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Determining influential factors for an air-spring fatigue life

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Abstract

This paper presents an analysis of factors essential to air spring and undercarriage design to determine their influence on air spring fatigue life. Two sets of factors were chosen for the investigation: those relating to the design of the air spring itself (cord angle, diameter difference), and those relating to the design of the undercarriage (lever length, eccentricity, lever eccentricity, inclination). A full factorial experiment, whereby interactions between all the factors are investigated, would demand 729 experiments. Using Taguchi methods the number of experiments required is dramatically reduced, using only 27 experiments to determine the influence of 6 factors and 3 interactions. Taguchi analysis shows here that factors inherent to air spring design have a much greater influence on fatigue life than those factors inherent to undercarriage design. Interaction between these factors has also revealed there is no optimal cord angle for all air springs.

Keywords: Spring; Fatigue; Composite; Finite element analysis; Life time assessment

Nomenclature: a	Shomate equation gas constant
b	Shomate equation gas constant
c	Shomate equation gas constant
c_p	heat capacity at constant pressure
d	Shomate equation gas constant
d_i	flex member inner diameter
D_o	flex member outer diameter
DF	degrees of freedom
e	Shomate equation gas constant
e_l	lever eccentricity
e_p	eccentricity
h	height
h_d	design height
IT	number of interactions
l	lever length

LV	number of levels
n	number of experiments
p	air pressure
R	stress ratio
$\bar{T}$	quality characteristic average
y_i	quality characteristic
α	inclination
ϑ	temperature
ϑ_0	absolute zero temperature
$\hat{T}$	prediction average
σ_{acp}	stress amplitude calculated with critical plane approach
σ_x	stress in the x direction
σ_y	stress in the y direction
τ_{xy}	shear stress
ϕ	cord angle

1. Introduction

A conventional rolling lobe air-spring is composed of a piston, flex member, upper plate and bumper (Figure 1). The flex member is filled with compressed air, which allows regulation of the spring's height. The bumper is used in operations such as kneeling of a bus and as a safety feature in case of flex member failure. The shape of an air spring piston directly determines the inner diameter d_i of the flex member and indirectly its outer diameter D_o too. With increasing and decreasing piston diameter the air spring's effective diameter D_{eff} is increased or decreased. An increase in the effective diameter means a more progressive stiffness characteristic of the air spring. This progressive spring stiffness characteristic allows a similar amount of comfort whether the vehicle is fully loaded or empty. This is the reason air springs are mainly used on buses and lorries, which operate with variable loads¹.

Figure 1. Air spring.

Air spring users and manufacturers have until now relied only on fatigue tests with a constant amplitude²⁻⁴. If a prediction of fatigue life is required, stresses on the air spring flex member must first be determined so that S-N curves can be calculated. Stresses can be determined using finite element analysis⁵⁻¹⁰, hence the design and operating conditions of an air spring can be changed using simulations in order to determine changes in the maximum stress. Such changes in maximum stress ultimately influence the fatigue life of an air spring¹¹⁻¹⁷.

Finite element analysis has previously been used to determine the influence of the cord angle on air spring fatigue life⁶. The cord angle is certainly among the most influential of design factors, but there are many other factors that might also have a significant influence on the maximum stress and hence fatigue life. Some of these factors are inherent to air spring design whilst others are important for undercarriage design. In reality, an air spring is not simply loaded in the axial direction. The undercarriage always imposes some deviation from a pure axial load which might have a significant impact on air spring life. With the help of experienced engineers, six factors were selected that might have a significant influence on the fatigue life (Figure 2). These factors are diameter difference ($(D_o - d_i)/2$), cord angle ϕ , lever length l , lever eccentricity e_l , eccentricity e_p and inclination α .

Figure 2. Influential factors: (a) diameter difference $(D_o-d_i)/2$, (b) cord angle φ , (c) lever length l , (d) lever eccentricity e_l , (e) eccentricity e_p , (f) inclination α .

If the relation between the factor of interest and the quality characteristic is linear, the quality characteristic need only be determined for two levels (minimum and maximum). If the relation between the factor of interest and the quality parameter is non-linear, the quality parameter must be determined for at least three levels (minimum, maximum and middle or the most important value). Air springs have progressive non-linear characteristics, therefore three levels for each factor are necessary (Table 1). The factor levels can be absolute values or can be calculated relative to other dimensions or characteristics. For lever eccentricity e_l (factor D) and eccentricity e_p (factor E), level values are determined relative to the design height h_D , and inner d_i and outer D_o diameters. All other factors have absolute level values.

If all the interactions for all six factors were to be examined with a full factorial suite of experiments there would be $2^6=64$ experimental conditions even if only two levels for each design factor were used. Since an air spring has a non-linear stiffness characteristic it would be better to examine three levels for each design factor but in that case a full factorial experimental suite would have $3^6=729$ experimental conditions. Since computer power is limited, it was decided to use Taguchi methods and limit interaction examination to only those three factors where the most interaction is expected¹⁸⁻²⁰.

Table 1. Levels of influential factors.

	Factors	Level		
		1	2	3
A	Diameter difference $(D_o-d_i)/2$	32.5 [mm]	52.46 [mm]	65.5 [mm]
B	Cord angle φ	30°	37.5°	45°
C	Lever length l	550 [mm]	900 [mm]	1500 [mm]
D	Lever eccentricity e_l	0	$h_D/2$	h_D
E	Eccentricity e_p	$-(D-d)/4$	0	$(D-d)/4$
F	Inclination α	-10°	0°	10°

2. Implementing Taguchi method on air-spring influential factors analysis

Once the most important factors and the number of levels have been chosen, the degrees of freedom DF must be calculated in order to apply the Taguchi method. To carry out an analysis of six factors F with three levels LV and three interactions IT the degrees of freedom can be calculated with the following equation¹⁸:

$$DF=DP \cdot (LV-1)+3 \cdot (LV-1) \cdot (LV-1)=6 \cdot (3-1)+3 \cdot (3-1) \cdot (3-1)=24 \quad (1)$$

Table 2. $L_{27}(3^{13})$ array

Test run	Factors												
	A	B	(AxB) ₁	(AxB) ₂	C	(AxC) ₁	(AxC) ₂	(BxC) ₁	D	E	(BxC) ₂	F	error
	1	2	3	4	5	6	7	8	9	10	11	12	13
1	1	1	1	1	1	1	1	1	1	1	1	1	1
2	1	1	1	1	2	2	2	2	2	2	2	2	2
3	1	1	1	1	3	3	3	3	3	3	3	3	3
4	1	2	2	2	1	1	1	2	2	2	3	3	3
5	1	2	2	2	2	2	2	3	3	3	1	1	1
6	1	2	2	2	3	3	3	1	1	1	2	2	2
7	1	3	3	3	1	1	1	3	3	3	2	2	2
8	1	3	3	3	2	2	2	1	1	1	3	3	3
9	1	3	3	3	3	3	3	2	2	2	1	1	1
10	2	1	2	3	1	2	3	1	2	3	1	2	3
11	2	1	2	3	2	3	1	2	3	1	2	3	1
12	2	1	2	3	3	1	2	3	1	2	3	1	2
13	2	2	3	1	1	2	3	2	3	1	3	1	2
14	2	2	3	1	2	3	1	3	1	2	1	2	3
15	2	2	3	1	3	1	2	1	2	3	2	3	1
16	2	3	1	2	1	2	3	3	1	2	2	3	1
17	2	3	1	2	2	3	1	1	2	3	3	1	2
18	2	3	1	2	3	1	2	2	3	1	1	2	3
19	3	1	3	2	1	3	2	1	3	2	1	3	2
20	3	1	3	2	2	1	3	2	1	3	2	1	3
21	3	1	3	2	3	2	1	3	2	1	3	2	1
22	3	2	1	3	1	3	2	2	1	3	3	2	1
23	3	2	1	3	2	1	3	3	2	1	1	3	2
24	3	2	1	3	3	2	1	1	3	2	2	1	3
25	3	3	2	1	1	3	2	3	2	1	2	1	3
26	3	3	2	1	2	1	3	1	3	2	3	2	1
27	3	3	2	1	3	2	1	2	1	3	1	3	2

Then an orthogonal array is chosen. The most suitable array is $L_{27}(3^{13})$ with 26 degrees of freedom (Table 2). In this array levels for 27 test runs are defined. The chosen array has one column too many, however this column will be used for the analysis error calculation. After the orthogonal array is selected, factors and interactions are assigned to appropriate columns with the use of linear graphs (Figure 3). For $L_{27}(3^{13})$ array several standard linear graphs are available, but only the one presented in Figure 3 has three interactions. The selected linear graph has interactions between factors A – diameter difference $(D_o - d_i)/2$, B – cord angle ϕ and C – lever length l and four unconnected factors. Since there are only three unconnected factors, the last node is used to calculate the analysis error.

Researchers have shown that the critical plane approach gives good predictions of the rubber components fatigue life^{5,6,10,21-23}. These investigations of natural rubber's fatigue life have shown that crystallization of natural rubber has a significant impact on its fatigue life²⁴⁻³⁰. Reinforcement by rubber crystallization occurs, when stress ratio $R > 0$. In all load cases of our analysis, R was less than 0 for the maximum amplitude node, and therefore rubber crystallization had no influence on our investigation.

Calculation of stress amplitudes σ_{acp} was based on the results of finite element analyses where air spring bellows were modelled with multilayer shell elements. With FEA plane stress components σ_x , σ_y and τ_{xy} were calculated for a load cycle with displacement amplitude $\pm 50\text{mm}$, while local nodal and elemental coordinate systems were adjusted at every substep of the load cycle. For all substeps of the load cycle, normal stresses σ_n on 128 evenly spatially distributed planes were calculated in all nodes. Maximum damage during the load cycle will take place in the node with maximum stress amplitude σ_{acp} in one of its planes. Since constant amplitude tests are still widely used for air spring validation tests¹⁰, the maximum stress amplitude σ_{acp} during the load cycle ($\pm 50\text{mm}$ displacement), was selected as the quality characteristic in the Taguchi analysis.

Figure 3. Linear graph $L_{27}(3^{13})$.

3. Determination of air spring quality characteristics with finite element analysis

The analysis of influential factors was based on three industrially produced air springs with different diameter differences. Measurements of pressure and spring outer diameter D_o were made during loading with a vertical load without any deviation. The same loading was also simulated using FEA. Because FEA gave similar results to the real measurements, this same FEA model was used for calculation of the quality characteristics σ_{acp} for all 27 test runs in the $L_{27}(3^{13})$ array.

3.1 FE Model

Finite element analyses were made in Abaqus using the explicit solver. The critical element of all air springs is the flex member. Therefore the emphasis of our finite element analysis will be on the flex member. Flex members are made of rubber reinforced with cord threads. They were modelled with linear homogeneous shell elements (S4R), with rebar layers used to model cord threads reinforcement. The stress strain characteristic for rubber and cord threads were obtained by uniaxial tensile stress experiments^{6,10}. Since both materials are hyperelastic, the Marlow hyperelastic material model was fitted to the experimental stress strain curve for both materials.

The air spring piston and the upper plate with lever are made of metal or reinforced plastic and are almost rigid compared to the flex member. Because they are not critical for the fatigue life assessment of an air spring, they were modelled with rigid elements (R3D4). Air spring pistons are hollow and filled with air. Instead of a complex shaped piston cavity, a cylinder with the same volume as the piston cavity was added to the assembly. During experimental verification of the FEA model, the air spring is loaded with a vertical load only, for simplicity. For the Taguchi analysis, a lever modelled with rigid elements (R3D4) was also added to the assembly (Figure 4)

Figure 4. FE model (a) for Taguchi analysis, and (b) for experimental verification.

A fluid cavity with adiabatic behaviour was used to model the air inside the air spring. The heat capacity at constant pressure c_p is determined with the Shomate equation^{6,31}:

$$c_p = a + b(\vartheta - \vartheta_0) + c(\vartheta - \vartheta_0)^2 + d(\vartheta - \vartheta_0)^3 + \frac{e}{(\vartheta - \vartheta_0)^2} \quad (2)$$

where a , b , c , d and e are gas constants, ϑ is the temperature and ϑ_0 is absolute zero temperature.

Contacts between the flex member and piston and the flex member and upper plate are defined as surface to surface contacts between rigid body surfaces (the piston and upper plate) and a deformable shell surface (the flex member). Contacts are defined as hard contacts in the normal direction and frictional contacts in the tangential direction. Modelling the whole air spring is not necessary. With a symmetry boundary condition half of the air spring can be modelled and realistic deformation of the air spring can still be applied whilst varying lever length, lever eccentricity, eccentricity or inclination (Figure 5)³².

Figure 5. Boundary conditions on FE model (a) for Taguchi analysis, and (b) for experimental verification.

The FEA model of the air spring is loaded in five steps (Figure 6). In the first step the air spring is modelled at the fully extended shape. During this step no movement occurs, only the flex member is filled with air at a pressure of 7 bar. In the second step the upper plate is moved to the design height while the air pressure remains constant. At the end of this step the quantity of air in the air spring remains constant, while pressure, volume and temperature are changed according to the adiabatic process. In the third step the upper plate is moved to the lower limit position and rotated for an inclination α . In the fourth step the upper plate is moved in a lateral direction until the required eccentricity is reached. In the fifth step the upper plate is moved to the upper limit position.

Figure 6. Load steps of the FEA: (a) unloaded air spring, (b) first load step, (c) second load step, (d) third load step, (e) fourth load step, (f) fifth load step.

From 27 separate analyses, stress tensors are obtained for 27 different versions of the FEA air spring with influential factors defined in the orthogonal array. The stress tensors are then transformed to an equivalent uniaxial stress response using the critical plane approach^{21,33,34}.

3.2. Experimental verification of FE model

Three air springs were tested on a Zwick HB250 testing machine. At the design height springs were filled with air at a pressure of 7 bar. All the springs were loaded from a minimum to a maximum load

and back to the minimum load. During this load cycle air spring pressure p and the flex member diameter D_o were measured (Figure 7).

Figure 7. Pressure and diameter measurement.

Hysteresis is observed in the pressure and diameter diagram (Figure 7). For a comparison of the measured pressure and diameter with those modelled by FEA, medium values of the hysteresis loop were calculated. Measured and calculated values were compared for the design height h_D ; $h_D + 50\text{mm}$; and $h_D - 50\text{mm}$. The differences between the measured and calculated values are presented in Table 3.

Table 3. The differences between the measured and calculated pressure and diameter.

Air-spring	Height h [mm]	Measured pressure [bar]	Calculated pressure [bar]	Difference [%]	Measured diameter [mm]	Calculated diameter [mm]	Difference [%]
1	205	9.37	9.22	-1.56	281.00	281.53	0.19
	255	6.87	7.00	1.86	275.00	269.88	-1.86
	305	5.27	5.56	5.60	270.00	258.08	-4.41
2	350	8.60	8.40	-2.29	334.00	353.00	5.69
	400	6.97	7.00	0.43	330.00	334.83	1.46
	450	5.80	5.95	2.54	303.00	307.76	1.57
3	335	8.61	8.57	-0.49	332.00	334.64	0.80
	385	6.97	7.00	0.38	327.00	319.62	-2.26
	435	5.78	5.88	1.78	304.60	295.11	-3.12

The maximum difference between the calculated and measured pressure is 5.6% while the maximum difference between the measured and calculated diameter is 5.69%. The differences are relatively low, therefore the FEA model is considered to be acceptable.

4. Results

4.1. Identification of strong effects

In most cases where Taguchi methods are used, several measurements are made for each test run. More than one type of Taguchi analysis is available: level average; smaller the better; larger the better; and nominal the best. The latter 3 consider the signal to noise ratio, however the experiments were simulated using FEA and had no experimental noise, therefore the most suitable type of Taguchi analysis for this investigation is level average analysis. The quality characteristic of the analysis was the stress amplitude σ_{acp} , calculated with the critical plane approach (Table 4)⁶. The smaller the amplitude the larger fatigue life, hence smaller amplitudes are optimal.

The results of this analysis are used to calculate the average stress amplitude σ_{acp} for each factor and interaction level. For example, the average of level 1 for factor A is:

$$\overline{A_1} = \frac{1.3553 + 1.2725 + 1.3123 + 1.0675 + 1.1349 + 1.1149 + 0.8606 + 0.8440}{8} = 1.090 \quad (3)$$

The average stress amplitudes for all the factors and levels are presented in a response table (Table 5). To determine which factor is the strongest, the difference between the level with highest quality characteristic average and the level with the lowest quality characteristic average is calculated. A larger difference means a stronger effect. The strongest effects are seen for diameter difference (factor A), cord angle (factor B), interaction between the diameter difference and cord angle (interaction AxB), and eccentricity (factor E). All other effects are lower than the error and hence are not significant.

Table 4. Stress amplitudes σ_{acp} for 27 test runs.

Test run	factor													Quality characteristic σ_{acp} [MPa]
	A	B	AxB ₁	AxB ₂	C	AxC ₁	AxC ₂	BxC ₁	D	E	BxC ₂	F	error	
1	1	1	1	1	1	1	1	1	1	1	1	1	1	1.3553
2	1	1	1	1	2	2	2	2	2	2	2	2	2	1.2725
3	1	1	1	1	3	3	3	3	3	3	3	3	3	1.3123
4	1	2	2	2	1	1	1	2	2	2	3	3	3	1.0675
5	1	2	2	2	2	2	2	3	3	3	1	1	1	1.1349
6	1	2	2	2	3	3	3	1	1	1	2	2	2	1.1149
7	1	3	3	3	1	1	1	3	3	3	2	2	2	0.8606
8	1	3	3	3	2	2	2	1	1	1	3	3	3	0.8440
9	1	3	3	3	3	3	3	2	2	2	1	1	1	0.8468
10	2	1	2	3	1	2	3	1	2	3	1	2	3	2.5499
11	2	1	2	3	2	3	1	2	3	1	2	3	1	2.5394
12	2	1	2	3	3	1	2	3	1	2	3	1	2	2.4957
13	2	2	3	1	1	2	3	2	3	1	3	1	2	2.9498
14	2	2	3	1	2	3	1	3	1	2	1	2	3	2.9373
15	2	2	3	1	3	1	2	1	2	3	2	3	1	3.0692
16	2	3	1	2	1	2	3	3	1	2	2	3	1	3.0764
17	2	3	1	2	2	3	1	1	2	3	3	1	2	2.9826
18	2	3	1	2	3	1	2	2	3	1	1	2	3	3.0531
19	3	1	3	2	1	3	2	1	3	2	1	3	2	2.3728
20	3	1	3	2	2	1	3	2	1	3	2	1	3	2.6193
21	3	1	3	2	3	2	1	3	2	1	3	2	1	2.3817
22	3	2	1	3	1	3	2	2	1	3	3	2	1	3.0818
23	3	2	1	3	2	1	3	3	2	1	1	3	2	2.7347
24	3	2	1	3	3	2	1	1	3	2	2	1	3	2.7468
25	3	3	2	1	1	3	2	3	2	1	2	1	3	3.1739
26	3	3	2	1	2	1	3	1	3	2	3	2	1	3.0709
27	3	3	2	1	3	2	1	2	1	3	1	3	2	3.0818

Table 5. Response table

Level	A	B	AxB ₁	AxB ₂	C	AxC ₁	AxC ₂	BxC ₁	D	E	BxC ₂	F	error
1	1.090	2.100	2.402	2.469	2.276	2.258	2.217	2.234	2.290	2.239	2.230	2.256	2.284
2	2.850	2.315	2.248	2.200	2.237	2.226	2.278	2.279	2.231	2.210	2.275	2.258	2.207
3	2.807	2.332	2.098	2.078	2.234	2.262	2.253	2.234	2.227	2.299	2.243	2.233	2.256
	1.760	0.232	0.304	0.391	0.043	0.036	0.061	0.045	0.063	0.090	0.045	0.025	0.077

4.2. Identification of optimum levels

When strong effects are identified, optimum levels for those effects must be determined. The smaller the stress amplitude σ_{acp} the larger fatigue life, hence smaller amplitudes are optimal. In the response table the optimum level for the eccentricity (factor E) without interaction can be easily identified (level 2). For diameter difference (factor A), cord angle (factor B), and their interaction (AxB), an interaction analysis must be made. In interaction analysis, the average response for all combinations of factors and levels are calculated (Table 6).

Table 6. Interaction table.

	B ₁	B ₂	B ₃
A ₁	1.3134	1.1058	0.8505
A ₂	2.5283	2.9854	3.0374
A ₃	2.4579	2.8544	3.1089

In the interaction table, the optimum levels are A₁B₃ (Table 6). If factors A (diameter difference) and B (cord angle) are examined without any interaction, the optimum level for both A and B is level 1. However, according to the response table the interaction between A and B is stronger than factor B, the optimum levels are interaction optimum levels A₁B₃. To better understand the interaction of factors A and B, an interaction graph is shown in Figure 8.

Figure 8. Interaction Graph.

In the interaction graph, B₃ (cord angle level 3) is indeed the best level for factor A₁ (diameter difference level 1), but for factors A₂ and A₃ (diameter difference, levels 2 and 3) the best level is B₁ (cord angle level 1). This indicates a strong interaction between both factors. In absolute terms, the combination A₁B₃ (diameter difference level 1, cord angle level 3) gives us the minimum value for the quality characteristic σ_{acp} , therefore this is the best combination. The optimum factor levels are thus A₁B₃E₂ (equating to diameter difference level 1 – 32.5mm, cord angle level 3 – 45°, eccentricity level 2 – 0).

4.3. Quality characteristic prediction

When the optimum levels for strong effects have been determined, the predicted quality characteristic can be calculated using equation 4¹⁸. The overall average is defined as $\bar{T}$ where:

$$\bar{T} = \sum_{i=1}^n \frac{y_i}{n} = 2.2491 \quad (4)$$

The prediction average $\hat{\mu}$ is defined as:

$$\hat{\mu} = \bar{T} + (\bar{A_1} - \bar{T}) + (\bar{B_3} - \bar{T}) + [(\bar{A_1B_3} - \bar{T}) - (\bar{A_1} - \bar{T}) - (\bar{B_3} - \bar{T})] + (\bar{E_2} - \bar{T}) = 0.8110 \quad (5)$$

Test run No. 9 has factor levels A₁B₃E₂ and a quality characteristic of 0.8468. The predicted quality characteristic is lower than the calculated one, but the difference is less than the estimated analysis error 0,077.

5. Conclusions

Taguchi analysis of six factors and three interactions was made to determine those factors with the strongest influence on air-spring fatigue life. If natural rubber's loading has a positive stress ratio, rubber crystallization has a significant effect on fatigue life. In all load cases of our analysis, stress ratios were negative, thus rubber crystallization does not influence air spring fatigue life. In our analysis, only conventional rolling lobe air-springs were analysed. Therefore our conclusions are relevant for conventional rolling lobe air-springs only. The quality characteristic used here was the stress amplitude σ_{acp} calculated with the critical plane approach. The FEA model results were compared with experimental measurements. Small differences between the measured and FEA results indicated that FEA is of sufficient quality. The Taguchi analysis has shown that those factors inherent to air spring design and their interaction (factor A – diameter difference, factor B – cord angle and their interaction AxB) have a much stronger effect on fatigue life than those factors inherent to undercarriage design. This is very useful information for air-spring and undercarriage

manufactures. If the fatigue life of an air- spring is too low, and undercarriage dimensions are within the limits defined in Table 1, the source of the problem must be the air spring itself.

The analysis presented here has shown that the most influential factor of an air spring is the diameter difference (factor A). Interaction between the diameter difference (interaction AxB) and cord angle (factor B) is also relatively strong. Interaction analysis has shown a strong interaction between diameter ratio and cord angle. This means there is no single optimum cord angle for all springs. For the strongest factors optimum levels were determined (diameter difference A1 – 32.5mm, cord angle B3 – 45°, eccentricity – factor E2 - 0) and an optimum quality characteristic prediction of 0.8110 was calculated (Eq. 5). The simulated experiment with optimum factor levels had a quality characteristic of 0.8468. Such difference between the predicted and simulated quality characteristics can be expected because the estimated analysis error is 0.077.

Since factors inherent to the air spring design have a much greater influence than factors inherent to the undercarriage design, further investigations could be carried out with emphasis on those factors. In such analysis, new factors such as the height of the air spring, cord diameter, cord material etc. should also be investigated. The aforementioned investigation would provide further information on the influence of the air spring design on fatigue life.

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Captions for figures and tables

Figure 1: Air spring.

Figure 2: Influential factors: (a) diameter difference $(D_o-d_i)/2$, (b) cord angle φ , (c) lever length l , (d) lever eccentricity e_l , (e) eccentricity e_p , (f) inclination α .

Figure 3: Linear graph $L_{27}(3^{13})$.

Figure 4: FE model (a) for Taguchi analysis, and (b) for experimental verification.

Figure 5: Boundary conditions on FE model (a) for Taguchi analysis, and (b) for experimental verification.

Figure 6: Load steps of the FEA: (a) unloaded air spring, (b) first load step, (c) second load step, (d) third load step (e) fourth load step, (f) fifth load step.

Figure 7: Pressure and diameter measurement.

Figure 8: Interaction Graph.

Table 1: Levels of influential factors.

Table 2: $L_{27}(3^{13})$ array.

Table 3: The differences between the measured and calculated pressure and diameter.

Table 4: Stress amplitudes σ_{acp} for 27 test runs.

Table 5: Response table.

Table 6: Interaction table.