# Modelling of a snow track vehicle

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**ABSTRACT**: In the present paper, a dynamic model of a snow track vehicle is presented. All the vehicle's components were modelled. The track is a tire driven rubber chain and its dynamic was reduced to five purposely kinematically related tires. Simulation was performed by introducing in the ADAMS code the characteristics of the suspensions and the power transmission, experimentally evaluated. The model allows to calculate vehicle strength and performances on ski test slopes.

## 1. INTRODUCTION

In order to analyse and evaluate the performances and the structural behaviour of a snow track vehicle a research program, in co-operation with a European Factory, has been activated.

A snowmobile is a transport vehicle on snow-covered slopes. Besides its purpose as a means of transport in the mountains for tourism, safety and logistical motives, a snowmobile, suitably equipped, is used as an operative automotive vehicle capable of snow handling and tracking on ski slopes. In this use it serves a double function: it gathers artificial and/or natural snow, distributing it on the slopes; it levels out and compacts the slopes.

The machine under examination is therefore particularly important for the economy of those regions which are characterised by prolonged periods of snow.

Currently all the important ski stations as well as those in charge of safety in the mountains, are equipped, or are becoming equipped, with skimobiles. The spread of winter sports and a more services demand, as well, bring about a predictable expansion in the market for the vehicles under examination.

In reality, in the technical literature there are existing studies dedicated to snowmobiles. Design and construction is based on the competence of the individual factories and are not drawn from a codified *corpus*.

The diffusion, still limited, and the specificity of the characteristics of these vehicles has meant that design and research in this sector has not benefited from the development and rationalisation utilised in other sectors like, for example, those of vehicle and earthmover machine construction. On the other hand, the specialised nature of the operativeness of the snowmobile does not permit a banal translation of the heritage of such sectors, but rather implies a re-elaboration of their techniques to adapt them to the sector in question.

The results we are looking for within this project are the reduction of design time and an improvement in Mobility, in Climbing Ability, in operating Speed and in Safety, all aimed at bringing snowmobile level up to that of the most technologically advanced land vehicles.

In particular the research project proposes, through an in-depth series of experimental tests, of Mechanical System Simulation codes (ADAMS) and of Finite Element Structural Analysis codes:

- a rationalisation of the design of snowmobiles obtained through the creation of a simulation model of normal operations of the vehicle, the conception of an integrated design system and the creation of an adequate data measuring system and self-diagnostics to be applied to each machine;
- an improvement in the operative and vehicular performance of snowmobiles obtained through an improvement of the weight-power ratio, in turn obtainable through a structural optimisation of the frame and a careful selection of the propulsion system.

In reference [1] some of the results already obtained during the research project are illustrated; in particular the work mentioned has had as its object the analysis and design of a new auxiliary multiple rope traction system and related hydraulic power plant.

In the present paper, a Snowmobile ADAMS Mechanical Simulation Model is shown.

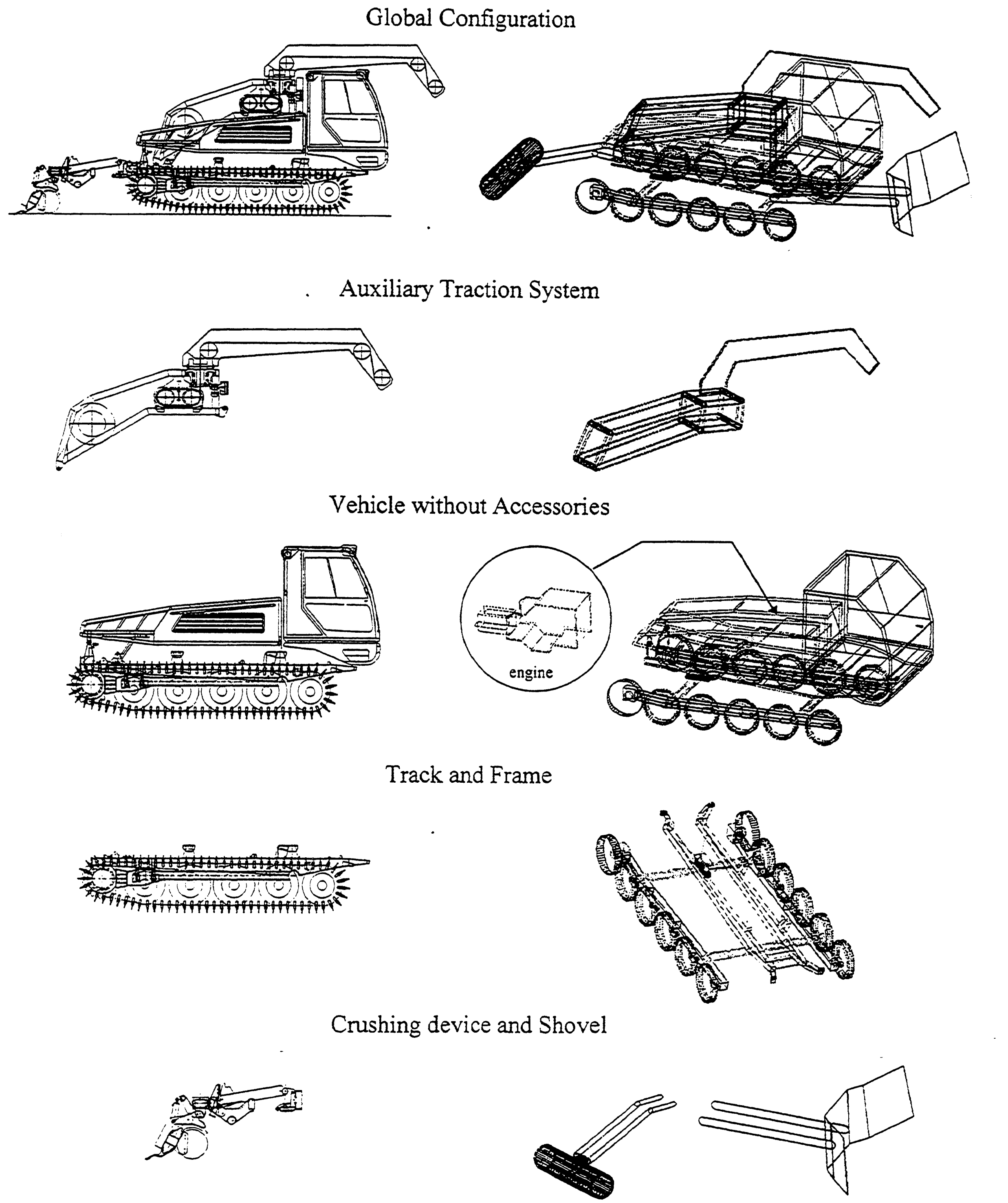


**Fig.1** - Comparison between vehicle and ADAMS model configuration

## 2. SNOW TRACK VEHICLE MODELLING

Generally speaking, in a snowmobile the feed of the principal traction system, of the auxiliary rope one and of the other accessories (the snow crushing device) is entrusted to hydraulic motors which utilise the power supplied by a combustion engine.
This solution allows the conveyance of energy in hydraulic form, thus avoiding the use of mechanical joints which would create considerable problems in the arrangement of the transmission equipment.
The vehicle analysed in the present paper, in particular, is characterised by a diesel combustion engine capable of supplying about 300 HP.
All components were modelled (Fig.1):

- the principal traction system,
- the frame,
- the upper structural components,
- the auxiliary rope traction system,
- the front snow shovel,
- the rear snow crushing device.

### 2.1. The principal traction system

2.1.1. Track modelling

The traction of the snowmobile is obtained through two tracks made of rubber chains, which are connected transversally to the aluminium climbing irons (Fig.1). The climbing iron itself houses in its tip a metallic insert, which guarantees a better hold on icy terrain and diminishes wear.
Control of the track is assured by a series of five wheels with tyres which guide and support the chain of climbing irons, drawn by a gear wheel fitted to the rear part of the longitudinal beam; in turn the wheels are constrained to the longitudinal beam through a torsional bar suspension.
In modelling this mechanism, for each track five *tyre* modules were used, which englobe a tyre-terrain contact model, reducing in this way the continuous grip of the track on the snow with a discretized one exerted by each tire (Fig.2,3).
The decision to utilise the *tyre* module for traction was motivated by the computational difficulty and onerousness of the track and of the track/snow contact modelling.

The ADAMS model used, therefore, differs from the vehicle not in its method of transmitting the load components to the frame, but rather in the realisation of the track/snow contact.
In fact, in the hypotheses of the model, contact with terrain is carried out in a number of discrete areas, characteristic to tyre deformation in the

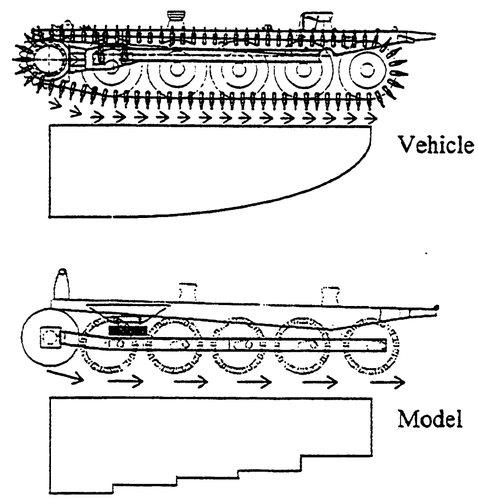


**Fig.2** - Axial force on track

frictional grip zone. Actually the contact does not take place in those areas but rather along a surface given by the area of the track in contact with the snow, with a distribution of force at each climbing irons dependent on the frictional grip conditions of the adjoining climbing irons.
In order to restore the congruence between the

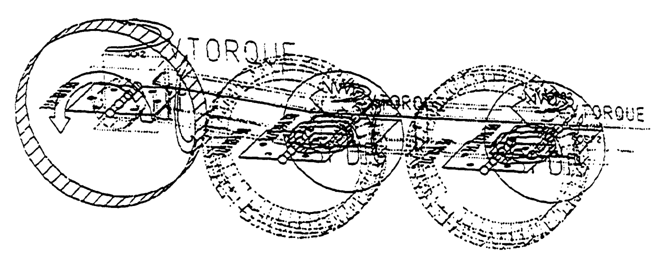


**Fig.3** - Suspension and Traction system model

model and vehicle behaviour, a constraint between the tyres and the gear wheel was realised. A kinematic one, in terms of angular speeds, was imposed, such that the peripheral velocities of the tyres, which composed the track, always coincided with that of the respective gear wheel. In this way, the power and total torque, made available to the driving wheel shaft, is distributed to the tyres relating to friction forces applying on these.

To do this it was necessary to control the normal deflection of each tyre and impose on it an angular speed $\omega_{tj}$ equal to:

$$\omega_{t_j} = \omega_{g_i} \cdot \frac{R_g}{R_{t_j}}$$

with $\omega_{g_i}$ the angular speed of the gear wheel,
$R_g$ the average radius of the gear wheel,
$R_{t_j}$ the loaded radius of the tyre

Since the deflection was small in comparison to the unloaded radius of the tyre, the choice to express this correlation utilising a first order expansion was considered acceptable:

$$\omega_{t_j} = \omega_{g_i} \cdot \frac{R_g}{R_{t_0}} \cdot \left(1 + \frac{d_j}{R_{t_0}}\right)$$

with $d_j$ the normal deflection of the j th tyre,
$R_{t_0}$ the unloaded radius of the tyre.

The UCOSUB routine was utilised and developed, as a function of the three lagrangian variables $\omega_{gi}$, $\omega_{tj}$, $d_j$ with i=1,2 and j=1:10 and, moreover, j=1:5 for i=1 and j=6:10 for i=2.

This routine imposes the above-mentioned constraint even in terms of derivatives of the lagrangian variables.

The normal deflection variable, not directly accessible in the tyre module during analysis, was made available in vectorial form in the routine by utilising a REQUEST file and saving it as a global system variable.

In order to render this a lagrangian variable, a fictitious group of SDOF systems, equal to the number of tyres and directly correlated to them, was created. The systems are of negligible mass and unitary stiffness. During the analysis, variable *single component forces*, with amplitude equal to the value of the relative global deflection variables, are applied to these systems. As a consequence, every system displacement, coinciding by realised hypothesis with the deflection of the respective tyre, becomes available as the lagrangian variable $d_j$ of the UCOSUB routine.

In this way, traction conditions equivalent to vehicle ones are realised.

The wheel/terrain reaction forces on each tyre are in this way dependent on the frictional grip conditions of the adjacent tyres, in congruence with what happens with the track climbing irons.

Moreover, the tyres separated from the ground (zero deflection), because of, for example, uneven ground, will in any case be obliged to rotate with an angular velocity proportional to that of the driving wheel with a coefficient of proportionality equal to the velocity ratio. Consequently, the torque will be distributed only on the wheels in contact.

Once the traction limit has been reached, tyres slip condition will occur which will persist until the sum of the torques exerted on the tyres by the friction forces is not greater or equal to the total driving torque. Therefore the traction limit depends on the number of wheels in contact


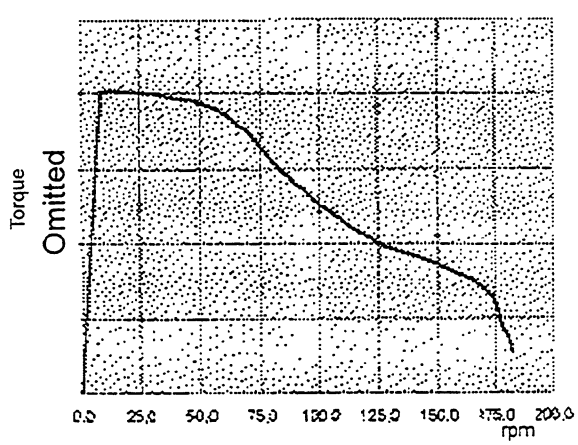


**Fig.4** - Power and Torque vs gear angular velocity characteristic
*(values were intentionally omitted because of vehicle's factory demand)*

simultaneously, analogous to what happens with a track which is increasingly able to drag the vehicle as the area in contact with the ground increases.
As far as the vehicle traction power is concerned, the total power and torque vs gear angular velocity (rpm) characteristics in conditions of maximum combustion engine inlet have been obtained from tests carried out at the engine bench (Fig.4).
Knowing, therefore, the experimental torque/angular velocity characteristic, the various dynamic simulations were conducted subjecting the model to the action of two *torque* functions, applied to the joints of the track driving wheel, and faithfully reproducing the actual characteristic (Fig.3).
To this end, the interpolating function AKISPL was utilised, with as an independent variable the angular velocity of the driving wheel calculated by ADAMS at each integration time step according to the motion conditions that the system presents in time instant, restoring (dependent variable) the value of the driving torque applied at the joint.
The motor braking action, necessary to reduce the maximum velocity that the model reaches in descent, was realised by extending the driving torque AKISPL function into the negative area. Therefore, a braking torque value, proportional to angular velocity, will be produced beginning with a limit track angular velocity.

2.1.2. Suspension modelling

As we said, each suspension is constructed of an oscillating arm and a torsional bar flanged to the longitudinal beam (Fig.5).
The suspension is also equipped with a limit deflection Puffer to reduce the bar torsion and impacts from large loads (Fig.6).
Suspension model (Fig.3) was done using two *parts*: one schematising the suspension arm and the other the track longitudinal beam, coupled by means of a *revolution joint*.
The stiffness of the bar was kept in mind by applying a *spring damper torque* to the revolution joint, to perform a reaction torque related to the rotation of the joint itself and its characteristics of stiffness and damping.
The viscose damping coefficient was determined by also evaluating the effect of the control exerted on the wheels by the climbing irons chain and the damping effect of the limit deflection realised by the rubber bumper puffer.

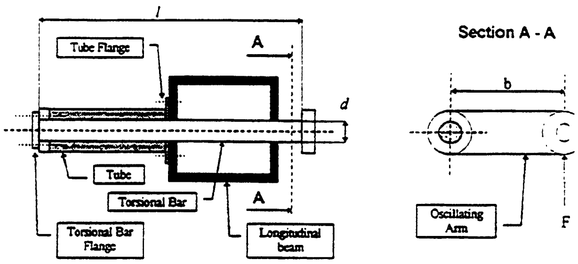


**Fig.5** - Vehicle suspension

The non linear load/deflection characteristic of the puffer (Fig.6, m curve) was realised with a

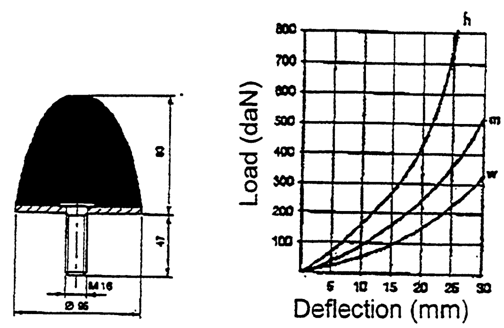


**Fig.6** - Puffer and its load/deflection characteristic

*torque* input function applied to the suspension joint related to its rotation.
In turn this function is realised with an AKISPL interpolating function that approximates the experimentally evaluated curve.

### 2.2. The frame

In substance the frame is composed of four fundamental components (Fig.7):

- upper frame (central beam),
- rear transverse beam,
- front transverse beam,
- left and right longitudinal beam.

The upper frame houses the entire propeller system (motor, coupler, pumps), the front and rear electro valve blocks, the batteries and partially supports the cabin and the storage plane. The *upper frame-rear transverse beam* joint system constitutes the longitudinal articulate joint of the frame. This type of joint allows the

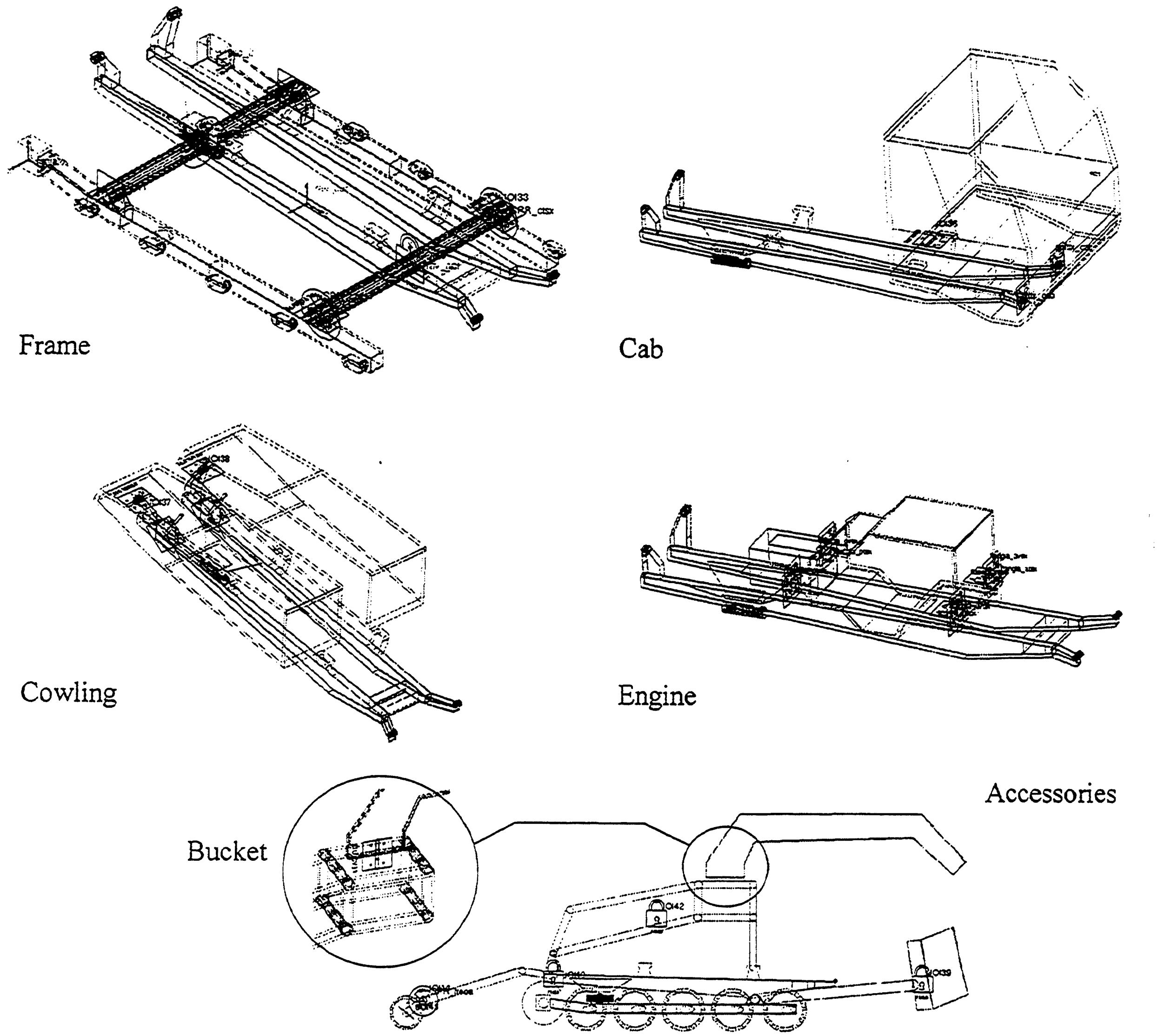


**Fig.7** - Principal components model and relative constraints

rear transverse beam to rotate related to the upper frame around the articulate joint longitudinal axis. In this way the storage plane does not feel the effects of any unevenness in the terrain encountered by the track, since this is compensated for by the rotation of the transverse beam. The storage plane which bears the combustion engine, coupler and pumps, remains within certain limits horizontal increasing the stability of the entire vehicle through the effect of diminution of the inertial forces related to the oscillating masses roll.

The *upper frame* is jointed to the *front transverse beam* by means of a closing semishell.

The *transverse beam-longitudinal beam* joint system is positioned at the end of the two transverse beams and is made using a coupler which permits the longitudinal beams to rotate around the their own axes.

All this contributes to the realisation of an articulated frame which permits an optimum adaptation of the track to the fluctuations and dissymmetries of the terrain.

The ADAMS modelling was carried out by reproducing the principal geometric shapes of the four elements composing the frame, connected in such a way as to faithfully reproduce the actual structural scheme.

To simulate the elastic behaviour of the rear longitudinal joints (*upper frame-rear transverse beam*) and that of the front transversal ones (*transverse beams-longitudinal beams), spring damper torques* were used, whose values of stiffness and damping were calculated according to the mechanical characteristics of the material composing the rubber rings (Fig.7).

### 2.3. Upper structural components

The upper structural components subjected to modelling were: the driver's cabin, the motor and the rear cowling (Fig.1,7).
Given the impossibility of determining the inertial properties for each structural component, the modelling was conducted in such a way as to obtain geometric shapes as similar as possible to the actual ones. In this way the evaluation of the properties of inertia of the various parts were entrusted to ADAMS routines.
More careful was the frame/components joints modelling; these, together with the suspension ones, are the principal transmission way for on frame load components.
As far as the control cabin is concerned, the coupling with the frame in the rear section is realised with a *hinge* with rotation axis oriented along a right-angled direction to the mean vertical plane; toward the front the cabin was coupled to the frame with rubber shock absorbing elements schematised with two *spring damper forces* applied between the cabin and the upper frame (Fig.1,7).
As far as the motor is concerned, the geometry modelling was intentionally schematic. The coupling with the upper frame is realised with four elastic supports blocked on the motor/frame semiflanges which assure a perfect force transmission and improved vibration isolation. The four supports were schematised in ADAMS with ten *spring damper forces* distributed in correspondence of the coupling flanges. The stiffness and damping coefficient were calculated from the elastic supports characteristics (Fig.1,7).
As far as the rear cowling is concerned, since it was not of interest to simulate the opening and closing action obtained with two hydraulic pistons, the realisation of the couplings was done in such a way as to eliminate any rigid motion conditions between the cowling and the upper frame by creating an isostatic structure; a *revolution joint* was used to schematise the rear hinge, and a *primitive joint inplane* (at single degree of freedom) to provide the support action of the hydraulic piston. This sort of modelling allows the load components which act upon the hinge and the hydraulic piston during vehicle operation to be determined (Fig.1,7).

### 2.4. The auxiliary rope traction system

The rope traction system (Fig.1), which can be acquired optionally, is of fundamental importance when the vehicle must operate on particularly steep slopes (Fig.8).
Generally a loss of vehicle grip occurs on snow-covered slopes with an inclination greater than 30° (limit variable according to the snow conditions). In such limit situations the winch allows a significant increase, within a radius of action of about 600 meters, in vehicle traction ability, supplying a notable contribution to operational safety both uphill and down.
The performance of the traction force system is described in terms of obtainable vehicle operative speed, of maximum force performed and of radius of action allowed.
In particular, the winch analysed was hypothesised with a maximum traction force of about 40 KN.
The modelling of this accessory (see Fig.7 and analysis no. 5) was conducted using two *parts*: one schematising the load-bearing frame, on which the winch and its storing drum are located, and the other the bucket, which orients the cable during traction force operations and which is connected to the load-bearing frame by means of a thrust bearing block. This part was constrained by means of a vertical axis *cylindrical hinge* to the frame's one, permitting it the related rotation, while the other, actually connected to the load-bearing structure by means of two cylindrical hinges with rotation axes right angled to the vehicle mean vertical plane, was completely constrained to the vehicle by means of a *fixed joint*. Winch action was schematised with a *single component force* applied on a *marker* fixed on the bucket free end, with direction determined during the motion by two marker, the bucket one and another related to the *ground*, positioned on the slant of the slope.
The influence of the auxiliary traction system operation on the principal one, which can be

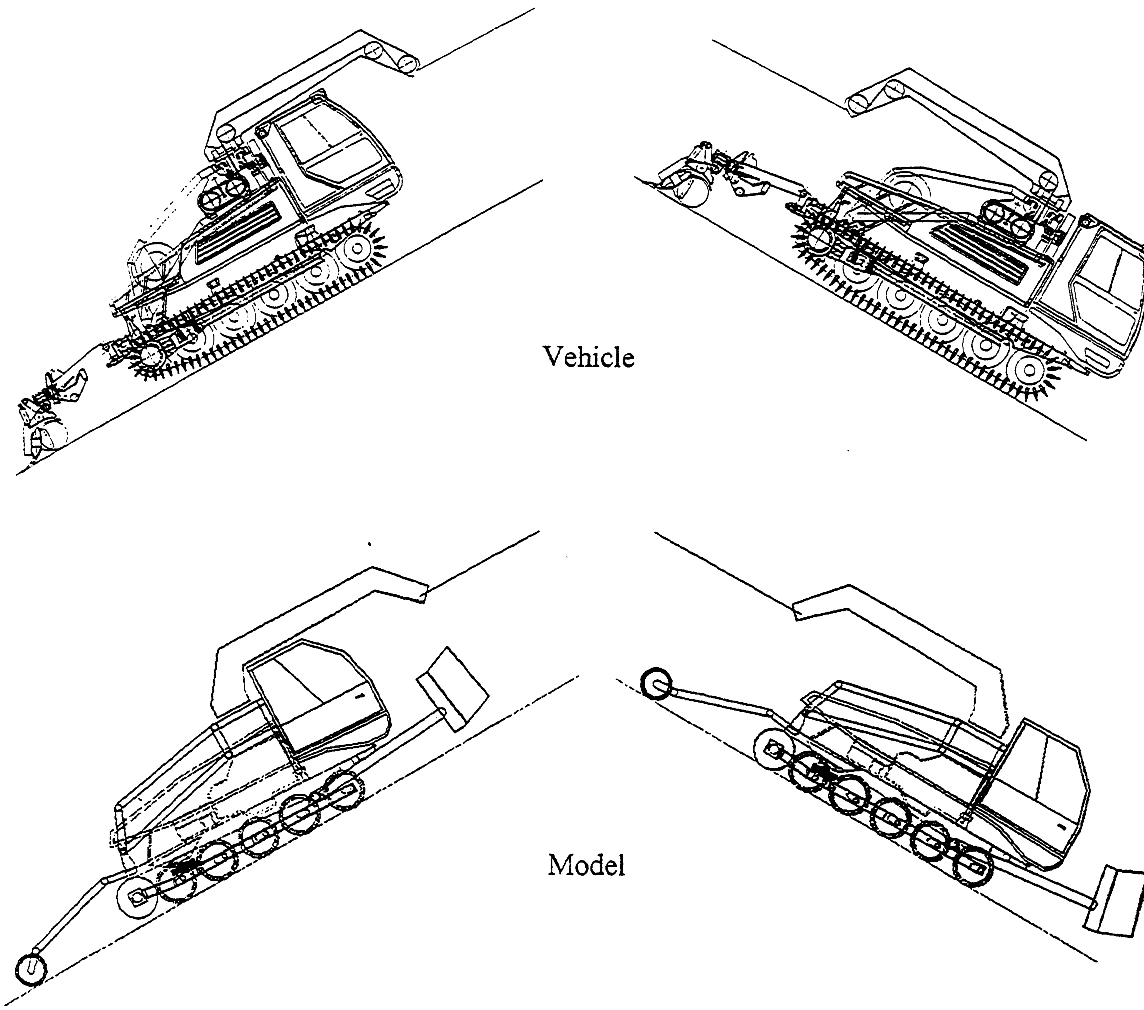


**Fig.8** - Vehicle and Model attitude during auxiliary traction system operation (30° slope)

noted with the lowering of track power, was taken into consideration by introducing a resisting torque into the gear *torque* function calculated according to the absorbed power, that is, according to the vehicle translation velocity and gear angular one.

### 2.5. The front snow shovel

The front-mounted shovel takes care of the task of snow handling. This tool is composed of a shovel and a bearing arm fixed to the front transverse beam of the vehicle. In this case as well, geometry modelling was essential; in fact, the principal goal was to examine the contribution of mass and inertia of the shovel system in order to analyse the behaviour of the vehicle during the phases of setting and snow handling and to point out the torque/angular velocity characteristic of the vehicle during handling by means of the application on the shovel of force which increases over time by a *step function* (see analysis no.1). This reasoning has led us to not reproduce the complete functioning of the articulated joint between the bearing arm and the shovel, modelling the entire system in a single *part* and constraining it with a *fixed joint* to the front transverse beam (Fig.7).

### 2.6. The rear s crushing device

The rear crushing device as a whole (rotary cutter and accessories) carries out the task of tracking the ski slopes, crushing the snow on passing in order to render it easily workable and, finally, compacting it. Schematically the device is composed of a cutter cylinder, an articulate power-lift and a central joint system (Fig.1). The modelling of the apparatus was done reconstructing just the central joint system without examining the articulate power-lift behaviour (Fig.7).

Then the geometry of the two parts, the lifter and the snow cutter cylinder, were modelled constraining them to each other through a *revolution joint* along transversal axis, in such a way to permit the rotation of the cutter, and attaching the part related to the lifter to the upper frame with a *fixed joint*.
The action of the rubber rings of the central joint system was reproduced with a *spring damper torque* at the revolution joint with stiffness and damping characteristics drawn from those of the elastic joint.
Finally, device/ground contact was introduced by coupling a *tyre*, of negligible mass and with the principal moment of inertia equal to that of the snow cutter cylinder, to the related part.
During the crushing operation, the tool's hydraulic plant is fed by an oleo-dynamic pump, driven by the combustion engine through the pumps coupler. As a consequence, part of the driving power supplied by the engine to the velocity is equal to about 60 kW (according to the conditions of the snow that the device must handle), through a reduction of the driving torque reproduced by the AKISPL function for each track. In this way, it is possible to determine during simulation the influence that the device exercises on the climbing ability of the snowmobile.

## 3. DYNAMIC ANALYSES

The analyses conducted were aimed at evaluating the ability of the model to permit the rationalisation of the structural design of snowmobiles through the evaluation of the stress state, under normal operation, of all vehicle components; a further goal was that of improving the operative and vehicular performance of the vehicle through the evaluation, under normal operation, of the performance limits in terms of surmountable slopes and power required. All of

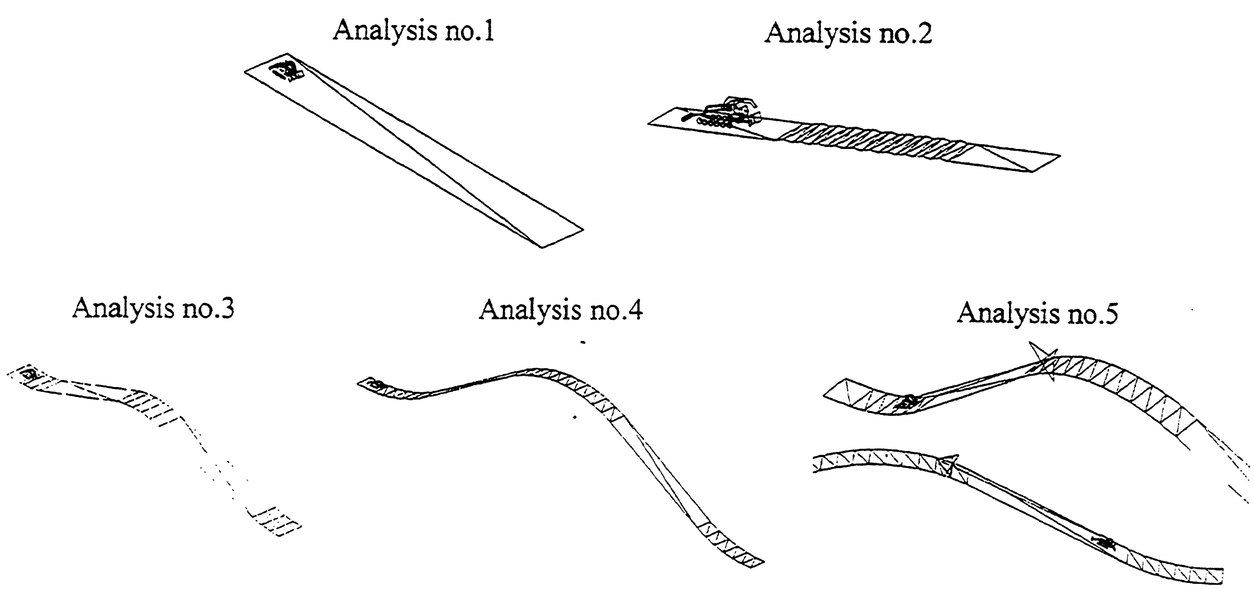


**Fig.9** - Analyses summary

principal traction hydraulic plant will be transferred to the crushing system. This loss of power to the track causes a reduction in traction ability with a diminution of the climbing ability of the vehicle.
In the ADAMS model, in relation to the simulations carried out with the snow crushing device in function, a percentile loss of power to the track was imposed which at maximum

this is clearly connected to the verification of the reliability of the numerical results with a series of experimental field tests. These tests, in fact, have the essential aim of verifying the actual conditions of the vehicle's frictional grip (static and dynamic friction coefficient), which to this point have been modelled by choosing values supported only by the experience acquired over the years by the manufacturer but which, in

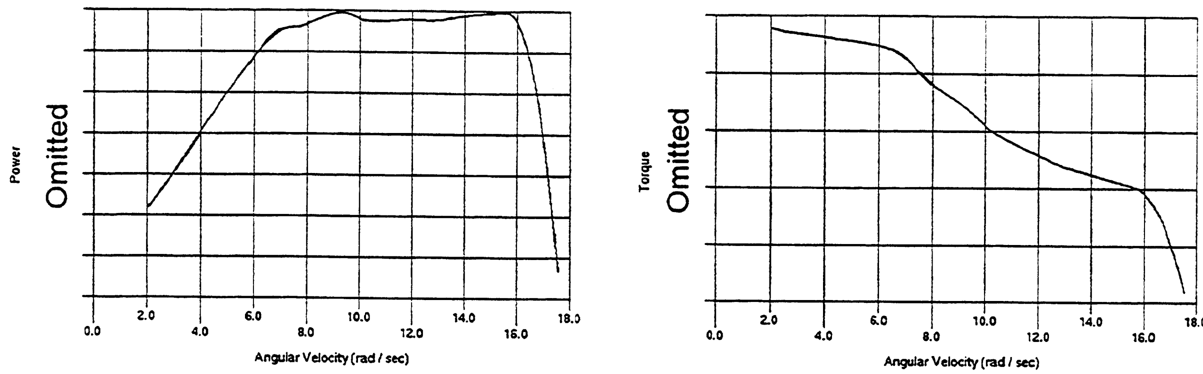


**Fig.10** - One track power and driving torque vs gear angular velocity characteristics (analysis no.1)

reality, are a critical point in the design process because of the considerable variability in snow-bed conditions, dependent on atmospheric ones.
Five analyses were carried out which demonstrated the correspondence of the results obtained with the experimental information in our possession (Fig.9):

no.1 analysis on plane ground;
(snow shovel operation)
no.2 analysis on rough ground;
(no accessory operation)
no.3 analysis on a 15° slope;
(crushing device operation)
no.4 analysis on a limit 30° slope;
(crushing device operation)
no.5 analysis on a limit 30° slope.
(crushing device and winch operation)

Thanks to the type of vehicle modelling carried out in ADAMS, it has been possible to reproduce the time histories of the load components transmitted to the frame at the principal connection joints: the connection joints of the torsional bar suspension to the longitudinal beam, the connection joints of the cabin to the upper frame, the connection flange of the combustion engine to the upper frame and the connection joints of the rear cowling to the upper frame. Also, the modelling of the tracked traction system through kinematically related tire elements has not compromised the evaluation of the axial force on the chain of climbing irons during the vehicle's work phases. In fact, from the knowledge of the wheel-terrain contact forces, which can be reproduced by the tyre module in SAE reference, it is possible to reach the normal stress value at the track. From the tight branch equilibrium of the climbing irons chain, in fact, one can see how the progress of the normal stress along the track is represented by the sum of all the friction actions which take place with the terrain. Therefore, utilising a user defined request file, the analyses allow us to reproduce the temporal progress of normal stress

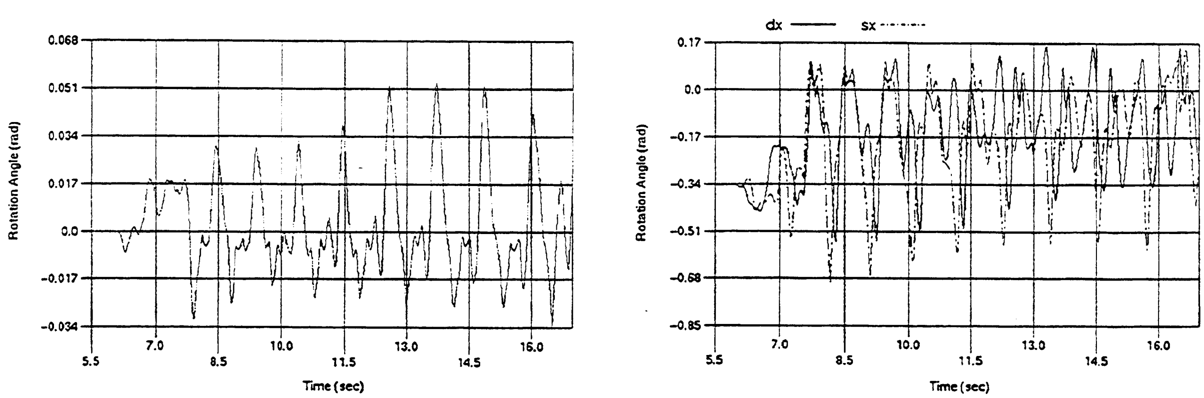


**Fig.11** - Frame Rotation along longitudinal axis, and rear suspensions rotation time histories (analysis no.2)

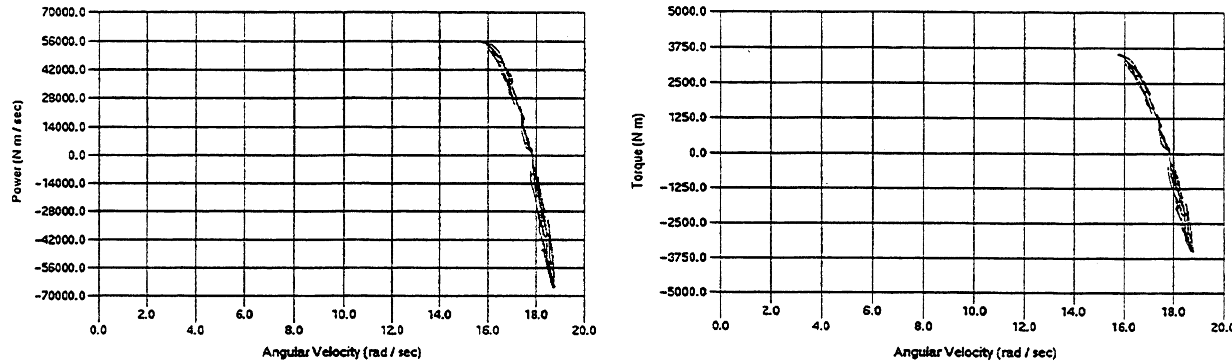


**Fig.12** - One track power and driving torque vs gear angular velocity characteristics (analysis no.3)

along the six parts of the climbing irons chain, defined by the five tyres and the gear wheel.

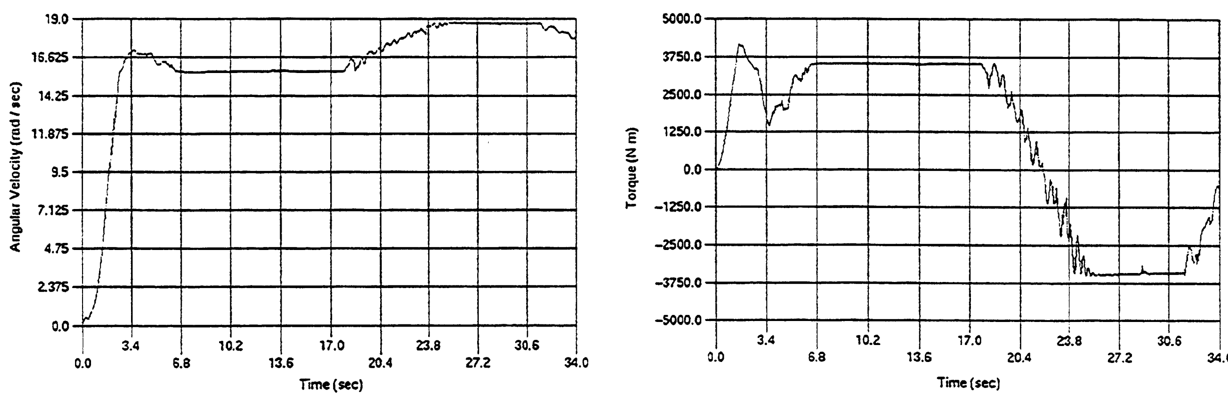


**Fig.13** - Gear angular velocity and driving torque time histories (analysis no.3)

As far as the evaluation of model performance is concerned, the climbing ability and maximum surmountable slope (theoretically imposed with the coefficients of static and dynamic friction of the tyres, chosen using field testing conducted by the manufacturer), the time histories of translation velocity and the power and driving torque vs gear angular velocity characteristics

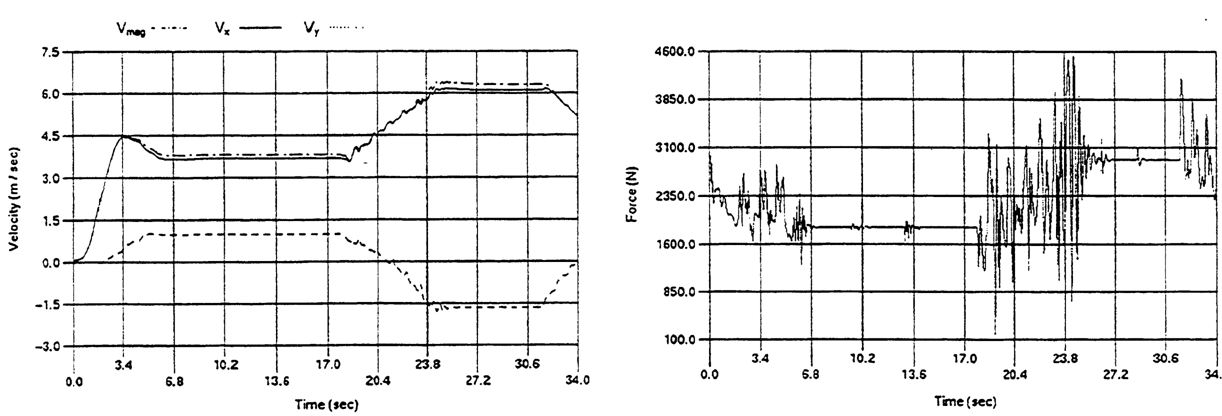


**Fig.14** - Vehicle Velocity and Cab dx spring force time histories (analysis no.3)

were determined.

In analysis number 1, the action of snow handling on a plane trail was reproduced by applying to the vehicle at the shovel a force that increased with time using a *step function*, of Fx component, in direction and opposite versus of the motion, with a maximum value of 75 KN and Fz component, vertical and directed downwards, with a maximum value of 7.5 KN. This simulation has allowed us to reproduce in an excellent way the model power and torque vs angular velocity characteristics, which resulted perfectly superimposable on the experimental results (Fig.10).

In analysis number 2, instead, an uneven trail was modelled in order to highlight the suppleness of the frame and the functioning of the wheel suspensions.

Figure 11 shows back transversal beam rotation along vehicle longitudinal axis and first dx and sx rear suspensions time histories.

Analyses 3, 4 and 5 concerned the behaviour of the snowmobile in situations more common to its actual use. Tracking of ski slopes with varying inclinations was simulated, in conditions, therefore, of utilisation of the rear crushing device and, in analysis number 5, also of the auxiliary traction system; all these three analyses have been carried out in maximum engine inlet conditions, that is, following the experimentally evaluated power and torque characteristics (Fig.4), reproduced by AKISPL function.

In particular analysis number 3 served to analyse the performance of the vehicle on slopes of average inclination (15°) in terms of attainable velocity and power required.

Low power and torque levels on slopes of this entity are evident (Fig.12); furthermore, it is interesting to observe, in this simulation, the motor braking action, as it results from the negative values that the driving torque takes on at the simulation time 22 s (Fig.13). In time intervals 6.8-18 s and 25.5-31.8 s, the vehicle runs on a constant slope track, corresponding to ascending and descending stretches.

Analysis number 4, instead, brought out the attainment of the frictional grip limit on limit slopes (30°) thus confirming our experimental experiences.

In Figure 15, the vehicle loss of traction in terms of velocity, friction coefficients and slip ratio is evident. This happens in a time interval from 10.5 to 18 s, as soon as vehicle runs on maximum slope (30°). This phenomenon is periodically repeating, with increasing frequency and decreasing amplitude, until an equilibrium condition, in the presence of a continuous slip, is established.

The last simulation, finally, analysed the use of the winch on this kind of limit slope both uphill and downhill. In the velocity time history of Figure no.17, the increase of the vehicle climbing ability is obvious, allowing it in slope crossings and keeping in such stretches (time interval 7.5-35.5 s), a velocity about 2.4 m/s (9 km/h). Moreover, in the same chart, the accessory contribution to the vehicle safety level

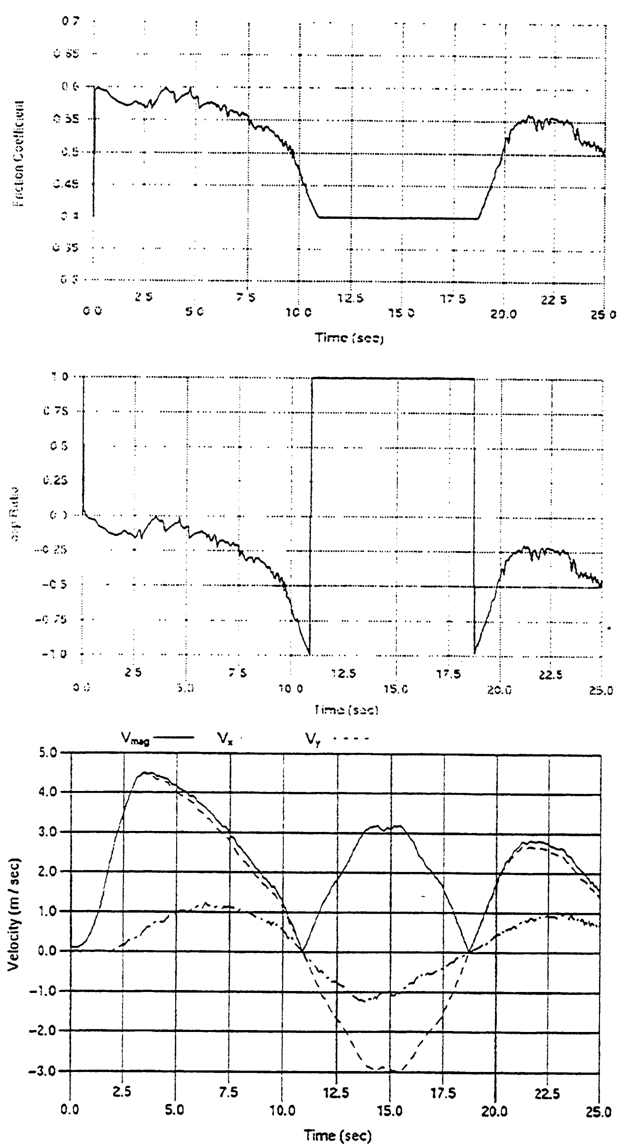


**Fig.15** - Friction Coefficient, Slip ratio and Vehicle velocity time histories (analysis no.4)

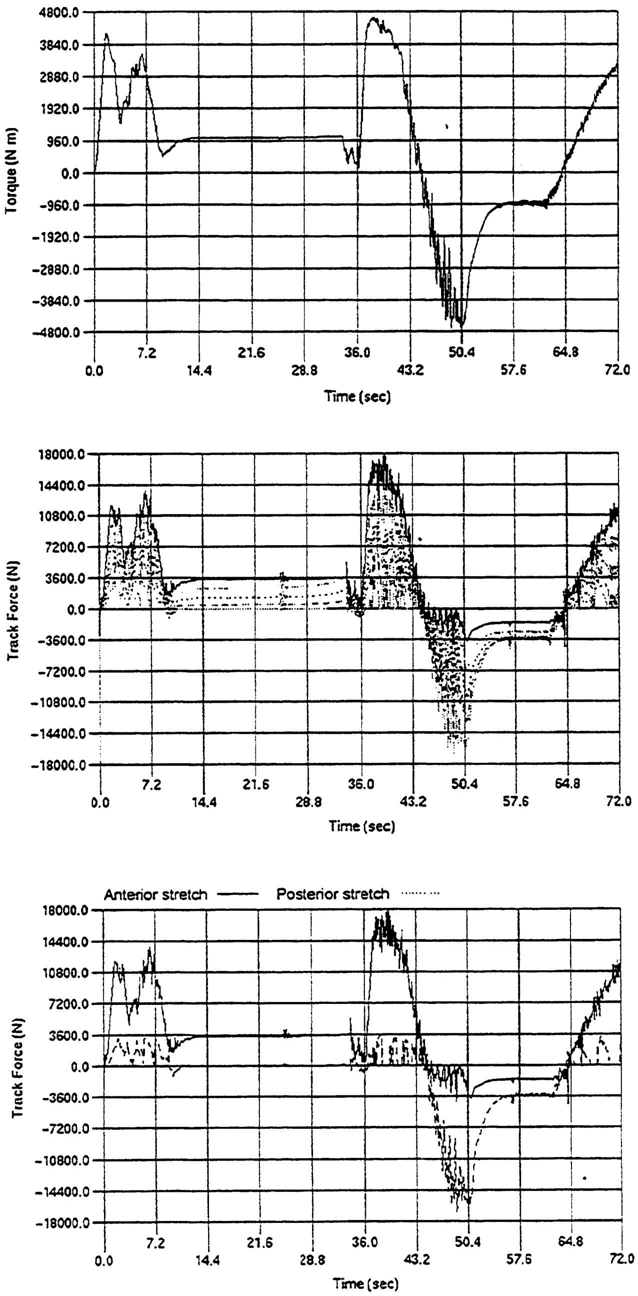


**Fig.16** - Gear driving torque, axial forces on dx track (all six parts) and only first and last part forces on dx track time histories (analysis no.5)

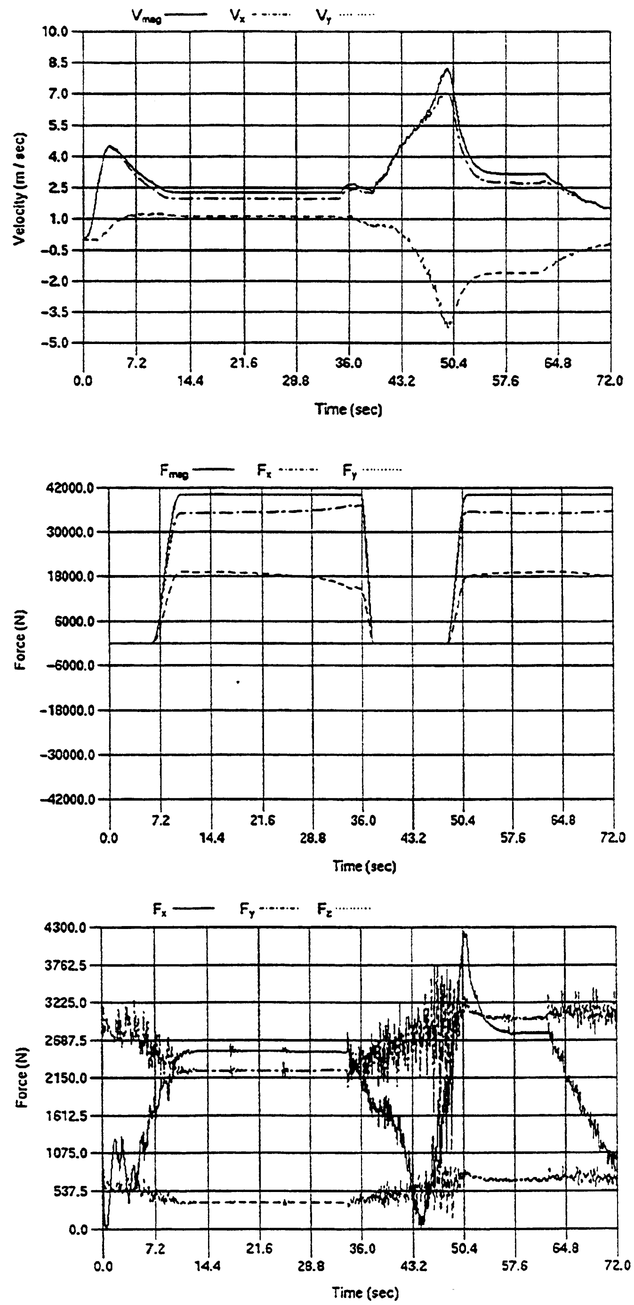


**Fig.17** - Vehicle velocity, auxiliary traction system force and engine anterior dx spring force time histories (analysis no.5)

during descending track can be observed; this performs in a velocity decrease and control (time interval 50.4-63 s) until a constant value about 3 m/s (11 km/h).

In Figure no.16, dx track axial load time histories, obtained by means of REQUEST file, are shown. In the middle chart, the traction in all stretches is shown whereas in the bottom one only the anterior and posterior are displayed. As hypothesised, the load amplitudes decreases toward the anterior stretch when a driving torque is applied (ascending track) and decreases toward the posterior stretch when a braking torque is applied.

## 4. CONCLUSIONS

The analyses carried out and the results obtained have demonstrated the correspondence of the ADAMS model with actual vehicle behaviour in terms of Mobility (speed attainable in all conditions of normal operations), of Climbing Ability (maximum inclination and the efficiency of the auxiliary traction system) and Safety (descent speed); this model's capacity to resolve by predictive functions the performance under normal operation of the snow track vehicle and, therefore, its usefulness in the rationalisation of the entire design process has also been demonstrated.

## 5. REFERENCES


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