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P.A.R.I.S. : Pendulum Acyclism Reducer Integrated System

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Evolution of automotive diesel engines has led to the production of smaller and smaller engines with increasingly higher torque. The reduction in external dimensions means smaller flywheels, and the increase of torque rates, higher cyclic irregularities that today's flywheels even with a dual mass techniques are no longer able to cope with.
Because of this, "damper" type absorber devices which control torque vibrations due to cyclic irregularities are no longer efficient to decrease rotating vibrations to an acceptable level.
The best solution to cope with these problems is a tuned rotating pendulum : P.A.R.I.S, (Pendulum Acyclism Reducer Integrated System). This vibration absorber system can be easily integrated into a standard engine flywheel.
P.AR.I.S. systems show a very high efficiency level. Normally, in order to decrease the 2nd harmonic level (mainly responsible for cyclic irregularities in 4 strokes 4 cylinders engines) to 60 %, a standard flywheel would have to have around 15 times its actual mass.
The mean torque level is not affected by P.A.R.I.S., only the instantaneous torque is counterbalanced by the pendulum.
In addition, the P.A.R.I.S. system device has the great advantage of being independent of engine speed. It is efficacious throughout the whole range of speed, as compared to damper vibration absorbers, which are tuned for a specific frequency.
On the other hand, P.A.R.I.S is not adapted for the whole harmonic range generated by the engine, but only for those harmonics it has been tuned for.
Another positive aspect of this solution is that it is a full mechanical system, needing no electronic controls, which consumes hardly any energy and these due only to small friction losses.
Application examples targeted by this patented system are :
torque regulation, due to reduction of cyclic irregularities in order to achieve smaller vibration levels,
idle speed decrease for consumption and pollutant reductions,
in camshaft applications - increase of the maximum rotating speed.

Keywords: Acyclism, torsional vibrations, pendulum.

INTRODUCTION

First appearance of vibration absorbers can be observed as early as 1929 by Kutzbach using fluid masses contained in U shaped channels. A patent in 1929 by Duesenberg describes the use of metal capsules partly filled with heavy fluid. Many forms of pendulum assembly were proposed and patented during the decade 1930 to 1940, mostly with solid pendulum masses. Over all of them, we can quote :

- Roll-form (Carter in 1929, Salomon in 1932),
- Bifilar link form (Sarazin in 1930),
- Ring-form (Sarazin in 1933),
- Bifilar suspension (Sarazin in 1935),
- Duplex suspension (Salomon in 1938).

Applications can be found since 1934 mainly in aeronautical engines to reduce torsional and flexural vibration stresses in crankshaft and vibratory stresses in airscrew blades, but also in in-line production for high and medium speed (Sulzer) both 2 and 4-stroke cycle types with either spark or compression ignition. Special arrangements were also developed for preventing the transmission of vibration from one part to another (from an engine crankshaft to the gearbox or the geared installation), reducing camshaft vibration (Ranger). Most of the cases were using pendulum in crankweb counterweight or in flyweel.

Even it was recognized that pendulum absorber was the most efficient device, its popularity began to decline during the latter part of the1950s, largely due to the successful development of a viscous damper using silicone fluid as a damping medium which could be used on all size of engine for smaller cost.

A different application is proposed, due to the increase of torque with decrease of engine size : reduction of engine acyclism. Thanks to this effect, improvements in torque regulation (because of cyclic irregularities reduction) could be achieved as well as reductions of vibrations levels of the car and idle speed for fuel consumption and pollutant decreases. This patented utilization of rotating pendulum vibration absorber (patents # 98 08826 and 99 06625) will be developed in this paper.

WAYS TO DECREASE ACYCLISM

Acyclism for a rotating speed, is inversely proportional to its mass moment of inertia and proportional to the amplitude of the harmonic torque which is the cause.

Angular acceleration of a rotor, at constant average rotational speed is proportional to its acyclism.

To decrease this angular acceleration there are only two ways :

- Decrease the amplitude of the external harmonic torque,
- Increase the moment of inertia.

Amplitude of the harmonic torque is linked to the dynamic of parts associated to the rotor and to the thermodynamic cycle.

Increase of the moment of inertia of the flywheel

An increase in flywheel dimensions would lead to an increase of its mass to achieve enough inertia.

Tuned vibration absorber

A system tuned on a frequency would allow to increase the apparent flywheel mass moment of inertia for the pulsation $q\Omega$ (Ω rotor average speed). If we consider the above model (Figure 1) in which, the disc 1, of inertia moment I_1, represents the rotor and its assembling parts. The disc 2, of inertia moment I_2 associated with the torsional spring of torsional rigidity k, represents the tuned vibration absorber.

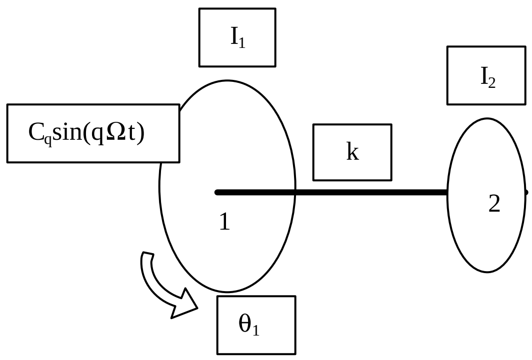

Figure 1 : 2 masses system

If we choose I_2 and k so that $w = \sqrt{\frac{k}{I_2}}$, then the amplitude θ_1 of disc 1 is null when applying an external harmonic torque $C_q\sin(q\Omega t)$.

The inconvenient of such system is that it is efficient only for one rotational speed Ω of the rotor. However, this system is efficient to decrease amplitudes of one vibration mode whose pulsation is independent of the rotational speed (principle of the vibration absorber type Damper).

Rotating pendulum vibration absorber

The only system that keeps its tuning for all rotating speeds is a simple rotating pendulum consisting of a weight (kg) assumed to be concentrated at the center of gravity of the bob, oscillating under the influence of gravity about the suspension point on a pendulum arm of length L (m), hinged to an arm of length R (m) which is rotating at constant angular velocity Ω (rad/s) as indicated in the following Figure 2.

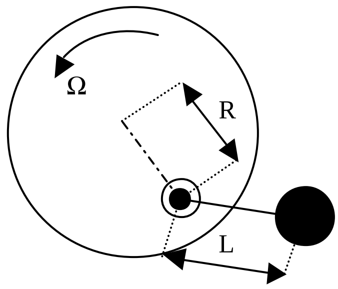

Figure 2 : Rotating pendulum

The phase velocity of natural vibration of this rotatingpendulum, is given by :

$$w = \sqrt{\frac{\Omega^2 R}{L}}$$

We assume the gravity acceleration negligible compared with the centrifugal acceleration Ω^2R.

When the carrier is solicited by an externally harmonic torque $C_q\sin(q\Omega t)$, pendulum can be tuned so that :

$$w = q\Omega$$

Thus : $$q\Omega = \sqrt{\frac{\Omega^2 R}{L}}$$

This tuning is realized in relation to the external harmo nic torque when : $q = \sqrt{\frac{R}{L}}$

Unlike to the system inertia-torsional rigidity, described before, the tuning of the pendulum for the harmonic excitation of order q is independent of the rotational speed.

When the pendulum is perfectly tuned to an harmonic order number q, the amplitude of the supporting carrier is null when solicited by a torque $C_q\sin(q\Omega t)$. Such a system will act as if the mass moment of inertia of the carrier and the pendulum became infinite and will render the system non responsive to the externally applied torque. When this condition is fulfilled, the pendulum mass automatically attains the amplitude and phase necessary for producing a torque which counteracts the externally applied torque.

This is effective only when the amplitudes of oscillation of the pendulum, in comparison with the support carrier, are small (only few degrees) to respect the hypothesis of small displacements which allow to give the natural pulsation expression.

This tuned system seems particularly well adapted to the given problem.

Physical explanation of rotating pendulum vibration absorber functioning

When the carrier, rotating at average speed Ω, without pendulum is submitted to an external harmonic torque $C_q\sin(q\Omega t)$, the rotating angle of the carrier is :

$$\alpha = at + a_0\sin(q\Omega t)$$ (where a_0 represent the acyclism).

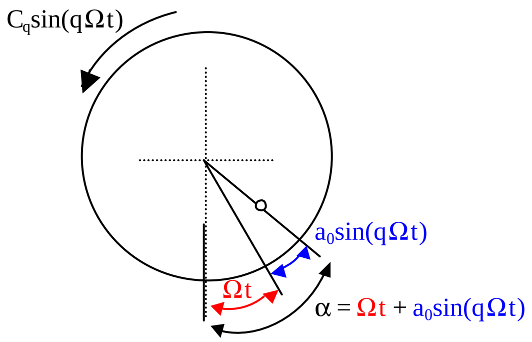

Figure 3 : Carrier angular positions

When the carrier submitted to the same external harmonic torque is supporting a pendulum, then the rotating angle of the carrier is :

$$\alpha = at + a\sin(q\Omega t).$$

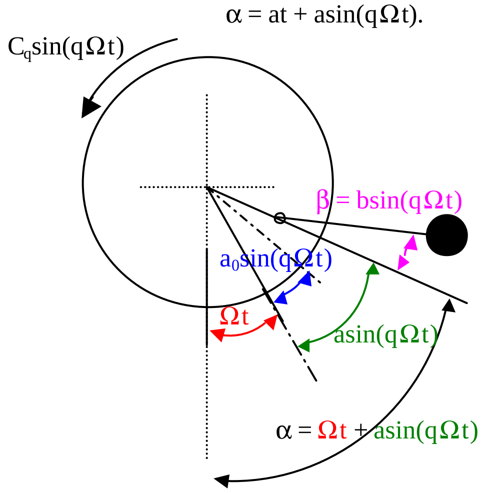

Figure 4 : Carrier + pendulum angular positions

When the amplitude "a" of the carrier is null, the sum of the moment about the axis of rotation of the efforts applied by the pendulum is equal and opposed to the external harmonic torque $C_q\sin(q\Omega t)$.

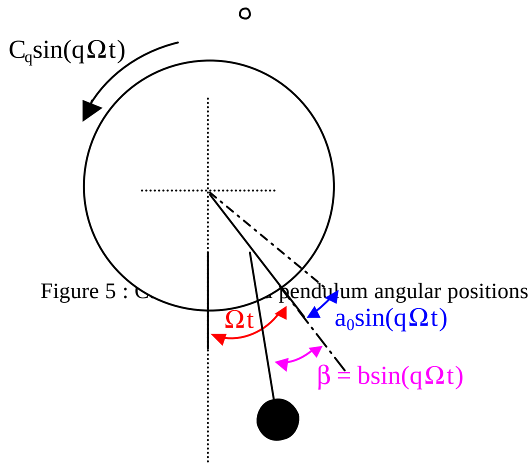

Figure 5 : C… pendulum angular positions

The moments of the efforts generated by the quantities of acceleration dues to the movement of the pendulum are equals to the external harmonic torque.

We can note that the angular rotation "b" of the pendulum presents a phase difference of π with the harmonic torque.

The pendulum acts as a carrier linked to the support by a torsional spring as indicated on the following Figure 6. The relative displacement of the carrier representing the pendulum regarding to the support rotating at the constant speed Ω is equal to "-a".

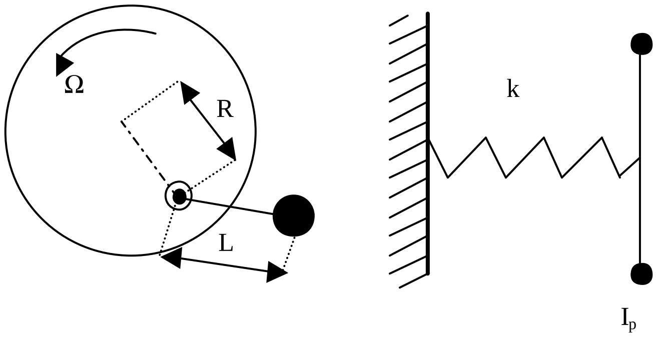

Figure 6 : Torsional analogy

Starting from this analogy, motion of pendulum has been put in equations ([1] and [2]).

SOFTWARE APPLICATION

According to the equations describing the motion of the pendulum ([1] & [2]), a SIMULINK application has been developed to help designer to dimension pendulum vibration absorbers. As usual, programming has been done by blocs (as it can be seen on the figure 7) :

- flywheel cinematic bloc : allows calculation of angle, speed and acceleration of the flywheel,
- pendulum cinematic bloc : allows instantaneous acceleration of the pendulum and by integration speed and angular position,
- resulting torque bloc : pendulum restoring torque then resulting torque are calculated,
- comparison speeds bloc : allows visualization of the pendulum effect on rotating velocity.

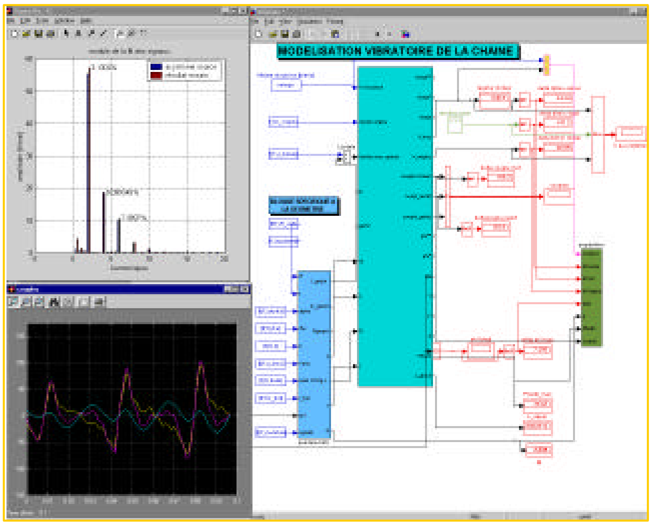

Figure 7 : Software application

A comparison between the calculation and the measurements in the case of a bifilar suspension pendulum absorber can be seen in the figure 8 (for acyclism) and 9 (for signals FFT modulus). The original flywheel's acyclism (which is used as imput data for calculation of the excitation's torque) is also represented.

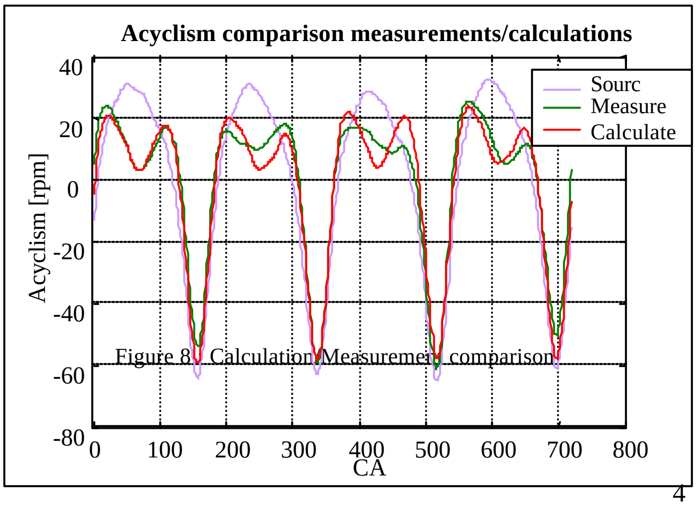

Figure 8 : Calculation/Measurement comparison

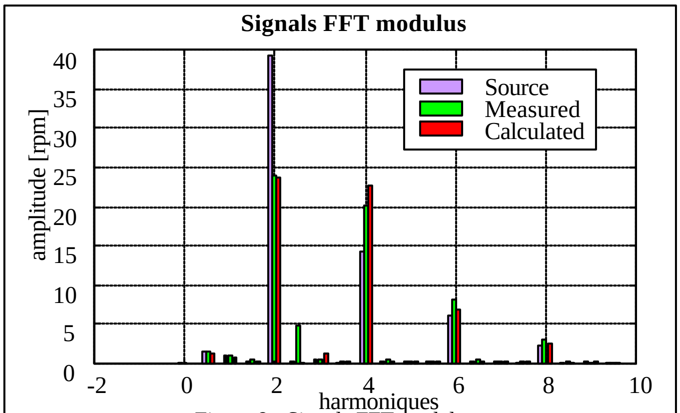

Figure 9 : Signals FFT modulus

The effect of the pendulum on the resulting torque can be seen in Figure 10 :

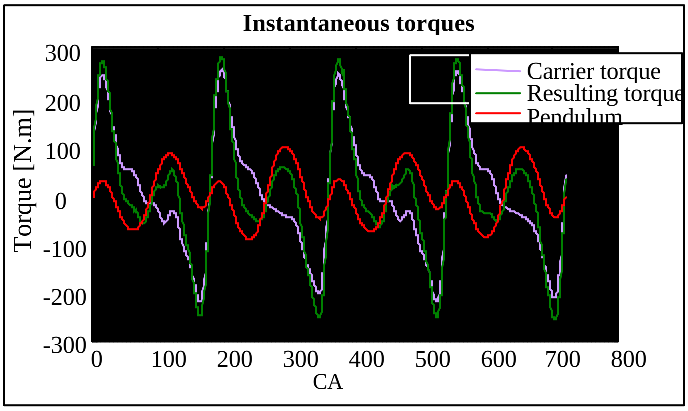

Figure 10 : Instantaneous calculated torques

Under hypothesis of small displacements, simulations gave correlations good enough with measurements to obtain design tool.

DESIGN

Nowadays, trends in cars applications leads to a reduction of the dimensions of the engine associated with an increase of output torques. Acyclism, as previously explained, follow the same tendency than the torque and the inertia (which decrease with the external dimensions of the engine) and began to be a problem that Dampers can not solve. Rotating pendulum absorbers have more chance of success in that battle against acyclism. The implantation of such device is mainly, in cars applications, located in the flywheel. The main stresses for this implantation are :

- Use of significant masses,
- Smallness of flywheel,
- Wear,
- Noise,
- Cost.

Indeed, the excitating torque leads to utilization of significant masses to counteracts him, the location of pendulum in the crankweb of crankshaft are not realistic for a non engine manufacturer company, so the implantation in the flywheel is the most realistic solution Thus, the device have to keep the same external dimension than the standard one to match with the gearbox and the clutch mechanism. The flywheel equipped with pendulum must have the engine life duration (no maintenance on the standard). The noise is also of great importance in passenger appreciation and during stop and go pendulum must be noiseless. The additional cost must be as low as possible and for this machining of the part must be as easy as possible.

D2T has developed experience in design and integration of rotating pendulum vibration absorbers matching the technical specifications described before, by means of the utilization of simulation tool functioning in closed loop with tests on engine.

TESTS RESULTS

Tests have been conducted upon a 4 cylinder 1.5L automobile diesel engine with several arrangements [3]. The tuning has been focussed upon the harmonic 2, mainly responsible of the acyclism in this configuration. At this phase of the development, only tests in test bed have been realized.

Results obtained, have in, a first step, helped to adjust the simulation tool and in a second step, validated design choices to improve the potential of acyclism reduction offered by rotating pendulum vibration absorbers and to adapt it to the engine environment stresses.

Reductions can achieve up to 60% on harmonic 2 providing 40% upon the acyclism in certain functioning conditions. However, good performances obtained, only a vehicle test will represent the real services conditions and is still to achieve.

The following figures 10 and 11 shows an example of the results obtained with the pendulum at idling.

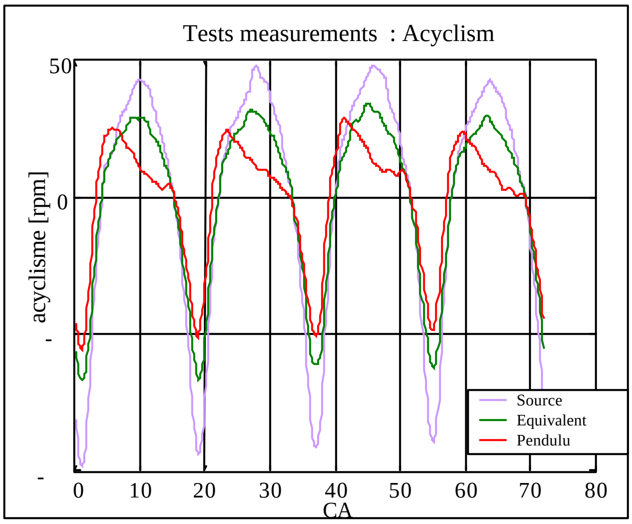

Figure 10 : Acyclism results

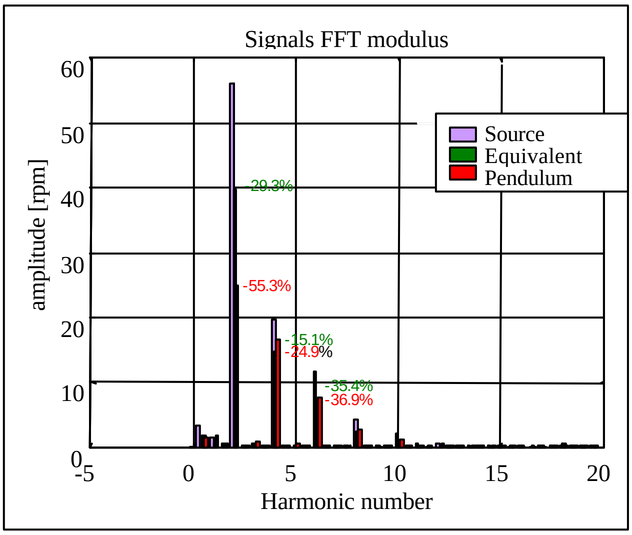

Figure 11 : Signals FFT modulus

CONCLUSION

Rotating pendulum vibration absorbers have been experimented at D2T to reduce acyclism of automobile engines.

The simulation tool developed, correlated with tests, has been of great help in arrangements design to match with the severe environmental services conditions of engine.

Results obtained give good perspectives for complete integration of such device on vehicles.

Different improvements are still in study to reduce size, cost and increase of efficiency.

Further developments will concern complete adaptation on vehicle to test the device in real service conditions.

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