Multi-Body Dynamic Modeling of the Expected Performance of Accelerated Pavement Testing Facilities

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ABSTRACT

Accelerated Pavement Testing (APT) facilities are nowadays considered fundamental for the thorough understanding of the performance of pavements. The amount of information that can be derived from APT investigations can serve as the basis for a more performance-related pavement design but also for the development of new pavement types and innovative materials.

In the design and use of such facilities, special care should be placed in the modeling of the loading systems which are employed to produce accelerated damage. Such an analysis is important both in the preliminary design phase, in which technological solutions are found in order to simulate the effects of heavy vehicles on the pavement, and in the various phases of investigation, when the recorded damage has to be clearly related with effective loading conditions. In this respect, modeling can be a valuable support to the evaluation of the data which can be acquired by means of a proper instrumentation of the facility.

In order to address such issues, the Authors have developed a design procedure which together with typical stationary calculations includes the adoption of Multi-Body (MB) and Finite Elements (FE) models of the testing system. As proven by the first implementation exercises of the design procedure, the use of MB and FE models allows the evaluation of the dynamics of the system in a wide variety of testing conditions. Thus, stresses and strains in the structure of the APT facility can be estimated and the dynamic forces and torques which arise during testing at the tire-pavement interface can be predicted.

Given the width of the design problem, this paper gives only a general overview of the structure of the proposed procedure, with its application to a specific case which has been studied in depth. Refinements are still under way and will hopefully yield a set of modeling methods which in the future will be available for the design of new APT facilities and for the assessment of the performance of existing systems.

INTRODUCTION

The design and construction of long-lasting pavements is one of the goals which highway engineers have continuously tried to reach for centuries. Such an effort has required experimental and theoretical research which in time has increased its level of complexity: while initial studies were essentially based on the observation of pavement sections subjected to traffic and on the index-characterization of the materials used in construction, current investigations are based on a more detailed description of material behavior and on the modeling of pavement performance under controlled or monitored loading conditions. In such a context, Accelerated Pavement Testing (APT) facilities have proven to be extremely powerful research tools, since they can simulate long-term effects of traffic loading in a relatively short time period, with the corresponding evaluation of the progressive changes in pavement response and distress. Quite obviously, the accelerated character of this kind of testing is a key element for the development of projects focused on the design and modeling of perpetual pavements, conceived in such a way to extend service life well beyond the levels associated to standard pavement structures.

APT programs are nowadays active worldwide and address almost all relevant aspects of pavement engineering. Valuable sources of information with respect to past achievements and future perspectives are provided by TRB Committee AFD40 (formerly A2B09) and by the European COST-Transport action COST 347.

When compared with the original prototypes developed at the beginning of the 20th century [1,2], current APT devices are quite complex, with a wide array of operative options and technological refinements which are implemented in order to maximize the simulation of the effects caused by actual heavy vehicles and/or aircraft landing gears. Moreover, whether of the linear of circular type, all APT systems are designed in such a way to make their use economically feasible, with a convenient limitation of energy consumption and testing time. In many cases research institutions have developed their own testing prototypes, conceived to suit specific needs; however, in other cases they have preferred to adopt systems already used in other countries which to some extent could be considered as standard devices with a rich set of test data available for comparative purposes.

In the Politecnico di Torino, the whole issue of vehicle-road interaction has been recently approached by combining the efforts and resources of the Pavement Engineering and Vehicle Dynamics research groups. Topics of interest which have been treated both experimentally and theoretically range from the evaluation of the effects caused by pavement roughness to the dynamic design of geometrical road elements. More recently, the joint research group has also started working on the design of APT facilities which is in fact a theme at the boundary between pavement and vehicle engineering: as illustrated in this paper, this has been done by developing a design procedure which together with typical stationary calculations includes the adoption of Multi-Body (MB) and Finite Elements (FE) models of the testing system.

As proven by the first implementation exercises of the design procedure, the use of MB and FE models allows the evaluation of the dynamics of the system in a wide variety of testing conditions. Thus, stresses and strains in the structure of the APT facility can be estimated and the dynamic forces and torques which arise during testing at the tire-pavement interface can be predicted. This second aspect of analysis is especially important when considering side and longitudinal forces, the importance of which is generally overlooked in the design of APT facilities even though it has been clearly recognized that they may greatly affect both the interaction between the components of the test system and the development of structural and functional distresses in the pavement. As shown in the design example illustrated in this paper, such a limitation can be overcome by means of an adequate modeling of the APT system: in fact, its results provide the basis for the definition of the location and type of on-board sensors used to measure shear forces applied to the pavement and of methods to be employed for their active control.

The development and use of the proposed design procedure requires a deep knowledge of tire and vehicle dynamics and should be supplemented by direct testing in order to make realistic assumptions of the many structure and materials parameters utilized in modeling. Given the width of the design problem, this paper gives only a general overview of the structure of the proposed procedure, with its application to a specific case which has been studied in depth. Refinements are still under way and will hopefully yield a set of modeling methods which in the future will be available for the design of new APT facilities and for the assessment of the performance of existing systems.

DESIGN PROCEDURE

Figure 1 gives a schematic illustration of the design procedure which is being developed by the Pavement Engineering and Vehicle Dynamics research groups.

The first step in design consists in performing the basic static calculations which are necessary to establish the main dimensions of the structure. The basic parameters of the APT device (circular or linear layout, engine power, number of wheels of the trolleys, static load acting on the wheels, etc.) have to be chosen at this step. As better explained in the following paragraphs, the adoption of an MB model in this first step of design would be too expensive in terms of time, due to the large number of uncertain parameters which should be evaluated and modified. In fact, an MB model considers the inertia of all the main components, which are connected by elements characterized by their own stiffness and damping coefficients.

After the definition of a first approximation layout, it is possible to implement the MB model of the APT facility. First of all, in semi-stationary conditions, it is necessary to make a comparison between the results of the static calculations and the output of the model, in order to assess its validity. Then it is possible to carry out an analysis of the behavior of the system in dynamic conditions, for example when the pavement surface is not perfectly flat, to investigate the forces and the torques acting at the tire-pavement interface and between the different components of the testing facility. The MB model can also be useful to optimize the structural design of the APT system and, as mentioned previously, to decide how to locate the sensors on its moving parts in order to measure or estimate the dynamic tire-pavement forces.

Finally, for an in-depth design of the structure of the APT facility, it is necessary to set up an FE model of the machine and to make it run in co-simulation with the MB model. This type of modeling is quite complex and should be performed on the final layout of the facility; the corresponding results can be used to better understand the performance of the APT system and to provide a continuous feedback to measurements carried out during testing.

This paper presents the results obtained in the design steps of Figure 1 which correspond to the static calculations and to the implementation and use of the MB model. The calculation examples refer to the specific case of a rotational APT system which has been thoroughly examined to test the applicability of the procedure and which could be actually built by employing the indicated technological components.

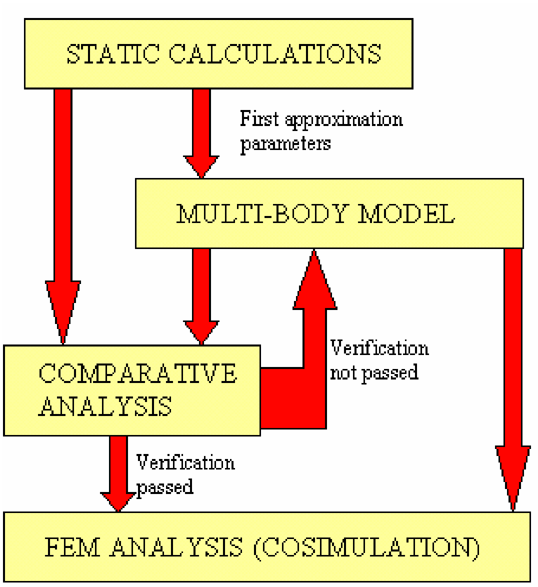

FIGURE 1 Flux diagram of the design procedure.

Preliminary Assumptions

The first choice for the designer of an APT facility consists in defining the configuration of the system, which can be either of the rotational or linear type. Both configurations have several advantages and disadvantages, so the choice should be made based on a balanced evaluation of a number of factors which may have different weights depending upon the intended use of the facility.

The linear configuration has the advantage of occupying a small area; furthermore, it is usually transportable and can be located in different sites depending upon specific needs. However, it has the limitation of permitting only small values of longitudinal speed of the loading trolleys. The inertial force necessary to obtain a speed of 40-50 km/h with an acceptable (not too large) spatial length of the test facility is too high both in terms of power required by the system of propulsion and of strains exerted on the structure.

The rotational configuration permits a very slow acceleration of the testing equipment (only during transients, necessary to make the machine start and stop) without significant dynamic forces or torques on the structure and on the system of propulsion. The power system of the testing facility can be designed as a function of the static load acting on the trolleys of the testing equipment.

Taking into account the observations illustrated above, the design procedure was applied to the case of a rotational APT facility, with a loading system composed of four trolleys connected by means of rigid horizontal arms to a rotating central shaft (Figure 2). In order to avoid the superposition of dynamic loading effects on the pavement, it was hypothesized that the trolleys should have a single axle configuration (Figure 3) with the possibility of using super single tires. Such a choice also allows a more simple modeling of pavement damage, which according to traditional approaches is always referred to single axles.

FIGURE 2 Sketch of the APT facility considered in the study.

For a system of the type indicated in Figure 2, the connection between the arms and the trailers should be made by employing ball recirculation bearings, characterized by a low friction. Their function is to prevent the rotation of each trolley around the axis of the wheels and to transmit the horizontal force necessary to make the system rotate along a circular trajectory.

In the preliminary phase of design some hypotheses had also to be made on the type of loading trolleys. Therefore, it was assumed that commercial suspensions, such as the one shown in Figure 3, should be utilized: this allows the dynamics of the loads to be very similar to the typical behavior of a heavy truck. The system represented in Figure 3 is a trailing arms suspension, with an integrated steering system based on a hydraulic power circuit. It can be equipped with a system of hydraulic springs, which can be used for the dynamic variation of the load acting on the pavement.

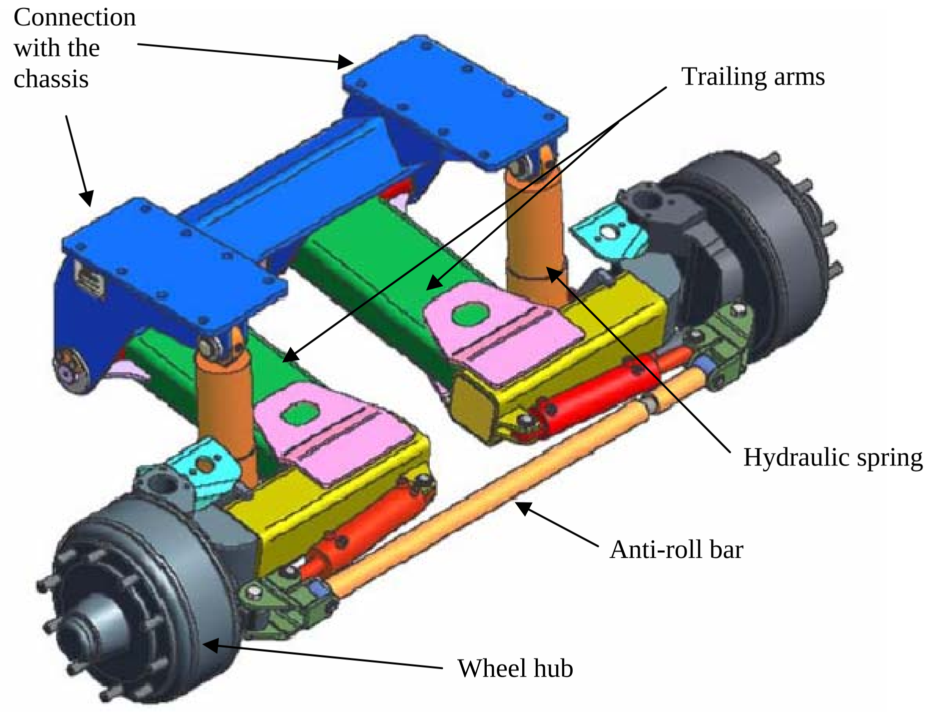

FIGURE 3 Sketch of the suspension system of the trolley.

With respect to the maximum static load acting on each trolley, in the calculations reference was made to a value of 120 kN, which can be obtained by means of a system of steel blocks piled on the each individual chassis (Figure 4). It is important to highlight the fact that by adopting the abovementioned solution for the trolley-arm connection, the vertical load, even in dynamic conditions, is entirely sustained by the tires and not by the bearings which connect the structure of the APT facility to the trolleys.

For the configuration schematically shown in Figure 4, centrifugal forces, which are generated by the motion of the trolleys, are absorbed by the lever arms (through the bearings) and not by the tires of the moving trolley. Only the sideslip angles due to the steering angle of the wheels give origin to a force which is transmitted to the structure of the trolley (suspension and chassis).

It has to be underlined that a commercial trailer for a truck can sustain a maximum level of lateral acceleration equal to 0.5g. A typical configuration (for example, with a diameter of 20 m) of a circular APT facility can easily give origin to a level of lateral acceleration greater than 1.5g, which corresponds to the force absorbed by the arms of the machine (and not by the suspensions). As a consequence, in designing the facility it is possible to refer to a level of lateral acceleration of the trolleys which exceeds the maximum value which it could have on the road. However, care should be taken in checking that the imposed steering angle does not give origin to a force which exceeds the structural resistance of the trolley.

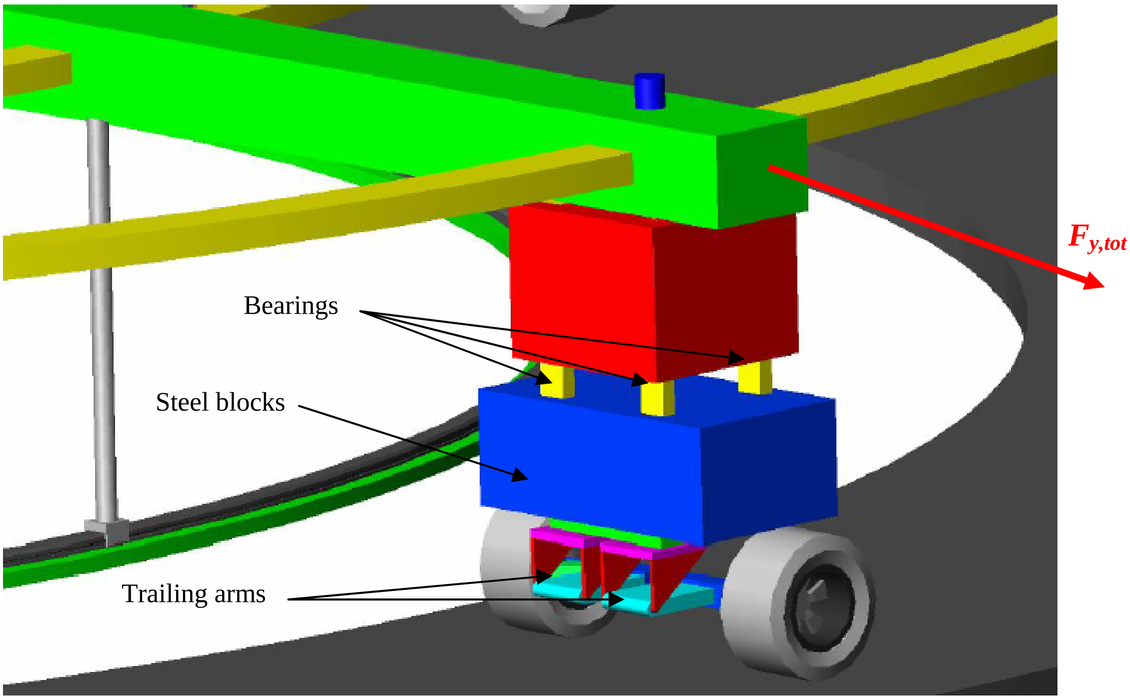

FIGURE 4 Sketch of a trolley and its connection to a lever arm.

Static Calculations

The first point to be addressed through static calculations is the evaluation of the forces acting along the axis of the arms of the testing machine ($F_{y,tot}$). These can be calculated as the sum of two components, $F_{y,\alpha}$ and $F_{y,centrifugal}$, respectively due to the sideslip angle α of the tires of the trolley [3] and to the centrifugal effects due to the radius of curvature of the trajectory of the trolleys during their motion around the pivot axis of the facility.

$$F_{y,tot} = F_{y,\alpha} + F_{y,centrifugal} \quad (1)$$

The equations necessary for the evaluation of the two components are the following:

$$F_{y,\alpha} = C_\alpha \cdot \alpha \quad (2)$$

$$F_{y,centrifugal} = ma_y = mV^2/R \quad (3)$$

where:

C_α is the equivalent cornering stiffness of the axle of the trolley (considered constant for low values of α);

m is the mass of the trailer, a_y is its lateral acceleration, V is its speed and R is the radius of trajectory.

It can be observed that since small values of the radius R imply large strains for the arms of the test machine, the choice of the size of the APT facility is also directly connected to structural issues. It should also be underlined that sideslip angle effects can be due both to the motion of the tie rod, which can be used to increase in controlled conditions the shear forces acting on the pavement, and to the elasto-kinematical behavior of the suspensions. Finally, the value of

$F_{y,tot}$ should be considered for a first estimate of the bending torque acting on the arms of the APT facility.

Another very important issue which has to be treated by means of static calculations is the evaluation of the power of the motor adopted to make the system rotate. This can be either an electrical motor or an internal combustion engine, but in any case it has to be correctly chosen to guarantee the expected performance of the APT facility.

First of all, it is necessary to consider the power contribution (for each trolley) due to rolling resistance. The following equation can be therefore used:

$$P_{rolling\_resistance} = W_{TOT} \cdot V \cdot (f_0 + KV^2) \quad (4)$$

where:

W_{TOT} is the vertical load acting on the axle of the trolley;

f_0 and K are the speed-independent and speed-dependent contributions to the rolling resistance coefficient of the tires.

Secondly, the power due to the aerodynamic resistance is considered according to the expression:

$$P_{aerodynamic} = 0.5 \cdot \rho \cdot S \cdot C_x \cdot V^3 \quad (5)$$

where:

ρ is air density, S is the front surface of the trolley, C_x is the aerodynamic drag coefficient.

Finally, additional power contributions should be calculated to take into account the effects due to tire sideslip angle and to the presence of longitudinal forces acting at the tire-pavement interface while running at constant speed. The corresponding equations are the following:

$$P_\alpha = C_\alpha \alpha^2 V \quad (6)$$

$$P_s = C_s s V \quad (7)$$

where:

C_s is axle equivalent longitudinal stiffness [3] and s is longitudinal slip.

The total power required to make a single trolley move at a constant speed V is therefore given by the following expression:

$$P_{tot} = P_{rolling\_resistance} + P_{aerodynamic} + P_\alpha + P_s \quad (8)$$

The different contributions in terms of power necessary for the motion of the system as a function of the speed of the trolleys can be observed in Figure 5, in the case of absence of longitudinal and lateral slips (P_α and P_s equal to zero). The static calculations demonstrate that the effect of the spin of the tires due to the curvature of the trajectory can be neglected. The contribution due to tire rolling resistance is prevalent on the contribution due to aerodynamics. By comparing equations (6) and (7), it is clear that, for reasonable values of α and s, the contribution of (6) is coherent with a value of power of the motor of the APT facility of some hundreds of kW, whereas contributions (7) easily exceed the power limit of existing motors. This suggests that it is economically feasible to design an APT system in which sideslip angles are imposed during testing in order to increase shear forces transferred to the pavement surface: this can be done through the controlled motion of the tie rods which can be hand-actuated before the beginning of the test or electrically or hydraulically actuated during the test. However, active control of longitudinal forces is not easily obtainable when operating in constant speed conditions.

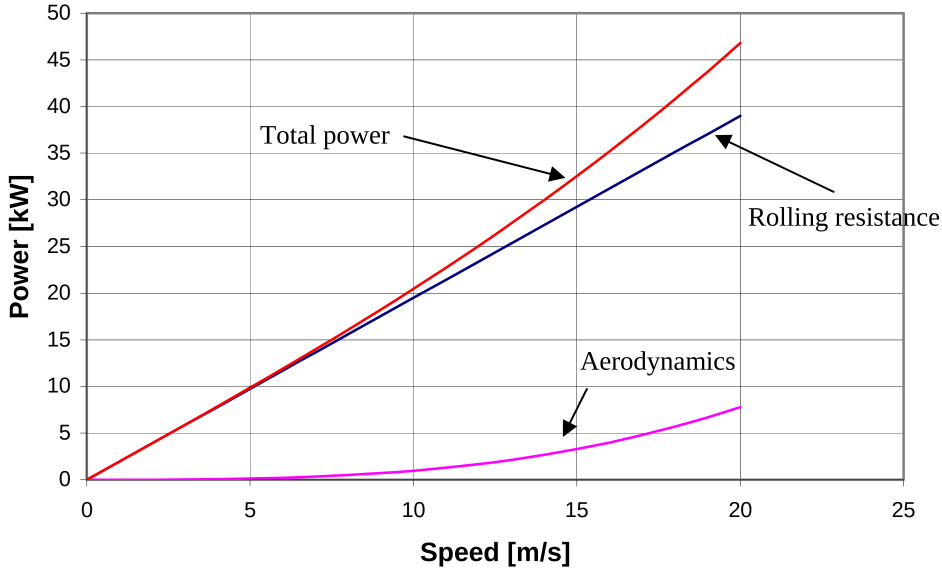

FIGURE 5 Power necessary for the motion of a trolley of the APT facility.

After choosing the motor and the driveline of the facility, it is necessary, through static calculations, to define the regulation diagrams of the system, fundamental both during the design process and for the final use of the APT facility in order to have the best performance of the system.

For example, it is useful to compute the maximum value of sideslip angle (α_{max}) as a function of motor power. This can be done by referring to the energy balance of the system which translates in the following equation:

$$\alpha_{\max} = \sqrt{\frac{\eta_t P_{motor} - P_{4,tot}(V)}{4C_\alpha V}} \qquad (9)$$

where:

η_t is the efficiency of the driveline of the facility, P_{motor} is the power of the adopted motor unit and $P_{4,tot}$ is the power contribution comprehensive of (4), (5) and (7) for the four trolleys.

Figure 6 is an example of such a kind of regulation diagram in which the longitudinal slip contribution has not been considered. The design target of a sideslip angle of 3° can be reached at a maximum speed of about 13 m/s, but much larger values of sideslip angle can be obtained at lower speeds.

In a similar way, it is possible to compute the maximum speed V_{MAX} of the trailers (having a total weight equal to W) of the testing facility as a function of sideslip angle, thereby obtaining the regulation diagram shown in Figure 7. In this case, the following equations, which do not consider tire longitudinal slip, can be employed:

$$\eta_t P_{motor} = AV + BV^3 \qquad (10)$$

where:

$$A = W \cdot f_0 + 4C_\alpha \alpha^2 \tag{11}$$

$$B = WK + 2\rho SC_x \tag{12}$$

To obtain V_{MAX}, it is possible to adopt Cardano's rule [4]:

$$V_{MAX}(\alpha) = A^* \left(\sqrt[3]{B^* + 1} - \sqrt[3]{B^* - 1} \right) \tag{13}$$

where:

$$A^* = \sqrt[3]{\frac{\eta_t P_{motor}}{2B}} \tag{14}$$

$$B^* = \sqrt{1 + \frac{4A^3}{27 P_{motor}^2 \eta_t^2 B}} \tag{15}$$

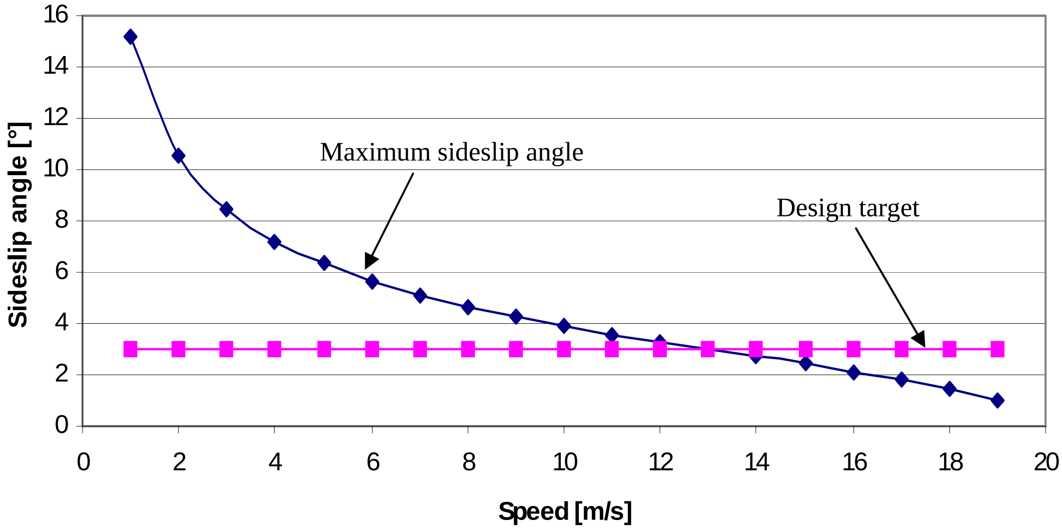

FIGURE 6 Typical α_{max}-V regulation diagram of the APT facility.

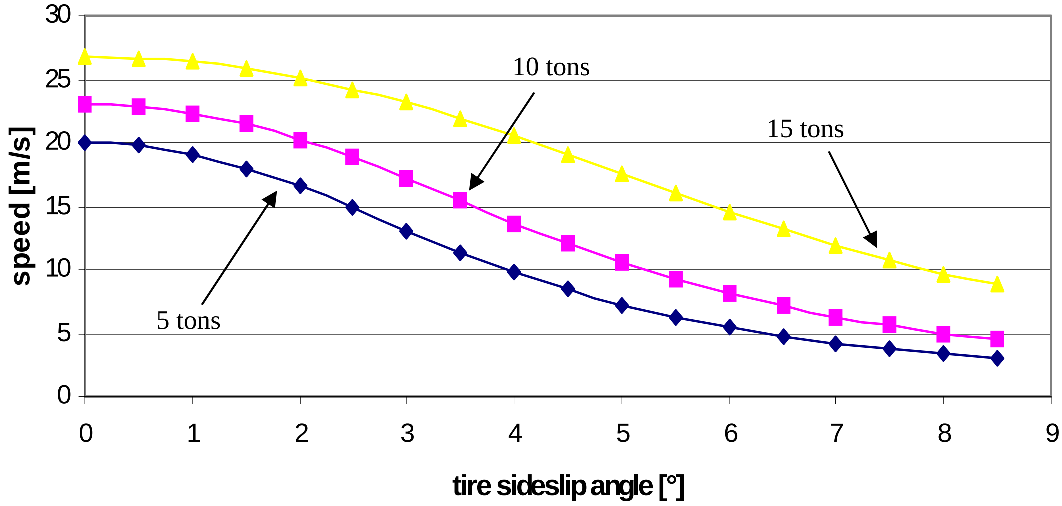

FIGURE 7 Typical V_{MAX}-α regulation diagram of the APT facility.

On the basis of regulation diagrams like those of Figure 6 and 7 and of equations (1), (2) and (3), it is possible to find the axial forces acting on the four arms of the test system as a function of the working parameters of the APT facility. Figure 8 shows the values of these forces (for their orientation, see Figure 4) plotted against the longitudinal speed of the trolleys, in the conditions of usage of the maximum power of the motor. It can be observed that the maximum value of sideslip angle (and of side force) decreases as a function of lateral speed according to a regulation diagram similar to the one of Figure 6.

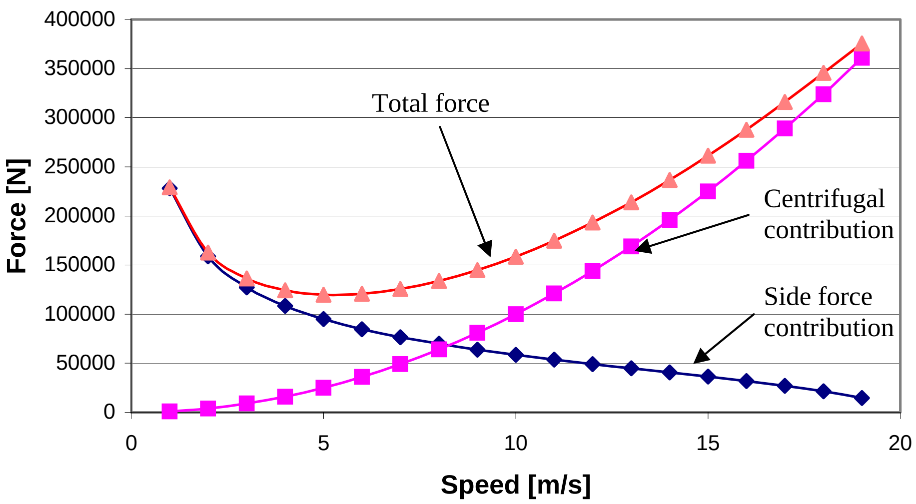

FIGURE 8 Forces acing along the axis of the arms of the APT device.

Multi-Body Modeling

Static calculations have to be followed by the development of a Multi-Body (MB) model which by means of the simulation of the dynamics of the system allows the evaluation of the forces and torques acting between the different components of the test machine and on the pavement surface. As anticipated previously, this kind of dynamic modeling can not only provide information on the general behavior of the test facility, but can also guide in selecting and positioning various sensors for the measurement and/or active control of forces and displacements.

In the specific case of the APT facility described in the previous sections, the MB model was implemented by making use of the software ADAMS View® [5]. Equations (4) and (5) were considered to estimate the resistance to motion of the test facility.

The tire model which was selected for implementation is based on Fiala's formulation [6], which takes into account the interactions between longitudinal and side forces. Side and longitudinal stiffness vary as a function of the working conditions of the tire and are not constant as hypothesized for static calculations.

In the MB model, the contributions to the power necessary for the motion of the trolleys related to longitudinal and lateral forces are not computed through equations (6) and (7), but directly as a function of the outputs of Fiala's model.

The elasto-kinematical characteristics of the suspensions of the trolley are taken into account by the MB model, which considers both unsprung and sprung masses, the geometry of the steering system, the stiffness of the bearings and the bushings of the components of the whole system. The arms of the testing facility, the trailing arms of the suspensions of the trolleys and the wheel hubs are considered infinitely stiff. As mentioned previously, the connection between the chassis of the loading trolleys and the arms of the testing device was hypothesized to be done by using low friction ball recirculation bearings. More specifically, in the MB model each trolley was connected to a horizontal arm by means of four rails, one for each corner of the trolley, with two bearings for each rail. This is shown in Figure 9, which gives a complete representation of the MB model of the loading trolley.

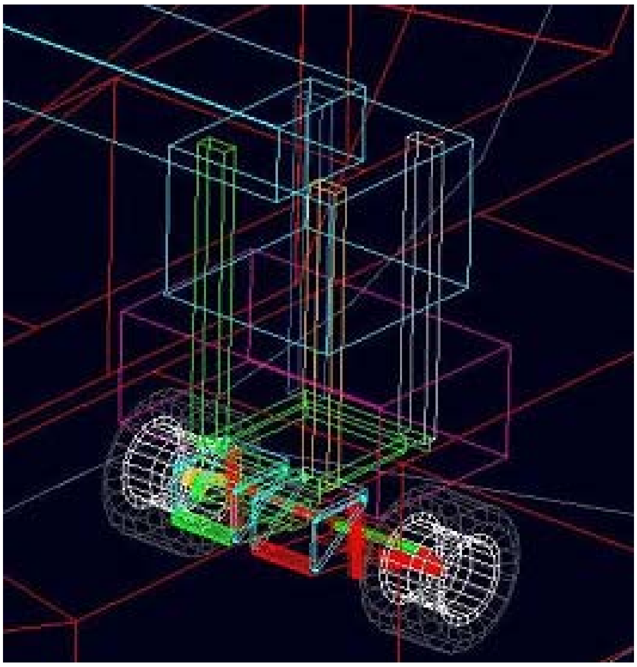

FIGURE 9 Multi-Body model of the loading trolley.

The MB models allows a wide set of data to be extracted from simulation runs, investigating the effects of critical factors on the performance of the APT facility and considering the relationships between relevant parameters. Examples of typical MB outputs are shown in the following paragraphs.

First of all, the dynamic forces applied to the pavement surface in standard stationary working conditions can be plotted for each wheel of the trolleys. This is shown in Figure 10, which refers to the condition of a perfectly flat pavement surface and a constant value of sideslip angle of 3°.

Secondly, the adopted model allows the evaluation of the forces exchanged between the elements of the trolley and the APT facility for each working condition of the system. For example, Figure 11 gives the time history of the force acting along the axis of the tie rod for different values of tire sideslip angle. These values are fundamental for dimensioning the actuation system which is necessary to impose given values of the steering angle.

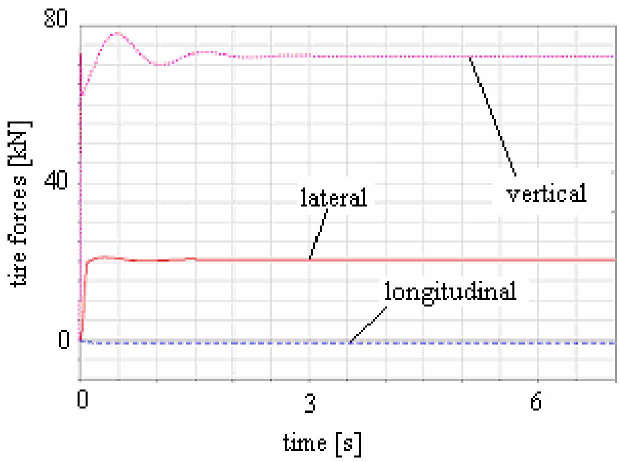

FIGURE 10 Forces acting on the pavement surface during APT testing.

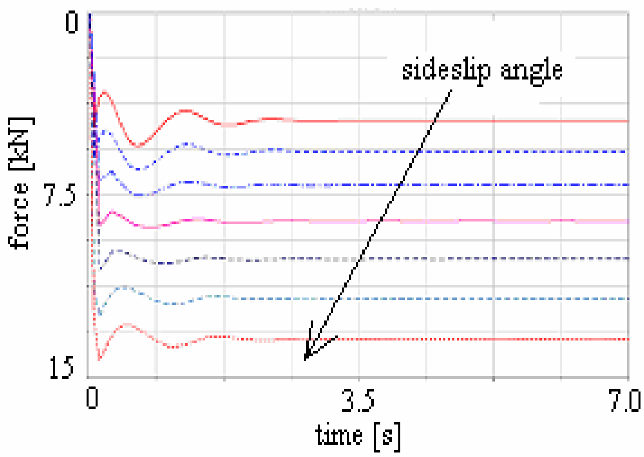

FIGURE 11 Forces acting along the axis of the tie rods during APT testing.

The MB model can also be used for the design of the ball bearings used to connect the trolleys to the horizontal arms of the APT device. In fact, the forces and the torques acting of each bearing can be calculated with a good level of approximation: this is shown in Figure 12, where the plotted data refers to simulations carried out by considering different values of the stiffness *K* of the ball bearings. Too stiff bearings can lead to a statically indeterminate structure, with abnormal values of torques or forces, due to the fact that the MB software does not consider the deformation of the rails or of the structure of the chassis.

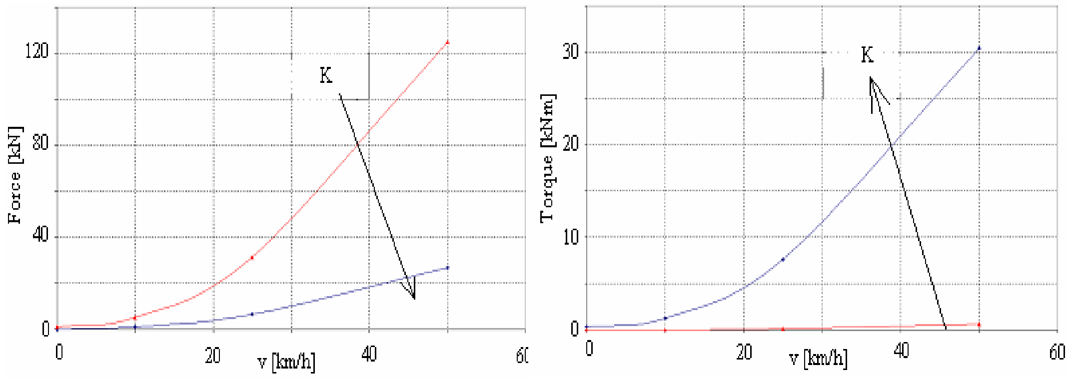

FIGURE 12 Forces and torques acting on the ball bearings.

Another interesting application of the MB model consists in its support to the optimization of the system of sensors with which the APT facility should be instrumented. For a system of the type considered in this study, the most expensive solution to measure the forces and torques acting on the wheels of each trolley consists in mounting a wheel hub equipped with sensors. However, this solution can be substituted by cheaper alternatives, which can be designed based upon the results of the MB model. For example, the estimation of tire sideslip angle can be performed on the basis of the measurement of tie rod displacement (the yaw rate motion and the sideslip motion of the trolley are limited by the arms of the APT facility). This is clearly shown in Figure 13, which highlights the existence of a precise correlation between the two dimensions estimated from MB simulations. Similarly, relationships can be obtained from MB simulations to link sideslip angle values to the side forces applied to the pavement surface. As a consequence, in a first approximation, side forces can be evaluated on the basis of tie rod position; in a second approximation (in any case cheaper than the solution of the sensors located directly in the wheel hub), side forces can be estimated as a function of the signals retrieved from force sensors located on the tie rods. Even in this case, by means of MB modeling, correlations between the forces at the tie rods and the side forces on the tires can be easily obtained.

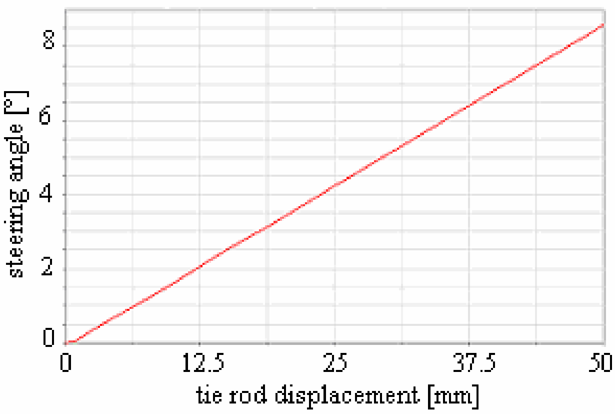

FIGURE 13 Tire steering angle as a function of tie rod displacement.

The last fundamental information which can be extracted from the MB model refers to the behavior of the APT system in extreme dynamic conditions which may be encountered in the case of rough pavement profiles. This is shown in Figures 14 through 17, which refer to different MB simulations.

Figure 14 shows the oscillations of the torques which as a result of a sinusoidal pavement profile were calculated around the three axes of the motor shaft. These data are especially important for the design of the central shaft of the APT facility and for the selection of its bearing system. During prolonged testing, any damage to this core element of the facility should be prevented and therefore the use of refined modeling tools which support structural calculations is totally justified.

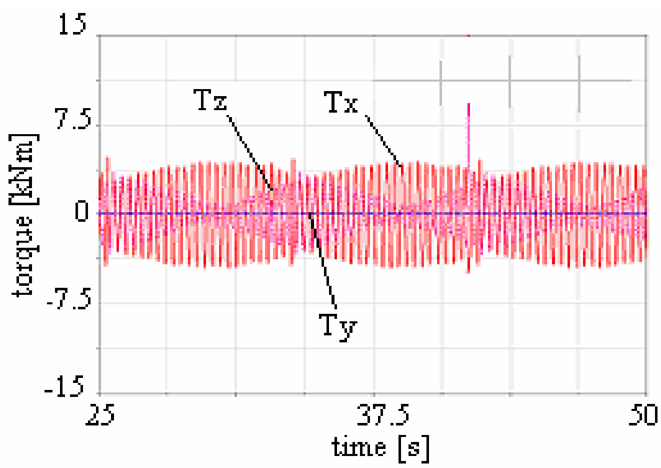

FIGURE 14 Torques on the bearings of the central shaft of the APT facility.

Figures 15 through 17 highlight the effects caused by rough profiles on the actual pavement loading history and prove the efficiency of the specific kind of suspension which was considered in the study. In all cases the trolleys of the APT device encountered a flat surface in the right wheel track and a sinusoidal profile (2.5 m of wavelength, amplitude of 50 mm) in the left wheel track. Simulations were carried out at a speed of 10 km/h.

Figure 15 contains plots of the vertical force and displacement of the left wheel only: as expected, oscillations are non negligible and are certainly bound to affect pavement performance. The data plotted in Figure 16 refer to the same simulations: however, it should be noted that the time history of loading of the right wheel is not affected by the roughness of the left wheel track. This proves that the adopted trailing arms suspension system decouples the oscillations between the two sides of the trolley. If the suspension system is modified, with the introduction of a stiff bar connecting the two arms, totally different results are obtained: this is shown in Figure 17, where it can be observed that the loading histories of both wheels is characterized by a continuous oscillation. Since the left wheel provokes the motion of the right wheel, both tires are subjected to dynamic forces, which would be difficult to quantify by making use of simple models based on mass-spring-damper systems, like those indicated in literature [7].

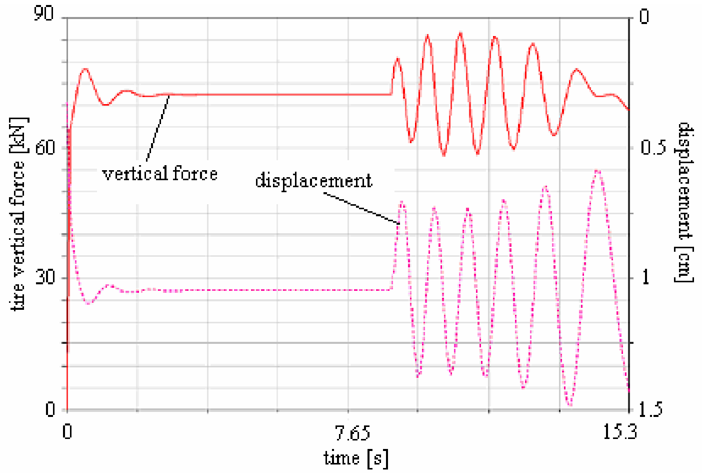

FIGURE 15 Wheel vertical force and displacement during a test on a sinusoidal pavement profile.

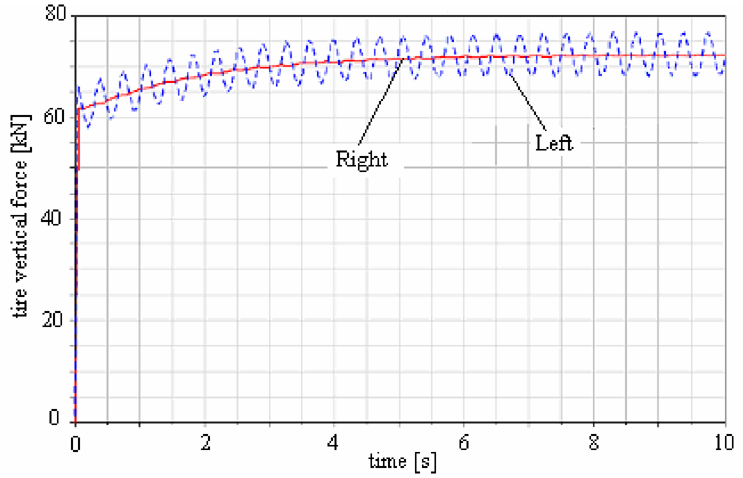

FIGURE 16 Tire vertical force in the case of trailing arm suspension and sinusoidal road profile for the left wheel.

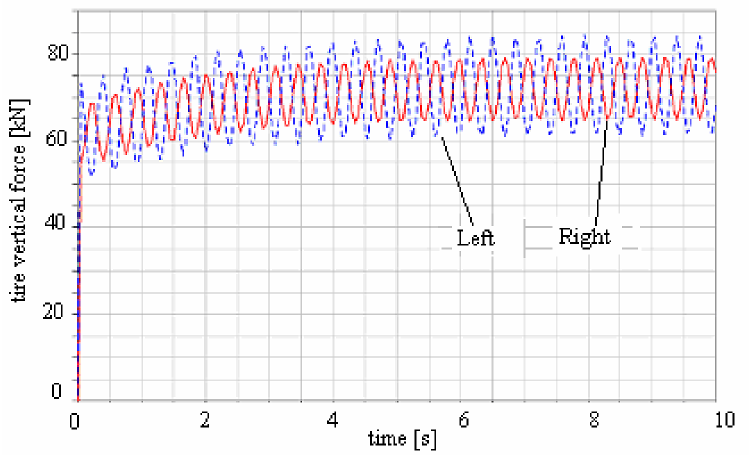

FIGURE 25 Tire vertical force in the case of modified suspension and sinusoidal road profile for the left wheel.

CONCLUSIONS

The paper presents a procedure for the design of APT facilities based on the integrated use of static calculations and Multi-Body models. Such a methodology enables designers to optimize the configuration of the APT system with the possibility of taking into account the effects caused on pavements by tires side and longitudinal forces, in addition to those caused by vertical forces. Moreover, the procedure permits to compute the regulation diagrams of the system, fundamental to use its full potential for all working conditions.

Examples of the advantages associated to the use of the MB model are provided in the paper. These are certainly relevant in the context of the structural design of the various components of the test facility and can be especially useful to optimize the choice and positions of various force and displacement sensors. By running MB simulations it is also possible to appreciate the effects caused by different configurations of the suspensions of the trolleys.

The next step which will be performed within the research which is currently being develop by the Pavement Engineering and Vehicle Dynamics groups of the Politecnico di Torino will consist in the full implementation of a FEM model of the structure of the APT facility to be run in co-simulation with the MB model. Furthermore, results derived from MB and FE modeling will be used as input data for the evaluation of the stresses and strains induced in pavements by different loading systems.

REFERENCES

1. NCHRP Synthesis 325, "Significant findings from Accelerated Pavement Testing", Transportation Research Board 2004, USA.
2. NCHRP Synthesis 325, "Application of Full-Scale Pavement Accelerated Testing", Transportation Research Board 1996, USA.
3. H. B. Pacejka, "Tire and Vehicle Dynamics", *SAE International*, 2002.
4. G. Cardano, "Ars Magna", Nurnberg, 1545.
5. ADAMS View® User's Manual, MSC Software, 2001.
6. E. Fiala, "Seitenkrafte am rollenden Luftreifen", VDI Zeitschrift, 1954.
7. B. D. Steven, J. J. De Pont, B. D. Pidwerbeski, G. Arnold, "Accelerated Dynamic Loading of Flexible Pavements at CAPTIF", *1999 International Conference on Accelerated Pavement Testing*.