried out simulations.

1. When designing the shaft coupling engine to the power receiver one should square requirements of avoiding the transmition of natural variability of engine generated torque with the limitation of torque vibrations caused by too high yield.
2. The torsional vibrations damper remains a very essential element of coupling system. Properly selected damper constrains the vibrations of coupling torque and converge the course of coupling torque to the torque generated by engine.
3. The engine to power receiver coupling torque close to the engine generated torque can be used for the analysis of various aspects of engine operation including definition of mechanical loss range that influences the engine total efficiency.

Results of efforts described in this paper brought about the development of a measuring–computational method which allows to determine the engine generated torque with sufficient accuracy. It is possible to determine this torque on the ground of measured engine to a power receiver coupling torque and the instantaneous value of angular velocity of both engine shaft and the shaft of power receiver [4].

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