Kinematic and dynamic analysis of a sit-ski to improve vibrational comfort

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Abstract

This paper reports both procedures for the kinematic design and the results of dynamic analyses of three models of sit-ski in order to evaluate their vibrational comfort. The model described in the US Patent US 6,019,380 has been assumed as a reference in the following kinematic and dynamic analyses of the new models proposed. The two main features to be considered while designing a sit-ski are the manoeuvrability and the range of vision of the driver. These two features appear to be important considering that the sit-ski is generally used by disabled people (e.g. paraplegic). Thus the kinematic design phase should account for the fulfillment of these requirements. On the basis of these, three new models have been proposed and herein named: Flex, modified Flex and eight-bar linkage (Smarrini 2003). Once the kinematic synthesis and analysis have been completed the dynamic performance of each model is investigated. For this purpose two kinds of slope profiles are simulated in order to reproduce the stresses which may likely occur during a downhill race. In particular the position and the acceleration of the seat have been monitored during the simulations. This is important in order to test the range of vision and handling requirements. The seat acceleration is also important to evaluate the vibrational comfort of the sit-ski according to the British standard BS 6841 (Griffin 1990). The dynamic simulation is first executed by means of the multibody dynamics code NumDyn3D, then the vibrational comfort analysis is performed using in-house developed software (Vita 2001, Valentini and Vita 2003).

Keywords: sit-ski, kinematic and dynamic analysis, vibrational comfort

Introduction

The number of users of winter sports equipment has increased in recent last years. Many new disciplines such as snowboarding, freestyle, eliski have grown. Among these different ways to practise skiing, 'sit-ski' has attracted special attention, not only as a new way of skiing, but, above all, for the possibility opened to disabled people to practise winter sports. 'Sit-ski' is an acronym for skiing in a sitting position and identifies the tool which makes it possible. A sit-ski is normally made of light alloys, similar to those used for bicycle frames (magnesium, titanium, aluminium up to composite materials), and is essentially composed of the following four parts:

1 A **board**, which allows the connection, by means of bindings normally used for ski boots, of the whole system to the ski. The shape of the board depends on the steepness of the slope .

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2 The **mechanism**, which regulates the motion of the seat, where the board and the shock absorber are fixed on.
3 The **shock-absorber**, used to minimize the effects of the slope roughness, as well as to offset the centrifugal force during the bend.
4 The **coupler** of a four-bar linkage, that acts as a support for the seat and the eventual footrest. Usually, the seat of the sit-ski is made of a single shell, or more often by two of these linked together. Plastic, carbon fibre or composite materials are often used. In the agonistic field it is made by moulds based on the anthropometric measures of the driver. The choice of the seat, made of two shells joined together, influences the movements of the upper part of the pilot as required for correct handling of the curves.

One of the features monitored during the kinematic analysis is the seat displacement. In fact, for an optimal sight range, the seat should always be kept, as much as possible, parallel to the ground.

This paper focuses on the kinematic design and dynamic analysis of three new models of sit-ski. The dynamic performances of these new models have been compared to the one described by the US Patent US 6,019,380. Thus the model described by the US Patent US 6,019,380 is considered as a reference. The kinematic design criterion of the new models is concerned with the optimality of the sight range and handling. During dynamic analysis the position and the acceleration of the seat are monitored. In particular the Fast Fourier Transform (FFT) of the vertical and horizontal acceleration of the seat are presented for each model. Finally the Vibration Dose Value (VDV), according to the British standard BS 6841, has been evaluated for each model of sit-ski.

Optimal features of a sit-ski

The bindings are usually located in the central area of a ski. Thus this area, apart from including the centre of mass of the ski itself, has also to contain the projection of the centre of mass of the skier. This grants an optimal handling of the ski. A conventional skier can move the centre of mass of his/her body by bending their trunk, and by bending their knees, transferring their weight to the front or the back of the skis as required. This increases the grip of the cutting edge on the snow, and also dampens the roughness of the ground and the changes in slant. It is important that this longitudinal movement of the centre of mass remains within certain boundaries (i.e. the area between the bindings). As a matter of fact, if the centre of mass of the skier moves, for example, too far backwards, the front of the ski would decrease the sharing of load with a consequent lack of control of the ski and a possible fall. Thus, another important feature to consider in the sit-ski design is the avoidance of this situation. Furthermore, it is necessary to account for the performances of the shock-absorber. The purpose of this unit is to smoothen and dampen the seat movement due to the roughness of the slope profile. A parameter that needs to be considered is the angle between the seat and the surface of the slope. The closer this angle is to zero, the better the comfort and the visibility of the slope. For these reasons four-bar mechanisms are generally basic elements of sit-ski structures. By means of an appropriate design configuration, four-bar linkage keeps the seat parallel to the ski and to the ground during movement.

To summarize, the key aspects to take in mind during the synthesis process are:

1 the position of the centre of mass and of its projection on the ski during motion
2 the handling and the manoeuverbility of the sit-ski (connected with the position of the centre of mass)
3 the angle between the seat and the surface of the slope, in order to grant a wide field of view.

State of the art

This section analyzes the main components of the sit-ski model described in the US Patent US 6,019,380 shown in Fig. 1.

These components are:

1 the board (22)[1] connected to the monoski (14) by means of normal bindings (18)
2 the jointed parallelogram made up of four links (56, 58 and the homologues on the opposite plane)
3 the spring-damper system (64)
4 the seat (42)
5 the foot rest (32).

[1] The bracketed numbers refer to the numbers reported in Fig. 1

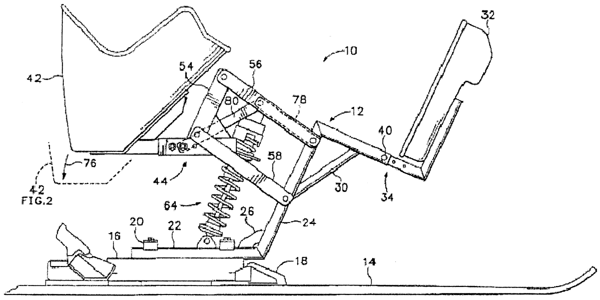

Figure 1 Sit-ski described in US Patent US 6,019,380

The parallelogram permits to absorb the roughness, the changes in slope and the centrifugal loads during the movement of the ski on the descent and during the bend. The spring force acts in opposition to the direction of seat rest (44) motion. During all the motion, the seat will always remain parallel to the ground (as shown by the dashed-line (42) and by the path (76)), ensuring a satisfactory sight range. In this model the movement of the connecting rod, and thus of the seat, is controlled only by the movement of the trunk of the user, leaving the legs and the feet linked to the frame. This choice is claimed by the inventor as an optimization of the load sharing on the centre of the ski, because of a decreased inertial contribution in the movement of the ski. On the other hand the angle between the trunk and the thighs of the passenger varies continuously during the movement, having an influence both on the comfort and on the overall sight range. There is no regulation system for the attachment point of the shock-absorber. It is possible instead to adjust the positions of the seat and foot rest. A 3D view of this model is shown in Fig. 2 where all the components previously described are included.

As shown in Fig. 3 the frame and the connecting rod are parallel to each other in order to grant the parallelism between the seat and the slope. As a consequence the range of sight for the user is always near to the optimal one. However, the centre of mass of the system is not bounded inside the area of the bindings. Table 1 reports the main dimensional features assumed for this mechanism and the vertical and horizontal range of motion of the seat.

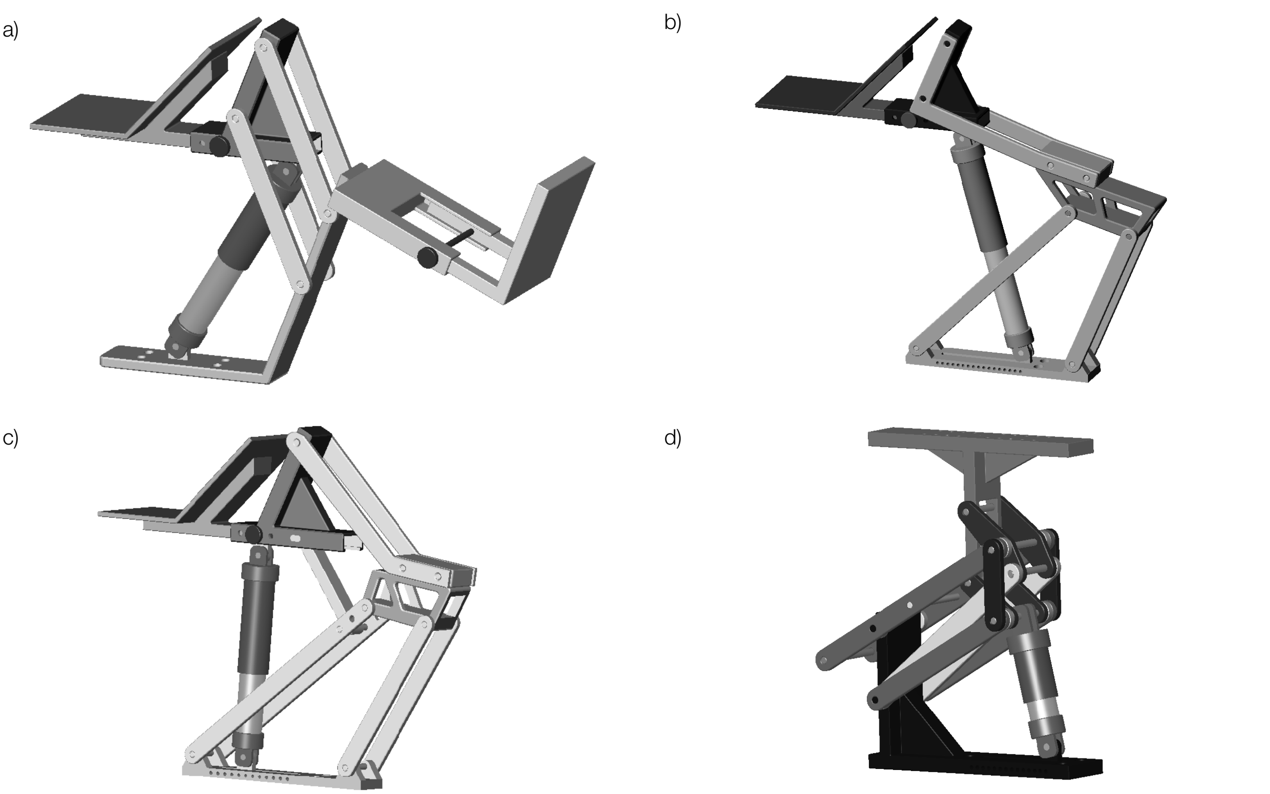

Figure 2 Sit-ski proposed models a) US Patent US 6,019,380 b) Flex sit-ski c) modified Flex sit-ski d) eight-bar linkage

Proposed mechanisms

This section describes the main features of innovative sit-ski structures proposed by the authors.

Flex mechanism

This solution is based on a double-rocker four-bar linkage, with the seat attached to the coupler (Fig. 2).

The optimality criterion adopted for the design of this model was to maintain the centre of mass of the system within the region of the bindings. Thus, the centre of mass G has been placed on the inflection circle of the coupler (Shigley 1980). In order to define the inflection circle in an analytical way the second expression of Euler-Savary has been used

$$(P_0M)^2 = \Omega M \times M'\boldsymbol{M} \qquad (1)$$

where P_0 is the instantaneous centre of rotation of the coupler, M is a generic point belonging to the coupler and Ω is its centre of curvature.

In such a manner the line joining G and inflection pole I is kept orthogonal to the longitudinal axis of the frame (A_0B_0), as shown in Fig. 4.

Table 1 Dimensional features

Component	Value
Bar length	250 mm
Connecting rod length	120 mm
Maximum rotation of seat	28°
Vertical range of seat motion	68 mm
Horizontal range of seat motion	104 mm

Simultaneously the projection point of G on the frame has been kept symmetrical with respect to the ends of the frame link. This model has also been designed considering the necessity of adjusting the frame length according to the features of the track and the personal attitudes of the driver. Thus the attachment point of the suspension on the frame could be displaced with a slight variation of the kinematic features previously mentioned.

As reported in Table 2, the maximum vertical displacement of the seat is greater than the one of the previous model of sit-ski. This leads to a high level of comfort and allows wider choice for the damping coefficient c and spring stiffness k.

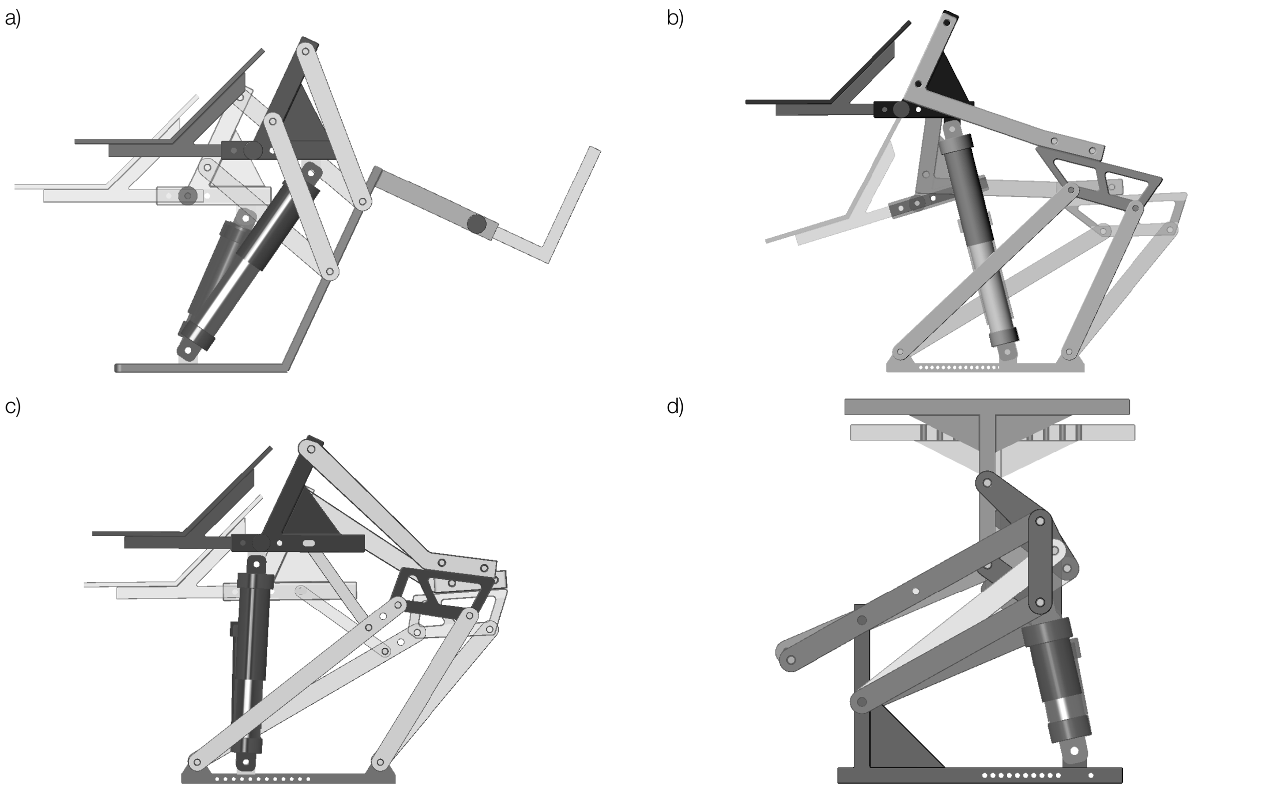

Figure 3 Trajectories of the moving parts a) US Patent US 6,019,380 b) Flex sit-ski c) modified Flex sit-ski d) eight-bar linkage

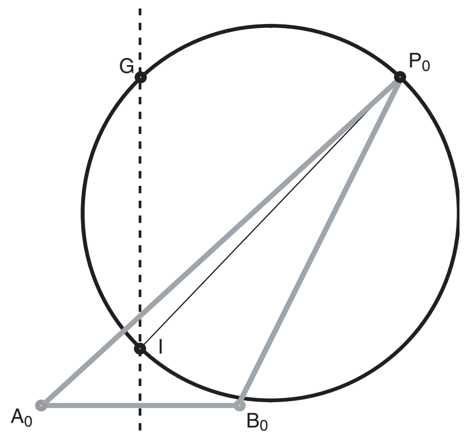

Figure 4 Inflection circle

Table 2 Dimensional features of the Flex model

Component	Value
Frame length	300 mm
First rocker length	418 mm
Second rocker length	281 mm
Crank length	120 mm
Crank angle[3]	14.25°
Max vertical range	17 mm

[3]w.r.t. the initial position of the frame

Modified Flex mechanism

This model is based on the same performance requirements adopted for the design of the previous sit-ski. Thus, apart from the main components already present, a new structure *guide-seat* has been introduced. This new element is essentially a swinging-block mechanism.

The function of this added mechanism is to maintain the seat parallel to the slope during all the motion of the sit-ski. This is the main difference with respect to the Flex model. A direct comparison of the two seat motions is shown in Fig. 5 (left-hand side shows the Flex model, and the right the modified Flex model).

The other advantage is to experience a vertical range of motion of the seat greater than with the previous model. This should increase the comfort of the sit-ski (Table 3).

Eight-bar linkage

This mechanism has eight hinged links, which can be split into two four-bar linkages. As a consequence, the whole system should have two degrees-of-freedom. However, a particular element called 'locker' (the element D_0A_1 in Fig. 6) has been introduced in order to decrease the number of degrees-of-freedom to one.

The second four-bar linkage has all moving links and shares the connecting rod of the first one.

The advantages of using the eight-bar linkage are summarized as follows.

1 Due to the kinematic structure and link dimensions, the seat is always parallel to the slope.
2 The centre of mass of the skier is always inside the area between the bindings. This improves the stability of the sit-ski.
3 The linkage ensures an optimal load/spring force ratio.
4 The whole mechanism is very compact.

Table 4 summarizes the dimensional features of the mechanism.

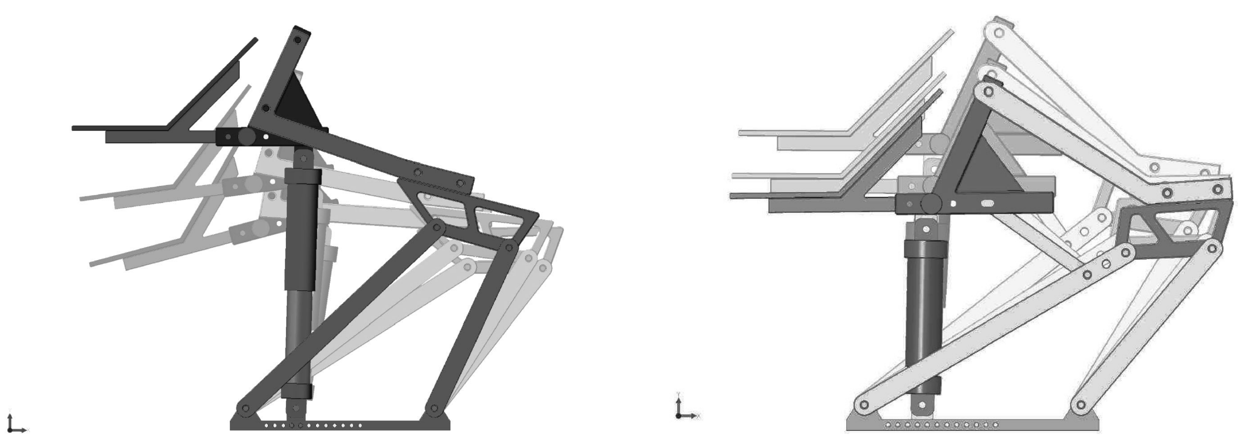

Figure 5 Comparison of seat motions

Table 3 Dimensional features of the modified Flex model

Component	Value
Frame length	300 mm
First rocker length	418 mm
Second rocker length	281 mm
Crank length	120 mm
Crank angle[4]	14.25°
Maximum vertical angle	182 mm

[4]w.r.t. the initial position of the frame

Figure 6 shows the four-bar linkage made up of the locker (D_0A_1), the frame (1)[2] (C_0D_0), the left rocker (1) (C_0A_1) and the crank (1) (A_1B_1). The function of this mechanism is to govern the motion of the second four-bar linkage and, as a consequence, of the seat in order to minimize the drift motion and to maintain the centre of mass inside the bindings area. This four-bar linkage has been designed by means of a kinematic inversion (Di Benedetto & Pennestrì 1993). In particular the following design data have been assumed:

1 the length of the frame of the first four-bar linkage C_0D_0;
2 the coordinates on the *xy* plane of the points A_1B_1 of the crank (1) and the frame (2): the values of the *x* coordinate of point B_1 are chosen in order to minimize the drift motion of the sit-ski. The values of these coordinates are reported in Table 5.

With reference to Fig. 7 a rigid triangle has been drawn between points A_1, B_1 and C_0. Then in the same way a triangle is drawn between points A_2, B_2 and C_0 and between points A_3, B_3 and C_0. By means of kinematic inversion these two new triangles are rigidly moved to the initial position (point A_1B_1) defining the position of the fixed hinge C'_0 and C''_0. The centre of the circle containing the three points C_0, C'_0 and C''_0 represents one of the moving hinges of the four-bar linkage. Applying the same procedure to the other fixed hinge D_0, the position of the second moving hinge is also evaluated. This graphical procedure allows the definition of the dimensions of all the links of the mechanism for the prescribed motion.

In order to better understand the motion of each model, in Fig. 3 the trajectories of the moving parts are depicted.

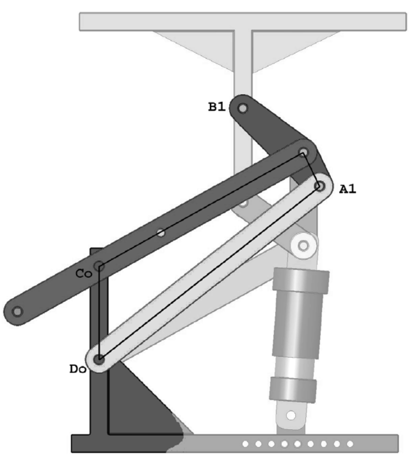

Figure 6 Front view of the eight-bar linkage

[2] The number in parentheses represents the four-bar linkage to which the link belongs.

Table 4 Dimensional features of eight-bar linkage model

Component	Value
Frame (1) length	100 mm
First rocker (1) length	249 mm
Second rocker (1) length	249 mm
Crank (1) – Frame (2) length	100 mm
First rocker (2) length	81 mm
Second rocker (2) length	81 mm
Crank (2) length	100 mm
Locker length	302 mm

Table 5 Coordinates of crank points

Points	X (mm)	Y (mm)
A_1	240	180
A_2	255	158
A_3	270	131
B_1	156	260
B_2	162	228
B_3	167	186

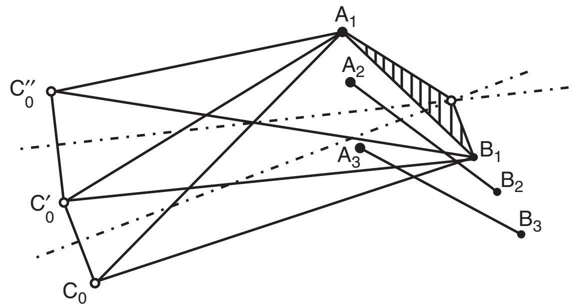

Figure 7 Kinematic inversion

Dynamic analysis

In this section the results of the dynamic analyses of the proposed models are presented. These analyses have been useful in order to define not only the dynamic behaviour of the model under investigation, but also the vibrational comfort according to the standard BS 6841 (1987) (Griffin 1990). The analyses have been executed simulating the profile of the track by means of two different analytical functions with the following features:

1 medium slope 45°
2 periodic variation of the amplitude of the profile: 400 mm along 20 meter of track
3 constant velocity of the sit-ski: 72 km/h.

In particular, the analytical expression of the first profile is

$$h = \frac{H}{\pi}\left[\frac{\pi t}{T_L} - \frac{1}{2}\sin\left(\frac{2\pi t}{T_L}\right)\right] \quad (2)$$

where $H = 400$ mm and t is the time.

In order to simulate the loads due to the presence of ice on the slope and imperfections of the track, the overall profile is obtained by blending profiles of type of Eq. (2) with straight-lines.

The characteristics of the cycloidal type profiles are summarized in Table 6.

The cycloidal type of slope profile is plotted in Fig. 8.

The second profile is of the harmonic type. This, when compared to the previous one, simulates harsher slope. The equation used for this purpose is:

$$h = H_1 \sin^3\left(\frac{\pi t}{T_L}\right) + H_2 \sin^3\left(\frac{\pi t}{T_L}\right) \quad (3)$$

where $H_1 = 200$ mm is the main amplitude,
$H_2 = -20$ mm is the noise amplitude and t is the time.

The characteristics of the harmonic type of slope profiles are summarized in Table 7. The overall profile is plotted in Fig. 8.

In both of the proposed slopes a null sollecitation is interposed between each profile (cycloidal or harmonic) at different frequencies. This is useful in order to have the system dampen out oscillations before receiving further inputs.

Table 6 Cycloidal profile

Sollecitation	Time	T_L [s]
Null	from 0 to 2 s	-
Cycloid	from 2 to 6 s	2
Null	from 6 to 8 s	-
Cycloid	from 8 to 10 s	1
Null	from 10 to 12 s	-
Cycloid	from 12 to 13 s	0.5
Null	from 13 to 15 s	-
Cycloid	from 15 to 15.5 s	0.25
Null	from 15.5 to 17.5 s	-
Cycloid	from 17.5 to 17.75 s	0.125
Null	from 17.75 to 20 s	-

Table 7 Harmonic profile

Sollecitation	Time	T_L [s]
Null	from 0 to 2 s	-
Harmonic	from 2 to 6 s	2
Null	from 6 to 8 s	-
Harmonic	from 8 to 10 s	1
Null	from 10 to 12 s	-
Harmonic	from 12 to 13 s	0.5
Null	from 13 to 15 s	-
Harmonic	from 15 to 15.5 s	0.25
Null	from 15.5 to 17.5 s	-
Harmonic	from 17.5 to 17.75 s	0.125
Null	from 17.75 to 20 s	-

The result of the dynamic simulation is the acceleration time history of the attachment point of the seat to the sit-ski for each model considered. The acceleration of this particular point has been monitored for two reasons:

1 in order to define the position of the seat during the use of the sit-ski on the track
2 in order to evaluate the Vibration Dose Value (Griffin 1990).

Thus the dynamic analyses have been executed for each model by changing the position of the attachment point of the suspension to the frame (where it was possible). This was in order to investigate the influence of the mechanism configuration on the overall response.

The first mechanism analyzed was the four-bar linkage, as described in the US Patent US 6,019,380. The attachment point of the suspension could not be

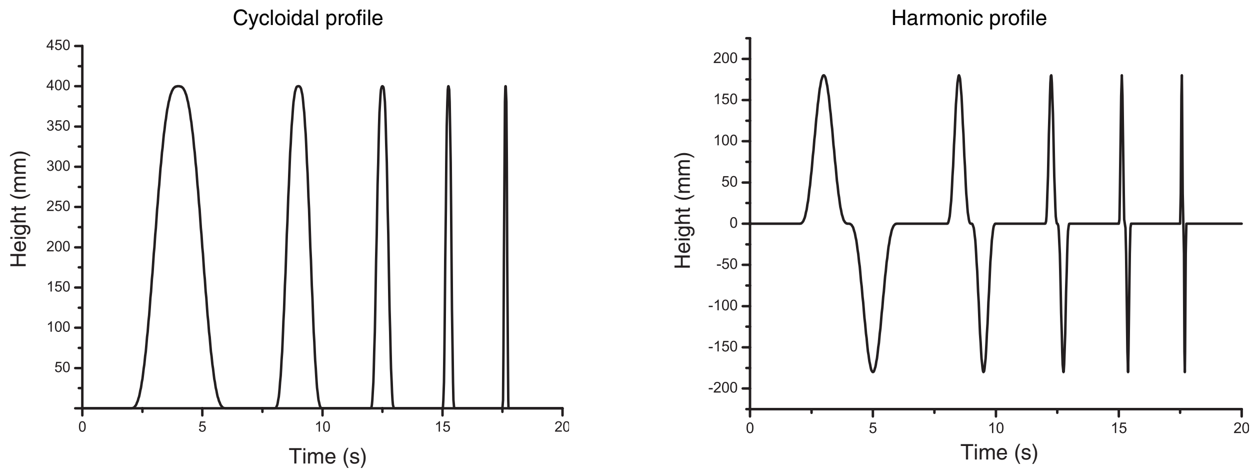

Figure 8 Cycloidal and harmonic profiles

modified in this mechanism and its configuration parameters and the shock absorber constants are summarized in Table 8.

In Fig. 9 are the FFTs of the acceleration time histories of the attachment point of the seat for both of the profiles simulated. In both cases there are two resonant frequencies. The first one, at 1 Hz, corresponds to the main frequency of the forcing input. The second one, at 3 ÷ 4 Hz, represents the resonant frequency of the structure. It is important to note that the FFT corresponding to the cycloidal slope has lower values of amplitude than the harmonic one, as expected.

The second model analyzed is the Flex with four different configurations of the attachment point of the suspension. Details of these configurations and of the shock absorber parameters are summarized in Table 9.

In Figure 10 are the FFTs of the acceleration time histories of the attachment point of the seat for both of the profiles simulated. Even in this case there are the same differences previously observed between the two track profiles. Considering instead the cycloidal profile only one main peak could be observed instead of the two peaks observed in the previous model.

The dynamic behaviour is also quite similar for each configuration of the suspension.

The third model analyzed is the modified Flex which has also four different configurations of the attachment point of the suspension. Details of these configurations and the viscoelastic parameters are reported in Table 10.

Table 8 Parameters for the dynamic analysis

Undeformed spring length	296 mm
Stiffness (k)	78 N/mm
Damping coefficient (c)	8 Ns/mm
Suspension configuration[5]	100 mm

[5]It is the distance of the attachment point from the border of the frame

Table 9 Parameters for the dynamic analysis

Stiffness (*k*)	60 N/mm
Damping coefficient (*c*)	10 Ns/mm
First configuration[6]	90 mm
Undeformed spring length	357 mm
Second configuration[6]	60 mm
Undeformed spring length	359 mm
Third configuration[6]	140 mm
Undeformed spring length	358 mm
Fourth configuration[6]	200 mm
Undeformed spring length	368 mm

[6]It is the distance of the attachment point from the border of the frame

Table 10 Parameters for the dynamic analysis

Stiffness (*k*)	60 N/mm
Damping coeff. (*c*)	10 Ns/mm
First configuration[7]	90 mm
Undeformed spring length	357 mm
Second configuration[7]	60 mm
Undeformed spring length	359 mm
Third configuration[7]	140 mm
Undeformed spring length	358 mm
Fourth configuration[7]	200 mm
Undeformed spring length	368 mm

[7]It is the distance of the attachment point from the border of the frame

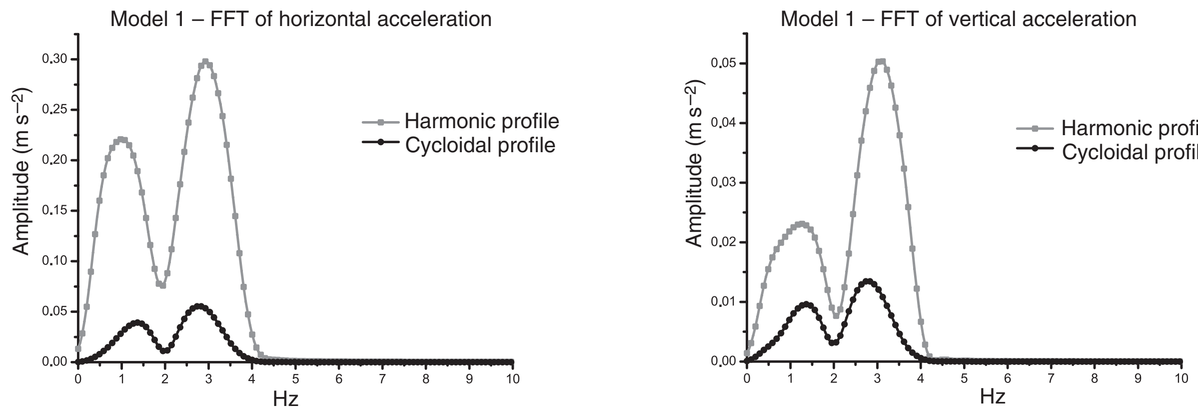

Figure 9 FFT of the accelerations of the first model

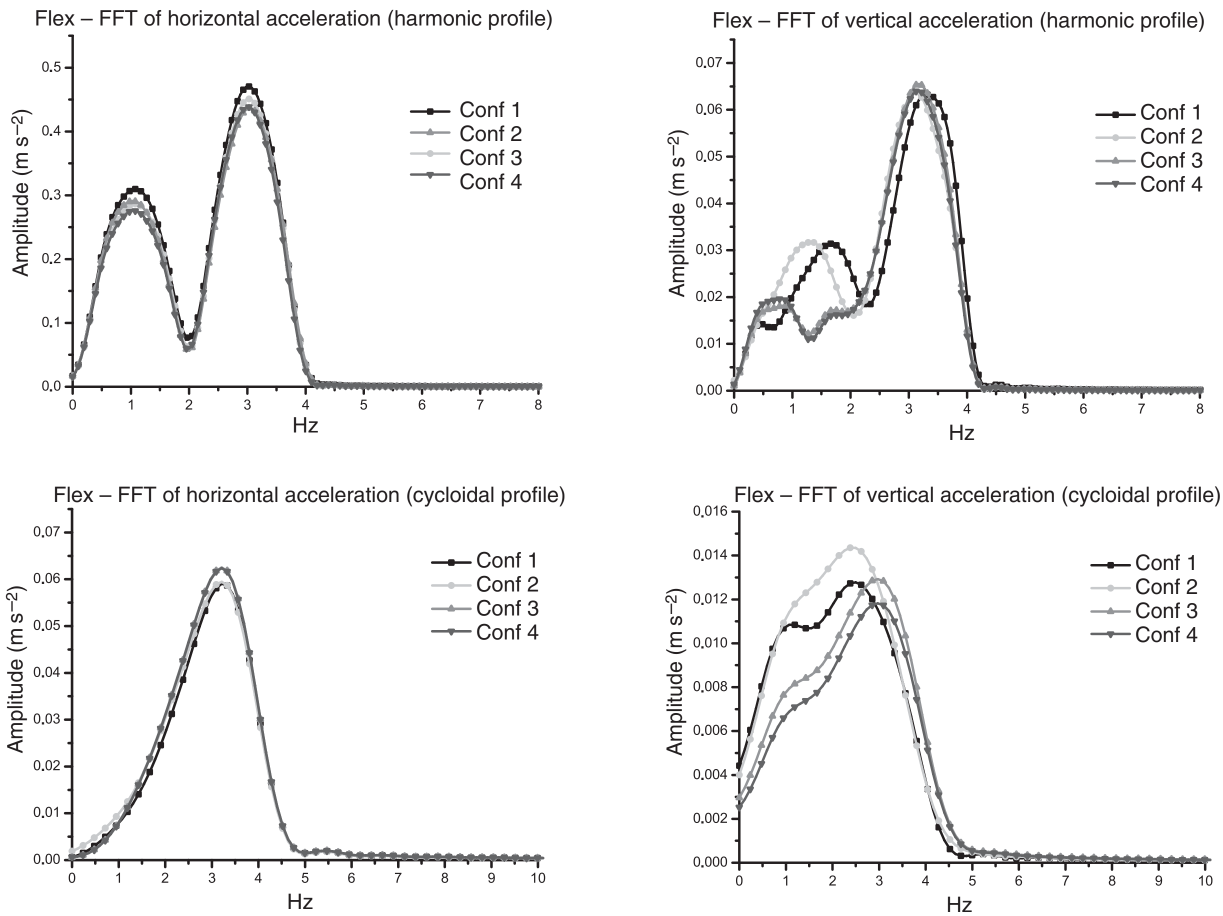

Figure 10 FFT of the accelerations of the Flex model

In Fig. 11 are reported the FFTs of the acceleration time histories of the attachment point of the seat for both of the track profiles simulated. The dynamic behaviour of the sit-ski has some differences when compared with the Flex model. With reference to the cycloidal slope, the FFTs of the horizontal accelerations of the two models are similar. The FFTs of the vertical accelerations appear more sensitive to the configuration changes of the suspension than the Flex model. Considering the harmonic profile both of the FFT characteristics (vertical and horizontal) are strongly dependent on the configuration of the suspension.

The last model analyzed is the eight-bar linkage in five different configurations of the suspension. The shock absorber parameters and details of these configurations are summarized in Table 11.

In Fig. 12 are reported the FFT of the accelerations of this last model. The dynamic behaviour is similar for each configuration except configuration number five which is more critical than the others. It should be observed that this model has the maximum value of acceleration amplitude (corresponding to the 'critical' configuration number five) lower than the other models. This feature affects also the vibrational comfort of the system as reported in the next section.

Vibrational comfort

Generally it is not easy to define the vibrational discomfort, because it is in some way a personal perception. What is apparent to anyone is that discomfort depends on the vibration frequency. At low frequencies the whole body responds as a rigid system,

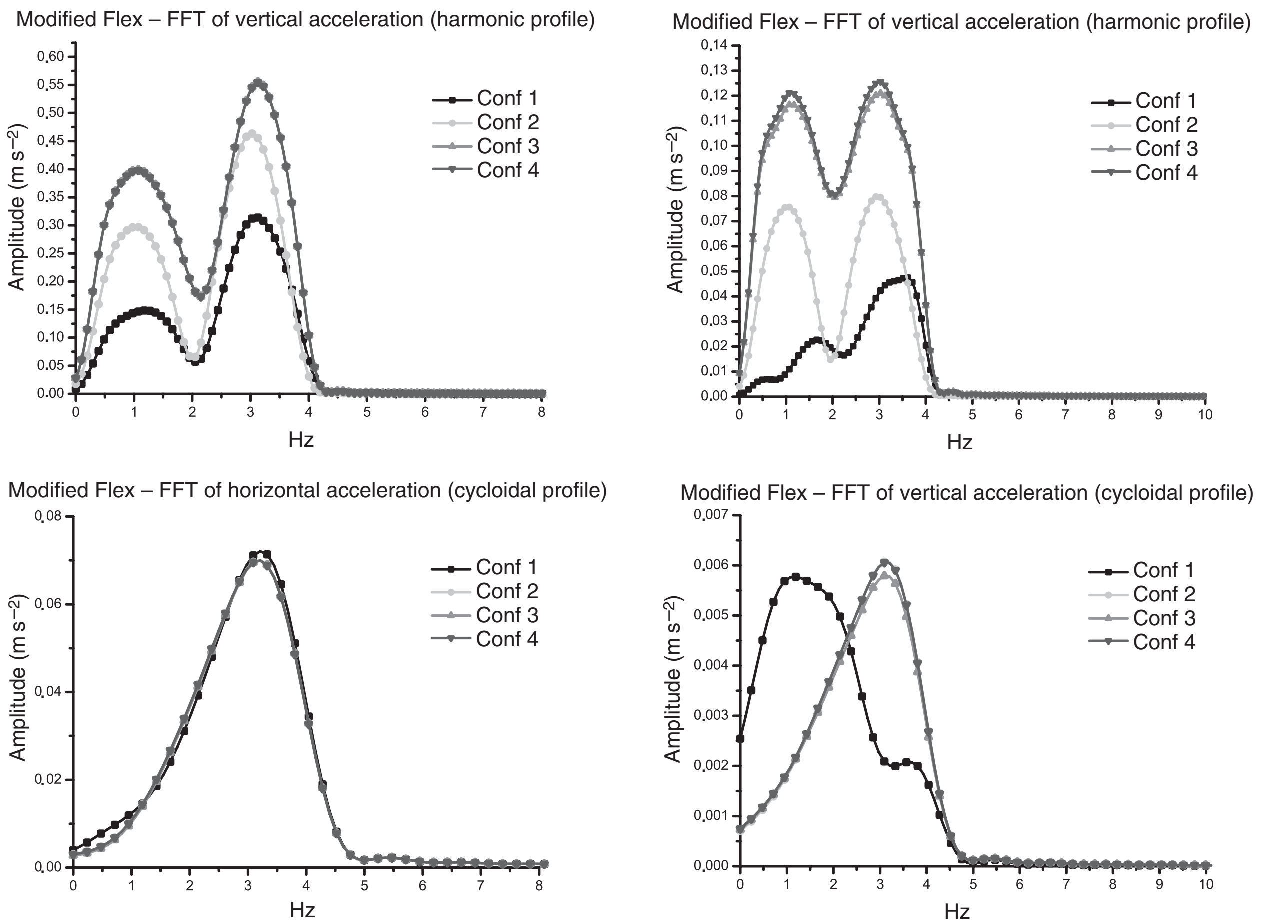

Figure 11 FFT of the accelerations of the modified Flex model

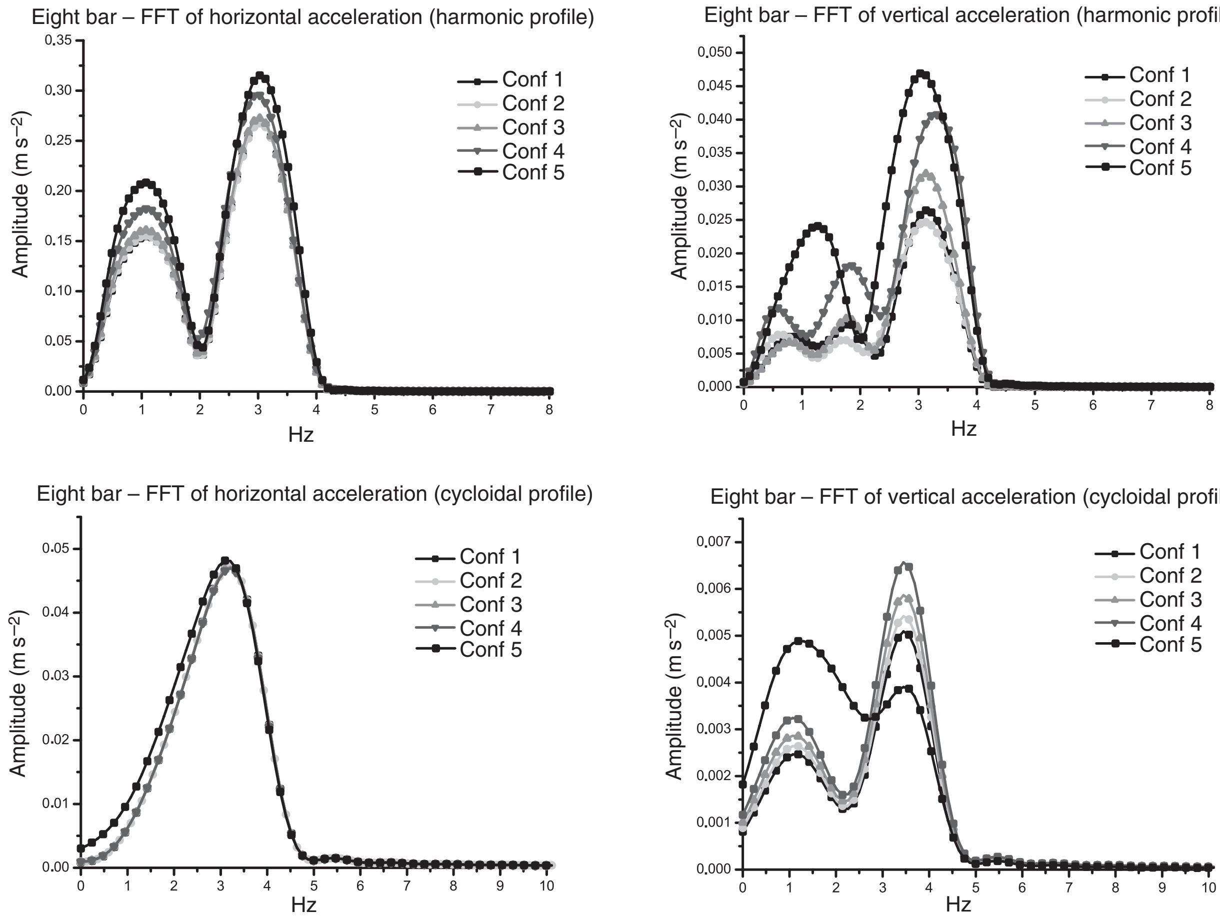

Figure 12 FFT of the accelerations of the eight-bar model

increasing the occurence of reaching resonance conditions for each part of the body (e.g. abdomen, thorax, shoulder and so on) and amplifying the discomfort. The other two main parameters are the magnitude of the vibration and the exposure time. The possible consequences of exposure to vibrational discomfort for long and frequent periods are tiredness, disturbed perception and several pathologies. For these reasons, it is important to define a reliable procedure for evaluating vibrational comfort. Many international standards try to give an objective evaluation of the vibrational comfort. ISO 2631(1974), slightly amended in 1982 (International Organization for Standardization 1982), suggested complex and restrictive methods for assessing some effects of vibration. It used an averaging method to evaluate vibration severity and was, therefore, unsuitable for assessing shocks, transients or non-stationary random vibration. These three last items could be managed by the British Standard BS 6841 (1987) which is applicable to the evaluation of all the types of oscillatory motion (British Standards Institution 1987). According to BS

Table 11 Parameters for the dynamic analysis

Stiffness (*k*)	60 N/mm
Damping coefficient (*c*)	10 Ns/mm
First configuration[8]	45 mm
Undeformed spring length	136 mm
Second configuration[8]	65 mm
Undeformed spring length	133 mm
Third configuration[8]	85 mm
Undeformed spring length	132 mm
Fourth configuration[8]	105 mm
Undeformed spring length	133 mm
Fifth configuration[8]	125 mm
Undeformed spring length	136 mm

[8]It is the distance of the attachment point from the border of the frame

6841, which is then the most advanced in this field, the standard parameter is the Vibration Dose Value (VDV). The expression for the VDV is

$$VDV = \left[\int_{t=0}^{t=T} a^4(t)dt \right]^{1/4} \tag{4}$$

where $a(t)$ is the frequency-weighted acceleration time history and T is the period of time over which vibration may occur. The 'maximum safe exposure' is set as a VDV value equal to 15 ms$^{-1.75}$. Above this limit a further exposure to vibration will be accompanied by an increased risk of injury.

In order to evaluate the VDV, the response of a virtual model of a dummy has been monitored. The dummy is made up of thirteen rigid bodies connected together by means of kinematic pairs. The code, written in Fortran language, was already experimentally validated for the vibrational comfort analysis of car occupants (Valentini and Vita 2003). The computed Vibration Dose Values for each model proposed are summarized in Table 12. These values refer to the harmonic profile which is the harsher one. Even if the limit value of VDV is far to be reached for each of the models of sit-ski analyzed, the last one (eight-bar linkage) is the best in regard to vibrational comfort and is in agreement with the results of the dynamic analyses.

Conclusions

This paper addressed the kinematic design and the dynamic analysis of sit-ski with classic and innovative kinematic structures.

In particular, three new models, called Flex, modified Flex and eight-bar linkage, have been herein proposed. The performance of the *classic* sit-ski model presented in the US Patent US 6,019,380 has been compared with the new structures.

Through numerical simulations the dynamic performance of different models has been compared by means of different quality indices. An in-house developed code, named DAViD (Dynamic Automotive Virtual Dummy), has been used for this purpose (Valentini and Vita 2003). The code is based on a multibody approach (Haug 1989). Each of the sit-ski models has been implemented together with a human body model. The virtual dummy is made up of twelve rigid bodies linked together by means of eleven kinematic pairs. Moreover three viscoelastic elements have been modelled to represent the muscular elasticity of elbow and knee articulations and torso. The output of the code concerns the acceleration time history of the attachment points of the seat to the sit-ski, of each body segment and, as stated by the BS 6841, the overall VDV. The input is represented by the track profiles as described in the dynamic analysis section.

The Vibration Dose Value (VDV), evaluated according to BS 6841 standard, is summarized in Table 12 for each model proposed. From the numerical simulations it appears that the eight-bar linkage model has superior capabilities with respect to the other structures tested. In particular it has the lowest values of acceleration FFT, for all seat configurations and for both of the tracks simulated and it is also confirmed by the overall VDV (3.35 ms$^{-1.75}$). This means that the vibrations perceived by the skier are dampened by the sit-ski structure. Moreover the angle between the seat and the surface of the slope is quite near to zero during the simulation. This means that an optimal visibility is ensured even for a harsh slope profile such as the harmonic one. These results obtained are in agreement with the intentions pursued during the synthesis phase.

Table 12 VDV of sit-ski models

Sit-ski	VDV
US 6,019,380	5.12 ms$^{-1.75}$
Flex	9.45 ms$^{-1.75}$
Modified Flex	4.21 ms$^{-1.75}$
Eight-bar linkage	3.35 ms$^{-1.75}$

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