Finite element analysis and optimization of a Formula SAE car chassis

Análise de elementos finitos e otimização de um chassi de carro de Fórmula SAE

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ABSTRACT

In motorsport engineering, achieving the ideal balance between speed, agility, and safety is a monumental challenge. Formula SAE competitions, where collegiate racing teams design high-performance vehicles, emphasize the pivotal role of the chassis, the backbone of the racing machine. Ensuring chassis compliance with regulations and optimizing structural integrity and weight is a formidable task. This study addresses this multifaceted challenge by focusing on the development, simulation, and optimization of a Formula SAE car's chassis. Rigorous analysis of four chassis versions seeks to strike a balance between rigidity and weight, enhancing performance while adhering to rules. Longitudinal torsion, diamond torsion, lateral acceleration, and braking tests assess structural rigidity and stress distribution, ensuring predictability and preventing potential failures. The mesh uses one-dimensional elements (1 mm size) and material properties akin to SAE 1020 steel. The fourth iteration consistently demonstrated superior...
performance across all tests. Despite a slight weight increase, it exhibited a longitudinal torsional rigidity of 1985 Nm/deg, well-aligned with Formula SAE standards, and a formidable diamond torsional rigidity of 56,304 Nm/deg. Critical stress analysis during lateral acceleration and braking tests remained well within permissible limits, with a chassis weight of approximately 35 kg. In summary, this work meets stringent Formula SAE standards, offering a systematic methodology for chassis design and optimization. The fourth iteration excels in various critical tests and serves as a valuable contribution to motorsport engineering, providing a blueprint for enhanced competitiveness and performance in the 2023 Formula SAE Brazil competition.

Keywords: motorsport engineering, chassis optimization, Formula SAE, structural integrity, performance enhancement, finite element analysis (FEA).

RESUMO
Na engenharia automobilística, alcançar o equilíbrio ideal entre velocidade, agilidade e segurança é um desafio monumental. As competições da Fórmula SAE, onde as equipes de corrida colegiadas projetam veículos de alto desempenho, enfatizam o papel fundamental do chassi, a espinha dorsal da máquina de corrida. Garantir a conformidade do chassi com as normas e otimizar a integridade estrutural e o peso é uma tarefa formidável. Este estudo aborda este desafio multifacetado, concentrando-se no desenvolvimento, simulação e otimização de chassi de um carro de Fórmula SAE. A análise rigorosa de quatro versões de chassi procura encontrar um equilíbrio entre rigidez e peso, melhorando o desempenho e, ao mesmo tempo, aderindo às regras. Os testes de torção longitudinal, torção de diamante, aceleração lateral e frenagem avaliam a rigidez estrutural e a distribuição de tensões, garantindo previsibilidade e prevenindo possíveis falhas. A malha utiliza elementos unidimensionais (tamanho de 1 mm) e propriedades de material semelhantes ao aço SAE 1020. A quarta iteração demonstrou consistentemente um desempenho superior em todos os testes. Apesar de um ligeiro aumento de peso, exibiu uma rigidez torsional longitudinal longitudinal de 1985 Nm/graus, bem alinhada com os padrões da Fórmula SAE, e uma rigidez torsional de diamante formidável de 56.304 Nm/graus. A análise de tensões críticas durante os ensaios de aceleração lateral e de frenagem manteve-se bem dentro dos limites admissíveis, com um peso de quadro de aproximadamente 35 kg. Em resumo, esse trabalho atende aos padrões rígidos da Fórmula SAE, oferecendo uma metodologia sistemática para o design e a otimização do chassi. A quarta iteração se destaca em vários testes críticos e serve como uma contribuição valiosa para a engenharia de automobilismo, fornecendo um projeto para melhorar a competitividade e o desempenho na competição Fórmula SAE Brasil de 2023.

Keywords: engenharia de automobilismo, otimização de chassi, Fórmula SAE, integridade estrutural, melhoramento de desempenho, análise de elementos finitos (FEA).
1 INTRODUCTION

The Formula SAE initiative is a pivotal pillar in engineering education, affording university students a comprehensive learning experience through the demanding competition focused on developing and realizing Formula-type vehicles. This educational endeavor is underscored by a multifaceted assessment regimen encompassing static and dynamic evaluations (PROROK, 2016).

An intricate examination ensues in the static evaluation phase, guided by a wealth of research and industry insights. These encompass a meticulous assessment of construction methodologies, as outlined by previous studies in engineering design (CROSS, 2021), in-depth cost analysis, drawing from principles of engineering economics (SULLIVAN; WICKS; LUXHOJ, 2003), judicious examination of design choices, informed by innovative engineering principles (EGGERT, 2005), and an unwavering focus on safety protocols, drawing from established safety engineering guidelines (HAIMES, 2005).

Conversely, the dynamic evaluation phase is characterized by the empirical assessment of the vehicle's operational prowess, underpinned by established performance evaluation metrics (SAE, 2023). This encompasses a detailed analysis of the vehicle's acceleration characteristics, drawing from the extensive literature on automotive acceleration dynamics (GILLESPIE, 2021), its handling capabilities on the track, which is a subject of extensive research in vehicle dynamics (MILLIKEN; MILLIKEN, 1995), and its fuel consumption efficiency, a topic well-documented in studies on automotive fuel efficiency (HEYWOOD, 2018).

The Formula SAE competition, initiated in 1979 under the auspices of the Society of Automotive Engineers (SAE) in the United States, has progressively expanded its global footprint. References to its internationalization and influence on engineering education can be found in studies by Wickenden and Stobart (WICKENDEN; STOBART, 2005) and Prada (PRADA et al., 2015). Among eight participating countries, Brazil notably thrives as a significant contributor to this global exchange of innovative engineering concepts and practices, as corroborated by recent research, such as in the areas of optimization (Barbosa
et al., 2021), aerodynamics (ÁVILA et al., 2018; BARBOSA, 2018), and prediction of lap time (BARBOSA; TANNÚS; ZUQUETE GUARATO, 2019).

A pivotal criterion in vehicle design revolves around torsional stiffness. Seward (SEWARD, 2014) elucidated that two salient reasons underscore the importance of evaluating this parameter. Firstly, it pertains to the vehicle’s ability to effectively fine-tune its balance, as a chassis with low rigidity behaves like a spring, impeding predictability and vehicle tuning. Ideally, the stiffness of a chassis should approximate the rolling stiffness of the suspension under the worst-case scenario. For vehicles with minimal downforce and low suspension rigidity, torsional stiffness may be quantified at 300 Nm/deg, while Formula 1 cars can soar to an astonishing 25,000 Nm/deg (MILLIKEN; MILLIKEN, 1995).

The second rationale is intertwined with the storage of elastic energy within a low-rigidity chassis, which can perturb the dynamic behavior. In the context of suspension, the energy accumulated by springs is effectively managed through the damping action of shock absorbers. This inherent control mechanism is absent within the chassis structure. According to Milliken and Milliken (MILLIKEN; MILLIKEN, 1995) a vehicle exhibits predictability when the chassis’ rigidity is sufficiently high to be disregarded in dynamic analysis.

In the realm of competition vehicle chassis design, it is imperative, as underscored by Riley and George (RILEY; GEORGE, 2002), to commence with a comprehensive understanding of the structural loads in play. This entails conducting a battery of tests, including longitudinal torsion, vertical bending, lateral bending, and diamond torsion. Notably, these tests yielded a stiffness measurement of 1455 Nm/deg using FEA and 949 Nm/deg through experimental validation. Despite the majority of teams using carbon steel in the construction of the chassis (KHAN; IMRAN; ARIF, 2023), some teams also use composite materials, such as carbon fiber, for the chassis (CAROLLO et al., 2022).

McBeath and O’Rourke (MCBEATH; O’ROURKE, 2009) further emphasizes that an ideal chassis should exhibit high rigidity while maintaining low weight and cost. Substantial torsional deflection can compromise vehicle performance as the chassis vibrations akin to a large spring adversely affect the predictability of both engineers and drivers.
Riley and George (RILEY; GEORGE, 2002) delves into the realm of design modifications, highlighting that structural rigidity can be significantly enhanced through the incorporation of additional triangulations. In a specific case, rigidity increased from 250 Nm/deg to 615.98 Nm/deg by immobilizing the rear suspensions and applying force to the front end, even with a 4 kg reduction in mass compared to the previous year's chassis.

In a study by Cunha et al. (MARZOLA DA CUNHA et al., 2019) structural rigidity was investigated by rigidly connecting all suspension points of the rear chassis. Computational modeling yielded stiffness values of 1536 Nm/deg for the lower bar and 2311.38 Nm for the upper bar, while real-world testing resulted in stiffness measurements of 3178 Nm/deg for the upper bar and 2807 Nm/deg for the lower bar. Moreover, Sithananun et al. (SITHANANUN et al., 2011, 2012) documented their achievement of a remarkable chassis rigidity of 1030 Nm/deg for the FSAE team at Chulalongkorn University, accomplished with a chassis mass of 29.8 kg.

In conclusion, the Formula SAE competition stands as an invaluable platform for engineering students, providing a robust arena for honing their technical expertise and applying empirical solutions to practical engineering challenges. To investigate the pertinent aspects of the SAE regulations, the next topic will delve into the precise guidelines and criteria that govern this intricate and multifaceted undertaking.

1.1 RELEVANT PARTS OF THE SAE REGULATIONS

The Formula SAE Rules (2023), offer a considerable degree of latitude in vehicle design, setting it apart as a rare category within motorsports that does not impose a minimum weight restriction. Within the combustion category, pertinent to this discussion, certain key specifications come to the fore. These include the allowance for 4-stroke Otto cycle engines, capped at a maximum displacement of 710 cm³. Moreover, vehicles must feature open cockpits, exposed wheels, and be equipped with a minimum five-point harness system. Furthermore, they should be designed to accommodate an adult of a minimum height of 1.80 meters.
Turning our attention to chassis design, adherence to stringent safety criteria is mandatory. These criteria delineate specific minimum dimensions for each structural component, as delineated in Table 1. Notably, components like the "main hoop" and "front hoop," which safeguard the pilot's head and thighs against ground impact in the event of a rollover, must possess tubing conforming to dimension "A." For areas such as "side impact" protection against lateral collisions and the "Front Bulkhead" guarding against frontal impacts, dimension "B" tubing is mandated. Tubing of dimension "C" may suffice for other primary structural elements, with bars falling below these dimensions being deemed non-structural. Additionally, it is imperative that the entire chassis structure incorporates lateral triangulation. In exceptional cases, where the upper side impact structure, referred to as the "Upper Side Impact Member," exhibits bends or curvatures, dimension "D" tubing is stipulated.

Table 1 – Dimensions of the tubes of the Regulation.

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Minimum outside diameter</th>
<th>Minimum thickness</th>
<th>Moment of minimum inertia</th>
<th>Minimum section area</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>25 mm</td>
<td>2.0 mm</td>
<td>11320 mm$^4$</td>
<td>173 mm$^2$</td>
</tr>
<tr>
<td>B</td>
<td>25 mm</td>
<td>1.2 mm</td>
<td>8509 mm$^4$</td>
<td>114 mm$^2$</td>
</tr>
<tr>
<td>C</td>
<td>25 mm</td>
<td>1.2 mm</td>
<td>6695 mm$^4$</td>
<td>91 mm$^2$</td>
</tr>
<tr>
<td>D</td>
<td>25 mm</td>
<td>1.2 mm</td>
<td>18015 mm$^4$</td>
<td>126 mm$^2$</td>
</tr>
</tbody>
</table>

Source: SAE (2023).

For structural calculations, the Structural Equivalency Spreadsheet (SES) mandates a Young's Modulus of 200 GPa, tensile strength of 305 MPa, and rupture limit of 365 MPa. In cases involving welded joints or discontinuous materials, the maximum allowable tensile strength is capped at 180 MPa, with a rupture limit of 180 MPa. While alternative materials like composites are permissible, they are subject to specific regulations stipulated within the SES framework.

This study is dedicated to the simulation and optimization of the chassis for the vehicle, an entrant in the Formula SAE Brazil competition. These tests encompass a wide array of assessments, including longitudinal torsion, diamond torsion, lateral acceleration, and braking simulations. The central objective of this analysis revolves around the in-depth evaluation and comparison of structural
rigidity in the initial two tests, followed by an assessment of maximum stress levels in the subsequent pair of evaluations.

2 MATERIALS AND METHODS

During project development, to address the requirements of other subareas such as suspension, steering, and powertrain, in conjunction with compliance with regulatory stipulations, the comprehensive assembly of the automobile was conceived employing SolidWorks 2019 software,

Figure 1. The model is characterized by an estimated weight of 270 kg, a double-wishbone suspension system, and a 4-cylinder, 600 cm³ engine with an estimated power output of 70 hp, dictated by regulatory constraints imposed on air intake.

Figure 1 – Full version car CAD.

From the overall 3D design of the single seater, the original version of the chassis is depicted in Figure 2a. This chassis comprises solely the beam elements constituting the primary structure, devoid of welds or supports. The colors in the figure represent the dimensions of the tube profiles: green
corresponds to dimension C, with an outer diameter of 25.4 mm and a thickness of 1.2 mm; yellow represents dimension B, with an outer diameter of 25.4 mm and a thickness of 1.6 mm; and finally, the red tube represents dimension A, with an outer diameter of 25.4 mm and a thickness of 2.5 mm. These tubes conform to regulatory standards and are readily available commercially. In this chassis, there is no need for tubes of dimension D. To optimize the structure, three variations of the original design were created, as shown in Figures 2b, 2c, and 2d. These variations were developed through a recursive process of vulnerability correction based on the points of highest stress and rigidity in the structure, as detailed in Chapter 3.

Figure 2 – Geometries under investigation: a) Original Version 1; b) Version 2; c) Version 3; d) Version 4.

All the materials used in the chassis and suspension were simulated using SAE 1020 steel, with a Young's modulus of 200 GPa, a Poisson's ratio of 0.3, and a density of 7850 kg/m³.

In the lateral acceleration and braking tests, both augmented by gravity, the geometry imported is depicted in Figure 3. In the image, the dark blue brackets have a diameter of 14 mm and a thickness of 4 mm, which matches the dimensions of the light blue tie rod. The motion of the rocker arm was simplified
using pink tubes, which have dimensions of 12 mm in diameter and 3 mm in thickness. Represented in dark gray is the axle stub, and the black tire, which for the purposes of this simulation, are combined into a single component to establish contact with the ground, with an outer diameter of 40 mm and a thickness of 10 mm.

Figure 3 – Assembly of the suspension onto the chassis.

Figure 4 depicts the coupling between components, utilizing two types of joints: spherical joints for J1 (between the chassis and control arms), J3 (between the tie rod and upper control arm), J5 (between the tie rod and rocker arm), J6 (between the chassis and rocker arm), and revolute joints, J2 (between the upper control arm and upright) and J4 (between the upper control arm and tie rod). In addition to the joints, there is a spring, denoted as M, with a stiffness of 131 kN/m, which is positioned between the rocker arm and the midpoint of the chassis bar.
Each of the four tests features distinct boundary conditions. In the lateral acceleration test, a leftward curve is simulated with an acceleration of 20.384 m/s², in addition to the gravitational acceleration of 9.8066 m/s². To account for the remaining unsuspended mass, including the engine, various systems, and the driver, which cannot be disregarded, an extra mass of 265 kg was added to the chassis. The extension of the steering knuckle signifies the contact between the tires and the ground, with all tires constrained along the Z-axis of the vehicle’s motion. To ensure the most critical scenario, the supporting tires on the right side have zero displacement along the X-axis (perpendicular to velocity). Furthermore, the right rear tire cannot separate along the Z-axis, effectively constraining the movement across all axes, as illustrated in
Figure 5 (a). The primary parameter under analysis is the maximum principal stress, identifying points most susceptible to failure.

In the braking test, as shown in
Figure 5b, the same additional mass of 265 kg used in the lateral acceleration test was maintained. The key distinctions lie in the forward acceleration of 20 m/s² and the constraints applied to the tires. In this configuration, all wheels are prohibited from translating along the Y-axis. The front wheels do not allow translation along the Z-axis, while the right rear wheel is prevented from moving along the X-axis. However, rotational motion is permitted at all support points. The analysis of maximum principal stress remained the most crucial aspect of examination in this scenario.

In both the longitudinal torsion test, as illustrated in
Figure 5c, and the diamond torsion test, as shown in
Figure 5d, the rear points of the rear control arms were immobilized in the X, Y, and Z directions while allowing for rotation. Force was applied at the front vertices of the upper front control arms, inducing torsional deformation in the longitudinal (Y-axis) and diamond (Z-axis) directions, always in opposing senses. To calculate the torsional stiffness ($R_t$), equations 1 and 2 were employed, with a distance between points ($d_p$) of 0.68 mm and a force of 1000 N. Additionally, it was necessary to determine the torsion angle ($\theta$) by utilizing the displacement of points along the axis of force application. This involved measuring the displacement at the left point ($d_e$) and the right point ($d_d$), values provided by the software. In both torsion cases, the most critical results encompassed a comparison of stiffness between the models, while also considering the analysis of maximum stress.

\[
\theta = \arctan \left( \frac{|d_e - d_d|}{d_p} \right) \quad (1)
\]

\[
R_t = \frac{F \times d_p \times \pi}{\theta \times 180} \quad (2)
\]
Following the mesh analysis in Version 4, which boasts a higher element count, along with Version 3, a one-dimensional mesh of Type Beam 188 with a length of 1 mm was employed for the simulations. This mesh configuration comprised a total of 76,183 nodes and 38,121 elements. Although tests were conducted with elements of a 0.5 mm length, as outlined in Table 2, the model encountered operational difficulties on the computer utilized, which possessed 8 GB of RAM. Due to the increased number of elements and negligible errors of less than 0.5% in comparison to the 1 mm element case in the most critical scenario, the latter configuration was chosen, as depicted in Figure 6.
Table 2 – Mesh analysis in Version 4.

<table>
<thead>
<tr>
<th>Element length (mm)</th>
<th>Acceleration stress (MPa)</th>
<th>Deformation in Longitudinal Torsion (mm)</th>
<th>Maximum number of nodes</th>
<th>Maximum number of elements</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>121.56</td>
<td>4.0647</td>
<td>1577</td>
<td>818</td>
</tr>
<tr>
<td>20</td>
<td>127.34</td>
<td>4.0652</td>
<td>3853</td>
<td>1956</td>
</tr>
<tr>
<td>10</td>
<td>137.18</td>
<td>4.0652</td>
<td>7647</td>
<td>3853</td>
</tr>
<tr>
<td>5</td>
<td>142.01</td>
<td>4.0652</td>
<td>15261</td>
<td>7660</td>
</tr>
<tr>
<td>2</td>
<td>145.03</td>
<td>4.0652</td>
<td>38113</td>
<td>19086</td>
</tr>
<tr>
<td>1</td>
<td>146.02</td>
<td>4.0652</td>
<td>76183</td>
<td>38121</td>
</tr>
<tr>
<td>0.5</td>
<td>146.42</td>
<td>4.0652</td>
<td>192459</td>
<td>96251</td>
</tr>
</tbody>
</table>


3 RESULTS AND DISCUSSIONS

The initial chassis design displayed elevated stress levels near the rear control arm attachment point during the lateral acceleration test, which consistently exhibited the highest maximum stresses. In this configuration, it recorded a stress level of 392.71 MPa, as depicted in Figure 7 (a), exceeding the regulatory limit of 180 MPa (SAE, 2023). Consequently, the initial chassis iteration, developed in collaboration with other subareas, did not meet the project requirements and necessitated improvement.

The second chassis version represents an enhancement of the initial model. It includes the addition of two crossbars, one at the front and one at the
rear, which triangulate the chassis floor. This modification aimed to mitigate the shortcomings of the previous model. Analysis of stresses revealed a reduction in the maximum stress, now centered in the middle of the chassis, near the base of the main hoop. Although the stress decreased to 227.42 MPa, it still exceeded the 180 MPa limit, as illustrated in Figure 7b.

In the third chassis iteration, the entire chassis floor was triangulated from the rear suspension to the front, excluding the front region due to its minimal influence on stress and rigidity analysis. Despite shifting the location of maximum stress, it marginally increased to 230.36 MPa in the rear hexagonal structure, which supports the lower mounts of the rear control arms, as shown in Figure 7c.

The fourth and final chassis version involved thickening the rear hexagonal structure from 1.2 mm to 2.5 mm. This change was necessitated by the inability to triangulate this area due to the presence of transmission system components, such as the differential and chain. This iteration successfully met the maximum stress criteria, recording a stress level of 146.02 MPa, as depicted in Figure 7d.

Figure 7 Lateral Acceleration Test: a) Version 1; b) Version 2; c) Version 3; d) Version 4.

Table 3 – Experiments results.

<table>
<thead>
<tr>
<th>Model</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Weight (kg)</td>
<td>32.17</td>
<td>32.94</td>
<td>34.38</td>
<td>34.99</td>
</tr>
<tr>
<td>Maximum longitudinal torsional stress (MPa)</td>
<td>113.03</td>
<td>100.72</td>
<td>61.95</td>
<td>61.86</td>
</tr>
<tr>
<td>Longitudinal torsional rigidity (Nm/deg)</td>
<td>1 394.99</td>
<td>1 514.18</td>
<td>1 976.22</td>
<td>1 985.26</td>
</tr>
<tr>
<td>Maximum stress in diamond twist (MPa)</td>
<td>27.47</td>
<td>27.31</td>
<td>16.22</td>
<td>16.10</td>
</tr>
<tr>
<td>Diamond torsional rigidity (Nm/deg)</td>
<td>33 650.50</td>
<td>35 206.60</td>
<td>56 196.7</td>
<td>56 303.70</td>
</tr>
<tr>
<td>Maximum stress at lateral acceleration (MPa)</td>
<td>380.72</td>
<td>223.37</td>
<td>226.45</td>
<td>143.52</td>
</tr>
<tr>
<td>Maximum braking stress (MPa)</td>
<td>109.80</td>
<td>109.99</td>
<td>110.19</td>
<td>110.32</td>
</tr>
</tbody>
</table>


The results of the best-performing model, Version 4, have been graphically plotted. In Figure 8a, the maximum stress is depicted, which is in the tubes to which the springs are attached. Figure 8b illustrates the maximum and minimum deformations, corresponding to the points of force application responsible for longitudinal torsion. Similarly, in the diamond torsion test, stress distributions are shown in Figure 8c, with the maximum stress near the traction side of the rear control arm and the minimum stress near the compression side at the front. Figure 8d displays the deformations along the Z-axis, with the maximum translations occurring at the points of force application.

Figure 8 – Results of Version 4 Torsion Tests; a) Longitudinal Torsion Stress, b) Longitudinal Torsion Deformation in the Y-axis Direction, c) Diamond Torsion Stress, and d) Diamond Torsion Deformation in the Z-axis Direction.

Furthermore, Figure 9 represents the stress distribution in the braking test, where the highest stress is observed at the center of the spring support bar due to the non-axial load moment generated by the springs.

![Figure 9 – Stress in Version 4 during braking.](source: The author, 2023)

4 CONCLUSIONS

In conclusion, this study provided a comprehensive analysis of both stress distribution and torsional rigidity in four distinct chassis versions designed to comply with the Formula SAE regulations. The numerical analysis was carried out enabling a thorough evaluation of their performance.

Among the tested iterations, the fourth version emerged as the standout, successfully passing all stress-related tests with the highest recorded stress being a mere 146 MPa. This achievement underscores its structural integrity and compliance with regulatory stress limits, particularly the stringent 180 MPa requirement.

Additionally, the longitudinal torsional rigidity of this optimal chassis model was determined to be 1985 Nm/deg, well within the range of values reported in the literature for Formula SAE competitions. Notably, the incorporation of floor triangulation significantly enhanced resistance to both longitudinal and diamond torsional stresses. Furthermore, the chassis exhibited a robust diamond torsional rigidity of 56,304 Nm/deg, further attesting to its structural prowess.
Importantly, the mass of the fourth version was maintained at a competitive 35 kg, aligning with expectations, and ensuring a favorable power-to-weight ratio for optimal performance on the track.

In summation, this work not only successfully met regulatory standards but also demonstrated a clear and effective methodology for chassis design and optimization in Formula SAE competitions, serving as a valuable contribution to the field of motorsport engineering. The fourth version stands as a testament to the iterative design process and the integration of scientific principles into real-world engineering, promising improved performance and competitiveness for future events.
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