MULTI-OBJECTIVE OPTIMAL DESIGN OF ROAD VEHICLE SUB-SYSTEMS BY MEANS OF GLOBAL APPROXIMATION

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SUMMARY

The paper presents a method for optimising road vehicle sub-systems. Provided that vehicle sub-system models are available for accurate simulations, the method allows to optimally tune the parameters of the considered sub-systems in order to improve their performances. The optimisation method is based on Multi-Objective Programming and on Global Approximation. A number of Global Approximation techniques are presented and their respective performances (in terms of computation time) are compared.
Two vehicle sub-systems are considered, namely the body and the engine suspension systems. The ride comfort, road holding and handling are studied and improved by a proper tuning of the suspension system parameters.
A new automated ADAMS®/Matlab® procedure for the preliminary design of engine suspension systems is introduced, relying on the presented method. The procedure has been implemented into a tool which has been fully integrated in the FIAT product development process and is currently applied for designing new vehicles.

1. INTRODUCTION

The paper introduces a method to support designers to find optimal & robust solutions pertaining to complex mechanical systems, with particular reference to vehicle sub-systems. Multi-Objective Programming (MOP) is taken as the appropriate framework for the definition and solution of the addressed problems. MOP (or Pareto-optimisation) may be considered as a tool providing a set of efficient solutions. The designer can choose one of them according to his motivated preferences [1, 2, 3, 4, 5, 6]. The set of efficient solutions is derived by a two-step numerical computation, called Global Approximation [see, e.g. 20]. At first one or more ADAMS® models are used to simulate, in a relatively small number of different cases, the dynamic behaviour of the vehicle sub-system under consideration. On the basis of the results of these simulations, a Global approximation model is derived in order to simulate the dynamic behaviour of the considered system, very quickly, in many different cases.

The presented method is applied for the optimal design of a car body suspension system. Both a very high number of parameters and a large set of driving situations could be taken into account, leading to a comprehensive optimisation. The presented optimisation, even if subject to further major refinements, seems to be one of the most extensive ever presented in the literature [1,8,16,17, 24, 26].
As a second application of the presented method, an automatic procedure for the preliminary design of the engine suspension system is developed. In the literature many efforts were devoted to the derivation of models more than to the synthesis design problem which is dealt with here

[10, 22, 23, 25]. The design procedure is completely automated and user-interfaced through customisation of ADAMS® and Matlab® commercial softwares. A wide variety of controls allow the user to manage the process and the huge amount of information involved. The effectiveness of the procedure is finally assessed through the study of different engine suspension layouts.

2. ADAMS® PHYSICAL SYSTEM MODELS

2.1. Full car system models

The reference car taken into consideration is a prototype, whose performances are close to the best in its class. To simulate the many different running/driving situations required for the design of the suspension system, three full vehicle ADAMS® models have been developed, validated and used.

Different tyre models have been considered depending on the simulated running/driving situation (ride comfort, road holding and handling). For example, Magic Formulae are used for handling, and complicated multi d.o.f. tyre models are used for simulating the passing over a cleat. The tyre models that have been exploited are summarised in Tab.2.

Every body constituting the vehicle is considered to be rigid, actually the flexibility of bodies can be neglected when the manoeuvres involve the “primary” and ”secondary ride” frequency range (0-30 Hz) [10]. The non-linear elastokinematics of the suspension system is accounted for. The modelling of the non-linearity of the suspension components are summarised in Tab. 1

Tab. 1. List of sub-systems composing the full vehicle model.

Vehicle Sub-system	**Layout**	**Non linear components**
Powertrain suspension	Three attachment points	Powertrain mounts
Front suspension	McPherson	Control arm bushings Damper top mount Shock-absorber Force/Velocity curve Shock-absorber friction Spring and bumpstop
Rear suspension	Trailing arm with subframe	Rear subframe mounts Damper bushings Shock-absorber Force/Velocity curve Shock-absorber friction Spring and bumpstop

Tab. 2. Tyre and road irregularity modelling as function of the simulated manoeuvre.

Manoeuvre	**Tyre/road model**
Passing over a cleat	Pirelli tyre model Semi-physical, 4 dof, planar Profile of the obstacle experimentally identified
Pothole symmetrical Pothole non-symmetrical	Pirelli tyre model Semi-physical, 2 dof, planar Analytical description of the road profile
Uneven road	Tyre modelled as a spring Identification of road profile based on experimental data

The three full car system models have the same body and different tyre models (Tab.2). These three models were used by FIAT during the design and development process of the considered vehicle. For the present application, they have been deeply modified in order to allow

- a parametric description of the non-linear components of the suspensions, (e.g. the control arm bushings of the front suspension, the shock-absorber damping ratio, etc.)
- user written subroutines for the calculation of the performance indexes
- automatic generation of the models/simulations and collection of the output data in formatted tables.

2.2. Engine suspension system model

The main engine suspension system layouts commonly used on FIAT vehicles are taken into account. Mechanical systems composed by two mounts + torque rod or three mounts with or without additional torque rod are modelled through ADAMS®. These models include a simplified but suitable representation of the powertrain mounting system. They can be used for the preliminary design of the engine suspension system. A new procedure has been implemented by means of a proper customisation of the ADAMS/View® environment with the aim of automatically generating the models/simulations required, as summarised in Table 3. The procedure accounts for three different engine suspension layouts, introduces a parametric description of the non-linear engine mounts and includes user written subroutines for the calculation of the performance indexes and for a suitable collection of the output data

Tab. 3. Reference manoeuvres for the preliminary design of the engine suspension

Manoeuvres	**Description**
Progressive braking	Steady manoeuvres. Constant acceleration. Inertial forces applied to the engine C.G.
Right cornering	
Left cornering	
3rd gear acceleration	
Uneven road	Multibody frequency-domain simulation Tyre modelled as a spring FRF calculation (linearised model)
Engine run-up	Calculation of the structure-borne 2nd order noise at driver's ear

The vehicle body structural response is modelled by means of proper acoustic transfer functions for calculating the structure borne engine noise. The noise spectrum due to the 2nd order excitation (in-line four cylinder engine) is calculated as a sum of every single engine mount contribution (*i*) in the *j* direction:

$$\overline{SPL} = \sum_{i} \sum_{j} \overline{disp}_{ij} * \overline{K}_{ij} * \overline{TFA}_{ij} \qquad (1)$$

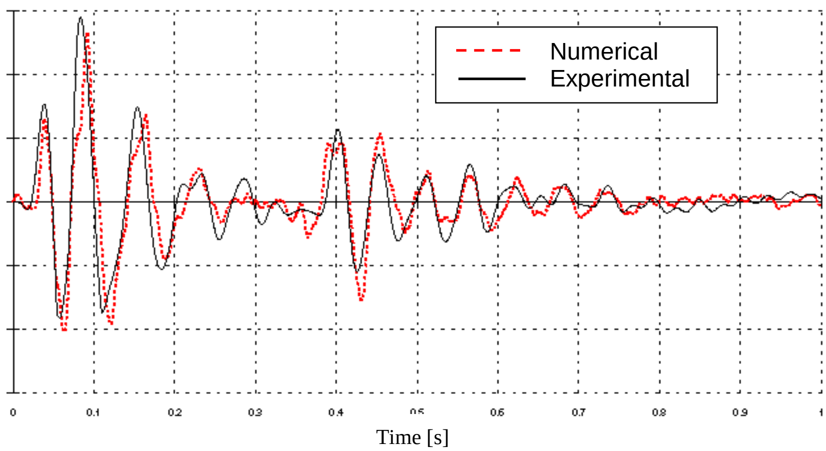
where *SPL [Pa]* = sound pressure level
disp [mm] = 2nd order dynamic displacement of the engine bracket (measured)
K [N/mm] = dynamic stiffness of the mount
TFA [Pa/N] = acoustic transfer function

The vertical transmissibility function is obtained from a standard multibody simulation (linearised full vehicle model). The non-linearity of the powertrain mounts have been taken into account only for validation purposes. A linearised model of the engine suspension system has proved to be accurate enough to perform the preliminary stage of the design process.

2.3. Validation

The validation of the physical models has been conducted thoroughly, taking into account both the time and the frequency domain response of the vehicle. Figure 1 and Figure 2 show examples of the satisfactory accuracy achieved by considering different manoeuvres.

Seat longitudinal acceleration – Time domain



Seat longitudinal acceleration – Frequency domain

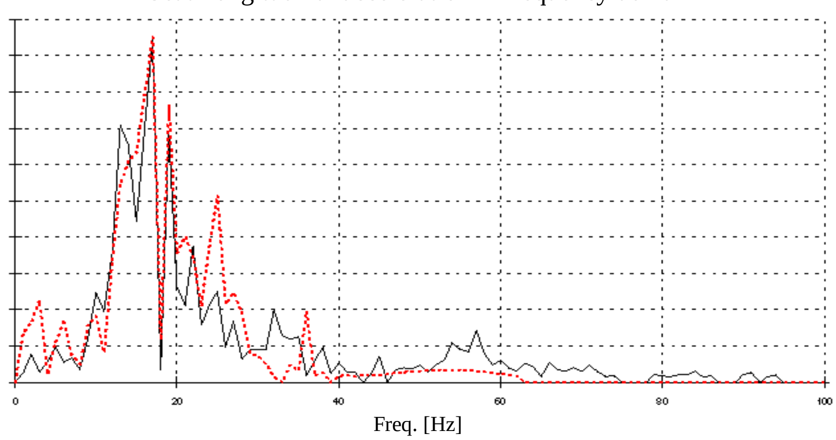


Fig. 1. Passing over a cleat 100x25 mm wide at a speed of 30 km/h.
Numerical vs. Experimental results for the reference FIAT car.

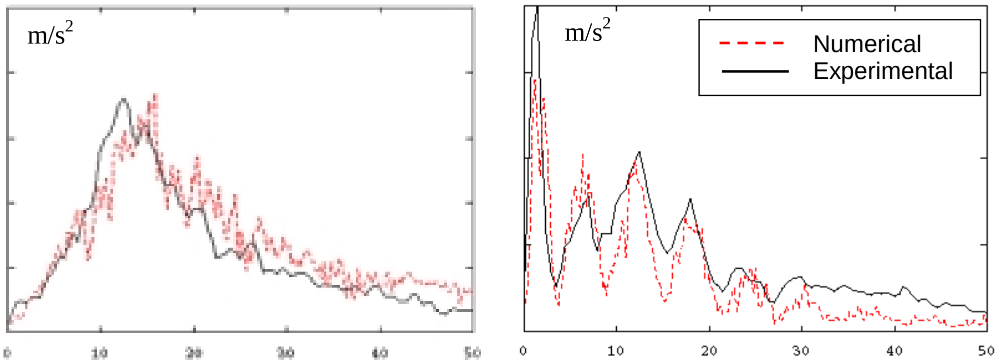


Fig. 2. Passing over a rough road at a speed of 60 km/h. Numerical vs. Experimental results.

The validation of the vehicle model considering the J-turn manoeuvre is shown in Table 4. The vehicle speed is 100 km/h and the steering wheel input is 90 deg.

Tab. 4. Model validation considering a J-turn manoeuvre (100 [km/h], steering wheel input 90 [deg], tyre size 205/55). The symbols are described in detail in section 3.

	Experimental	Computed
Slip angle at c.g. β [deg]	1.4	1.4
Roll angle θ [deg]	3.2	3.2
$\dot{\psi}$ steady-state yaw rate [deg/s]	12	13
Delay time *D50-Acc-y* [s]	0.12	0.11

The models referring to the engine suspension system have been validated and the results have been already presented (see [21]).

3. VEHICLE SUSPENSION OPTIMISATION: PROBLEM FORMULATION

DESIGN VARIABLES

Thirty-six design variables (system model's parameters) are tuned which refer to:

- the suspension elastic characteristic (front and rear)
- the characteristics of the rubber suspension bearings (front and rear)
- the characteristic of the hydraulic dampers (front and rear).

The design variables are described in detail in the following subsections.

3.1. Hydraulic dampers

Three design variables (r_{bls}, r_{hs}, r_{rls}) are employed to describe the non-linear shock absorbers characteristic (Fig. 3). The transition speed (V_0) is kept constant during the optimisation procedure.

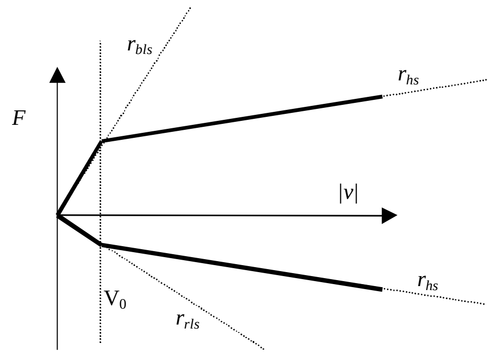


Fig. 3. Non-linear hydraulic damper characteristic, force *F* vs. damper speed *v*.

3.2. Suspension springs and bump stops.

The non-linear elastic spring-bump stop characteristic is shown in Figure 4.
Three design variables summarise the non linear behaviour

K [N/mm]	the spring stiffness
X_{it} *[mm]*	bump stop engaging position
i [-]	exponent of the non-linear characteristic of the bump stop.

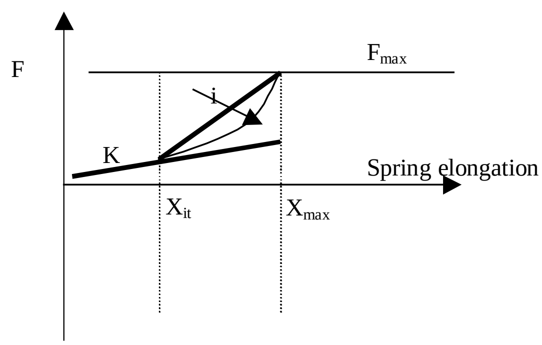


Fig. 4. Non-linear spring and bump stop characteristic.

The total load on the suspension is given by

$$F = K \cdot X + F_{max}\left((X - X_{it})/(X_{max} - X_{it})\right)^{i} \qquad (2)$$

where *X [mm]* is the spring elongation and *Fmax [N]* is a known parameter.

3.3. Suspension rubber bearings

The bushing characteristics play a fundamental role in the definition of the vehicle dynamic behaviour, particularly at high acceleration levels and in presence of step inputs. Each bushing characteristic listed in Table 5. has been optimised using three design variables: the linear stiffness *K [N/mm]* , the linear zone upper limit X_{it} *[mm]* and the exponent of the non-linear zone *i [-]*.
The analytical expression of the total load on the bearings is identical to (2)

Tab. 5. List of the bushings whose characteristics have been varied during the optimisation.

Front suspension: wishbone arm front bushing (x direction)
Front suspension: wishbone arm front bushing (y direction)
Front suspension: wishbone arm rear bushing (x direction)
Front suspension: wishbone arm rear bushing (y direction)
Rear suspension: sub-frame front bushing (x direction)
Rear suspension: sub-frame front bushing (z direction)
Rear suspension: sub-frame rear bushing (x direction)
Rear suspension: sub-frame rear bushing (z direction)

RUNNING SITUATIONS AND PERFORMANCE INDEXES

In the following application example, a total number of ten different manoeuvres and running conditions (driving situations) are introduced.

We define performance indexes the quantities given by the following expression

$$PI = f(x_v, x_f) \tag{3}$$

where: PI is the performance index
x_v is the vector of the design variables (varied during optimisation)
x_f is the vector of the parameters (fixed during optimisation)

The performance indexes refer to a single running condition, the objective functions are the performance indexes evaluated at different parameters values (for instance different vehicle speeds). Obviously, there are hundreds of metrics that may be derived from vehicle tests. The list has been reduced to the minimum. The goal of the designer is to meet all of the performance targets and produce a balanced vehicle.

Fifteen performance indexes have been introduced, assuming best values when they are *minimised*, i.e.:

1. *P-P(Xg)* Peak-to-peak value of the driver seat longitudinal acceleration
2. *P-P(Zp)* Peak-to-peak value of the driver seat vertical acceleration
3. *RMS(SP)* RMS value of the sound pressure inside the vehicle
4. *RMS(RH)* RMS value of the vertical force acting between road and wheel
5. *P-P(TH)* Peak-to-peak value of the pitch acceleration of the vehicle body
6. *P-P(RO)* Peak-to-peak value of the roll acceleration of the vehicle body

7. *RMS(TH)* RMS value of the pitch acceleration of the vehicle body
8. *RMS(RO)* RMS value of the roll acceleration of the vehicle body
9. *IQV* Discomfort coefficient given by the weighted sum of the RMS value of the filtered driver seat vertical and longitudinal acceleration

Five additional performance indexes refer to the J-turn manoeuvre.

10. *Acc-y* Vehicle body lateral acceleration at steady state.
11. β Slip angle at centre of gravity at steady state.
12. θ Roll angle at steady state.
13. *D50-Acc-y* Delay between lateral acceleration (at 50% of the steady state value) and ramp steering wheel input (at 50% of the steady state value).
14. *D50-*$\dot{\psi}$ Delay between yaw rate (at 50% of the steady state value) and ramp steering wheel input (at 50% of the steady state value).
15. *D50*-θ Delay between roll angle (at 50% of the steady state value) and ramp steering wheel input (at 50% of the steady state value).

The last five performance indexes assume best values when the deviations from a reference value are *minimised* (goal programming). The indexes have been computed for different speeds as indicated in Table 6, together with. the description of different running conditions. The performance indexes considered for the different driving conditions are summarised in Table 7.

Tab.6. Running conditions.

Running condition	Vehicle Speed	Description
Passing over a cleat	30 – 50 km/h	Obstacle 100 x 25 mm
Passing over a symmetric pot-hole	40 – 80 – 120 km/h	-
Passing over a non-symmetric pot-hole	40 – 80 – 120 km/h	-
Rough road (random profile)	60 km/h	-
ISO 7401 J-turn	100 km/h	90 deg steering wheel input

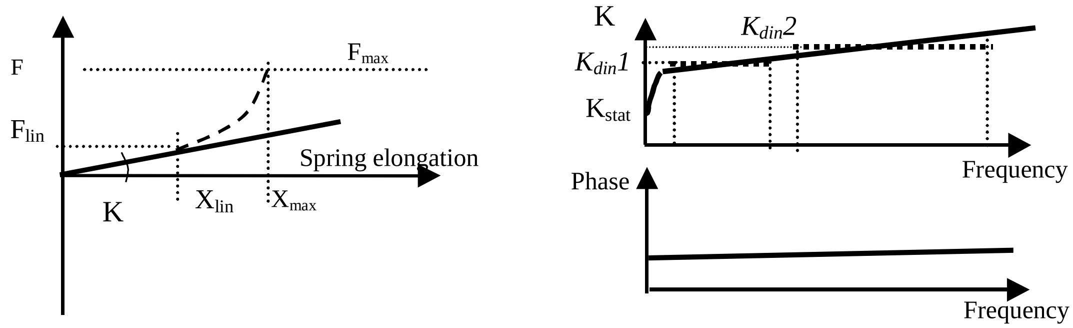
Tab.7. Summary of the defined performance indexes.

Passing over a cleat	P-P (Xg)	MAX RMS(RH)	RMS (SP)			
Passing over a symm. pot-hole	P-P (Zp)	MAX RMS(RH)				
Passing over a non-symm. Pot-hole	P-P (Zp)	MAX RMS(RH)	P-P (TH)	P-P (RO)	RMS (TH)	RMS (RO)
Rough road	IQV	MAX RMS(RH)				
J-turn	*Acc-y*	β	θ	*D50-Acc-y*	*D50-*$\dot{\psi}$	*D50*-θ

4. ENGINE SUSPENSION OPTIMISATION: PROBLEM FORMULATION

DESIGN VARIABLES

Depending on the suspension layout under investigation, *from seven to ten* design variables are tuned, which refer to the linear stiffness of each powertrain rubber mount in each X,Y,Z direction and possibly to the axial linear stiffness of each rod.



K_{stat} = static stiffness
$K_{din}1$ = approximated dynamic stiffness for ride manoeuvres
$K_{din}2$ = approximated dynamic stiffness for 2nd order noise calculation

Figure 5. Schematic representation of the engine mounts characteristics

With reference to Figure 5, in standard operating conditions the forces acting on each mount in each direction are assumed not to exceed the "linear" value $F_{lin}[N]$: this is a correct assumption that allows a dramatic reduction of the number of design variables that have to be tuned contemporarily. The dynamic stiffness is considered to be a function of the static stiffness value, constant in the frequency domain. The nearly constant phase shift of the rubber component is assumed to be related to a viscous damping.

RUNNING SITUATIONS AND PERFORMANCE INDEXES

Six different manoeuvres are introduced, as sketched above in Table 3. Three performance indexes assuming best values when they are *minimised* have been calculated:

1. *Mean(Spl)* Mean value of the spectrum of the 2nd order noise [dB] at driver's ear (50-200 Hz frequency range)
2. *Peak(Spl)* Peak value of the spectrum of the 2nd order noise [dB] at driver's ear (50-200 Hz frequency range)
3. *Peak(Zp)* Peak value of the vertical transmissibility function [-] between the tire contact patch and the driver's feet in the 5-20 Hz frequency range (where dynamic interaction between engine and suspension modes occurs).

In addition to these main performance indexes, a number of constraints have to be set. The engine mounts/rod displacements *X, Y, Z* and the engine rotations *Rx, Ry, Rz must not exceed* *proper threshold values* during the four different steady manoeuvres. Thus from *forty to fifty-two* additional constraints have to be considered as summarised in Table 8.

Tab. 8. Summary of the engine suspension performance indexes (bold) and constraints.

Manoeuvre	Mount 1	Mount 2	Mount	Torque rod	Engine
Braking	*X,Y,Z*	*X,Y,Z*	*X,Y,Z*	*X*	*Rx, Ry, Rz*
Right cornering	*X,Y,Z*	*X,Y,Z*	*X,Y,Z*	*X*	*Rx, Ry, Rz*
Left cornering	*X,Y,Z*	*X,Y,Z*	*X,Y,Z*	*X*	*Rx, Ry, Rz*
3rd gear acc.	*X,Y,Z*	*X,Y,Z*	*X,Y,Z*	*X*	*Rx, Ry, Rz*
Uneven road	***Peak(Zp)***				
Engine run-up	***Mean(Spl)***		***Peak(Spl)***		

Considering the previous formula (1), its evident that engine mounts must be as compliant as possible in order to minimise the structure borne engine noise during run-up manoeuvres. At the same time they have to be stiff enough to satisfy the wide set of displacement constrains. Additionally, the stiffness distribution on the attachment points must ensure the minimum interaction between the engine and front suspension rigid-body modes, while running on rough roads.

5. GLOBAL APPROXIMATION

In order to save computation time, the *physical* ADAMS® models are substituted by purely *numerical* models. This procedure is called Global Approximation as the responses of the physical models are approximated by purely numerical models. The type of numerical model that should be used depend on the problem under consideration. With reference to the vehicle suspension optimisation problem (section 3), a study is presented for assessing the accuracy of the approximation obtainable by different mathematical models. The results are summarised in Table 9 where the performances of linear interpolation, of quadratic approximation [20], of kriging modelling [19,20] and of artificial neural networks [14,15,20] are compared. The errors have been computed on a 'validation set' constituted by 100 solutions not belonging to the training set. The performances of the artificial neural network are evidently superior.

Tab. 9. Mean and standard deviation of the absolute value of the errors on all the objective functions, using 950 exact evaluations.

	Mean ABS Err.	Std ABS Err.
Linear Interp.	>20%	>20%
Quad. Approx LS	12.1%	15.9%
Kriging	2.6%	2.4%
Neural Net	1.9%	1.7%

Additionally, by employing an artificial neural network the computation time can be reduced typically up to *one thousand* times with respect to an equivalent multi-body calculation. The main challenge is to generate a reliable approximation without an excessive number of exact evaluations performed by means of the physical model.

6. SEARCH METHOD

A search based on a number of uniformly distributed parameter vectors [4, 5, 13] (within their respective variation ranges) can be very effective.

6.1. Performance indexes reduction

The set of simulations generated by means of a uniformly distributed sequence has been analysed in order to reduce, if possible, the number of performance indexes that have to be taken into account in the subsequent optimisation stage. This is done by using the 'Spearman rank correlation coefficient' [7]. If strongly correlated performance indexes are found (even non-linear correlations can be detected), the existence of a redundancy is discovered and the number of performance indexes can be reduced.

6.2. Pareto-optimal solutions

Optimal solutions can be obtained by resorting to the Pareto-optimal solution definition. Given a multi-criteria minimisation problem defined by n design variables and k objective functions, a solution x_i is Pareto-optimal (i.e. non-dominated) if a solution x_j doesn't exist such that

$$\begin{cases} f_n(x_j) \leq f_n(x_i) \quad n = 1,2,3,..,k \\ \exists l : f_l(x_j) < f_l(x_i) \end{cases} \tag{4}$$

A large number (>1.0 E5) of Uniformly Distributed solutions can be generated by the Neural Net approximation within a short time. The condition (4) can be checked and the Pareto-optimal solutions can be stored.

6.3. Robust design

The fundamental principle in robust design is to minimise variation in performance caused by variations in design variables, thus providing insensitivity to design variables uncertainty.
The minimum sensitivity method has been employed for achieving robustness [12]. No probability distributions are needed and it can be proved that minimum sensitivity implies greater robustness.

7. VEHICLE SUSPENSION OPTIMISATION: - NUMERICAL RESULTS

One thousand uniformly distributed design variables vectors have been generated and used to compute the exact responses using the physical ADAMS® models. The set of generated solutions has then been analysed by using the 'Spearman rank correlation coefficient' and redundancy between strongly correlated (*Rs>0.95*) performance indexes has been eliminated. The correlation analysis has allowed to isolate a subset of only 27 of the 36 initial objective functions that can be considered as fully representative of the vehicle dynamic behaviour with limited loss of information.

Tab. 10. Improvements of a number of performances of an optimised vehicle with respect to the reference prototype car and normalised variations considering the J-turn manoeuvre.

Driving Situation	Vehicle Speed	Performance Index	Improvement with respect to reference car
Passing over a cleat	30 km/h	Longitud. Accel.	-10%
		Road Holding	0%
		Sound Pressure	-1%
	50 km/h	Longitud. Accel.	-6%
		Road Holding	-1%
		Sound Pressure	0%
Pothole symmetrical	40 km/h	Vertical. Acc.	-22%
		Road Holding	-28%
	80 km/h	Vertical. Acc.	-18%
		Road Holding	-9%
Pothole non-symmetrical	40 km/h	Vertical. Acc.	-22%
		Road Holding	-18%
		Roll Acc.	-27%
		Pitch Acc.	-24%
	80 km/h	Road Holding	-18%
		Roll Acc.	-9%
		Pitch Acc.	-3%
	120 km/h	Road Holding	-7%
		Roll Acc.	-11%
		Pitch Acc.	-2%
Randomly uneven road	60 km/h	Discomfort	-11%
		Road Holding	-4%
			Variation with respect to target car
J-turn manoeuvre	100 km/h	Lateral acceleration	3%
		Slip angle at c.g.	3%
		Roll angle	8%
		Delay on lateral acc.	0%
		Delay on yaw rate	14%
		Delay on	1%

On the basis of the 1000 ADAMS® calculated solutions, three Neural Networks have been trained, up to a mean error lower than 2%. The pareto-optimal set has been computed according to the procedure described in the previous sub-section, by applying the definition (4).

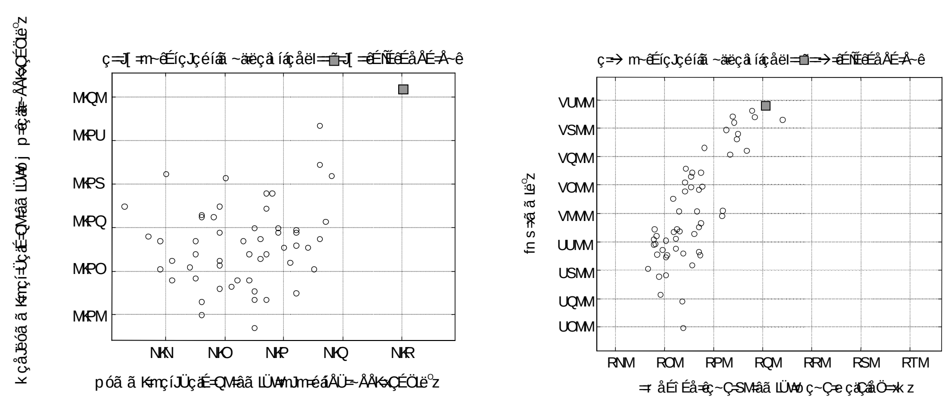


Fig. 6. Pareto-optimal set projections onto the roll/pitch acceleration and IQV/road holding domains.

Figure 6 shows two bidimensional projections of the Pareto-optimal set. The comparison of the performances of the candidate optimal solutions with those of the reference vehicle stresses the sensible improvement that can be obtained for almost all the objective functions.
Only minor improvements can be observed for sound pressures.
Almost all the Pareto-optimal solutions are robust according to the minimum sensitivity check.

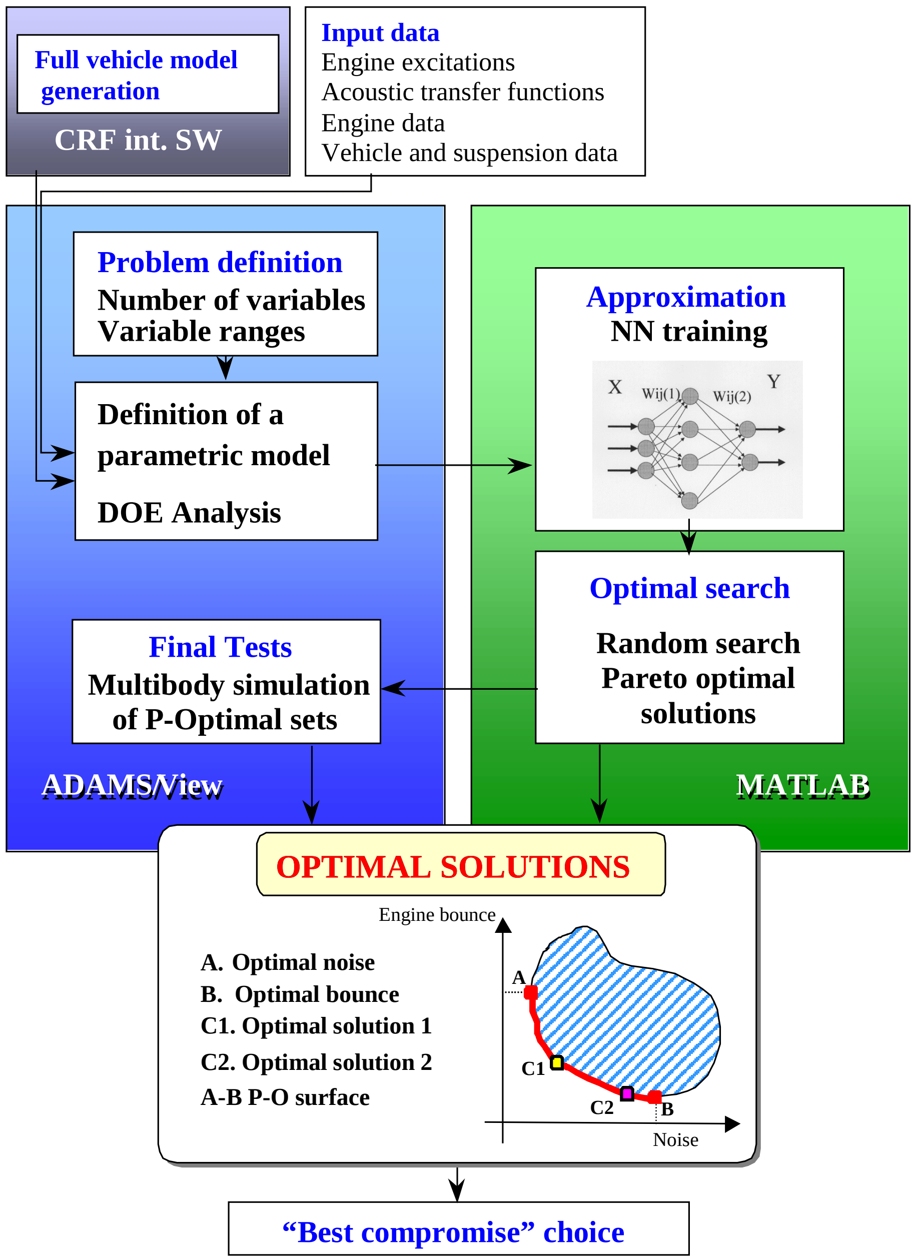


Fig. 7. Multi-objective procedure for the optimisation of the engine suspension system

The improvements of one optimal solution with respect to the reference prototipe car are discussed in detail and shown in Table 10. The average improvement (on all of the 27 performance indexes taken into consideration) is about 12%, while some of the performance indexes prove to be 30% better than the corresponding of the reference car. The improvement of the handling behaviour (J-turn manoeuvre) is comparable.
The time requested for the application of the optimisation procedure is reduced from several weeks to some days, including the simulation time needed for the neural network training.

8. ENGINE SUSPENSION OPTIMISATION: RESULTS

The concepts outlined in the previous pages have been applied to the definition of the design procedure synthesised in Figure 7. The whole set of sequential operations has been completely automated and user-interfaced through customisation of commercial software.

The user is required only to provide a suitable ADAMS® model and the input data needed for the different simulations and then

- select the engine suspension layout to be analysed
- choose a suitable range of variation of the design variables
- properly set the thresholds on the maximum displacements allowed to the engine system
- monitor the process evolution.

No correlation analysis (ref. Sub-section 6.1) is performed, because the optimisation problem has been designed in order to avoid redundancy in the number of performance indexes.
Figure 7 shows an additional stage of multi-body simulations. For each set of parameters leading to a Pareto-optimal solution a corresponding ADAMS® simulation is performed in order to increase the accuracy of the numerical results. The user can also graphically compare this Pareto-optimal surface with the results of the artificial neural network evaluations.
The approximation error obtained by using the neural network is about 2% (mean of the absolute value of the errors on all the objective functions, using 1150 exact evaluations) with a 5% maximum error on single simulations. The errors have been computed on a 'validation set' constituted by 150 solutions not belonging to the training set.

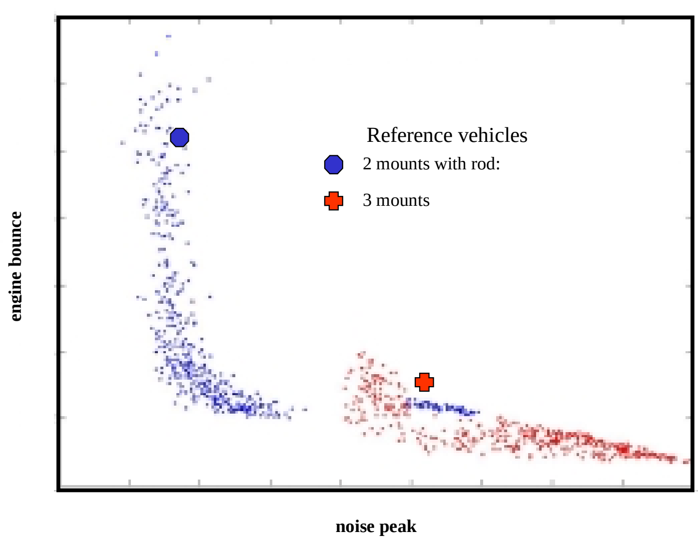


Fig. 8. Comparison of Pareto-optimal sets evaluated for different suspension layouts

As a case study the procedure has been used to compare the overall performances of two different layouts of engine suspension. A three mounts suspension and a two mounts+rod engine suspension are compared for a given geometry of the attachment points, considering a reference 4 cylinders in line gasoline engine.
After the training of the neural network 10^6 evaluations were performed in 20 hours of CPU time on an SGI R10000 Octane. About two hundred solutions satisfying the Pareto-optimality definition (4) for each suspension layout have then been verified through standard ADAMS® simulations.

Figure 8 shows the projection of the resulting Pareto-optimal sets onto the noise/engine bounce domain, stressing the sensible improvement that can be obtained for the two objective functions with respect to realistic reference vehicles. The potentiality of each suspension layout can be compared.

CONCLUSION

A method for designing car chassis sub-systems has been presented. A Multi-criteria approach has been followed. According with the Global Approximation procedure, the physical sub-models of the vehicle (derived by ADAMS®) have been substituted by corresponding artificial neural networks. By this way the relationships between chassis design variables and performance indexes could be computed accurately and very quickly. Pareto-optimal solutions have been computed within an acceptable time. To show the efficiency of the proposed method, two application examples have been presented. The dynamic behaviour of a car has been optimised and a proper trade-off among comfort, road holding and handling has been achieved. An automated ADAMS®/Matlab procedure for the preliminary design of the engine suspension systems has been developed, relying on the described method. The procedure has been implemented into tool which is currently in use during the product development process of FIAT cars.

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